Proceedings of the Second U.S.

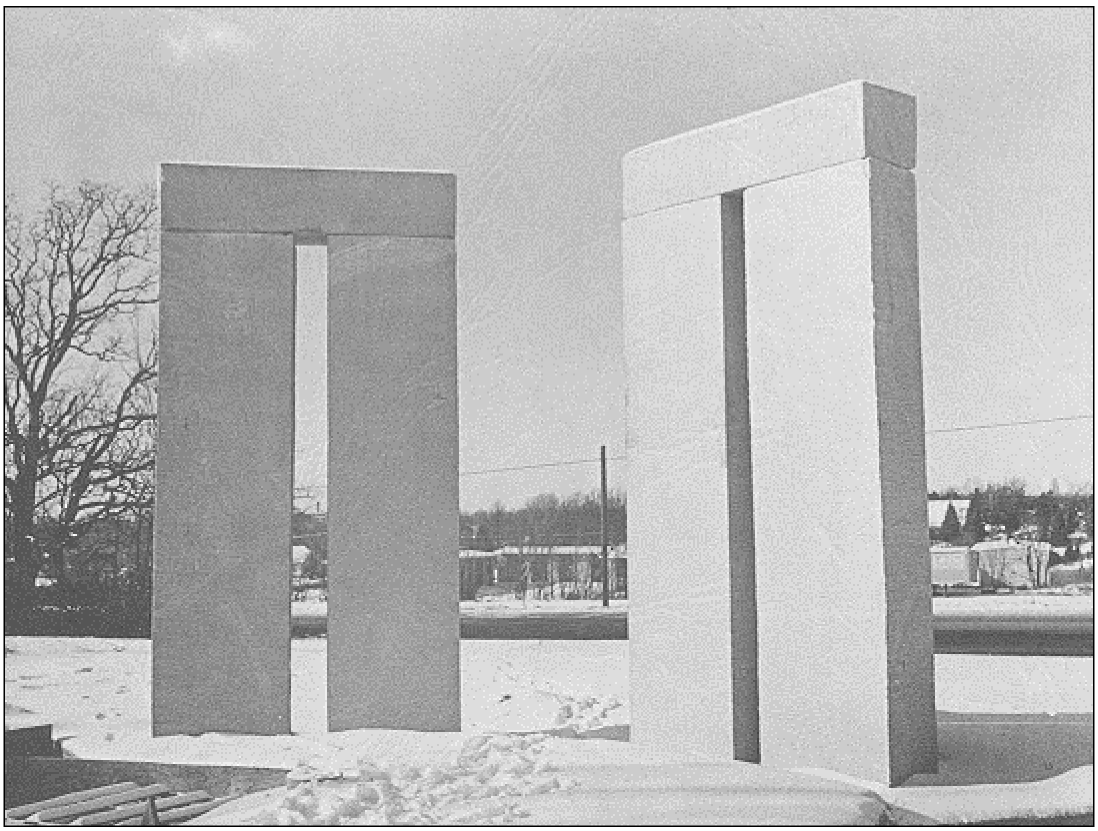
WATER JET CONFERENCE

May 24-26, 1983
Rolla, Missouri

Edited by: David A. Summers and Frank F. Haston

Sponsored by School of Mines & Metallurgy, University of Missouri-Rolla

Published by: University of Missouri-Rolla, Rolla, Missouri 65401



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DIMENSIONLESS PIPE LENGTH ANALYSIS FOR JET MODULATION SYSTEMS

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ABSTRACT

A dimensionless pipe length analysis has been developed to summarize the possible responses of a modulator system for any combination of pipe lengths. The parametric summary has as many variables as pipes in the modulator system. The transfer matrix method with pipe friction neglected is used for the response equations. The results are compared to the more exact method of characteristics.

INTRODUCTION

The pressure transient produced by the impact of the leading edge of a liquid jet or droplet with a solid surface has been estimated to be several times the stagnation pressure of the moving liquid [4, 5]. In order to increase the number of leading edge transients and thereby increase the fragmentation efficiency, conventional jets have been interrupted, pulsed or modulated to form a series of discrete water bunches each producing a water hammer pressure pulse [7, 8, 9]. Large interruption frequencies on the order of several thousand cycles per second are needed to maximize the number of liquid particle impacts while still producing a water package large enough to produce significant local fragmentation. Periodic jet flows with frequencies in this range have been produced by a periodic variation in the flow resistance [4], an oscillating volume in the flow circuit, or a self-excited oscillation [6]. With the proper frequency and wave shape, large water drops can be made to form near the impact surface producing repetitive water hammer pressure transients. One can speculate on the possibility of controlling the spacing between stress waves in such a way as to reinforce incident and reflected waves.

In producing a modulated jet, the liquid compressibility and the elasticity of the piping system can cause the modulation response to be very different from the input signal. For example, it is not uncommon for the modulation response to be completely attenuated by the piping system at certain frequency intervals. A theoretical study of a periodic flow resistance modulator [3] predicted that a significant distortion of the wave shape, producing unwanted secondary droplets, is produced for all frequencies except those that place odd harmonics on both sides of the oscillator. Therefore, analytical methods that can predict modulator response can be very beneficial in assessing the relative merits of various oscillator and piping configurations.

The transfer matrix method [2] and the method of characteristics [1] can both be used to predict modulator response. However, when the response wave shape is required, the method of characteristics must be used. This method retains all of the non-linear frictional and piping component relationships and calculates the pressures and flow rates at selected points along the piping system for each time step from the start of the transient until a steady oscillatory condition is realized. Since a hundred or more oscillations are often required to reach the steady oscillatory state, considerable computer time is required to calculate a complete response spectrum for a single modulator configuration.

During the design of a modulator system it is beneficial to obtain a system response spectrum showing the amplitude of the output oscillations over the entire range of driving frequencies. These relationships predict the frequencies for which the modulator operates most efficiently and regions where the oscillation is attenuated by the piping system. The transfer matrix method, which solves the governing equations directly for the steady state amplitudes of oscillation provides an economical means of obtaining the response diagrams. Even though approximations are made to linearize the viscous terms and the non-linear boundary conditions, this technique can be used to identify areas of interest for which more accurate calculation can be made with the method of characteristics.

The possibility of developing a modulator that greatly amplifies the input signal by resonating the piping system is of interest in minimizing the power required to drive the oscillator. However, assessing the potential for a given modulator configuration to produce a large amplification of output would require a response spectrum for each combination of pipe length, pipe diameter, wave speed and boundary conditions. Even using the transfer matrix method, where only one closed form calculation is required to obtain a response at each driving frequency, the number of calculations required to obtain a clear picture of the modulation possibilities quickly becomes very time consuming if not prohibitive as the modulator complexity increases.

In this study the transfer matrix method is used to predict the system response. However, the governing equations are developed in terms of dimensionless pipe lengths and parametric calculations are made in terms of these quantities. The resulting summary curves greatly reduce the number of calculations required to assess the modulation possibilities of a given system of components. Specific dimensions required to produce a particular condition predicted by the summary curves can be easily obtained.

THE TRANSFER MATRIX METHCD

Since the dimensionless pipe length analysis uses the transfer matrix method for the system response equations, a brief discussion of the method is included. Complete details of the method are presented by Chaudhry [2]. This technique develops linear equations relating pressures and flow rates at a point upstream of the particular modulator component to these quantities at the downstream end of the component. Then the relationship between flow rates and pressures at the upstream boundary of the entire system is obtained in terms of these quantities at the downstream boundary by an ordered multiplication of the equation matrices for each component in the modulator system. Examples of the component matrices for a pipe section and a pipe junction will be presented.

The equations of motion for a section of pipe are solved by assuming that the instantaneous pressure, H, and discharge, Q, are mean quantities (H_o, Q_o) plus deviations from the mean (h, q). The deviations are assumed to be sinusoidal with respect to time. The flow rate and pressure on the left side of the (i+l)th section are obtained in terms of these quantities on the right side of the ith section as follows.

$$q_{i+1}^{L} = \cos\frac{L_i\omega}{a_i}\, q_i^R - \frac{j}{C_i}\sin\frac{L_i\omega}{a_i}\, h_i^R \qquad (1)$$

$$h_{i+1}^{L} = -\,jC_i\sin\frac{L_i\omega}{a_i}\, q_i^R + \cos\frac{L_i\omega}{a_i}\, h_i^R \qquad (2)$$

and

$$C_i = \frac{a_i}{gA_i} \qquad (3)$$

where a_i is the water hammer wave velocity in the ith pipe, g is the acceleration due to gravity, A_I is the area of the pipe, L_i is the length of the pipe and ω is the driving frequency. The equation matrix for a length of pipe is composed of the coefficients of q_i^R and h_i^R in Equations (1) and (2) above.

Similar equations are obtained for each component in the modulator piping system. For example, at a series pipe junction where only the pipe size changes, the flow rate and the pressure are normally assumed to be constant. Therefore, the equations relating upstream quantities to those downstream are given by:

$$q_{i+1}^{L} = q_i^L + O\left(h_i^L\right) \qquad (4)$$

$$h_{i+1}^{L} = O\left(q_i^L\right) + h_i^L \qquad (5)$$

The equation matrix for a series pipe junction is the identity matrix since the coefficient in the equations above are either zero or one. When equation matrices for each component in the modulator are obtained they are multiplied in the proper order to produce two equations relating pressure and flow rate at the upstream boundary of the system to these quantities at the downstream boundary. At the system boundaries normally two of these quantities are known and the two equations can be used to solve for the remaining two unknowns.

When the method is applied to the branch piping system shown in Figure 1, the following equations result

$$q_3^R = \frac{\bar{q}}{\sqrt{R^2 + I^2}} \qquad (6)$$

$$R = \frac{a_1}{A_1}\ \frac{A_2}{a_2}\tan\frac{L_1\omega}{a_1}\ \sin\frac{L_2\omega}{a_2}\ \cos\frac{L_3\omega}{a_3} + \frac{a_3}{A_3}\ \frac{A_2}{a_2}\sin\frac{L_2\omega}{a_2}\ \sin\frac{L_3\omega}{a_3} - \cos\frac{L_2\omega}{a_2}\ \cos\frac{L_3\omega}{a_3} \qquad (7)$$

$$I = \frac{2H_o}{Q_o}\left[\frac{gA_1}{a_1}\tan\frac{L_1\omega}{a_1}\ \cos\frac{L_2\omega}{a_2}\ \cos\frac{L_3\omega}{a_3} + \frac{gA_2}{a_2}\sin\frac{L_2\omega}{a_2}\ \cos\frac{L_3\omega}{a_3} + \frac{gA_3}{a_3}\cos\frac{L_2\omega}{a_2}\ \sin\frac{L_3\omega}{a_3}\right] \qquad (8)$$

where q is the amplitude of the input oscillation of flow, H_o is the mean pressure at the nozzle, Q_0 is the mean flow rate and a_i , A_i , L_i and ω are the wave speed, pipe area, pipe length and modulation frequency respectively. This equation can be used to calculate the response flow rate at the nozzle q_3^R for a given input oscillation of flow q for any modulator frequency ω.

THE DIMENSIONLESS PIPE LENGTH APPROACH

The arguments, Θ_i of the trigonometric functions in Equations (7) and (8) are given by:

$$\Theta_i = \frac{L_i\omega}{a_i} = \frac{2\pi F L_i}{a_i} \qquad (9)$$

where F is the driving frequency in cycles per second. Since the quantities in Equation (6) are squared, the trigonometric functions repeat their values every π radians. Therefore a dimensionless pipe length, M_i can be defined as:

$$M_i = \frac{\Theta_i}{\pi} = \frac{2FL_i}{a_i} \qquad (10)$$

where all of the response information from Equation (6) is obtained as the dimensionless pipe lengths vary from zero to one. Since there are three dimensionless pipe lengths, the parametric study will be three-dimensional. Equation (6) was written in terms of the dimensionless pipe lengths and the amplitude of the modulated flow at the nozzle q_3^R non-dimensionalized with the mean flow Q_0 was calculated for the full range of dimensionless lengths. Contours of modulation response are presented versus dimensionless pipe lengths in Figures 2 through 6. The input oscillation was 20% of the mean flow for all of the calculations. Figures 3 and 5 show relatively large areas of modulation in excess of 20%. Contours up to 51% were plotted.

To illustrate how this maximum area of amplification can be realized by the modulator of Figure I at any chosen driving frequency, we first observe that the region of interest occurs where M_1 is 0.64, M_2 is 0.2 and M_3 is 0.4. If we assume that the wave speed, a_i is 4500 feet per second and the desired driving frequency is 450 cycles per second, Equation (10) can be used to determine the lengths of each pipe to produce the 51% modulation. The pipe lengths are 3.2, 1.0 and 2.0 feet respectively. Recalling that the modulation response repeats as the dimensionless pipe lengths are increased by an integer, M_1 could be taken as 1.64, M_2 as 1.2 and M_3 as 1.4 giving pipe lengths of 8.2, 6 and 7 feet respectively. Combinations of these lengths as well as those resulting from larger integer multiples would also produce the same response.

The transfer matrix and the method of characteristics were both used to calculate a response spectrum for the modulator designed above and the results are shown in Figure 7. The frequency has been non-dimensionalized with the theoretical frequency F_{th} which is defined by:

$$F_{th} = \frac{1}{4\sum_{i=1}^{n} \frac{L_i}{a_i}}$$

where only pipes in the main pipeline are included.

The modulator in this example has a theoretical frequency of 268 cycles per second. One can observe that the 51% modulation occurs at a dimensionless frequency of 1.7 which corresponds to a driving frequency of 450 cycles per second. Although this response spectrum has several points of amplified modulation, they all occur at very narrow frequency ranges. However, this example illustrates how other configurations could be analyzed for a possible broader band of amplification. One should also notice the similarity between Figures 2 and 6 and Figures 3 and 5. In both cases the figures are identical when rotated 180^o . Therefore additional condensation of the response curves can be accomplished.

Figure 7 illustrates the agreements between the transfer matrix method and the method of characteristics. The transfer matrix method can be seen to have its greatest deviation in areas where the piping system is attenuating the modulation.

The series piping system shown in Figure 8 can be completely described with a single response summary curve shown in Figure 9. This response contour curve shows that there are no areas of modulation amplification. In fact, only the corner points appear to approach full modulation. All other areas indicate significant attenuation. This curve is also identical to the response for the branch system of Figure I with the dimensionless length of branch 3 equal to zero. The response spectrum for the series system shown in Figure 8 is given in Figure 10. The points of full modulation occur at dimensionless frequencies of 6 and 12. All other points in the spectrum experience considerable attenuation of the output. The transfer matrix method and the method of characteristics are both used to calculate the system response in order to illustrate how well the more approximate transfer matrix method predicts the response.

CONCLUSIONS

Parameters in the modulator response equations have been defined in terms of dimensionless pipe lengths allowing one set of curves to summarize all of the responses for any combination of pipe lengths. The resulting parametric summary has as many variables as pipes in the modulator system. The summary response curves exhibit symmetry and mirror imaging which suggests that additional reduction in response calculations might be possible. Simpler modulator configurations can be analyzed as special cases of a more general system by allowing one or more of the dimensionless lengths to be zero. The dimensionless pipe lengths can be used to calculate actual lengths required to produce a desired response at any chosen frequency.

The transfer matrix method of calculating modulator response is compared to the more exact method of characteristics. The agreement between the two methods justifies the use of the highly efficient transfer matrix method in predicting the response of a modulator system. The greatest deviation in the two methods occurs in regions of the highest attenuation. However, the transfer matrix method assumes a sinusoidal wave shape and only predicts the amplitude of the oscillation. Therefore, it cannot be used to predict the resulting wave shape.

ACKNOWLEDGEMENTS

This research was funded by a Southern Illinois University at Carbondale Coal Extraction and Utilization Research Center Special Research Contract with the U.S. Department of Energy.

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NOTATION

a	water hammer wave velocity.
A	pipe area.
F	modulation frequency.
g	acceleration due to gravity.
H_0	mean pressure.
h	amplitude of the deviation in pressure from the mean.
i	denotes the ith pipe or point.
j	imaginary component.
L	pipe length.
L	superscript denotes the left side of a pipe or point.
M	dimensionless pipe length.
Q_O	mean flow rate.
$\underline{q}$	amplitude of the deviation in flow rate from the mean.
q	amplitude of the volumetric oscillation at the modulator.
R	superscript denotes the right side of a pipe or point.
Θ	argument of trigonometric functions.
ω	radial frequency of the modulator.

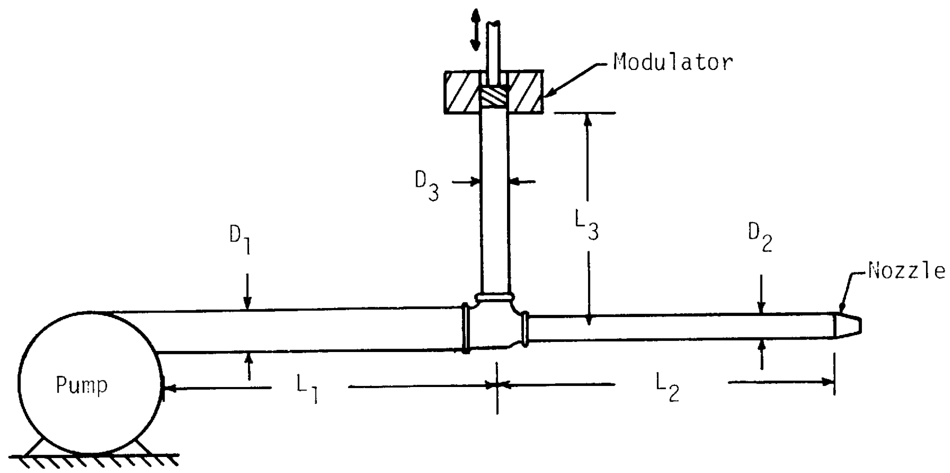

Figure 1. Branch System Modulator

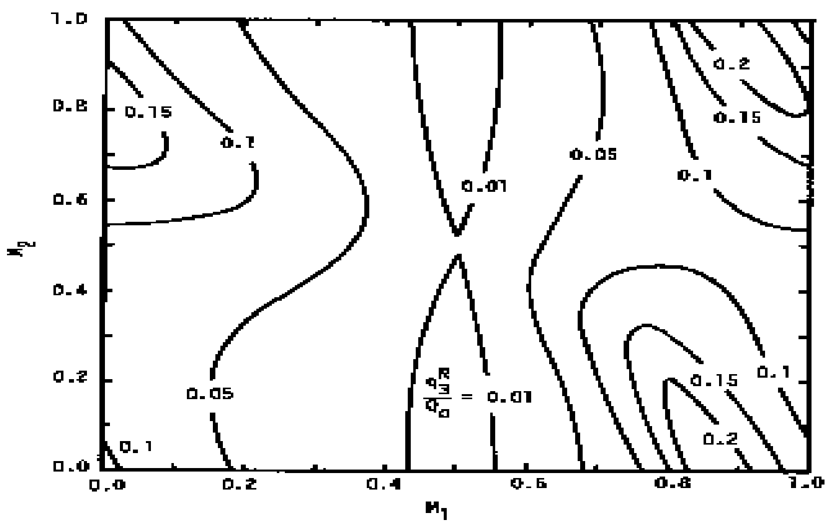

Figure 2. Modulation Response for Branch System (M_3= 0.2).

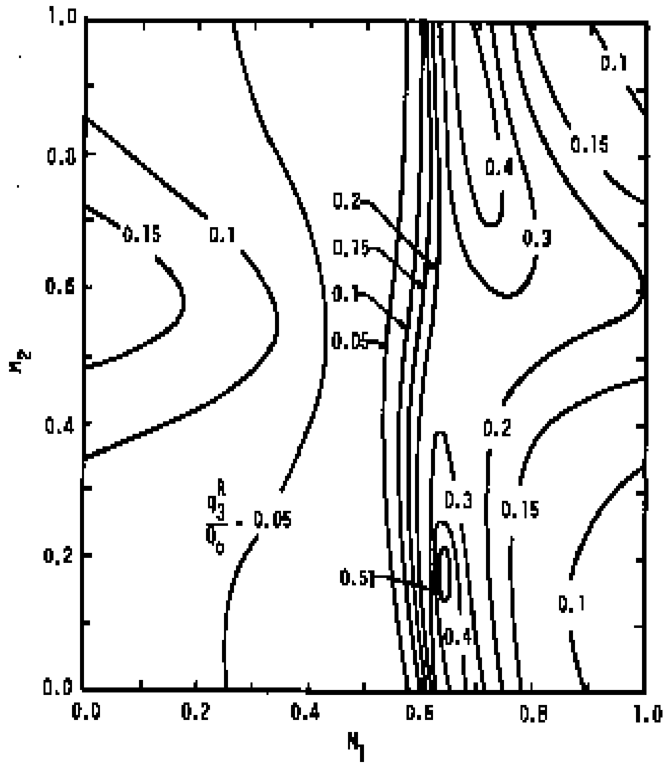

Figure 3. Modulation Response for Branch System ($M_3 = 0.4$).

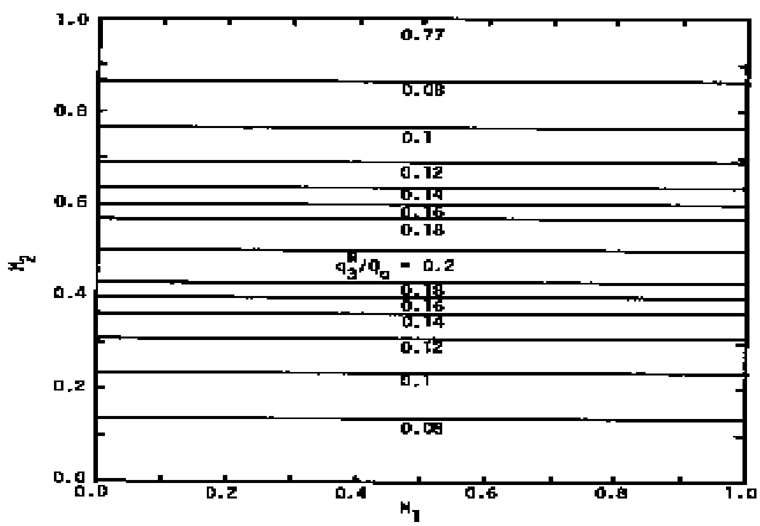

Figure 4. Modulation Response for Branch System ($M_3 = 0.5$).

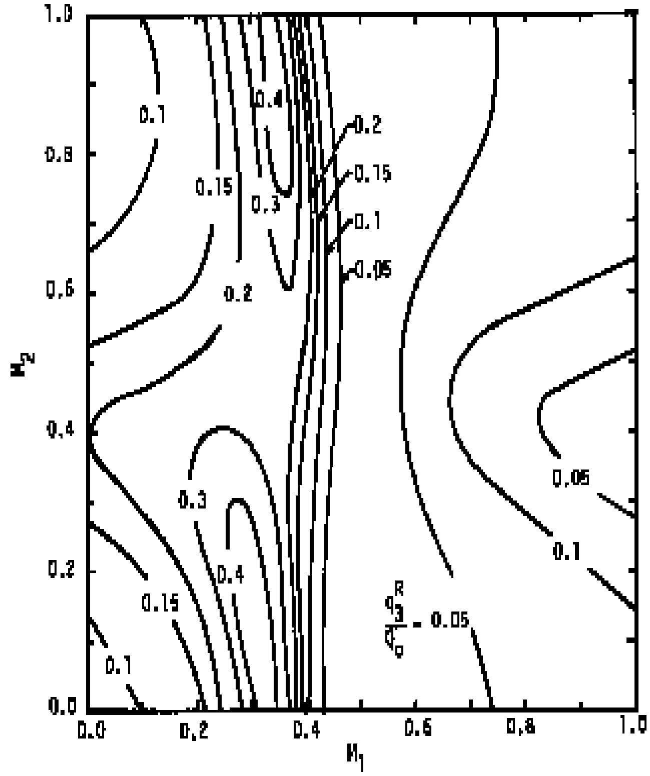

Figure 5. Modulation Response for Branch System (M_3=0.6)

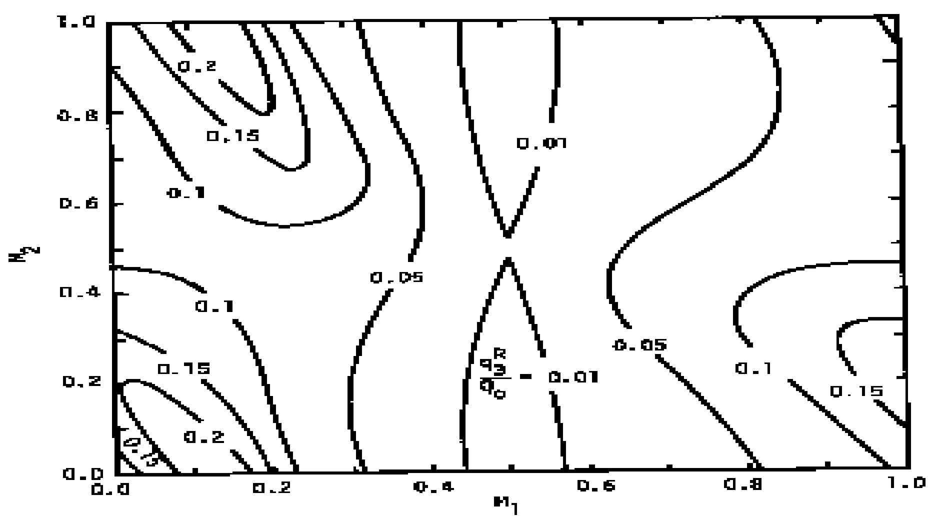

FIGURE 6. Modulation Response for Branch System (M_3=0.8)

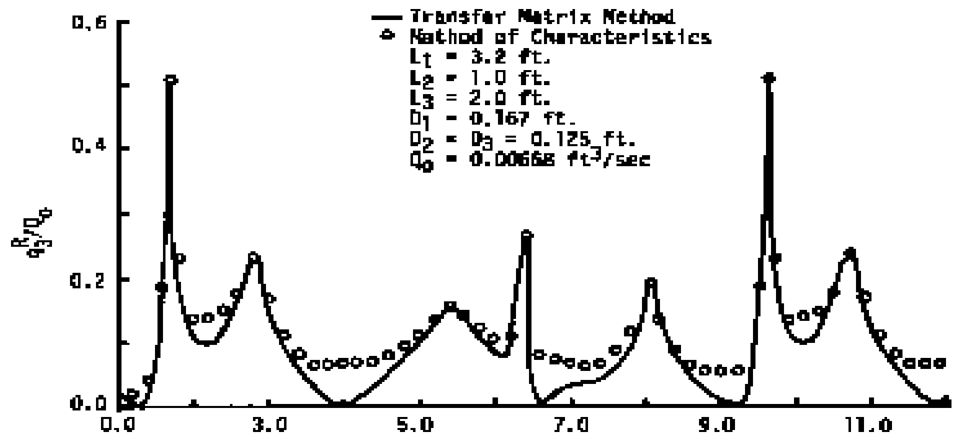

Figure 7. Response spectrum of branch system.

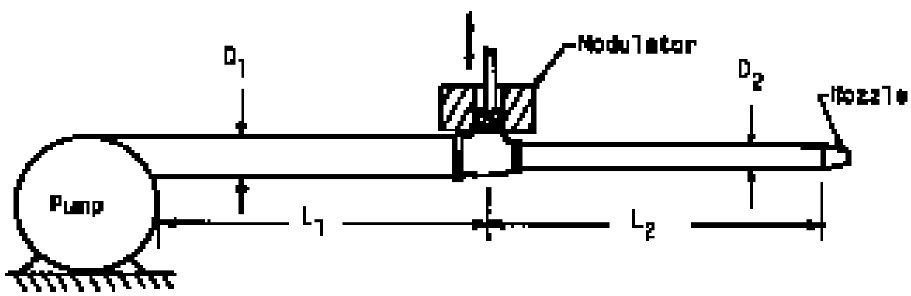

Figure 8. Series Modulator.

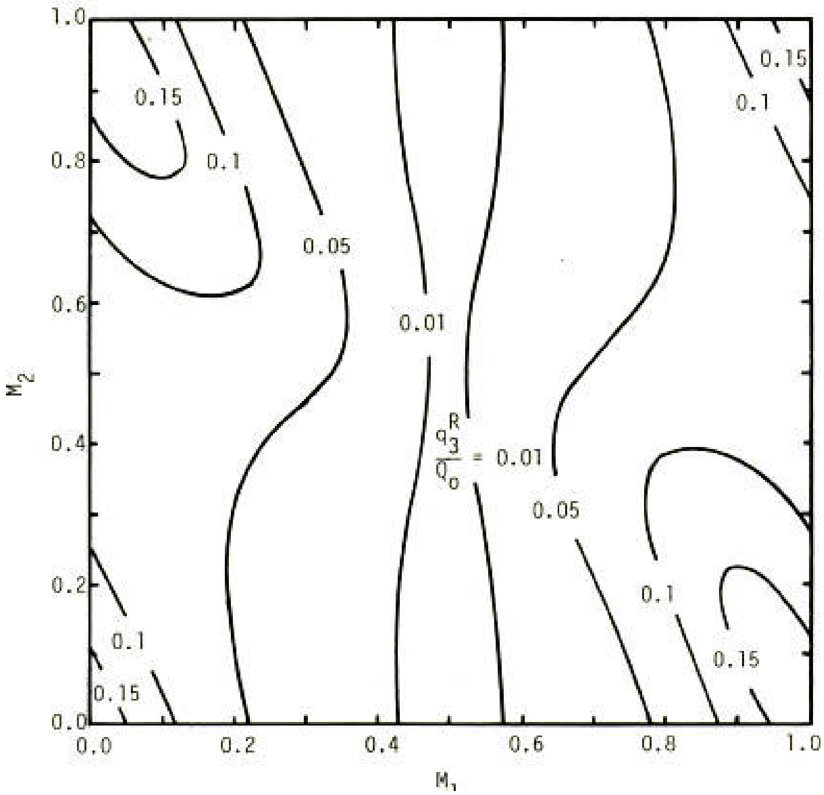

Figure 9. Modulation response for series system.

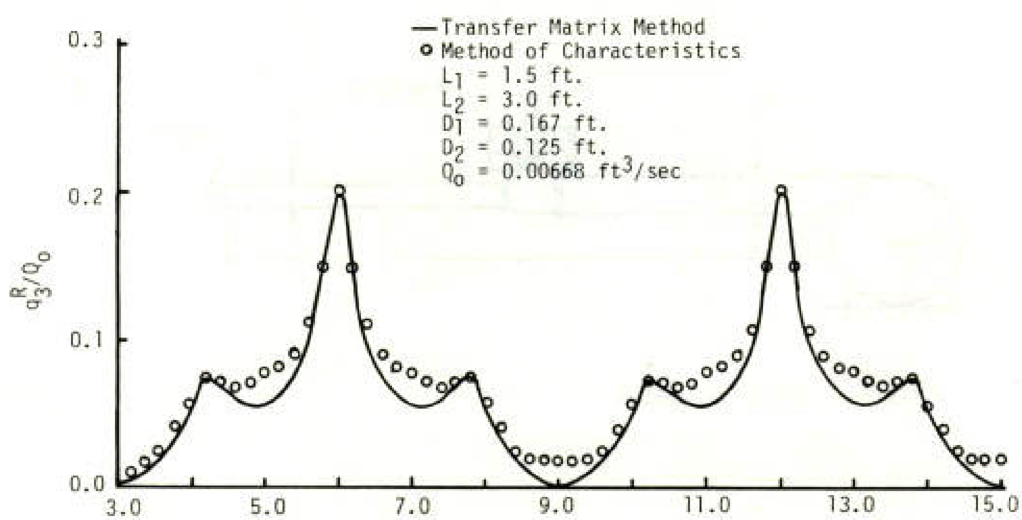

Figure 10. Response spectrum for series ystem.

DISCUSSION

NAME: Eugene B. Nebeker
COMPANY: Scientific Associates, Inc.
QUESTION TO: J. L. EVERS

SESSION l: THEORETICAL
QUESTION: "What are the limitations in the type of modulation input function that this approach will accept? For instance, are only sinusoidal inputs acceptable? How would you accomodate more complex inputs?"

ANSWER: The method of characteristics can handle any form of input. This method was used in our paper that was presented at the 1st U. S. Water Jet Conference to describe the modulation produced by a sawtooth shaped resistance. The transfer matrix method will onlyhandle sinusoidal input in its present form. However, it might be possible to express a complex shape in a Fourier series and apply the method to the series.

NAME: Mohamed Hashish, Senior Research Scientist
COMPANY: Flow Industries
QUESTION TO: J. L. EVERS

SESSION l: THEORETICAL
QUESTION: l. "What is the range of parameters (pressure and flow rate) for which the analysis in the paper is valid? It seems that for low flow rates of water, typically encountered in jet cutting applications, modulation is not going to be effective, do you agree?"

ANSWER: There is nothing in these analyses that would limit either pressure or flow rate. However, the governing equations have been linearized in developing the transfer matrix method. This would limit the degree of modulation that could be described with accuracy. The method of characteristics retains all of the nonlinear terms and thus has no such limitation.

I have not tried the analysis on very low flow rates that one might encounter in cutting with small jets. This is an interesting point that should be checked.

QUESTION: 2. "For effective modulated jet, the wave shape of the pulse should be square. The transform matrix method assumes sinusoidal response and hence is not the right method to use....what is your comment on that?"

ANSWER: The transfer matrix method in its present form will only handle sinusoidal input. However, the method of characteristics, even though more costly in computer time, can be used for any form of input. In fact we studied sawtooth shaped waves in a previous paper.

QUESTION:3. "Did you try other piping configurations that might be more effective in producing better modulation rations?"

ANSWER:We only used one piping configuration to illustrate the method. The method will now be used to identify the properties of effective piping systems for modulators.

NAME: Dr. M. Vijay
COMPANY: National Research Council
QUESTION TO: J. L. EVERS

SESSION l: THEORETICAL
QUESTION:l. "What is the reason for the discrepancy between your experimental data, and the theoretical curves (method of characterization)?"

ANSWER: The experimental set-up will require additional refinement to insure a better control over the boundary conditions, and the properties of the flowing fluid. For example, air trapped in the system as well as air entrained in the fluid can drastically change the compressibility of the fluid. Piping components such as valves, pumps and ties introduce other complexities that have not been adequately described. An attempt will be made to eliminate as many of these as possible.

QUESTION:2. "How do you protect the pump from the pulsation?"

ANSWER: This analysis has not addressed the effect of pulsation on the pump. However,the level of the pressure pulses required to modulate a jet is usually low.

AN ANALYSIS OF ONE POSSIBILITY FOR PULSATING A HIGH PRESSURE WATER JET

Dr. M. Mazurkiewicz,
Assistant Professor of Mechanical and Aerospace Engineering,
University of Missouri-Rolla,
Rock Mechanics and Explosives Research Center,
Rolla, Missouri

ABSTRACT
On the basis of the present state-of-the-art, the author presents a discussion of a new method for producing a jet made up of distinct liquid segments.

In the paper, a theoretical analysis of the potential for use of laser beams to cut a regular jet into separate segments is presented. The level of laser beam frequency, power consumption and the anticipated shape of the leading edge of segment are also discussed.

SIGNIFICANCE OF THE LIQUID - SOLID IMPACT EVENT

The last decade has seen high pressure water jet systems being utilized commercially as a very effective cutting, cleaning, and excavation tool. This could only have happened with a considerable advance in the available high pressure operating equipment. Commercial pumps and fittings, now available can be operated in the range of 15,000 to 20,000 psi pressure and above. Such large pressures generate liquid jets with extremely high energy densities, sufficient to cut most construction materials. Further progress is possible either by increasing the operation pressure or by taking greater advantage of the specific mechanics of the liquid on the solid surface.

When the motion of a liquid segment of a regular coherent jet is abruptly stopped by a solid body, a very high pressure in the vicinity of the contact area is created for a very short duration. The initial momentary high pressure results from a "water hammer" effect, the compressibility of the liquid giving an instantaneous pressure of the order:

$$\rho_0 \cdot C_0 \cdot v$$

Pressure then decays to the ordinary hydrodynamic stagnation pressure given for incompressible flow by the equation:

$$\rho_{st} = \frac{1}{2} \cdot \rho_0 \cdot v^2$$

where:
ρ_0 = the density of the incompressible liquid
v = the normal component of the collision velocity.

The duration of the peak pressure is estimated to be:

$$\frac{2l}{C_0}$$

with a cylindrical shape to the jet segment, where:

l - the length of the jet segment

C_0 - the acoustic speed of the liquid.

The conventional "water hammer" equation which was originally proposed by Saint Venant must be corrected to allow for the compressibility of the liquid, which occurs at higher impact velocities.

Huang and Hammitt (1) found that use of the equation:

$$\frac{C}{C_0} = 1 + 2(\frac{v}{C_0}) - 0.1(\frac{v}{C_0})^2$$

gives less than a 3% error over the range of impact velocity ratios

$$\frac{v}{C_0} \leq 3$$

Beyond this level "water hammer" pressure is:

$$p_{wh} = \rho_0 \cdot C \cdot v$$

where:

C - is the shock-wave velocity of the liquid.

Comparing the effective pressure on the target surface developed by a coherent discontinuous jet (stagnation pressure) with the pressure imposed by a cylindrical jet slug ("water hammer pressure") it can be seen that a significant advantage may be obtained by making cuts with a jet of discrete slugs, rather than a continuous jet. The results of such calculations are shown in Figure 1. It is clear that the stagnation pressure developed by the jet moving with 500 m/sec is about 9% of the pressure developed by the impact of the head of the cylindrical jet segment. In other words, the level of the "water hammer" pressure could only be reached by the continuous jet, if the pump pressure is increased about 11 times, requiring a power increase of about 37 times. One can easily anticipate that this would not only be very expensive but would also create many technical problems.

A more economic way is to find a solution involving a jet interrupter which would break a coherent jet into discrete segments, one following the other and each developing a "water hammer" pressure level on the target. In order to maintain this pressure level, the frequency of impact has to be high enough to maintain a small amplitude of variation.

From a technical point of view the problem is quite difficult, because of the high jet velocity and the aggressive cutting ability of the jet. Several authors have tried to develop a working solution to this problem (Table 1). Any of these may produce an interrupted jet, but the quality of the leading edge of the jet slugs produced, from a geometric point of view, is not high enough to create the "water hammer" effect. Further, it is very difficult to obtain a truly discretely segmented jet by method 2 and 3 before the jet hits the sample. The reason is that the fluctuating velocity of the jet is not high enough to form a clear separation of the jet. In order to develop a "water hammer" pressure level, the break between segments has to be long enough to allow a segment to hit the surface and then leave before the following segment impacts.

The method used by Summers and Lichtarowicz, in which rotating plate with holes, cuts the jet, will create splashing and a deviation of the jet. The major problem, however, is in the rapid wear of the interrupter blade and the high frequency noise it generates.

PROPOSED SOLUTION

From this author's point of view, one practical solution to the problem, which would eliminate the indicated weaknesses of existing designs, is that a laser beam used to break the jet. The laser action on the water jet will evaporate the required portion of the jet thereby forming a discontinuous water jet (Fig. 2).

Taking into consideration a water jet diameter d (m); laser beam diameter L (m); ρ - density of the water (kg/m^3); C_p - heat of water vaporization (5.4 x 105 cal/kg); C_w ~ specific heat of the water (103 cal/kg deg); t the temperature difference between boiling point and existing jet temperature (60 deg C); X - dissipate ratio (2); and A conversion factor (0.24 cal/J), the necessary energy for evaporating such a segment of the jet can be found from the equation:

$$E = \frac{\pi d^2}{4} \cdot l \cdot \rho \cdot (C_p + \Delta t \cdot c_w) \cdot x \cdot \frac{1}{A} (J)$$

For a jet moving with a velocity v, the laser beam action must occur in a period of time

$$t = \frac{l}{v}$$

This means that the power consumption will be expressed in the following equation:

$$p = 3925\ d^2 v \text{, assuming } l = d$$

The results of these calculations are presented in Figure 3, for different jet diameters and where the jet is moving with a velocity in the range of 250 m/s to 3000 m/s. The range of power found from this equation is low enough that commercially available lasers of this level are available.

In order to remove regular segments having length t continuously from a water jet moving with velocity v, the laser beam must pulsate at a frequency

$$f = \frac{v}{6l}$$

The plots for such models are presented in Figure 4 and Figure 5. It is easy to conclude that for larger jet diameters the range of laser beam frequency is narrow. For a small jet diameter this range is very wide, and for very fast jets it is impractical.

The gap between the successive water segments must be long enough for each segment to impact on an uncovered surface. This means that the segments have to have time to flow from the target area by lateral flow.

THE SHAPE OF THE WATER SEGMENTS

The shape of the head of a jet segment is of great importance. This shape will depend on the duration of time between vaporization and impact of the target.

In this time, because of the drag and the surface tension, the shape of the segment or the shape of the head of the segment will vary. Still it is possible to influence the shape, changing the distance at which the laser beam cutting plane lies above the target surface.

Using the proposed technique, which is patented (5) it is possible to influence the shape of the segment head. For example, by focusing the laser beam on the center of a jet, it is possible to make a concave head. A concave head could give us the Monroe jet (6) which is, in essence, a very power concentrated micro jet.

This head shape requires a high speed camera for accurate analysis studying and we are still in the process of acquiring this facility.

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6. Mazurkiewicz, M. Barker, C.R., and Summers, D.A., "Adaptation of Jet Accumulation Techniques for Enhanced Rock Cutting," ASTM Symposium, Vail, Colorado, October 1977.

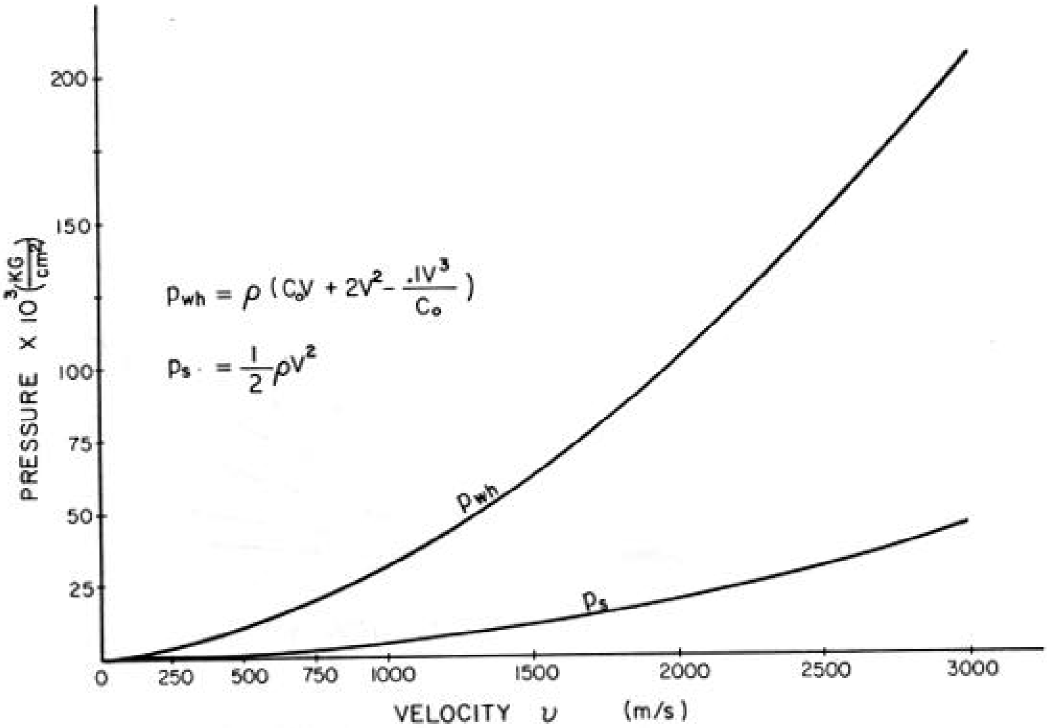


Figure 1. Water Hammer pressure (p_{wh}) and stagnation pressure (p_s) vs velocity.

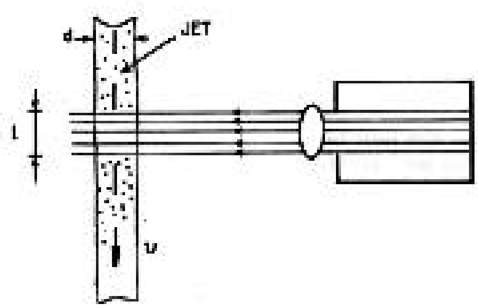


Figure 2. Interaction of water jet and laser beam.

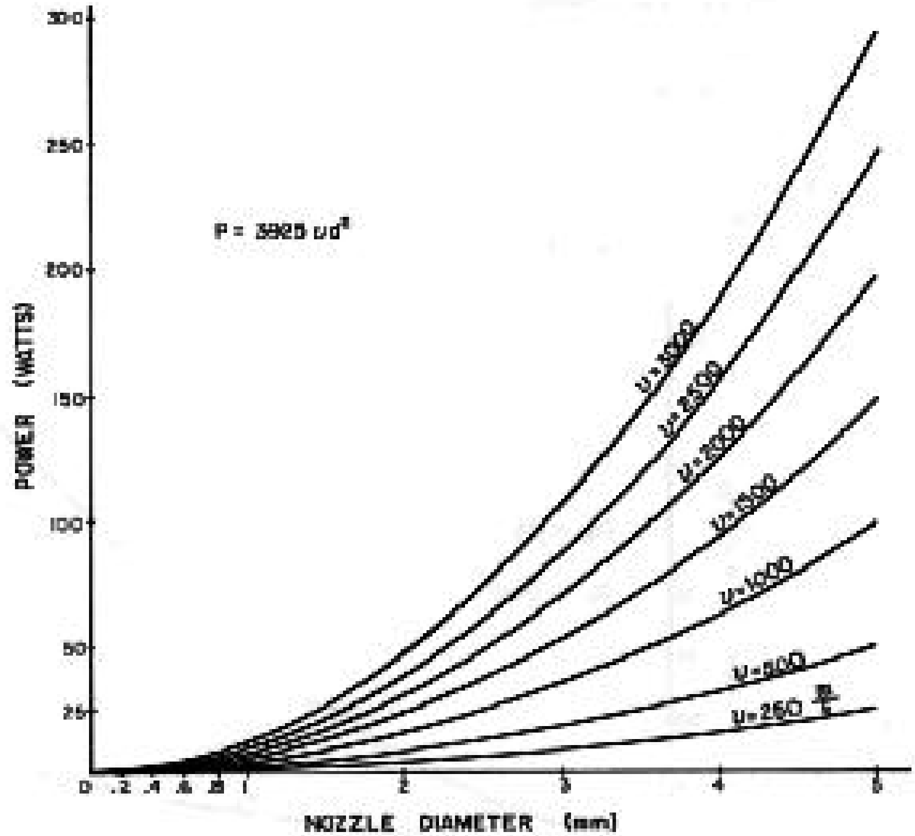


Figure 3. Power vs nozzle diameter and velocity of the jet.

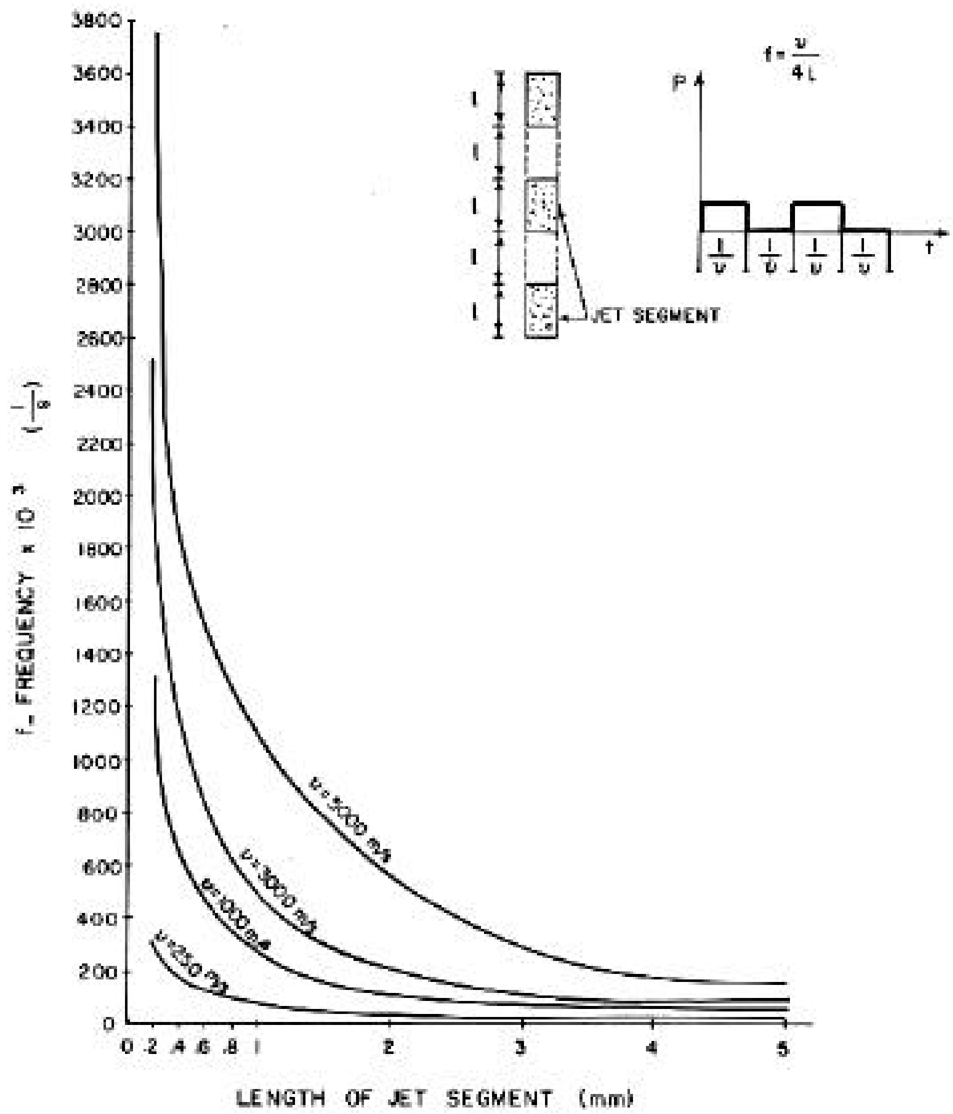


Figure 4. Frequency vs length of jet segment

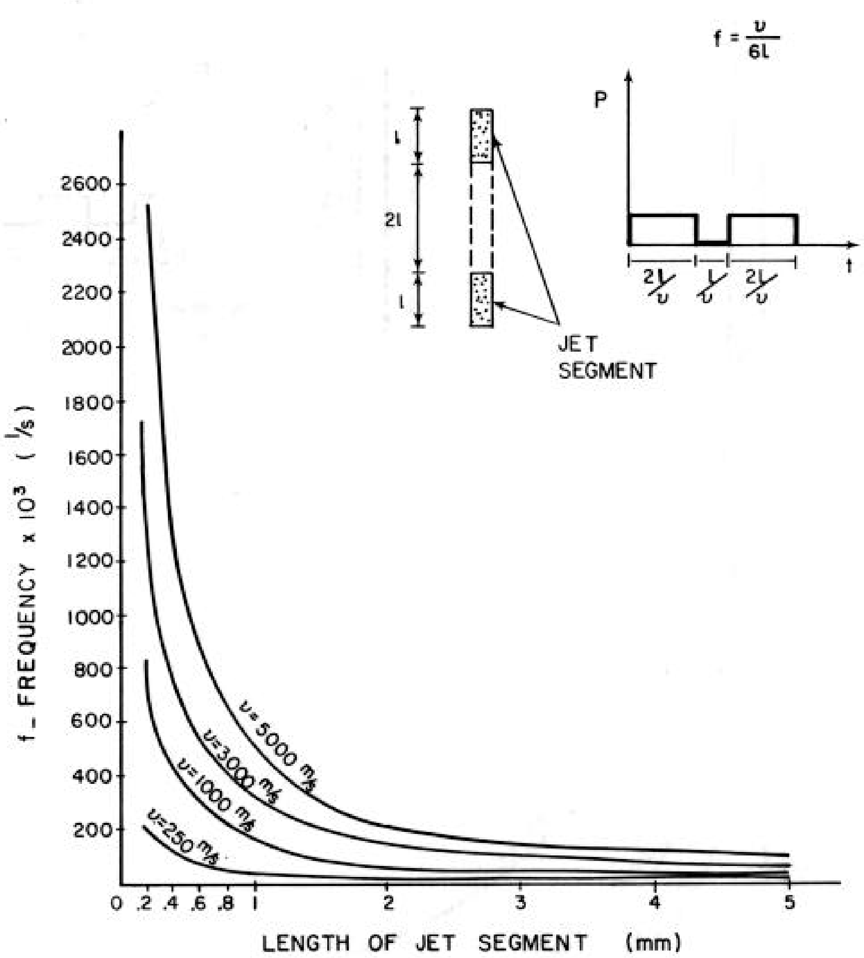


Figure 5. Frequency vs length of the jet segment

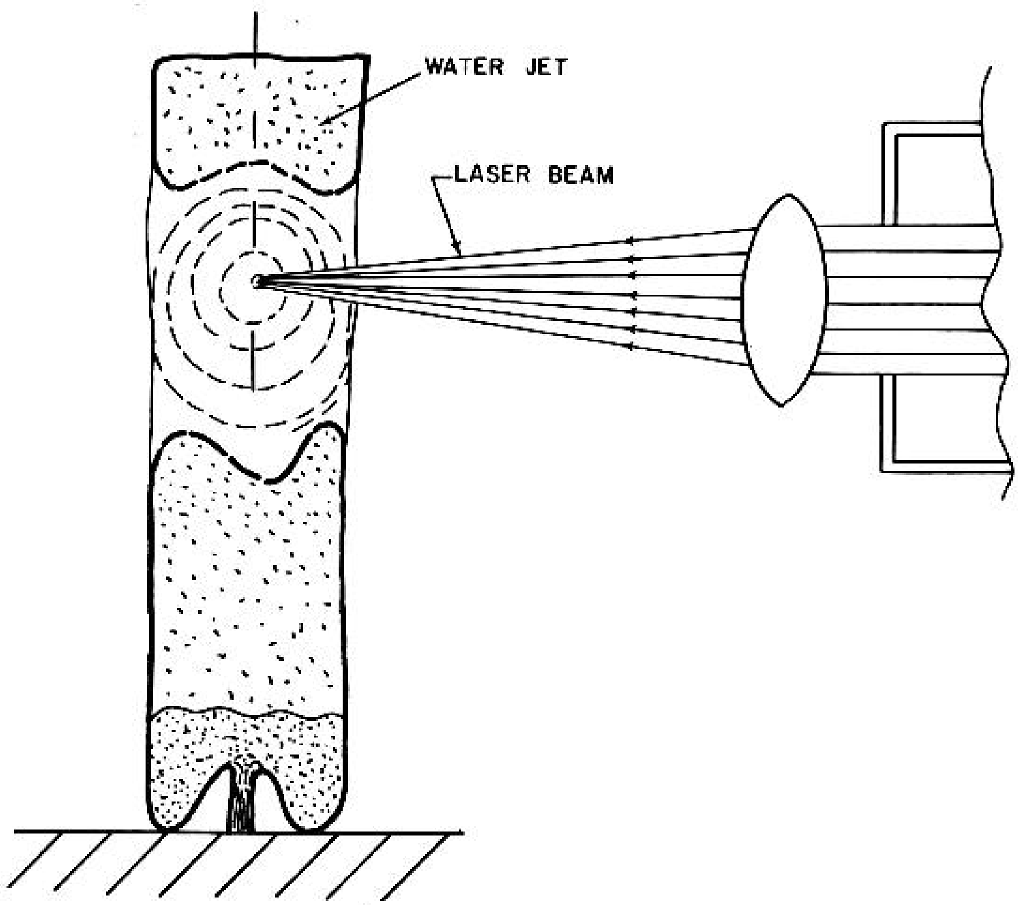


Figure 8. Laser beam concentrated on the center of the water jet and segment action on cutting material.

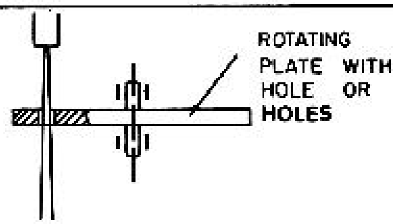
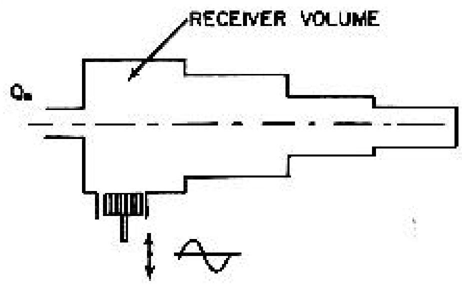
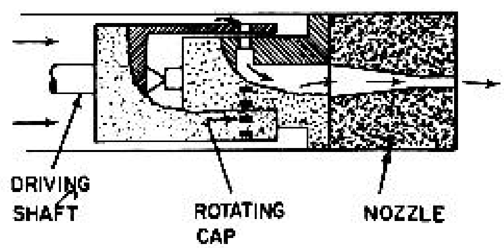
	SOLUTION	AUTHOR
1		A. LICHTAROWICZ G. NWACHUKWA (2) D. A. SUMMERS
2		E.B WYLIE (3)
3		E.B NEBECKER S.E RODRIGUEZ (4)

Table I. WATER JET BREAKERS

DISCUSSION

NAME: Gerald Zink
COMPANY: StoneAge, Inc.

QUESTION:
The power needed to vaporize half of the jet flow would seem rather high. When a continuous jet is caught and all its energy converted to heat, the water is a few tens of degrees warmer than the upstream temperature. To vaporize the water would take considerably more power.

ANSWER:
I checked my calculations once more, and I have no reservations. For evaporating 1 kg. of water in 100°C, we need 2260 10^3 Joules, which means that for evaporating a portion of water having 0.1 mm diameter and length of 0.1 mm, we need 1.775 x 10^3 Watts. This can be done in 0.001 sec. For example, the power consumption will then be 10775 Watts, for 0.0001 sec. evaporating time will be 0.1775 Watts. These results proved my calculations. It is the same range results.

NAME: William Lindenmuth
COMPANY: Tracer Hydronautics

QUESTION:
When calculating laser power, did you include efficiencies for: 1) creating the laser, and 2) laser/jet evaporation?"

ANSWER
Yes, I did. I took into consideration pretty high value of the dissipate ratio (X = 2). Possibly, further corrections can be made after further experiments.

NAME: Eugene Nebeker
COMPANY: Scientific Associates, Inc.

QUESTION:
Bear in mind that you are destroying an appreciable amount of energy and momentum when you interrupt the jet with the laser. Also the optimum shape of the front of your slug may not be flat. When you proceed into the experimental portion of your program, some insight may be obtained relative to these comments."

ANSWER:
This is true. I do destroy a portion of energy and momentum by cutting out a segment of jet; but also, I clearly developed single segments of jet, which do much more effective work than solid jets. Besides, my expectation is, that concave heads of jet segments can be created and additional gain can be achieved by Monroe jets: The experimental trials are absolutely necessary for an evaluation of a proposed method.

STANDOFF DISTANCE IMPROVEMENT USING PERCUSSIVE JETS

Eugene B. Nebeker, Ph.D.
Scientific Associates, Inc.
Santa Monica, California

ABSTRACT

A fundamental modification of the fluid mechanics of an ordinary, continuous water jet was shown to increase the standoff distance or cutting range. This paper indicates how this was accomplished by using the Percussive Jet. Examples of the results are presented.

In static firings, the maximum distance from the nozzle the Percussive Jet could cut cement block targets was 30 to 60 percent greater than an ordinary jet. At distances closer to the nozzle, the Percussive Jet often did 15 to 20 times the damage to the target material than the ordinary jet. These results will be useful to the Department of Energy high-volume and borehole hydraulic mining systems as well as to the other jet systems where cutting range is important.

Percussive Jets apply force to rock as a sequence of high-frequency impacts, rather than steadily. These Jets can be produced with a conventional, continuous flow system by modulating the discharge, i.e., cycling the discharge rate by a small amount above and below its average. With modulated discharge, the faster part of each cycle overtakes the slower in the free stream. If this process is done properly, the stream becomes bunched instead of uniform, and a Percussive Jet is produced.

The fluid mechanism by which the Percussive Jet can increase standoff is briefly discussed. Reliable and simple hardware was produced which could effectively create and control these Jets. A variety of interesting test specimen failures resulting from Percussive Jet loading were observed.

INTRODUCTION

Hydraulic coal mining has been used in various parts of the world for many years, having been used in the Soviet Union, for example, since 1935. More recently, Kaiser Resources, Ltd., Canada, has operated a most successful hydraulic mine in British Columbia and has planned to design others. The reasons for the increasing popularity of hydraulic coal mining include less capital and operating cost per unit of coal cutting capacity and production, greater safety and reduced health hazards for operators, increased equipment simplicity and reliability, etc.

Realizing these aspects of hydraulic mining, the Bureau of Mines and Department of Energy have sought to develop two basic types of large-diameter hydraulic mining systems: high-volume and borehole systems. The role of the Percussive Jet is similar in each system. The Percussive Jet technique will be used to improve the range of the jets or to improve ease by which the coal is cut at any distance from the nozzle. In practice, the conventional nozzle will be removed from these systems and replaced by a Percussive Jet modulatornozzle assembly. The ordinary water jet issuing from the nozzles will be transformed into Percussive Jets, as illustrated in Figure 1.

Earlier attempts to improve the standoff distance of water jets have consisted of improving nozzle design, conditioning or smoothing the flow upstream of the nozzle, using polymer additives in the water, and employing air-injected or water-injected shrouds around the Jet. Brown is a bibliography which cites much of this past work. This Percussive Jet program may be the first attempt to extend standoff distance by significantly altering the fluid mechanics of an ordinary, continuous water jet.

BASIC CHARACTERISTICS OF PERCUSSIVE JETS

A more detailed background of Percussive Jet in general and the more important technical considerations in particular, such as impact characteristics, acoustic considerations, fluid mechanics, modulator characteristics, etc. are given in earlier publications (Nebeker and Rodriguez, 1973, 1976, 1979). For a more detailed discussion of the work discussed here, which applies primarily to long standoff distance applications, refer to Nebeker (1981 and 1983).

However, a first approximation can be readily obtained to explain Percussive Jets (Nebeker and Rodriguez, 1979) by considering only small-amplitude modulation and disregarding various factors, e.g., aerodynamic forces, surface tension, etc., which influence the process. The small-amplitude assumption imples that second-order items in the axial and radial velocity perturbations generated by the modulation may be neglected and that pressure remains essentially uniform within the jet. This assumption is relatively acceptable in practice because in the process envisioned above, a small discharge modulation acts progressively with distance to establish the Percussive Jet.

The results obtained for this limiting case may be summarized as follows. Let the modulated discharge velocity be given by

$$U = U_{SO}[1 + F(ft)] \qquad (1)$$

U_{SO} is the average or steady component of the velocity. F(ft) is a dimensionless modulating function which varies cyclically or periodically with time t at frequency f. This modulating function could, in principle, have any waveform, e.g., sinusoidal, sawtooth, etc., but, by assumption, its amplitude is small and compared to unity. Then, the diameter D_B of the jet varies with distance x from the discharge according to

$$\frac{D_B}{D_0} = \exp[\frac{1}{2}\frac{x}{L}F'(ft - x/L)] \qquad (2)$$

D_o is the diameter of the discharge. The length L = U_{S0}/f. The prime on F' is cyclic like the parent function F. Hence, the argument (ft-xL) shows the jet envelope as a forward traveling wave with velocity USO and wave-length L. The length L, an important parameter of Percussive Jets, is best considered as a "bunching length." It simply corresponds to the length of jet stream associated with each complete cycle of the discharge modulation. Within each bunching length of jet, the progressive process of central growth and thinning away from neighboring bunches repeats itself.

The function F' in equation 2 implicitly contains the influence of modulation amplitude and waveform. The direct effect of amplitude is self explanatory. As to waveform, the appearance of the derivative of the modulating functions indicates that the jet surface responds to the suddenness or steepness of modulation changes. For given amplitude and frequency, different waveforms will imply different rates of velocity change with corresponding differences in jet behavior.

According to equation 2, the amount of jet deformation obtained at standoff distance x depends on x/L, i.e. on the number of bunching lengths contained by the standoff. This property may be understood by considering that as bunching length decreases, the velocity variation along the jet is compressed axially and will generate proportionately faster radial growth. With given discharge, bunching length decreases as frequency increases ($L = U_{S0}/f$). Hence, in addition to amplitude and waveform, the modulation frequency also influences the rate of development of stream bunching and Percussive Jet character.

The basic modulation and bunching process is affected to various degrees by many secondary factors. The interaction of the jet with the surrounding air appears most important owing to the large jet velocities involved. One aerodynamic effect is promoting the growth of modulation bunching because air pressure deficiency or suction is produced at any jet surface protuberance, such as bunch edges, whereas increased pressure is obtained in surface depressions between bunches. This effect depends in relatively complex manner on aerodynamic pressure, air acoustic speed, and surface disturbance size and shape (Weber, 1931) and Fenn and Middleman (1968). In general, however, aerodynamic pressures should help to produce Percussive Jets.

Another aerodynamic effect is the shearing off of the outside surface of the core of a jet by aerodynamic drag. This major cause of eventual destruction of a high-speed jet. This shearing action also creates water droplets which prevents direct visual observation of the core. Since the main objective of this program was to develop a jet that would cut at greater distances from the nozzle than an ordinary jet, this problem of aerodynamic shear was considered. It was minimized by operating the Percussive Jet in a very particular way which is discussed later.

Many other factors can affect the modulation bunching process, e.g., fluid friction, turbulence, velocity profile, rotational flow components, etc. Fluid friction effects are probably among the least important since the jet flow is unconfined; for example, studies of surface-tension instability indicate a very weak effect of viscosity on surface disturbance growth (Donnelly and Glaberson, 1966). One effect of velocity profile is to create a radial outflow as the nozzle exit profile decays in the free jet, thus reinforcing bunching outflow due to modulation.

DISCUSSION OF FLUID MECHANICS

The intent of this work was to modify an ordinary, continuous jet for the purpose of improving cutting ability at long distances from the nozzle. The Percussive Jet was an example of a fundamental modification of such a jet.

The knowledge which has been accumulated so far is not complete but represents a first investigation into this area. The information was taken from infra-red transmitted light photography, scattered visible light photography, force or impact gauge measurements, visual inspection of Jet impacts on concrete block test specimens, and a limited amount of mathematical modeling.

The Percussive Jet can take an ordinary jet and transform it into a bunched jet several times as wide. An example is shown in Figure 3 where the outside diameter of the Jet is increased five-fold (from 1/2-inch to 2 1/2 inches. Since aerodynamics is the principal mechanism acting to destroy ordinary jets, making a jet "fatter" would seem to be the appropriate action to take to protect the jet so it can operate at long standoff distances.

However, the actual mechanism which occurs in Percussive Jets is more complicated and interesting than this simple explanation. In actuality, a protective shroud of water droplets seems to be created around the core and bunches of a Percussive Jet. The real novelty is that this shroud is created and maintained at the average velocity of the Jet, U_{SO} .

Figure 2 is useful in visualizing this phenomenon. In this sketch, the Jet is travelling at average velocity U_{SO} from left to right. Wavelength is L considering f as the predominant frequency of modulation. The original diameter of the Jet is D_o. As discussed earlier, the faster portion of the Jet overtakes the slower region and a radial velocity component U_R is formed and produces radial bunch growth. Aerodynamic drag shears off some water from the outside extremities of the bunch. Since the bunch is travelling with axial velocity U_{SO}, this "sacrificial" water is sheared from the bunch at this velocity. This water which is sheared from the bunch forms a shroud around the Jet to help protect it from the disruptive influences of aerodynamics. An estimation of the axial velocity profile is shown. Although no distinct boundary between air and water droplets surrounding the Jet exists, a shroud envelope is sketched to illustrate the radial extent of a water-rich area. The central core of the Jet between bunches is "tight" in diameter and relatively quiescent with some minor protuberances resulting from Rayleigh instability, noise in the modulation signal, and feedline dynamic effects.

From a fluid mechanics point of view, two points are most interesting. First, the shroud is created by water sheared from the outside edges of the bunches which are travelling at velocity U_{SO}. Therefore, the shroud is travelling at a velocity approximately U_{SO}. Hence, this gives an example of a Jet shrouded by water drops which is travelling along and being constantly recreated (200 - 1000 times per second) at the velocity of the Jet. (Shrouds which have been discussed in the literature are created by expelling water or air in the vicinity of the discharge nozzle only. These shrouds rapidly lose momentum and are slowed down by aerodynamic drag.)

Second, the presence of vortices or eddys downstream of the bunches cannot be detected. Such a phenomenon would be expected from the fluid mechanics of blunt bodies travelling through a viscous medium. The absence of such effects is considered to be due to the sacrificial loss of water from the bunch to the shroud and to the repetitive nature of the Jet.

With this brief explanation as background, examination of Figure 3 is illustrative. This photograph shows an infrared photograph of a Percussive Jet discharged at 500 psig and travelling from left to right at 330 feet/second. The bunches are most optically dense and appear dark. The flow around the Jet is caused by scattered light from the mist surrounding the Jet. The Jet appears to be starting to separate just upstream (to the right) of the bunch because optical density is diminished in this area. Ligaments are being sheared off the outer edges of the bunches. These ligaments are not swirling or forming vortices but are coming back in gently curving lines. Although some protuberances appear on the central core of the Jet, the central core still maintains a "tight" 1/2 inch.

In Figure 4, the discharge pressure has been increased to 2000 psig. The Jet is now travelling at 500 feet/second. The mist around the Jet is confined to a more restricted radial dimension, as would be expected with higher velocities and increased aerodynamic drag. The angle the sheared ligaments make with the core of the Jet is much less than that in the previous photograph. Again, no vorticing or eddy flow can be seen. Separation of the core of the Jet from the leading edge of the bunch is more evident. The central core of the Jet looks "tight" and relatively undisturbed.

If Percussive Jets are slowed to about 200 feet/second so that the mist surrounding the Jet can be reduced, these jets appear to the eye to be quite different than ordinary jets. An ordinary jet looks like a homogeneous mist, whereas a central core or line can clearly be discerned in the Percussive Jet stream.

As an example of a Percussive Jet operating at extremely high amplitudes, Figure 5 is a photograph taken using scattered light. This view is helpful in understanding the fluid mechanism which allows the Percussive Jet to persist to larger distances from the nozzle than ordinary jets. The Jet is travelling from left-to-right in this photograph. The Jet velocity was extremely low so that the mist that surrounds high-speed Jets would not interfere with photography. however, the amplitude of the Percussive Jet was set very high to illustrate this effect (the amplitude is so great that these conditions would have questionable practicality in the field).

This photograph shows the bunching process creating protective "umbrellas" around the Jet. As these umbrellas move away from the nozzle, they grow in diameter and aerodynamics gradually shears off ligaments of water at their extremities. These umbrellas are about four inches in diameter and are five inches apart. The central core of the Jet is 1/2-inch.

This process serves to protect the Jet inner core from the disruptive influence of aerodynamic drag. These photographs were made by backlighting the Jet. Since water does not absorb light very well at the wavelengths of light used, the film was exposed primarily by scattering light off the Jet surface. In some of these views, the central core of the Jet can barely be seen, whereas the umbrellas are clearly visible. The central core of the Jet is protected by the outer surface of the umbrellas which is very turbulent and wavy as a result of the aerodynamic interaction. The core itself is relatively quiescent and smooth in comparison. Low speed photographs like this show the same type of mechanism as those taken of high-speed Jets using infrared techniques.

Heretofore, the fluid mechanics of Percussive Jets operated for the purpose of cutting at long distances from the nozzle has been briefly discussed. Some perspectives can be obtained by comparing the flow fields of the Percussive and ordinary Jets.

Figure 6 is a schematic representation of the Percussive and ordinary Jets discharging at 500 psig from a 1/2-inch diameter nozzle. Only the dimension along the Jet axis are drawn to scale. This information was obtained from the fluid mechanics literature of ordinary jets as well as our extensive photography of both ordinary and Percussive Jets under these conditions. Discharge velocity is about 330 feet per second.

The ordinary jet has a central core which essentially disappears and becomes discontinuous at 12.5 feet (300 nozzle diame ters) from the nozzle. Aerodynamic drag has sheared off the outer layer of the central core until nothing remains. Downstream of this point, statistically varying blobs and blotches of water, entrained with air, move downstream. The ordinary jet appeared similar to that reported in Metcalfe and Davies (1978). In our concrete cutting tests, the ordinary jet was unable to cut at distances 35 feet (840 diameters) from the nozzle.

The Percussive Jet did not show a readily discernible disappearance of the core at 300 diameters. In fact, the dominant modulation frequency could still be detected at 28 feet or 670 diameters from the nozzle. Cutting ability of the Percussive Jet decreased to a negligible level at 55 feet (1320 diameters) from the nozzle.

CONCRETE IMPACT TESTS

The purpose of this testing was to ascertain the performance of the Percussive Jet on a material that approximately simulated the properties of bituminous coal and to provide a basis upon which to compare the cutting performance of the Percussive and ordinary Jets. In this testing, two pressures were used: 500 and 20O0 psig. Modulation frequencies ranged from 100 to 1000 cycles per second (hertz). Standoff distances generally varied between 25 and 80 feet (600 to 1920 nozzle diameters).

Nebeker (1981) has reported some early work on this project. This reference describes the experimental equipment and hardware. A turbine pump was used for low pressure testing at 500 psig. For testing in the range of 2,000 psig, a Halliburton Services HT-400 pump equipment with operating crew was rented to raise the pressure to 2,000 psig. Figure 7 shows these units ready for test. The building at the left in the photograph was 45-feet long. An open end was provided because the Percussive Jet eventually strikes the ground 125 feet from the nozzle.

The firings were always static because the project as originally defined and funded did not provide for traverse facilities. The runs were 5-seconds in duration because this was about the shortest time interval that was required to turn the Jet on and off in a consistent and reproducible manner. Usually, the concrete blocks were moved away from the nozzle until the ordinary jet could no longer make an appreciable cut on the concrete but could only roughen the surface. This was the point at which the ordinary jet was considered to be unable to cut the block. The Percussive Jet was then fired on the same concrete face of the block and the relative effects photographed and measured. Then the block was moved further away from the nozzle until the Percussive Jet could no longer cut the block in 5-second duration runs.

Several concrete mixes were poured and evaluated as test specimen targets. Concrete typically consists of a mixture of sand (and gravel), cement, and water. For our purposes, a homogeneous mixture was desired, so no aggregate or gravel was used. Curing time and the water/cement ratio are the main considerations which determine the strength of concrete. In the construction industry, the water/cement ratio often is in the range of 0.6 to 0.9. Since rather weak cement was desired in order to simulate bituminous coal, the concrete mixes which were evaluated for use in this program ranged between 1.3 and 2.6.

However, the mixture that was finally chosen was suggested by Barker (1980). This mixture seemed to function satisfactorily and was also chosen so the results of this program could be more easily compared to those of others. The mixture used was prepared on a volume basis as

5 parts washed masonry sand,
1 part masonry cement
2 parts water

For Jet impact tests, the blocks were cast in cubic molds with 21-inch sides. Care was taken to eliminate any voids or air spaces. The insides of the mold were lined with polyethylene film to produce a smooth surface texture. The resulting water/cement ratio was 1.3. The blocks were allowed to cure at ambient temperatures for a period of only two weeks before cutting tests were performed. Additional curing,, would have increased the strength. During this program, 18 tons of sand were consumed making concrete block test specimens.

Sample cylindrical molds, commonly used for concrete testing in the construction industry, were poured and taken to a testing laboratory. The molds were allowed to cure alongside the test blocks for the same period of time. The compressive strength of these samples was about 440 psi..

For test purposes, these concrete blocks were transported and held in the bucket of a front end loader attached to a Kubota L245DT tractor. The blocks weighed about 500 pounds each. This arrangement gave a steady platform for testing, and the four-wheel drive feature of the tractor allowed the blocks to be easily moved through mud and water.

RESULTS

Based on the procedure discussed earlier for low pressure testing at 500 psig, the standoff distance at which the ordinary jet was no longer able to cut was 35 feet. The Percussive Jet's cutting ability deteriorated to the same level at a standoff distance of 55 feet. This represents an increase in cutting distance of 60% by this criteria.

However, at intermediate distances, the difference in Percussive Jet and ordinary jet performance was more significant. An example is shown in Figure 8. In this test, the two jets were discharged side-by-side under identical conditions with the concrete test block placed 25 feet (600 nozzle diameters) from the nozzle. As shown, the Percussive Jet firings removed about twice as much material as the ordinary jet. Hence, all other conditions remaining the same, the Percussive Jet required about half the specific energy as the ordinary jet.

Most interesting was the shape of the Percussive Jet craters. Each was about 2 1/2inches in diameter but had a deep central hole less than one-inch in diameter. In approximately one-half of the tests, these deep central holes wandered off in one direction or the other in the concrete blocks. The presence of these deep central holes gives further evidence of the presence of a "tight," relatively undisturbed central core of the Percussive Jet even at large standoff distances.

The performance of the Percussive Jet was optimized when the 3/4-inch diameter bypass was used in the modulator to control the modulation amplitude. At these low pressures, an optimum in modulation frequency was not found. Performance increased monotonically as frequency increased to 400 hertz.

During high pressure testing at 2,000 psig using procedures outlined earlier, the ordinary jet failed to cut the test blocks at about 60 feet (1440 diameters) from the nozzle, whereas the Percussive Jet could extend out to 75 - 80 feet (1900 diameters). This represents an improvement of about 30%.

At closer distances to the nozzle, the results were again more apparent. Figure 9 shows an example of a concrete block that was impacted by a Percussive and ordinary Jet at 45 feet (1080 diameters) from the nozzle. As illustrated, the Percussive Jet removed about 15 times as much material as the ordinary jet. Hence, the specific energy requirements of the Percussive Jet are in the range of an order of magnitude less than an ordinary jet under these conditions.

These high pressure tests illustrated the same type of crater as evidenced in low pressure testing: a crater with a deep central hole. In the example of Figure 9, the small central hole is 2 inches in diameter in the center of a larger crater 9 inches in diameter.

Hence, this high-pressure testing gave further evidence supporting the existence of a "tight" central core.

The optimal performance of the Percussive Jet discharged at 2,000 psig was obtained using the same 3/4-inch diameter bypass used to produce the best low pressure cutting results. However, in contrast to the low pressure results, an optimal modulation frequency of about 230 hertz was found. The reason for the existence of a maximum in cutting performance with frequency is subject to speculation at this time. Most likely, this maximum has its origin in the fluid mechanics of the free jet rather than in the interaction of the Percussive Jet with the material.

Several interesting observations were made upon examination of the test specimens after they had been impacted by the Percussive Jet. Such observations were made repeatedly. These are listed below and briefly discussed. None of these effects was ever observed while using an ordinary jet.

Percussive Jet Craters. Large craters with deep central holes discussed earlier.

Cracking In Concrete Blocks With Smooth Faces. Figure 10 shows an example of the impacted surface of a concrete block 60 feet away from the nozzle when hit by a Percussive Jet discharged at 2000 psig for 5 seconds. The surface of the block was originally smooth. Such a result also occurred during a demonstration before Department of Energy and Bureau of Mines personnel. These results cannot be explained in detail at this time but probably result from the complicated pattern of repetitive stress waves set up in the material by the Percussive Jet.

Cracking In Concrete Blocks With Surface Imperfections. Results similar in appearance to Figure 10 were obtained when the Percussive Jet impacted within a few inches of a shallow surface groove or scratch in the concrete surface. Again, the detailed explanation is a complicated one but perhaps the Percussive stress wave pattern was "concentrated" near the crack. Stresses are known to increase like $1/\sqrt{r}$ as they approach the tip of a crack (where r is the distance from the stress wave to the tip of a crack) (Jasper and Cook, 1971).

Large-Scale Breakout of Nonhomogeneous Surface Materials. In cases where the surface of the concrete had aggregate or other non-homogeneous substances cast in or near the surface, stress wave propagation dislodged large areas of the material. A typical example is shown in Nebeker (1983). Presumably, this type of result is due to the compression-tension mechanism discussed in Nebeker and Rodriguez (1973).

ACKNOWLEDGEMENT

This activity was performed on U. S. Department of Energy Contract DE-AC01-79ET14280. Currently, the technical project officer is Mr. Robert J. Evans. Earlier, the technical project officers were Mr. Hamilton Reese, Jr. and Mr. Howard Parkinson assisted by Dr. Sedat Sami.

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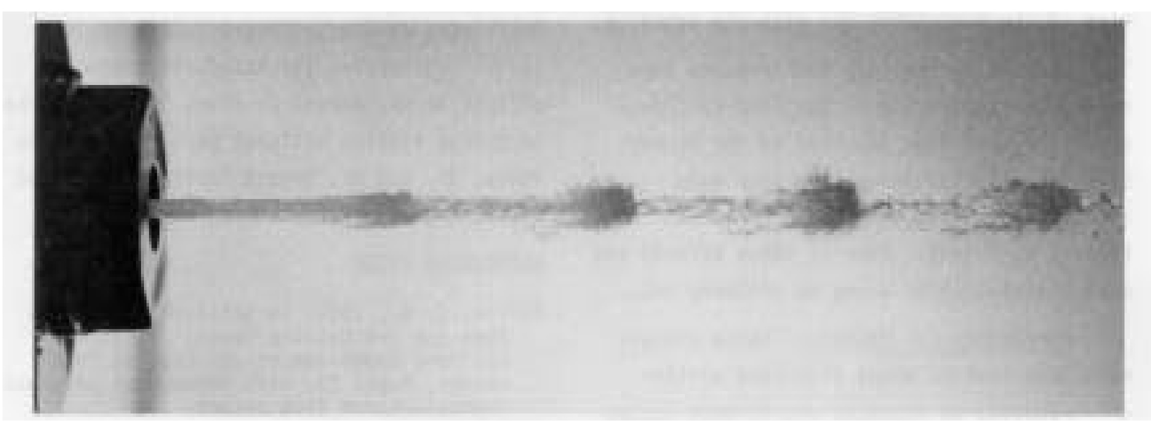

Figure 1. EXAMPLE OF PERCUSSIVE JETS. A small cyclic variation or modulation impressed on a steady discharge of water causes the free jet to become bunched. Consequently, the jet will strike a target in a sequence of sharp impacts rather than steadily.

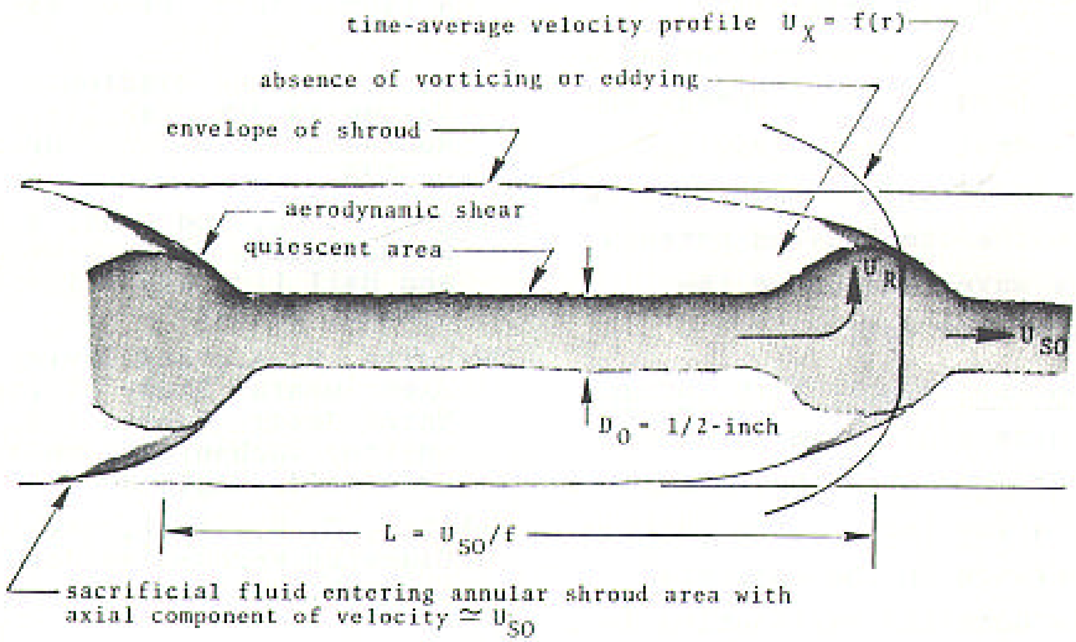

Figure 2. MECHANISM OF PERCUSSIVE JET PERSISTENCE TO LONG STANDOFF DISTANCE

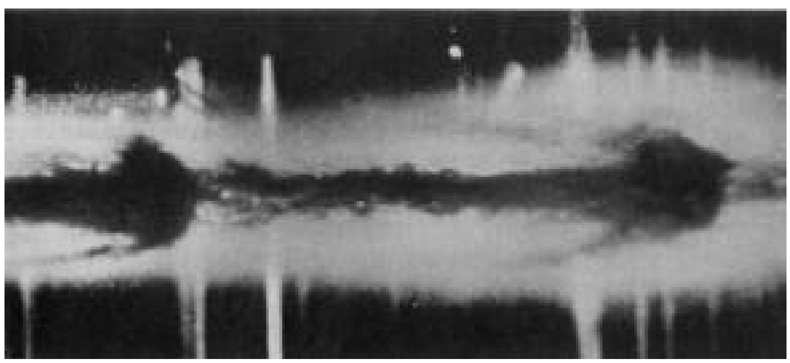

FIGURE 3. INNER CORE OF PERCUSSIVE JET DISCHARGED AT 500 PSIG. Using infrared photography technique developed during this contract, mist surrounding the central core of a high-speed jet can he penetrated. Bunches have formed and are ahout 2 1/2 times the diameter of the original jet (l/2-inch). Surrounding the central core is a sheath which is the mist which has been illuminated by scattered light. The Jet is travelling at about 33n feet/second and the modulation frequencv is 300 cycles ner second. The wavelength or distance between bunches in this rhotooraph is about 12 inches. Exposure time was 3 microseconds.

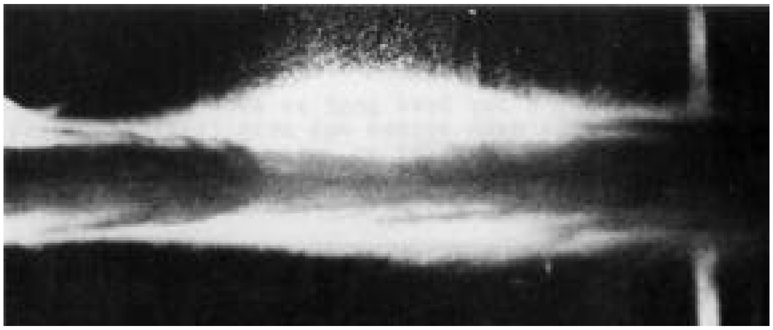

FIGURE 4. INNER CORE OF PERCUSSIVE JI.T DISCHARGED AT 2000 PSIG. This photograph is identical to Figure 3 except the Jet is now travelling at 500 feet/ second. At these velocities, the edges of the bunches were sheared back more drastically than at the slower Jet conditions.

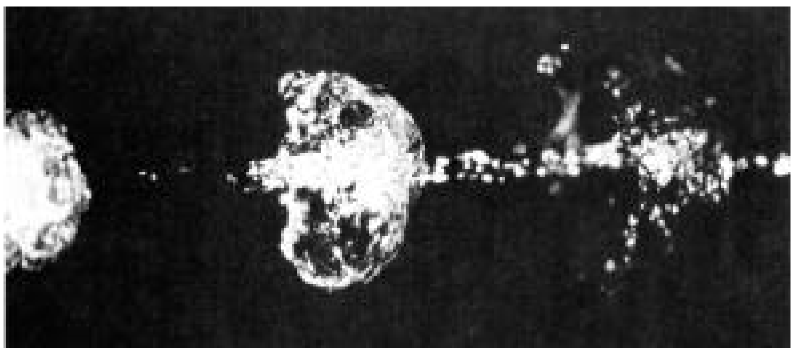

FIGURE 5. SCATTERED LIGHT PHOTOGRAPHS OF PERCUSSIVE JET. This view is helpful in understanding the fluid mechanism which allows the Percussive Jet to persist to larger distances from the nozzle than ordinary jets.

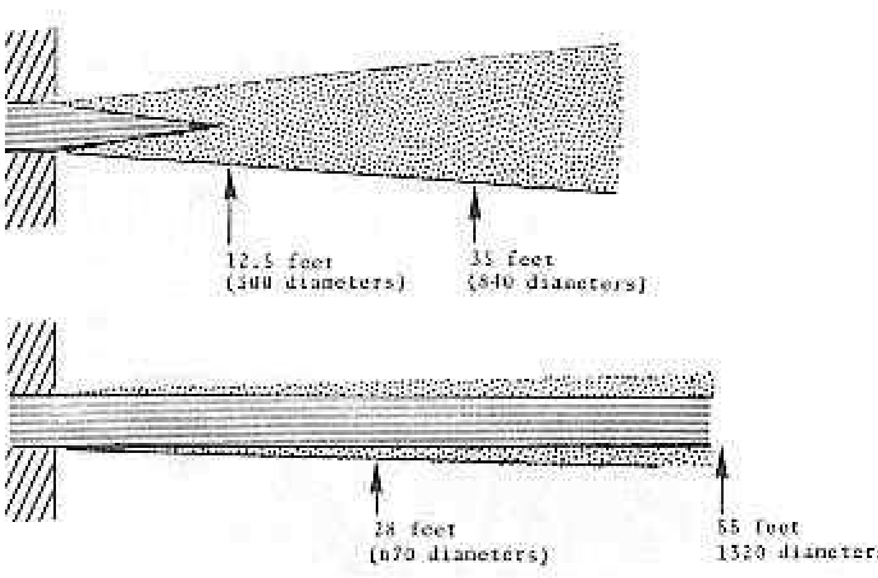

FIGURE 6. COMPARATIVE SCHEMATIC DRAWINGS OI ORDINARY (top) AND PERCUSSIVE JET (bottom) DISCHARGED AT 500 PSIG. This illustrates the relative distance discussed in the text. The ordinary jet becomes discontinuous at 12.5 feet and will no longer cut concrete at 35 feet. The periodic pattern of the Percussive Jet frequency can be seen at 28 feet, whereas the Percussive Jet no longer cuts at 55 feet.

FIGURE 7. HALLIBURTON SERVICES HT-400 PUMP EQUIPMENT READY FOR HIGH-PRESSURE TESTING. Enclosed test range is at left, and concrete specimens are in foreground.

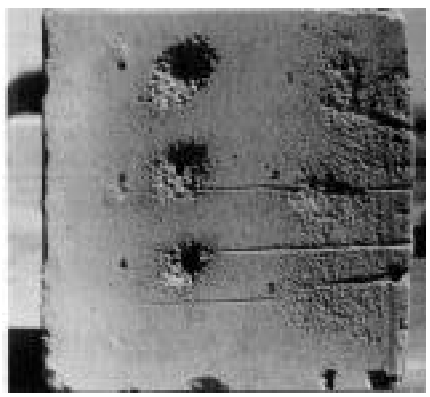

FIGURE 8. CONCRETE BLOCK IMPACT TEST. Concrete placed 25 feet (600 nozzle diameters) from nozzle discharge. Ordinary jet impacts shown in runs 1, 2, and 3 produced spotted damage about 4 inches in diameter, ranging from 1/16- to 3/16-inches in depth. Percussive Jet impacts 4, 5, and 6, were much deeper 11/16- to l-inch in depth and 'tighter," about 2 1/2-inches in diameter. Each Percussive Jet crater has a deep central hole. Firings lasted 5 seconds and were produced by 200 gpm discharges at 500 psig from a 1/2-inch diameter nozzle. Percussive Jet frequency was 600 cycles per second.

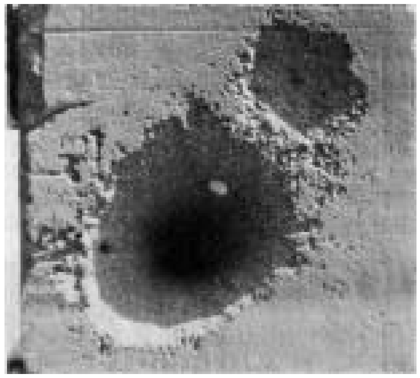

FIGURE 9. EXAMPLE OF HIGH-PRESSURE TESTING USENG CONCRETE BLOCKS. Five-second duration firings of ordinary and Percussive Jet were made side-by-side at 2000 psig discharge pressure. The block was 45 feet (1080 nozzle diameters) from the nozzle. The small hole in the upper right was made by an ordinary jet and was 3/4-inch deep hv 6 inches in diameter. The large hole was produced by a Percussive Jet under identical conditions and is 5 9/16inches deer and about 9 inches in diameter.

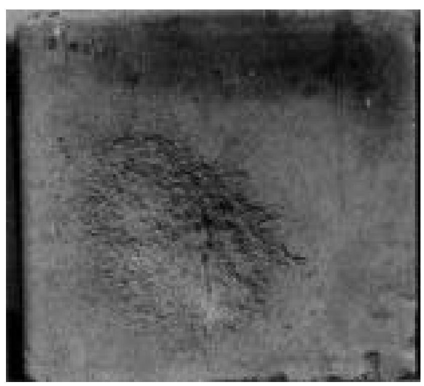

FIGURE 10. EXAMPLE OF CRACKING lN CONCRETE TEST BLOCKS. A crack appeared in this homogeneous test block with no apparent surface discontinuities prior to testing. This block was 60 feet (1440 diameters) from the nozzle when impacted by a Percussive Jet discharge at 2000 psig for a duration of 5 seconds.

DISCUSSION

NAME: David Eddingfield
COMPANY: SIU

QUESTION: "In a computer study at SIU-C, we have been computing the shape of the jet subjected to a sine wave modulation. We have found with surface tension but no air resistance the jet shape is evolving to clumps which are pancake shaped, not round clumps, as most people draw them.

not this, but this

I feel that air resistance then folds back these edges into a parachutetype shape. There parachute shapes are common in droplet breakup what is a function of the relative Weber number? Do you agree with my comments with respect to shape?

ANSWER: In response to your question, Dave, I have several comments which may be of interest:

1. A "pancake-type" bunch can be formed using a sinusoidal input function. Not only can these forms he produced using a computer, but can be calculated using closed form solutions by properly segmenting the flow field.

2. At this time, sinusoidal waveforms are difficult to produce in jets using large power but are easy to construct in laboratory-scale. For practical cutting jets, a sinusoidal input is not a particularly good waveform to use.

3. Since a sinusoidal input is not close to optimum, we do not use it. The mechanism described in the paper is what we feel is occurring. No evidence exists for the occurrence of pancake shapes except at low velocities, and perhaps, waveforms with large amplitudes.

4. I hoped to show in this paper, and you have touched on it again, that many forms are possible in addition to the customary bunching one.

5. Please bear in mind that what waveform is created in the modulator is altered depending on a variety of considerations such as upstream dynamics, modulator fluid mechanic nozzle absorption and free stream fluid mechanics. The job of the design is to take all of these factors into consideration to produce a jet for a useful purpose.

NAME: John E. Wolgamott

COMPANY: StoneAge, Inc.

QUESTION:"The work you have presented dealt primarily with relatively large flow rates, large diameter nozzles and slow velocities. Do you have experience with creating percussive jets with smaller nozzles and/or higher velocities? If so, could you comment on the observed benefits in comparison to steadyflow jets?"

ANSWER: Our earlier publications referenced in this paper dealt with work using nozzles in the 0.060-inch diameter range at velocities of about 1,000 feet second. We are currently working at higher velocities. In essence, improvements over steady-flow jets are quite striking as a result of:

1. The increased ratio of impact area to water volume obtained with Percussive Jets is generally beneficial;

2. Percussive Jets repeatedly provide initial impact effects-waterhammer and high lateral velocities--which enhance fracturing and erosion;

3. Cyclical unloading of the target material in percussive impact produces absolute tension which promotes brittle fracture;

4. The short duration of percussive impact stresses tends to reduce losses of energy within fractured material, and therefore, the specific energy for excavation;

5. Stream bunching acts to conserve jet velocity against air drag, and thus, increase the maximum range of a jet. We are currently working at much higher velocities. In general, higher velocities imply higher modulation frequencies.

NAME: Andrew Conn
COMPANY: Tracor Hydronautics

QUESTION: "In our work with pulsed jets at Tracor Hydronautics (see paper in Session 4 of this conference), we have found that certain nondimensional parameters are very useful in predicting jet performance. For instance, the nondimentional standoff distance, X/d, for optimum bunching can be predicted by:

$$\frac{x}{d} = \frac{1}{S_d} \cdot \frac{V}{2V} \qquad (1)$$

Where: the strouhal number: S_d is:

$$S_d = fd/V \qquad (2)$$

The various parameters in these equations are: f; frequency, d: nozzle diameter, V; mean jet velocity, V; amplitude of modulation of the jet velocity. From Figure 4 of your paper: f = 300Hz, V = 330 ft/s, and elsewhere you cite: S/d ~1000 with d = 0.5 in. Thus, from (2) = S_d ~0.04, and hence from (1), it would seem that your modulation, V/V is about 1 to 2 percent. Is this prediction (from Eqns. (1) and (2) consistent with your experimental observations for jet modulation? Do you agree that such an analytical approach should be useful in trying to understand these pulsed

ANSWER: Andy, many important problems cannot be solved by completely theoretical or mathematical methods. Naturally, one way of attempting a problem for which no mathematical equation can be derived is empirical experimentation. The procedure is laborious, and difficulty is encountered in organizing or correlating the results into a useful relationship for calculations.

Dimensional analysis is a method intermediate between formal mathematical development and a completely empirical study. Such an approach is based on the fact that if a theoretical equation does not exist among the variables controlling the phenomenon, the equation must be dimensionally homogeneous. As a result, many factors can be grouped into a smaller number of dimensionless groups of variables. A dimensional analysis cannot be made unless enough is known about the physics of the situation to decide what variables are important in the problem and what basic physical laws would be involved in a mathematical solution if one were possible.

I am pleased equation (1) is useful to you. However, such an equation would predict a poor operating point for our Percussive Jet in most cases. Certain variables very important to the process do not appear in this equation. We operate over such a large range of variables that "optimum bunching" is redefined for each case. For example, sometimes surface tension is important but often it is of no concern.

You might try to do a more formal mathematical analysis using a computer for solution. We feel we can process more variables, include more details, and generally arrive at a more fundamental solution than by using dimensionless groupings.

THE FOCUSED SHOCK TECHNlQUE FORPRODUCING TRANSIENT WATER JETS

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University of Colorado, Boulder, Colorado 80309
(Visiting from University of Lulea, Lulea, Sweden

ABSTRACT

This paper presents experimental work on a new type of transient water jet device. A spark discharge at one focus of a two-dimensional, elliptical, water filled cavity creates a shock wave which converges towards the other focus. There it enters an axisymmetric duct and is converted into a plane shock wave which moves out from the plane of the ellipse. When the plane wave reaches the free surface at the front of the nozzle a water jet is emitted. The Schliereren technique is used to study the process of wave convergence in the cavity. Resulting jet speeds form a simple non-optimized device are also measured. The fundamental idea appears to work and the results are promising, but difficulties remain.

INTRODUCTION

In the areas of rock cutting and erosion by liquid impact, continued interest has been devoted to the use of transient water jets. The reason for this is the property of this kind of jet to strongly erode a surface on which it is impinging. This destructive behaviour has been explained by the very high pressures that develop in the contact zone between the liquid drop and the solid surface, pressures that exceed the water-hammer pressure at the edge of the contact zone.

There are several ways of producing a transient liquid jet. Brunton introduced the technique of generating a shock wave in a liquid filled cavity (Bowden and Brunton, 1958). When this wave reaches a free surface of the liquid, i.e. at the front of a nozzle, a small amount of the liquid near the surface is emitted in the form of a slug. This shock wave can be generated in different ways, of which the most popular has been to mechanically impact the liquid mass in the cavity. This can be done using a movable piston or slug from, for example, an air gun. However, in applications where a high repetition rate is desired, this method is less suitable because of the inertia associated with the moving parts. In such cases an alternative is to generate the shock wave by a discharge between two electrodes submersed in the liquid.

Regardless of the way in which the shock wave is produced, it is desirable to convey as much as possible of its energy to the jet. It is thus of interest to examine the possibility of focusing the energy in the shock at the orifice of the generator. This paper presents experiments that have been carried out on a new type of transient water jet device. It operates with electrical discharges to create the shock wave, and has a cavity geometry which forces the major part of the wave to converge towards the nozzle area.

TRANSIENT WATER JET DEVICE

The liquid filled cavity has a two-dimensional elliptic shape. It consists of three layers of flat plates, of which the one in the middle has an elliptical hole. At one focus of the ellipse an electrode is mounted in each of the side plates. At the other focus one of the side plates has a conical hole, and one has a cone attached to it. The cone and the conical hole together form an axisymmetric converging channel with its axis directed normal to the plane of the ellipse. Fig. 1 shows a crosssection sideview of the cavity.

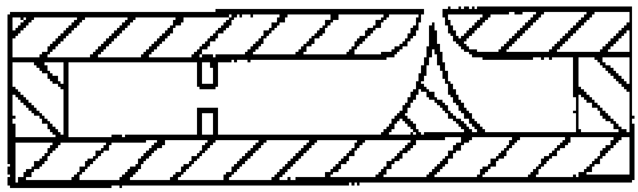

Fig. 1. SCHEMATIC DRAWING OF THE ELLIPTICAL CAVITY WITH THE ELECTRODES AT ONE OF THE FOCII AND THE AXISYMMETRIC CONVERGING CHANNEL AT THE OTHER.

An electrical discharge across the electrodes creates a shock wave which travels away from the electrodes in all directions. If the spark gap is considerably smaller than the distance between the side plates, the shock wave will have a shape close to spherical. If, however, the spark gap is the same, or nearly the same, as the distance between the side plates the shock wave will have an almost cylindrical shape. Furthermore it will also be weak if the amount of energy given to the liquid by the discharge is comparatively small. The shock then travels with approximately constant speed through the liquid, i.e. the speed of sound in the liquid. When the different parts of the wave reach the wall they will be reflected, and due to the specific geometry of the cavity the reflected wave will be a negative image of the outgoing wave, with its axis of symmetry at the second focus towards which it converges. Fig. 2 shows an illustration of the reflection process.

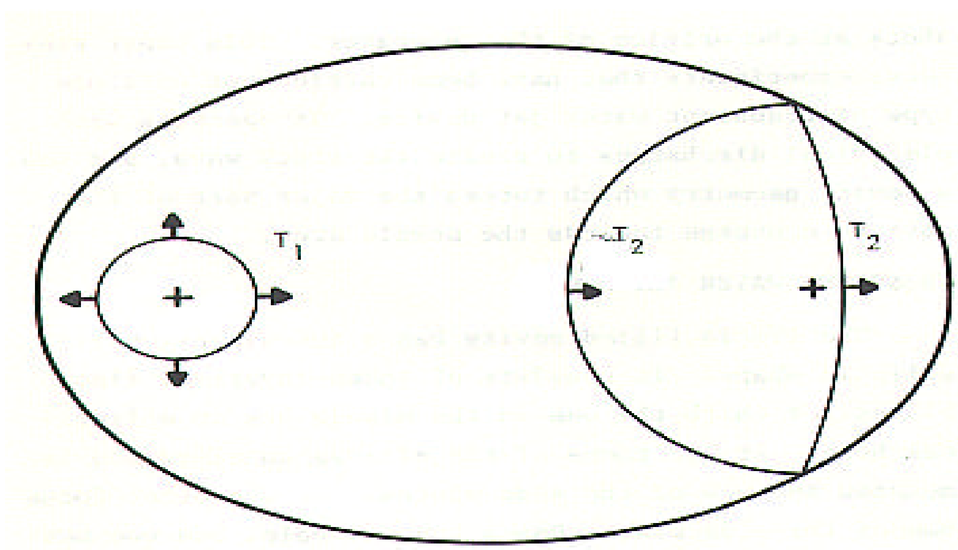

Fig. 2. THE WAVE PATTERN IN THE ELLIPSE AT TWO DIFFERENT TIMES, T2 > T1. AT TIME T, NO PART OF THE OUTGOING WAVE HAS YET BEEN REFLECTED.

The reflected wave is thus also cylindrical and converges towards the focus where the converging channel is. When the shock wave passes through this channel its direction of travel is turned 90 degrees and it is converted into a plane shock wave.

EXPERIMENTS

Optical studies of the shock wave motion in the cavity in the absence of a converging channel have been conducted. This was made possible by using side plates made of glass whereby photographs could be taken with the use of a Schlieren system.

One glass plate was permanently attached to the center plate, and the other was squeezed tight in between the center plate and a cover plate. The latter two, which were made of steel, were held together by bolts along their edges. To prevent leaks, the removable glass plate was sealed with a silicone rubber gasket. The center plate and the cover plate both had elliptical holes with lengths of the major and the minor axes of 150 mm and 120 mm respectively. The thicknesses of the center plate and the glass plates were respectively 20 mm and 10 mm. To avoid the problems associated with holes in the glass plates, the electrodes were mounted in brass holders which were cemented onto the inside of the glass plates at one focus. The electric wires were led out of the cavity through grooves in the center plate. This arrangement made it necessary to adjust the distance between the electrodes before mounting the removable glass plate. The electric circuit for the experimental set-up is shown in Fig. 3.

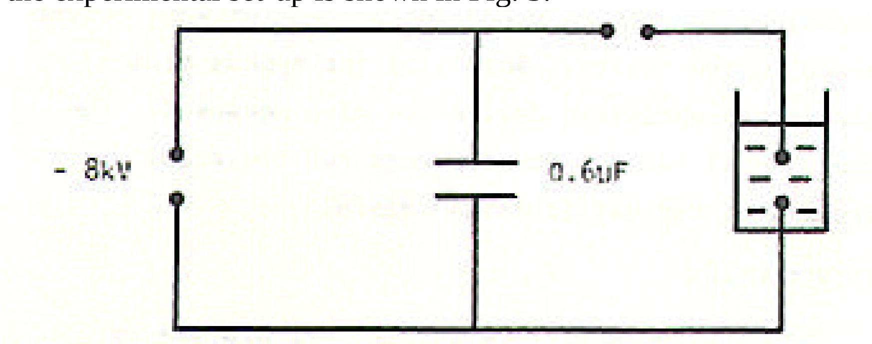

Fig. 3. CIRCUIT FOR THE ELECTRICAL DISCHARGE.

The discharge energy was supplied by an 8kV constant DC source charging a 0.6 μF capacitor, to which the electrodes in series with an air spark gap were connected. Both pairs of electrodes, in liquid and air, were cylindrical with a diameter of 2.4 mm and made of tungsten. The air spark gap was necessary in order to get a discharge, and by changing the distance between its electrodes the breakdown voltage could be varied.

With the water filled cavity placed in the parallel light section of a Schlieren system, photographs of the wave motion were taken. The light source was a Fischer-Nanolite, coupled to a delay unit which was triggered by the air spark gap. By changing the delay time between the discharge and the light flash the waves could be photographed at different positions ir the cavity. The recordings were made in a dark room with a Hasselblad 500 EL/M camera, equipped with a 500 mm lens with open shutter, To suppress as much disturbing light from the liquid spark gap as possible, the parts of the glass plates that covered the electrodes were painted black on the inside.

RESULTS AND DISCUSSION

A series of photographs of th econvergence porcess, taken at different times after discharge, is shown in Fig. 4.

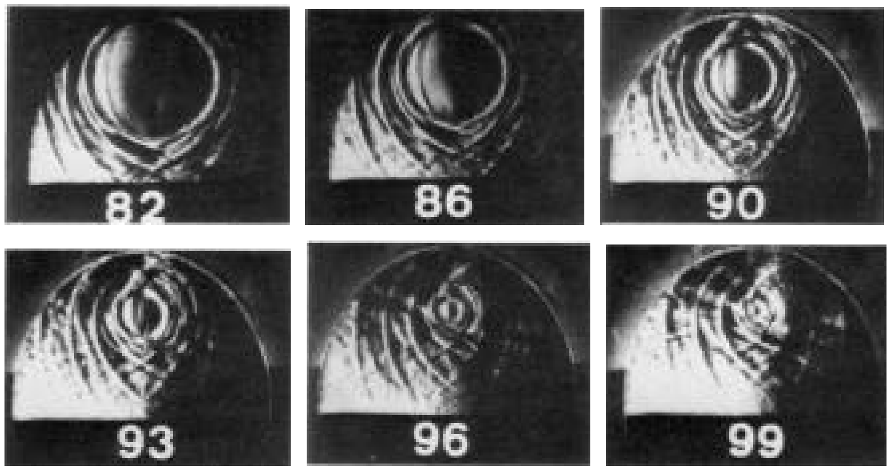

Fig. 4. WAVE PATTERN IN THE ELLIPSE. EACH PICTURE IS FROM A DIFFERENT EXPERIMENT, AND THE NUMBERS INDICATE TIME IN MICROSECONDS AFTER DISCHARGE.

The spark gap in the air was 3 mm, and the liquid spark gap was < 1 mm. The latter distance was difficult to measure exactly since it depended on how strongly the removable glass plate was pressed against the center plate. Practical considerations required flat ended electrodes. Electrodes with conical ends eroded very rapidly, which made it necessary to open the cavity after only a few discharges to adjust the distance between them. No differences between the wave patterns produced by the two different types of electrodes could be detected.

The photographs reveal a complex pattern of waves. The pattern was very repeatable. The almost circularsymmetric wave front is the leading wave from the discharge. In the photograph taken at 82 μs after discharge its reflection from the cavity wall is just completed. It converges to zero radius in 100 μs, which shows that it moves with a speed of 1500 m/s and hence is approximately acoustic. The slight disturbance at its bottom is caused by the electrodes and their wires. Inside it is a lens-shaped wave which is a Mach wave originating at the wall, caused by the difference in sound speed between steel and water. The almost equally spaced waves behind it are the reflections of the leading wave from the glass side plates. These are present because the distance between the electrodes was smaller than the distance between the side plates, as explained earlier. Behind the circular wave are also many other waves of apparently irregular shape, of which some are Mach waves from the reflections in the side plates, and some probably produced by oscillations of the compression ring that surrounds the electrodes for some time after discharge (Frungel, 1980).

For the objective in question, namely to produce one coherent wave, only the leading circular wave is desirable. The reflected waves from the side plates and the disturbance to the leading wave can be eliminated by a suitable design of the electrodes as mentioned earlier. The Mach waves and the waves from the compression ring are more difficult to suppress but can, however, in view of their relative weakness probably be neglected. The introduction of a cone and a converging channel will inevitably also disturb the leading wave, since a part of it passes them before it is reflected from the wall. By using a cone and a channel with small lateral dimensions compared to the dimensions of the cavity, this disturbance call however be made small.

The conclusions that can be drawn from these experiments are that the electrodes should be placed in the plane of the side plates, and that the cone and the converging channel should have small lateral dimensions compared to those of the cavity. In addition, it is also advantageous to make the thickness of the center plate small, since this will result in a converging wave which has a more two-dimensional behaviour and will bend more easily in the converging channel.

A few experiments were also carried out with a device with steel side plates, equipped with a cone and a converging charnel. It was not optimized in the way outlined above, but was merely built to get a rough estimate of the jet speeds attainable. The lengths of the major and minor axes of its cavity were respectively 30 mm and 24 mm, and it had a center plate with a thickness of 9 mm. A high-speed camera capable of taking a series of pictures of one single jet was used to measure the jet speeds. With the use of the same electrical circuit as in the previous experiments, Fig. 3, typical jet speeds attained were almost 100 m/s.

The experiments carried out established that the shocks in the elliptical cavity converge to the second focus and that the principles employed essentially work. The discharge in water leads to a complicated system of waves. This is undesirable as it means that the discharge energy is spread over the resulting wave system rather than being concentrated in the leading pulse. Previous tests with air in the cavity gave a "clean" discharge with a single observed shock.

To achieve practical jet speed levels considerable work will be required to optimize the device. The directions in which this should proceed are relatively clear. These are:

1. Improvement of the external spark generating circuit and system to prevent the "ringing" discharge;
2. Optimization of the cone:orifice geometry to reduce the diffraction of the incident wave on the first pass and decrease the reflection coefficient of the converging cylindrical wave, and
3. Optimization of the nozzle attached to the cavity (Lovgren, 1983; Lovgren and Gustafsson, 1983).

In conclusion it is felt that the basic idea has considerable promise insofar as the fundamental principles involved appear to work as expected, but that more must be done to produce water jets with interesting properties.

ACKNOWLEDGMENT

This work was supported by the National Swedish Board for Technical Development (STU).

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DISCUSSION

NAME: George Savanick
COMPANY: U. S. Bureau of Mines

QUESTION: "Please give details regarding the photographic technique used."

ANSWER: A presentation of the Schlieren technique can be found in most textbooks on compressible flow theory, e.g., Elements of Gasdynamics by Liepmann and Roshko.

NAME: H. S. Stevens
COMPANY: BHRA

QUESTION: "Is there a practical limit of the proposed shock technique to the size of jet it is supplied to?"

ANSWER: There is no fixed limit, but the relative losses in the process can be expected to increase with larger cavity dimensions and stronger shock waves.

Sufficiently strong shocks also display a nonlinear behavior, and this further decreases the efficiency by imperfect focusing of the converging wave.

NOZZLE DESIGN FOR COHERENT WATER JET PRODUCTION

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ABSTRACT

This paper describes some of the work undertaken during the past six years on the fundamental aspects of hydraulic mining of china ciay. The complete study has covered the three inter-related areas of jet production, jet anatomy and jet impact. The present paper addresses itself to jet production. Subsequent papers will deal with jet anatomy and jet impact.

A survey of the literature on nozzle design and its influence on jet characteristics reveals a substantial gap between theory and practice and it is the intention of this paper to show how this gap may be bridged for the particular case of clay mining monitors.

The prime objective of the present work has been to maximize jet momentum flux at an excavation face. A study of jet anatomy shows that delayed jet beeak-up produces an increase in jet impact momentum flux and that careful nozzle design leads to more coherent jets.

This paper describes a systematic method for nozzle design which should lead to maximum possible jet impact momentum fluxes. The method may be applied to any particular water jet production system when due allowance is made for the constraints of that system. The constraints within which the specific design herein must operate are discussed by Davies and Jackson (1982) and by Jackson (1983).

1. NOMENCLATURE

$AR = Lt/D$	nozzle aspect ratio
B	blockage factor
C_c	coefficient of contraction
C_d	coefficient of discharge
C_p	coefficient of pressure
CR	contraction ratio
d_o	nozzle diameter
d_j min	minimum diameter of contracting jet
D	barrel or pipe diameter
$Fr = U^2/d_o\, g$	Froude number
f	function
g	gravitational acceleration
h	radial grid spacing
h_j	actual vertical jet trajectory
H	shape factor

H'	mean shape factor
H_j	theoretical vertical jet trajectory
k	axial grid spacing
K	number of grid points in r-edirection
K_l	loss coefficient
K_p	pressure gradient parameter
L	pipe length
L_b	barrel length
L_f	length of nozzle final taper
L_s	length of settling section
L_t	total length of nozzle
L_p	length of nozzle parallel extension
m	mass flow rate in boundary layer
M	mass flow rate per unit nozzle length
N	number of grid points in x-direction
P	static pressure
P_m	jet centre-line dynamic pressure
P_o	nozzle pressure
P	monitor or pipe flow pressure drop
Q_o	nozzle flow rate
r	radial distance
r_c	radius defining boundary layer
R	nozzle wall contour radius
$\mathrm{Re}_D = \dfrac{\overline{U}D}{\nu}$	pipe flow Reynolds number
$\mathrm{Re}_i = \dfrac{U_i D}{\nu}$	nozzle inlet Reynolds number
$\mathrm{Re}_\theta = \dfrac{U_i \theta}{\nu}$	mimentum thickness Reynolds number
$\mathrm{Re}_e = \dfrac{U_e d_0}{\nu}$	nozzle exit Reynolds number
R_p	relaxation parameter
S	sublayer thickness
T_i	turbulence intensity
u'	x-direction turbulent velocity
u	x-direction velocity
U	x-direction wall velocity
$\overline{U}$	mean velocity
v'	redirection turbulent velocity
v	redirection velocity
v*	frictional velocity
V	r-direction wall veloeitv
x	axial distance
Xi	jet initial region length
Xc = Xi/do	dimensionless initial region length
X_b	jet breakup length

y	distance normal to nozzle wall
$We = \dfrac{U^2 \rho d_0}{\sigma}$	Weber number
ρ	nozzle half angle
δ	boundary layer thickness
$\delta *$	displacement thickness
δ_t	turbulent layer thickness
ε	boundary layer parameter
$\bar{\varepsilon}$	CLA roughness
η	entrainment parameter
η_j	jet efficieney
θ	momentum thickness
μ	viscosity
υ	dynamic viscosity
τ_w	wall shear stress
ϕ	entrainment function
φ	stream function
σ	surface tension

NOZZLE FLOW ANALYSIS

Introduction

Figure 1 indicates that basic monitor flowsystem design parameters and regions of jet development whilst figure 2 indicates the basic flow regions within the nozzle. For convenience of analysis flow in the monitor may be divided into two main regions.

i) An inviscid core-flow region where the effects of viscosity are negligible and radial velocity gradients are relatively small.

ii) A boundary-layer region adjacent to the wall within which the effects of fluid viscosity are confined. In this region the velocity gradients are high and the velocity increases from zero at the wall (no slip condition) to 99% of the free-streamor local core velocity at the edge of the region

Although the purpose of the nozzle is to accelerate the mean flow by converting potential energy associated with the static head to kinetic energy, the turbulent kinetic energy of the fluid is also increased. The increase in turbulent kinetic energy however, is not nearly as large as that of the mean flow kinetic energy and as a result both radial and axial turbulence intensities decrease within the contraction.

In the presence of a strongly favourable (negative) pressure gradient, associated with fluid acceleration in the contraction, the boundary-layer thickness decreases.

Boundary-layer development is influenced by the value of the local pressure gradient at the edge of the boundary layer given by the potential or inviscid core flow solution. The value of the local pressure gradient is determined by the shape and length of the contraction contour and by the overall contraction ratio.

Required streamline curvature at the inlet and exit of classically shaped wind-tunnel and venturimeter contractions leads to regions of unfavourable (positive) wall pressure gradient. Wall pressure overshoot occurs near the inlet and the possibility of boundary-layer separation arises whilst wall pressure undershoot near the exit leads to the possibility of cavitation in the region of low static/high dynamic pressure. Additional flow disturbance will be created if the boundary-layer flow is not hydraulically smooth i.e. the height of individual roughness elements on the contraction wall are not well within the laminar sublayer thickness. If this requirement is not met then the possibility of roughness induced cavitation also exists adjacent to the nozzle exit. Surface roughness elements cause wakes in which the static pressure is low and because of the higher velocities close to a smooth wall relative to a rough wall, isolated roughness elements may cause cavitation at lower velocities than for continuous roughness of the same maximum height. Cavitation considerations imply that the flow should still be accelerating as it leaves the nozzle exit and provide additional incentives to ensure that hydraulically smooth flow conditions are achieved.

Relaminarization phenomena may also occur with resultant changes in the nature of both wall and jet flows.

On exit from the nozzle the jet fluid will contract if the final nozzle contour is non-parallel and the diameter of the jet reaches a minimum before subsequent jet development. In order to describe jet flow development in highly turbulent jets it is convenient to define two main regions linked by a transition region (figure 1).

i) Initial region. The initial region of length X_i is defined as the region within which the jet centre-line pressure, P_m remains equal to the nozzle pressure P_o.
ii) Transition region. In highly turbulent jets, initial disturbances present on the surface of the jet and associated with fine scale turbulent eddies, are propagated as transverse or sinuous wave-like disturbances. These disturbances are driven by a classical Helmholtz instability mechanism and are amplified and modified by the action of aerodynamic drag forces. The propagation of surface waves leads to entrainment of surrouding air. As a result the jet dynamic pressure is reduced and the initial region eroded. When the wave amplitude approaches the local jet radius, entrained air reaches the jet centreline and the jet begins to disintegrate into discrete 'packets'.
iii) Main region. In the main region further development of these largescale structures occurs. Further aeration of the packets occurs due to secondary breakup. Atomisation at the jet surface leads to the development of a region of fine droplets surrounding the broken core. More effective packet separation occurs until the jet is defined to be fully broken ($X=X_b$) and air is present on the jet centre-line 50% of the time. A considerable increase in drag forces and reduction in time average jet momentum flux associated with the packet/droplet or twophase structure leads to the onset of jet deceleration. Eventually, far downstream, atomization and axial deceleration becomes complete.

In this paper attention is not centred around the mathematical aspects of flow solution techniques and accuracy but is confined to establishing methods of analysis for estimating the relative performance of nozzle designs and the surface finish required to achieve hydraulically smooth flow. The objective is to provide working engineering design criteria for practical nozzles.

Inviscid Flow Solution

Previous analytical studies of flow in axisymmetrical ducts has centred around the problems of wind tunnel and diffuser design and much attention has been given to the mathematical aspects of the analytical 'inverse problem' where the solution of the wall shape is derived from specified wall or axis velocity distributions.

Such theoretical solutions are either not readily available or not readily adaptable to the solution of wall velocity and velocity gradient distribution in nozzles of simple arbitrary shape. For the purposes of this study a curvilinear finite-difference approximation has been adopted by the authors.

For the axisymmetric flow of an inviscid, incompressible irrotational fluid

$$\frac{\partial u}{\partial r} - \frac{\partial v}{\partial x} = 0 \qquad \text{(irrotationality)} \qquad (2\text{-}1)$$

$$\frac{\partial (vr)}{\partial r} + \frac{\partial (ur)}{\partial x} = 0 \qquad \text{(continuity)} \qquad (2\text{-}2)$$

Defining a stream function so that

$$ru = \frac{\partial \psi}{\partial r}, \qquad rv = -\frac{\partial \psi}{\partial x} \qquad (2\text{-}3), (2\text{-}4)$$

and combining (2-1) and (2-2) gives

$$\frac{\partial^2 \psi}{\partial x^2} - \frac{1 \partial \psi}{r \partial r} + \frac{\partial^2 \psi}{\partial r^2} = 0 \qquad (2\text{-}5)$$

Consider the curvilinear finite-difference grid scheme for internal flow points shown in figure 3.

Adopting the usual finite-difference approximations the derivatives in equation (2-5) may be formulated.

$$\frac{\partial^2 \psi}{\partial r^2} \quad (\psi n - 2\psi p + \psi s) / h^2 \qquad (2\text{-}6)$$

$$\frac{\partial \psi}{\partial r} \quad \frac{\psi n - \psi s}{h} \qquad (2\text{-}7)$$

$$\frac{\partial^2\psi}{\partial x^2} \quad (\psi e - 2\psi p + \psi w)/k^2 \qquad (2\text{-}8)$$

Substituting (2-6), (2-7), (2-8) back into (2-5) and rearranging in terms of Qp gives

$$\psi p = \frac{\psi n.(1-A) + \psi s(1+A) + B(\psi e + \psi w)}{2(1+B)} \qquad (2\text{-}9)$$

Where A=h/r, B=(h/k)^2

ψ_e and ψ_w are obtained by performing 2nd order Taylor expansions about Q_e and Q_w respectively (figure (3)).

$$\psi w = \psi w' - S1\frac{\partial\psi w'}{\partial r} + \frac{S1^2}{2}\frac{\partial^2\psi w'}{\partial r^2} \qquad (2\text{-}10)$$

$$\psi e = \psi e' - S1\frac{\partial\psi e'}{\partial r} + \frac{S1^2}{2}\frac{\partial^2\psi e'}{\partial r^2} \qquad (2\text{-}11)$$

The derivatives in equations (2-10), (2-11) are obtained by consideration of 2nd order Taylor expansions for $\psi nw', \psi sw', \psi ne'$ and $\psi se'$ about $\psi w'$ and $\psi e'$ respectively. For exammple:

$$\psi nw' = \psi w' + hw\frac{\partial\psi w'}{\partial r} + \frac{hw^2}{2}\frac{\partial^2\psi w'}{\partial r^2} \qquad (2\text{-}12)$$

$$\psi sw' = \psi w' - hw\frac{\partial\psi w'}{\partial r} + \frac{hw^2}{2}\frac{\partial^2\psi w'}{\partial r^2} \qquad (2\text{-}13)$$

Independently eliminating 1st and 2nd order derivatives from (2-12), (2-13) gives:

$$\frac{\partial\psi w'}{\partial r} = \frac{\psi nw' - \psi sw'}{2hw} \qquad (2\text{-}14)$$

$$\frac{\partial^2\psi w'}{\partial r^2} = \frac{(\psi nw' + \psi sw' - 2\psi w')}{hw} \qquad (2\text{-}15)$$

similarly

$$\frac{\partial\psi e'}{\partial r} = \frac{\psi ne' - \psi se'}{2he} \qquad (2\text{-}16)$$

$$\frac{\partial^2 \psi e'}{\partial r^2} = \frac{(\psi ne' + \psi se' - 2\psi e')}{he} \qquad (2\text{-}17)$$

Substituting equations (2-14), (2-15), into (2-10) and equations (2-16), (2-17) into (2-11) gives:

$$\psi w = \psi w' - S1\frac{(\partial nw' - \psi sw')}{2hw} + \frac{S1^2}{2}\frac{(\psi nw' + \psi sw' - 2\psi w')}{hw} \qquad (2\text{-}18)$$

$$\psi e = \psi e' + S1\frac{(\partial ne' - \psi se')}{2hw} + \frac{S1^2}{2}\frac{(\psi ne' + \psi se' - 2\psi e')}{he} \qquad (2\text{-}19)$$

The solution scheme is presented as a flow chart in figure 4. Boundary conditions are set by assuming a uniform velocity profile upstream of the nozzle inlet and downsteam of the nozzle exit and by specifying the wall contour shape. The values of thestream function along the streamline is arbitrarily set to unity and the value of the stream function along the nozzle centre-line set to zero. Solution is achieved by the relaxation method and the new local value of stream function is determined according to:

$$\psi p = \psi p(old) + Rp(\psi p(new) - \psi p(old)) \qquad (2\text{-}20)$$

where $\psi p(old)$ is the value calculated from (2-20) in the previous iteration and R_p is the relaxation parameter

The complete computational algorithm is given by Jackson (1983).

Once the streamline curvature solution has been obtained the wall velocity distribution may be obtained from:

$$U = \frac{1}{r}\frac{d\psi}{dr} wall \qquad (2\text{-}21)$$

$$V = \frac{-1}{r}\frac{d\psi}{dr} wall \qquad (2\text{-}22)$$

Consider the grid scheme for points adjacent to the wall shown in figure 3. Second order Taylor expansions for ψ p and ψ s about ψ n gives

$$\psi p = \psi n - h\frac{\partial \psi n}{\partial r} + \frac{h^2}{2}\frac{\partial^2 \psi n}{\partial r^2} \qquad (2\text{-}23)$$

$$\psi s = \psi n - 2h\frac{\partial \psi n}{\partial r} + 2h^2\frac{\partial^2 \psi n}{\partial r^2} \qquad (2\text{-}24)$$

eliminating $\dfrac{\partial^2 \psi n}{\partial r^2}$ from equations (2-23), (2-24) gives:

$$\frac{\partial \psi n}{\partial r} = \frac{-4\psi p + 3\psi n + \psi s}{2h} \tag{2-25}$$

2nd order Taylor expansions for ψ nw and ψ ne about ψ n gives:

$$\psi nw = \psi n - k\frac{\partial \psi n}{\partial x} + \frac{k^2}{2}\frac{\partial^2 \psi n}{\partial x^2} \tag{2-26}$$

$$\psi ne = \psi n + k\frac{\partial \psi n}{\partial x} + \frac{k^2}{2}\frac{\partial^2 \psi n}{\partial x^2} \tag{2-27}$$

eliminating $\dfrac{\partial^2 \psi n}{\partial x^2}$ from equations (2 26), (2-27) gives:

$$\frac{\partial \psi n}{\partial x} = \frac{\psi ne - \psi nw}{2k} \tag{2-28}$$

ψ ne and ψ nw may be obtained from Taylor expansions about ψ e' and ψ w'

$$\psi nw = \psi w' + (hw - S1)\frac{\partial \psi w'}{\partial r} + \frac{(hw - S1)^2}{2}\frac{\partial^2 \psi w'}{\partial r^2} \tag{2-29}$$

$$\psi ne = \psi e' + (he - S2)\frac{\partial \psi e'}{\partial r} + \frac{(he - S2)^2}{2}\frac{\partial^2 \psi e'}{\partial r^2} \tag{2-30}$$

Invoking the usual finite-difference approximations for the derivatives and noting that ψ wall = 1 i.e. ψ ne = ψ w = ψ nw = 1 gives:

$$\psi nw = \psi w' + \frac{(hw - S1)(1 - \psi sw')}{2hw} + \frac{(hw - S1)^2(1 - 2\psi w' + \psi sw')}{2hw^2} \tag{2-31}$$

$$\psi ne = \psi e' + \frac{(he - S2)(1 - \psi se')}{2he} + \frac{(he - S2)^2(1 - 2\psi e' + \psi se')}{2he^2} \tag{2-32}$$

The wall velocity gradient is obtained by applying the central difference approximation about obtained values of wall velocity.

In the limit as h and k tend to zero, the finite-difference scheme yields the exact solution. As a check on the uncertainty associated with practical finitedifference approximations, solutions were obtained using successively finer meshes. The maximum percentage change between successive solutions was determined and recorded together with other relevant information in table 1. Nozzle design (2) of figure 13 was used with U_i=5 m/s.

TABLE 1

N	K	$\Delta X/L$	$(\Delta u/U)_{max}$	No. iterations	approx. cpu time
12	12	0.016		69	2 mins
20	20	0.010	+0.39%	69	8 mins
32	32	0.00625	+0.12%	130	15 mins
40	40	0.005	+0.06%	210	28 mins

As an adequate compromise between time and uncertainty for the purpose of this study, A_x= 0.0025 and h/k = 4 were selected. This selection provided sufficient wall points for the boundary layer analysis of section 2.5. Resultant potential flow solutions were within 0.5% of the exact solution. The value of the relaxation parameter was set to 1.75 throughout and is suggested as sufficiently close to optimum.

Figure 5 illustrates typical examples of the comparison of the wall velocity distribution with that for simple one-dimensional flow. Figure 6 indicates typical examples of the radial velocity distribution at the nozzle exit. The figures (5,6) highlight the differences in velocity distribution between nozzle contractions terminating in parallel cylindrical sections and those terminating in straight taper sections. The well documented regions of velocity undershoot which occur in both cases and the region of wall velocity overshoot in the vicinity of the joint between the taper section and the parallel section may be observed. A notable difference between the two examples is the velocity defect near the wall associated with the nonuniform exit velocity in the case of the straight taper exit section nozzle as opposed to the nearly uniform exit velocity distribution associated with the cylindrical exit section nozzle. It must be stated however that for contractions terminating in non-parallel exit sections the flow contracts on leaving the exit plane. The fluid accelerates towards uniform parallel flow associated with a minimum in jet flow diameter, dj min., before subsequent downstream expansion.

Thus in the case of the non-parallel exit section nozzle exit velocity profile non-uniformity is subsequently relaxed and perfectly uniform flow is achieved downstream of the nozzle exit. A jet contraction coefficient may be defined as $C_c = (d_j \text{ min.}/d_o)$. The associated hydraulic advantages are that additional viscous drag due to fluid contact with the contraction boundary is not incurred and finite, favourable pressure gradients exist at the nozzle exit. In addition it is well known that in the absence of strongly favourable pressure gradients e.g. in flat plate or developing pipe flow, boundary-layer growth is rapid.

Strictly the potential flow solution of the contracting jet influences the upstream nozzle flow solution to some degree. The exit boundary condition of uniform flow an arbitrary distance downstream of the nozzle exit is somewhat artificial but provides an approximation to the actual flow situation. A complete solution is not attempted here but some indication of jet flow contraction coefficient is required.

For the purpose of this study an indication of Cc values is obtained from the report of Lohn and Brent (1976) who use the 2-D method reviewed by Gurevich (1965).

Turbulence Transformation

An indication of the effects of nozzle contraction ratio and contour shape on turbulence intensity in nozzle flow may be obtained from the work of Ramjee and Hussain (1976).

Experiments conducted for a range of conditions and a variety of contraction designs (figure 7) using air as the working fluid, yield results which contradict those predicted by previously accepted theories. Linear theory considers the homogeneous strain applied to vortex filaments by the contraction, assumes rapid distortion of the filaments and predicts that whilst the kinetic energy of longitudinal turbulence decreases, that of lateral turbulence increases. Experimentally the kinetic energy of longitudinal turbulence was shown to increase in the latter part of the contraction with the overall result that prediction of the effect of contraction ratio on turbulence intensity transformation is poor (figure (8)). The figure 8 together with other data indicates that for CR < 10 further reduction of CR is associated with a drastic increase in core flow exit turbulence intensity. Comparatively little benefit, in terms of exit turbulence intensity, is derived from using CR > 40.

Turbulence transformation in the nozzles was shown to be well correlated with mean flow characteristics. Nozzles that produced the thinnest exit boundary-layers also produced the thinnest exit 'turbulent-layers' (defined, δt s.t $u'/U < 0.04$) with associated lowest maximum turbulence intensities within the turbulent-layer (figure 9). In general these effects were shown to be associated with nozzles that provided a more gradual acceleration in the initial region of the contraction (less tendancy towards separation) and higher pressure gradients at the nozzle exit. In addition it may be that the lower initial rate of strain of vortex filaments associated with the more gradual acceleration in the best nozzles has an important effect in reducing exit turbulence intensities.

Core flow turbulence intensities remained independent of contraction shape for a given contraction ratio.

Figure 9 indicates that in order of decreasing performance were nozzle designs 1,2,3 and the nozzle contour described by a cubic equation resulted in the best exit turbulent-layer characteristics. It is interesting to note that the results obtained for a disc nozzle (design 4, figure 7), although indicating a maximum turbulence intensity comparable with design 1 had an associated turbulent-layer thickness an order of magnitude greater.

Increased fluid entrainment rates and hence expansion rates were obtained llsing nozzles producing thicker exit turbulent-layers during subsequent experiments with submerged air-jets.

Relaminarization Phenomena

Reviews of relaminarization phenomena occuring in highly accelerated turbulent boundary layers have been conducted by Narasimha and Sreenivasan (1973) and Au (1972).

Reversion to laminar-like flow or relaminarization in strongly favourable pressure gradients is essentially due to the domination of pressure forces over the slowly responding Reynolds stresses in an originally turbulent flow. The generation of a new, inner laminar boundary layer accompanies the above process and is stablized by the favourable pressure gradient. Boundary layer retransition to turbulence quickly follows the onset of instability in the laminar layer.

Many criteria for the onset and development of laminarization exist and although no unified approach has been adopted the majority of the criteria may be interpreted as a form of Reynolds number with different choices of the relevant length and velocity scales.

A simple, convenient and hence widely adopted criterion for relaminarization is the pressure gradient parameter:

$$Kp = \frac{\nu}{U}\frac{du}{dx} \qquad \text{(2-30)}$$

Although not considered the most fundamental and hence not likely to be the most accurate for a wide range of flow situations, it is considered appropriate for the purpose of discussing nozzle flow phenomena in this paper. The parameter has the advantage that is may be determined from a consideration of the potential flow solution alone.

Au (1972) cites the work of Schraub and Kline (1965) which suggests that for Kp < 0.5 x 10 the flow is turbulent and that for 0.5 x 10 6 < Kp <3.0 x 10 detectable changes in the structure of the turbulent boundary layer are noticed, turbulent bursting ceases and the boundary layer is in transition from turbulent to laminar with the sole characteristics of neither. For Kp > 3 x 10 a fully laminar-like boundary layer results and complete relaminarization occurs.

In general it is assumed that a finite, but as yet unquantified, time is required for retransition to a developing turbulent boundary layer once the stabilizing pressure gradient is released. Analogies have been drawn to the transition of a laminar boundary layer under zero pressure gradient conditions.

Boundary Layer Analysis

In the wake of the celebrated Stanford competition, Kline (1968), a number of relatively simple and reliable integral boundary layer analysis techniques for flows in adverse and mildly favourable pressure gradients have become well established.

Integral methods are conveniently employed since they avoid local turbulence assumptions and require only the solution of ordinary differential equations. All such methods utilize the well documented Karman integral relation and a correlation for the skin friction coefficient but are closed using a third auxiliary relation which may take a variety of forms and relies entirely or partly on empirical observation.

According to the literature, the only readily adaptable method which takes into account, empirically, relaminarization phenomena is the method for the prediction of axisymmetric turbulent boundary layers in conical nozzles due to Au (1972). A version of the momentum equation for axisymmetric incompressible flow is developed which is shown to be a more general form of the two dimensional Karman equation utilized by previous workers. Closure is achieved using an extended version of Heads' entrainment equation (1958) and resultant entrainment functions are formulated empirically from the experimental results. Good agreement with experiment for boundary layer flow in venturi-meters over the inlet Reynolds number range $1 \times 10^5 < Re < 5 \times 10^5$ was obtained.

In figure 10 the flow element is bounded by the wall surface BC and an imagined inner surface AD of radius r_c. It is assumed that flow is axisymmetric and the velocity at the edge of the boundary layer ($r = r_c$) is given by the potential flow solution at the wall.

From continuity the mass flow across the annular surface AB is equal to the sum of the mass flows from the cylindrical surface AD and the annular surface CD. The mass flows have associated momentum fluxes and the change of momentum flux in the x-direction in the element equals the sum of all the applied forces on the element. These applied forces consist of pressure forces on the surfaces AB and CD, the x-direction pressure force due to the sloping wall and x direction force due to wall friction.

The x-direction momentum equation may be written:

$$\frac{d}{dx}[\int_{r_c}^{R} \rho u^2 r dr] - U\frac{d}{dx}[\int_{r_c}^{R} purdr] + \frac{1}{2}\frac{dp}{dx}(R^2 - r_c^{\,2}) + R\tau_w = 0 \qquad (2\text{-}31)$$

Applying the one-dimensional Euler equation and assuming incompressible flow, equation (2-31) becomes:

$$\frac{d\theta}{dx} = \frac{R}{R-\theta}\{[\frac{\theta^2}{2RU}(H^2+2) - \frac{\theta}{U}(H+2)]\frac{dU}{dx} - \frac{\theta}{R}\frac{dR}{dx} + \frac{Cf}{2}\} \qquad (2\text{-}32)$$

where $H = \delta * / \theta$ (2-33)

$$Cf = \tau_w \setminus \frac{1}{2}\rho U^2 \qquad (2\text{-}34)$$

Au uses the Ludwieg-Tillmann equation (1949) for skin friction coefficient:

$$Cf = 0.246(10)^{-0.678H}\,\mathrm{Re}_\theta^{\;-0.268} \qquad (2\text{-}35)$$

White (1974) reports that (2-35) correlates available data to within +10% and that equation (2-36) is a more accurate correlation (+3%). Equation (236) is adopted in this analysis.

$$Cf = 0.3\exp(-1.33H)(\ln \mathrm{Re}_\theta)^{(-1.74-0.31H)} \qquad (2.36)$$

Evaluation of the shape factor, H invokes the auxiliary entrainment equation derived from a consideration of the mass flow rate in the boundary layer. The mass flow rate in the boundary layer at any point is given by

$$\dot{m} = 2\pi \int_{r_c}^{R} \rho u r dr \qquad (2.37)$$

Using this definition, the mass flow rate in the boundary layer per unit circumferential length of nozzle wall is then:

$$\dot{M} = \rho U H' \theta \{1 - \frac{(H' + 2H)}{2R} \qquad (2\text{-}38)$$

where

$$H' = \frac{\delta - \delta^*}{\theta}$$

The rate of entrainment is then given by:

$$\frac{d\dot{M}}{dx} = \frac{d}{dx}[\rho U H' \{1 - \frac{(H' + 2H)}{2R}\}] \qquad (2\text{-}39)$$

Thus $$\frac{d\dot{M}}{dx} = f(\rho, U < H', \theta, \varepsilon, \mu)$$

where $$\varepsilon = 1 - \theta(H' + 2H) / 2R \qquad (2\text{-}40)$$

expressing (2-39) in non-dimensional form we have:

$$\frac{d}{dx}(\rho U H' \theta \varepsilon) = \rho U \phi(H' \varepsilon) f(\mathrm{Re}_\theta) \qquad (2\text{-}41)$$

With reference to the analogous case of turbulent jets and wakes where the viscosity plays no controlling part in the spreading of the turbulent region, it is assumed that the entrainment of the boundary layer by the core flow will be essentially independant of Reynolds number. For incompressible flow therefore:

$$\frac{d}{dx}(U\theta\eta) = U\phi(\eta) \qquad (2\text{-}42)$$

Where $$\eta = H'\varepsilon \qquad (2\text{-}43)$$

Au determined $\phi(n)$ and η empirically and related $\phi(n)$ to a single argument of shape factor, H.

$$\phi(H) = A_1 H^3 + A_2 H^2 + A_3 H + A_4 \qquad (2\text{-}44)$$

$$H = B_1\eta^3 + B_2\eta^2 + B_3\eta + B_4 \qquad (2\text{-}45)$$

where:

$A_1 = 0.0090507$

$A_2 = -0.041917$

$A_3 = 0.0688713$

$A_4 = -0.0379579$

$B_1 = 8.2342 \text{ X } 10^{-5}$

$B_2 = 0.00636019$

$B_3 = -0.131286$

$B_4 = 2.04229$

The entrainment equation may be written:

$$\theta \frac{d\eta}{dx} + \eta \frac{d\theta}{dx} + \frac{\theta\eta}{U}\frac{dU}{dx} - \phi = 0 \qquad (2\text{-}46)$$

A modified Adams-Moulton predictor-corrector technique was used to solve the boundary layer flow equations. A flow chart indicating the general method of solution is shown in figure 11.

The wall velocity distribution is obtained from the potential flow solution derived in section 2.2 and is used to solve the boundary layer equations. Iteration to allow for the modification to the potential flow solution due to the finite boundary layer displacement thickness was thought unnecessary due to the very small boundary layer thickness present in nozzle flows. Solutions were carried out on a Prime 750 Computer and run times were of the order of 30 seconds.

The boundary layer solution yields the skin friction coefficient distribution along the nozzle wall from which the laminar sublayer thickness may be determined. The sublayer thickness S. is defined such that:

$$y+ = 5$$

$$i.e. \frac{Yv^*}{\nu} = 5$$

where $\quad v^* = \sqrt{\tau_W / \rho}$

Using the definitioon of skin friction coefficient (2-34)

$$S = \frac{5\sqrt{2\nu}}{U\sqrt{Cf}} \qquad (2\text{-}47)$$

The majority of the uncertainty associated with the prediction of the laminar sublayer thickness is due to uncertainty associated with the prediction of the shape factor, H. From a general observation of available boundary layer solution techniques it is apparent that good agreement between methods for prediction of momentum thickness is achieved but that prediction of shape factor is not so consistent nor accurate. This is primarily due to lack of experimental data for flows in strongly favourable pressure gradients which cause considerable modification of shape factor. Fortunately the form of equation (2-47) determines that S values are relatively insensitive to C_f values predicted by equation (2-36) and larger changes result from variation in inlet conditions.

In fully relaminarized boundary layers the inner or viscous dominated region may be considerably wider than for turbulent flow under equivalent conditions. It is likely therefore, that sublayer thicknesses defined by equation (2-47) represent a lower limit for some nozzle flows.

Separation

The approximate separation point in boundary layer flows is determined as the point where the velocity gradient is zero and classically the onset of separation is recognized as $C_f \rightarrow 0$. A well known criteria for boundary layer flow separation is that due to Stratford (1959) and has been used as a design criteria by many wind tunnel and diffuser designers. The criteria does not require knowledge of the boundary layer solution itself but does require the virtual origin of the boundary layer to be known.

A more accurate and more conveniently utilized method for the present purpose is that due to Weber (1978) and is given by:

$$Kp\ Re_{\theta}^{1.163} = -0.0319 \qquad (2\text{-}48)$$

The above criteria requires knowledge of the previously discussed pressure gradient parameter, K_p and the local momentum thickness Reynolds number and is similar to the criteria reported by Kutateladze (1964) given by:

$$Kp\ Re_{\theta} = -0.00668 \qquad (2\text{-}49)$$

The two equations yield the same value of K_p for separation at $Re_{\theta} = 15{,}000$.

Coefficient of discharge, C_d

Many attempt have been made in the past to predict discharge coefficients. In spite of more rigorous formulations of C_d in terms of exit boundary layer parameters for nozzles and venturimeters by Benedict and Wyler (1978) and Au (1972) respectively, the approximation to the original equation of Rivas and Shapiro (1956) used by Hall (1959), is sufficiently accurate for many engineering purposes.

For a parallel exit section nozzle:

$$Cd = (1 - 4\ \delta^*/do) \qquad (2\text{-}50)$$

In the case of nozzles terminating in nonparallel exit sections, account must be taken of the non-uniformity of the flow at the nozzle exit and the tendency of the flow to contract as discussed in section 2.2. However, in such cases, values of cut have a small effect (< 0.5%) on coefficients of discharge. C_d values may be taken to a good approximation to be those given by coefficients of contraction, C_c. C_d and C_c values are related according to:

$$C_d = C_c^{\ 2} / [1 - (\frac{C_c d_o}{D})^2]^{\frac{1}{2}} \qquad (2\text{-}51)$$

APPLIED NOZZLE DESIGN

Design Constraints

With a view to satisfying the criteria for design indicated by both the literature survey and E.C.L.P. Ltd. operating requirements, a general nozzle form consisting of a cubic arc matched to a final straight taper cone (CA+T) (figure 12) is recommended. The objective of the theoretical study of such nozzle forms is to determine the minimum acceptable nozzle length and to provide an engineering estimate of the quality of surface finish required to provide hydraulically smooth flow conditions.

Given D, d_o, A, L_t constant, L_f was chosen to avoid a point of inflection in the cubic arc portion of the nozzle contour. This constraint leads to a relatively simple nozzle shape and eliminates the possibility of separation and/or cavitation at the entrance to the final taper section. Figure 12 indicates that selecting L_f too large or too small leads to a point of inflection in the cubic contour.

A nozzle diameter of 31.75 mm (1.25 inches) was selected for the study since the majority of experimental test work has been conducted with this diameter nozzle. From consideration of nozzle inlet conditions, an acceptable contraction ratio of 10.25 in conjunction with a 101.6 mm (4 inch) barrel diameter is optimum. The reader is referred to Jackson (1983) for more details of design constraints.

Nozzle flow analysis was conducted according to the method outlined in section 2. In this section appropriate inlet conditions for and results from theoretical studies of nozzle flow in the four nozzles shown in figure (13) are presented and discussed.

Nozzle Inlet Boundary Layer Parameters

To commence nozzle flow boundary layer analysis, starting values of boundary layer momentum thickness, and boundary layer shape factor, H are required. Assuming that the origin of a developing boundary layer is taken as the exit of the stream forming device and that core flow is approximately uniform at this point, the degree of flow development in the following 4 to 8D settling section may be estimated. Recent reviews of turbulent developing pipe flow are provided by Ward-Smith (1980) and Klein (1981). Experimental results presented in these reviews indicate that within the first 4 to 8D of developing pipe flow the blockage coefficient, B defined as:

$$B = \frac{2\delta^*}{R} \qquad (3.1)$$

attains values of 0.02 to 0.09 for $2 \times 10^5 < Re_i < 6 \times 10^5$. Fully developed pipe flow in the above reported Reynolds number range is achieved after approximately 50D and 0.14 < B

< 0.16 may be expected. Development of shape factor is more rapid (1.5 < H < 2.0), H attaining its fully developed flow value of between 1.3 and 1.4 after only lOD.

Combining equation (3.1) with the definition of shape factor gives:

$$\theta = B\,R/2H \qquad (3.2)$$

Evidence from the literature suggests that increased turbulence intensities and pipe wall roughnesses accelerate flow development. Table 2 indicates the approximate range of corresponding B. H. likely to be encountered in practical flow situations and hence used as starting conditions in subsequent flow analysis. The corresponding values of η from (2-45) are also indicated.

TABLE 2

B	H	θ	η
0.02	2.0	0.0005	0.3
0.04	1.9	0.001	1.1
0.07	1.8	0.002	2.0
0.10	1.6	0.003	4.1
0.12	1.5	0.004	5.4
0.14	1.4	0.005	7.1

Initial boundary layer conditions corresponding to θ = O.002 have been selected for boundary layer calculations. Flow solutions were obtained for a representative range of flow rates corresponding to $3 \times 10^5 < Re < 6 \times 10^5$. The range of nozzle pressures due to the above Reynolds number range varies according to nozzle design since C_c values and hence C_d values are functions of nozzle design. As discussed in section 2.7 values of Cc reported by Lohn and Brent (1976) give the best available indication of the C_d values for the nozzle designs of figure 13. Values of C_c and C_d are indicated in table 3.

TABLE 3

β	C_c	C_d
30°	0.902	0.814
15°	0.946	0.895
12.5°	0.957	0.916
7°	0.973	0.947

Values of flow rate and nozzle pressure based on the C_d values of table 3, corresponding to $3 \times 10^5 < Re < 6 \times 10^5$ are indicated in tables 4 and 5 respectively.

TABLE 4

Re_i x 10^5	U_i (m/s)	Q_o (m^3/hr)	Q_o (gpm)
3	3	87.6	321
4	4	117	429
5	5	146	535
6	6	175	642

TABLE 5

	Re_i x 10^5							
β	3		4		5		6	
	Po		Po		Po		Po	
	Kpa	psi	KPa	psi	KPa	psi	Kpa	psi
30°	714	104	127.	184	1980	288	2850	414
15°	590	85.6	1050	152	1640	238	2360	342
12.5°	564	81.7	1000	145	1570	227	2250	327
7°	527	76.5	937	136	1460	212	2110	306

RESULTS

Figures (14,15) indicate the effect of nozzle aspect ratio on exit momentum and displacement thicknesses respectively and confirm the result of Lohn and Brent (1976) that a reduction in nozzle wall length leads to a significant reduction in exit values of δ^* and θ. Values of exit displacement thickness are an order of magnitude lower than values inferred from the C_d measurements common in the literature due to the provision of hydraulically smooth exit taper sections with $\beta < 7°$. It may be inferred from equation 2-50 that the small values of exit displacement thickness determine that, to within ±0.5%, C_d values are equivalent to those given by the jet contraction ratio (section 2.7).

Boundary layer exit momentum and displacement thickness decrease with increasing upstream Reynolds number due to the correspondingly higher exit wall velocities and hence are decreasing functions of nozzle pressure. Exit boundary layer parameters are also a function of inlet boundary layer parameters and figure 16 indicates the effect of changes in inlet conditions, corresponding to changes in values of θ_i on exit momentum thickness. In general nozzle exit boundary layer parameters are weak functions of inlet momentum thickness Reynolds number and the relative effects of nozzle design are preserved.

Much can be determined from the inviscid flow solution alone if presented in terms of the pressure gradient parameter, K_p. Both separation and relaminarization phenomena may be considered by examination of the distribution of K_p along the nozzle walls shown in figure 17. K_p increases with increase in pressure gradient and decreases with increase in velocity so that a maximum in K_p exists in the profiles. The finite values of K_p at the nozzle exit are indicative of the finite, nonzero values of exit pressure gradient responsible for the relatively thin exit boundary layer dimensions associated with figures 14,15.

The minima in pressure gradient parameter K_p min. at the entrance of the nozzles correspond to points of maximum unfavourable pressure gradient and hence are the points at which separation is most likely to occur. As previously discussed in section 2.6, the separation criteria of Weber (1978) and Kutateladze (1964) are most conveniently applied. Strictly the application of equations (2-48) and (2-49) requires a knowledge of the local value of Re_θ and hence the boundary layer solution. However, an estimate of the onset of separation may be obtained using the upstream initial conditions. In fact the local velocity will be less than the upstream value due to velocity undershoot at the inlet and the local value of θ larger due to the tendency of the boundary layer to grow in an unfavourable pressure gradient.

Figure (18) indicates values of K_p min. as a function of nozzle aspect ratio. K_p values corresponding to the onset of separation for Re_θ = 5,000, 10,000, 15,000 are indicated. In view of the range of operating conditions likely to be encountered and the fact that nozzle inlet boundary layers are likely to be in the early stages of development (Re_θ < 10,000), it is thought likely that separation occurs in designs 1 and 2 but that it is avoided in designs 3 and 4 of Fig. 13.

Applying the relaminarization criteria of section 2.4 then figures 17, 19, 20 suggest that some degree of relaminarization occurs in all nozzles. For nozzle design 1 with AR=1 a fully relaminarized boundary layer results for nearly all flow conditions and it is likely that the process of retransition to turbulence is little developed at the nozzle exit. For designs 2 (AR=1.5) and 3 (AR=2) however, the pressure gradient parameter begins to relax closer to the nozzle inlet, maximum values are considerably less and the process of retransition likely to be further developed at the nozzle exit. For high exit velocities retransition at the nozzle exit may be complete. In design 4 (AR=3.5) little relaminarization of the boundary layer occurs within the nozzle and a turbulent exit boundary layer is likely for all flow conditions considered.

Throughout the literature it is well documented that nozzles with high final tapers produce jets whose surfaces, for a short distance downstream of the nozzle exit, are perfectly smooth and exhibit a glass like appearance. Instability waves on the jet surface which mark the end of this region and the onset of retransition have been photographed by Hoyt and Taylor (1977). In contrast the jet surface from nozzles with cylindrical extensions is described as frosty and milky white in appearance.

From consideration of exit boundary layer displacement thicknesses, the onset of separation and relaminarization phenomena, it is suggested that a nozzle design with AR=2 is close to optimum. A relatively high degree of exit velocity profile uniformity is achieved whilst separation is avoided and for high velocities retransition within the nozzle is likely. Some margin of error in prediction of separation is incorporated in this selection and shorter nozzles may be tolerated in practice. The adverse effects of mild separation may be more than offset by the greater degree of exit velocity profile uniformity achieved. lt is unlikely however, that AR < 1.5 may be tolerated.

Nozzles for other barrel diameters and nozzle diameters which are very nearly geometrically similar would be expected to exhibit similar relative performance for the same range of inlet conditions. Recommended nozzle design parameters are summarized in table 6.

TABLE 6

$D(mm)$	$d_o(mm)$	$L_t(mm)$	$L_f(mm)$
	31.7	200	100
101.6	28.58	207.46	107.46
	25.40	214.94	114.94
152.4	34.93	329.87	179.87
	38.10	321.49	171.49
	41.28	314.32	164.32
	44.45	307.17	157.17
203.2	47.63	435.79	235.79
	50.80	428.64	228.64
	60.00	407.94	207.94
	63.50	400	200

From equation (2-47) (section 2.5) nozzle exit laminar sublayer thicknesses have been calculated as a function of local wall velocities for the designs under consideration and are shown in figure 21. Calculated sublayer thicknesses are compared with sublayer thicknesses for equivalent smooth pipe flow and are shown to be between 20% and 30% less. Sublayer thicknesses decrease with increasing local wall velocity as might be expected. This suggests that a higher quality of surface finish is required in nozzles operated at higher pressures. The relatively small values of sublayer thickness calculated for nozzles with taper exit sections indicates that a higher degree of surface finish is required than hitherto thought necessary by, for example, Murakami and Katayama (1966).

Figure 22 indicates that variation in S values of up to +20% from values calculated with $\theta_i = 0.002$ may be associated with changes in nozzle inlet boundary layer parameters. This together with the requirement that all roughness element heights should be well within the laminar sublayer thickness, suggests that a safety factor of 10 or more should be incorporated into levels of required surface finish. In addition the work of Cocks (1968) suggests that this is particularly important for surfaces where APV roughnesses are considerably greater than CLA roughnesses.

CLA surface finishes of between 0.18 μm and 0.22 μm (7 and 9 μinch) are suggested as adequate for the recommended designs of table 6. However, material type and method of manufacture may dictate lower CLA values and a careful check on maximum roughness heights should always be performed.

Figure (23) illustrates the sublayer profile along the wall of the recommended design and suggests as may be expected that the very high degree of surface finish is only required adjacent to the nozzle exit where wall velocities are highest. Ideally any joint in the nozzle wall as a result of construction method should be flush to within a tolerance of the order of the local value of S.

Combining equations (2-36) and (2-47) yields an estimate of surface finish requirements at the nozzle inlet. Values of sublayer thickness are approximately independant of inlet boundary layer parameters and for Re = 5×10^5 S is approximately 82 μm (3 thous.). The work of Schofield (1981) suggests that upstream wall roughness may cause flow downstream over hydraulically smooth walls to retain rough flow characteristics and emphasises the importance of maintaining smooth flow conditions far upstream of the nozzle inlet. Thus attention should be given to barrel surface finish and prevention of corrosion. Allowing for a developing boundary layer, barrel roughness element heights should be less than 13 μm (0.5 thous.).

NOZZLE DESIGN AND JET PERFORMANCE

The basic flow regimes which describe the development of highly turbulent water jets are outlined in section 2.1. It is evident that such flows are essentially two-phase in nature and complete description is the subject of another part of the study of large scale water jets conducted by Exeter University. In this paper consideration of jet flow analysis is confined to a simple description in terms of axial jet-dynamic pressure decay, the chosen indicator of jet-performance used here.

According to available literature, Russian workers appear to have been the first to assess jet-performance by measuring the decay of time average jet dynamic pressure along the jet using a pitot tube device e.g. Shavlovskii (1966). Results have been documented for both small and large scale jets and some recorded data are presented in figure 24. The data indicate that pressure decay trends are independent of scale and to a good approximation are correlated according to:

$$\frac{Pm}{Po} = 1, \qquad X < Xc$$

$$\frac{Pm}{Po} = (c1do / X)^k, \qquad X > X_c \tag{5-1}$$

where $X_c = C_1 = A_l - B_l.Re$, $k = 0.85$ and A_l, B_1 are constants determined by nozzle form. Hence the value of region length, X_c decreases with increasing Re and varies according to nozzle design, the best nozzles having the largest value of $C_1 = X_c$.

The reduced pressure decay of the small nozzles relative to the large scale nozzles, by virtue of increased initial region lengths may be noted. This is a reflection of the higher contraction ratios associated with the small nozzles (lower exit turbulence intensities) and the probably lower inlet turbulence intensities associated with straight feed-pipe configurations as opposed to high turbulence generation monitors.

CONCLUSIONS

A procedure has been outlined which allows a nozzle to be designed with the object of producing the most coherent water jet for use in the hydraulic mining of china clay. For other applications, with different constraints, other aspects of nozzle design may require more emphasis, but the following general conclusions should be kept in mind:

1. The precise mathematical definition of the nozzle contour itself is not important provided that the following requirements are complied with.
2. Nozzle inlet flows should have a minimum turbulence intensity, be free of large scale tu'rbulent eddies, and be free of any rotational or swirl components.
3. Joints between inlet piping and nozzle should be concentric in order to avoid asymmetric boundary layer development and associated rotational motion of the flow. Protruding gaskets must be avoided. If steps are unavoidable they should be

expanding rather than contracting and ideally step height should be restricted to the order of the local laminar sublayer thickness.

4. Nozzle exit core flow turbulence intensities should be minimized by providing a sufficient contraction ratio of $10 < CR < 45$.
5. Nozzle exit boundary thickness should be minimized by ensuring:
   i) some degree of nozzle exit taper i.e. $\beta > O$ and hence avoiding parallel, cylindrical extensions.
   ii) minimum nozzle aspect ratio.
   iii) hydraulically smooth wall flow from nozzle approach section to nozzle exit.

6. Minimization of boundary layer exit thickness leads to larger values of jet initial region lengths and hence from the simple correlations adopted for jet axial pressure decay, results in increased impact pressures for a given set of nozzle conditions.

7. The above conclusion (6) implies that for a given nozzle design, the higher the value of C_d the better the jet performance. In comparing different nozzle designs however, account must be taken of the values of coefficient of contraction. It may be that a nozzle with a low C_d value performs well since both δ^* and C_c values are relatively small. Thus the general statement that high C_d values are always associated with the best jet performance is seen to be fallacious. In addition low C_d values are not detrimental in terms of pumping power requirements, since although identical diameter nozzles with different values of C_d require different nozzle pressures to achieve the same flow-rate, nozzle diameter may be adjusted to achieve identical flow-rate and nozzle pressure.

8. Boundary layer separation at the nozzle inlet and subsequent generation of large-scale turbulence in resultant recirculation zones should be prevented.

9. Cavitation near the nozzle exit due to sharp changes in wall curvature and isolated roughness elements should be prevented.

10. Safety factors of approx. 10 should be applied to required surface finishes assessed by CLA roughness measurements.

11. Precise definition of the nozzle exit plane must be ensured.

12. Retransition of a relaminarized boundary layer should occur within the nozzle.

ACKNOWLEDGEMENT

The authors wish to acknowledge the financial and practical support made available by ECLP Ltd., St. Austell, Cornwall, U.K. and the SERC, U.K.

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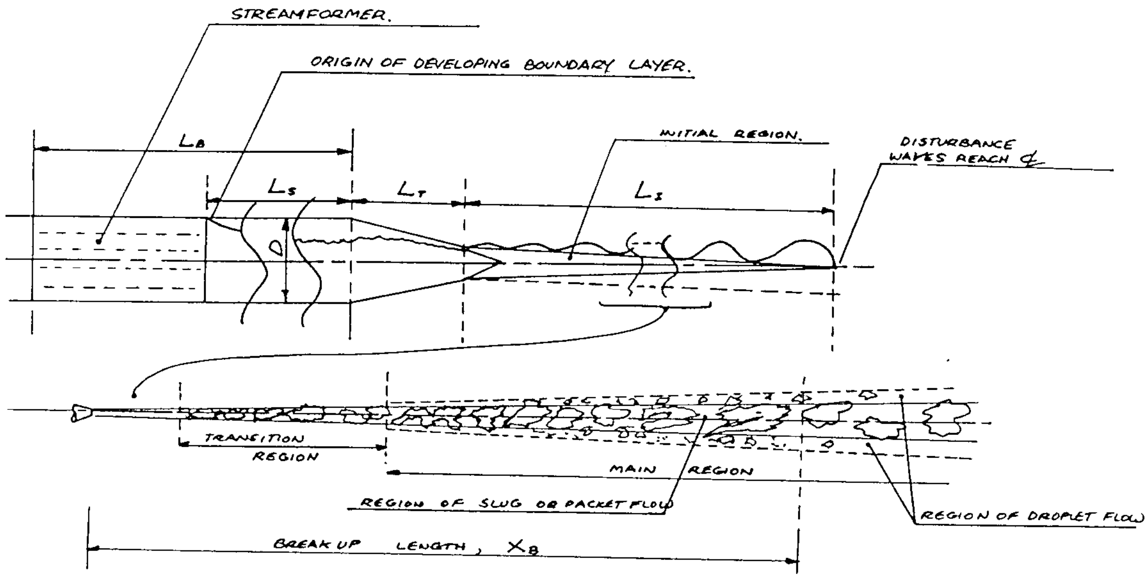


Figure 1. Monitor and jet flow regions.

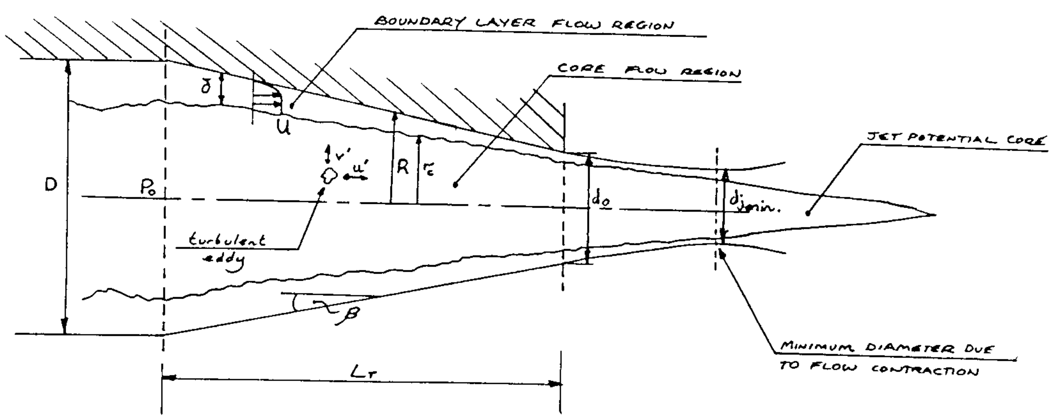


Figure 2. Regions of nozzle flow.

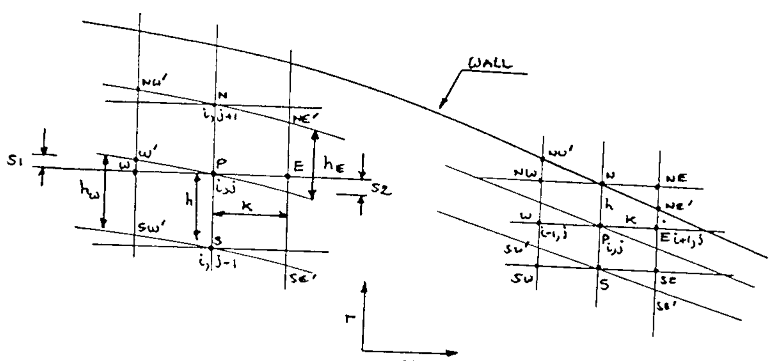


Figure 3. Curvilinear Finite Difference scheme.

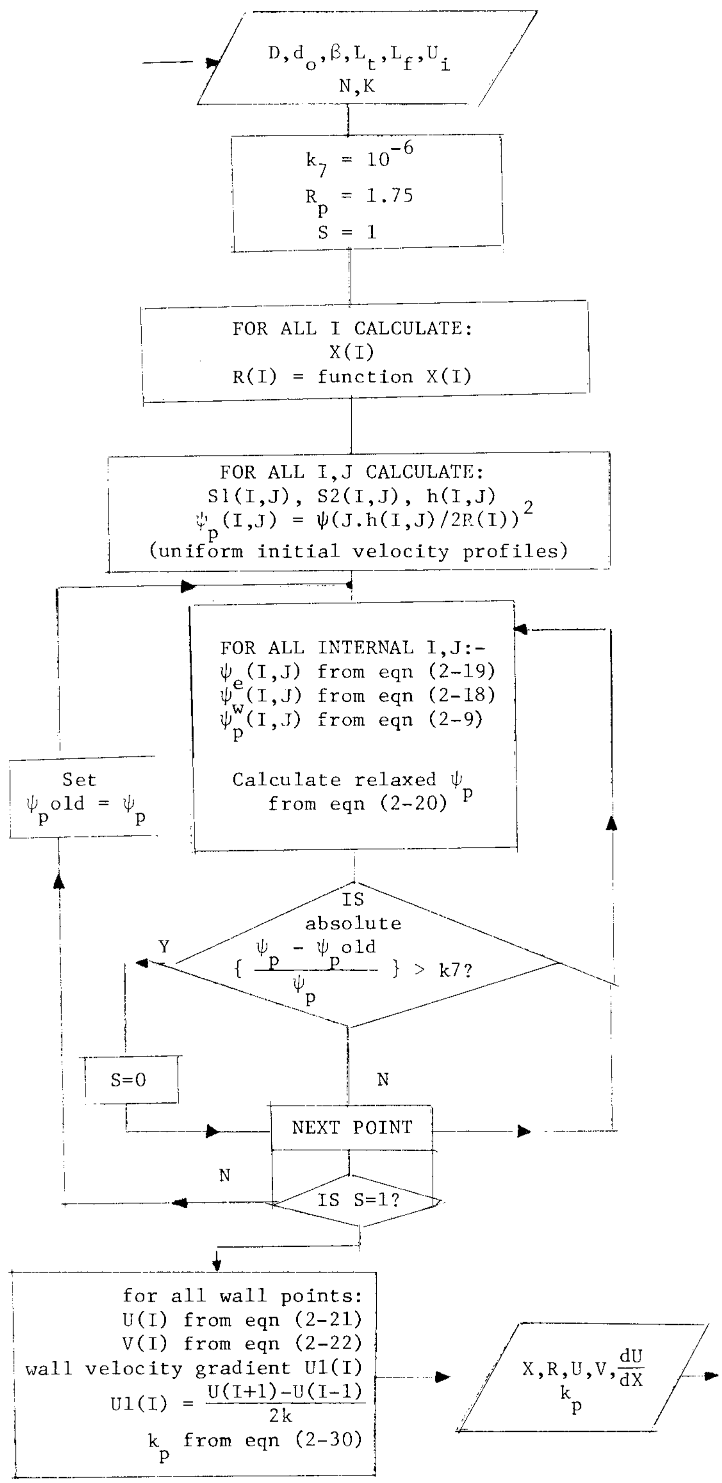


Figure 4. Potential Flow solution scheme

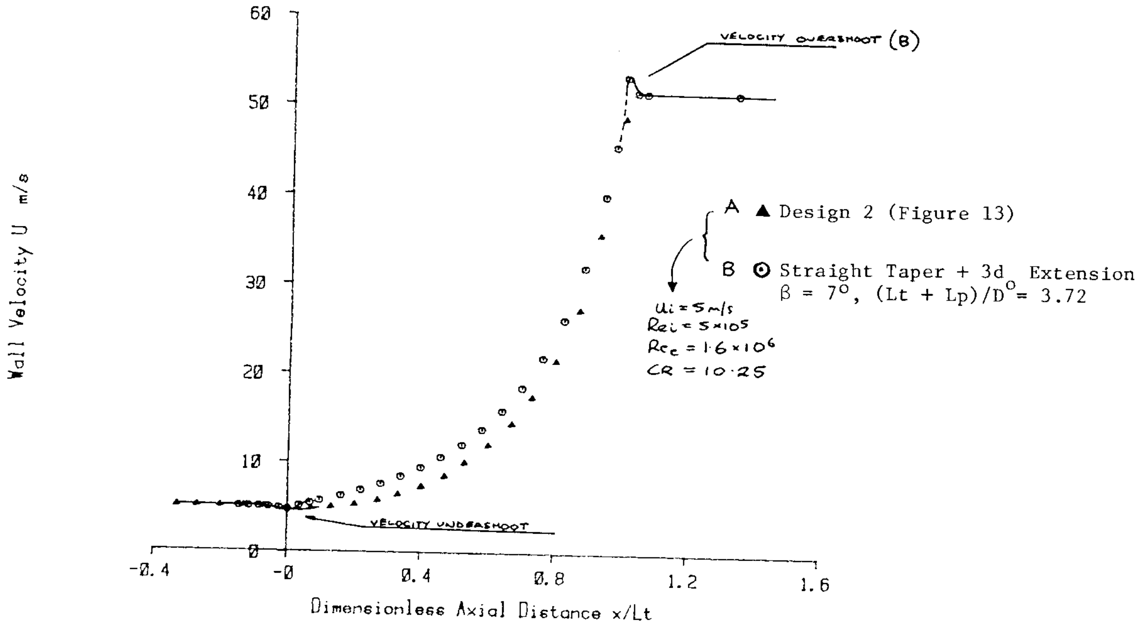


Figure 5. Wall velocity distribution

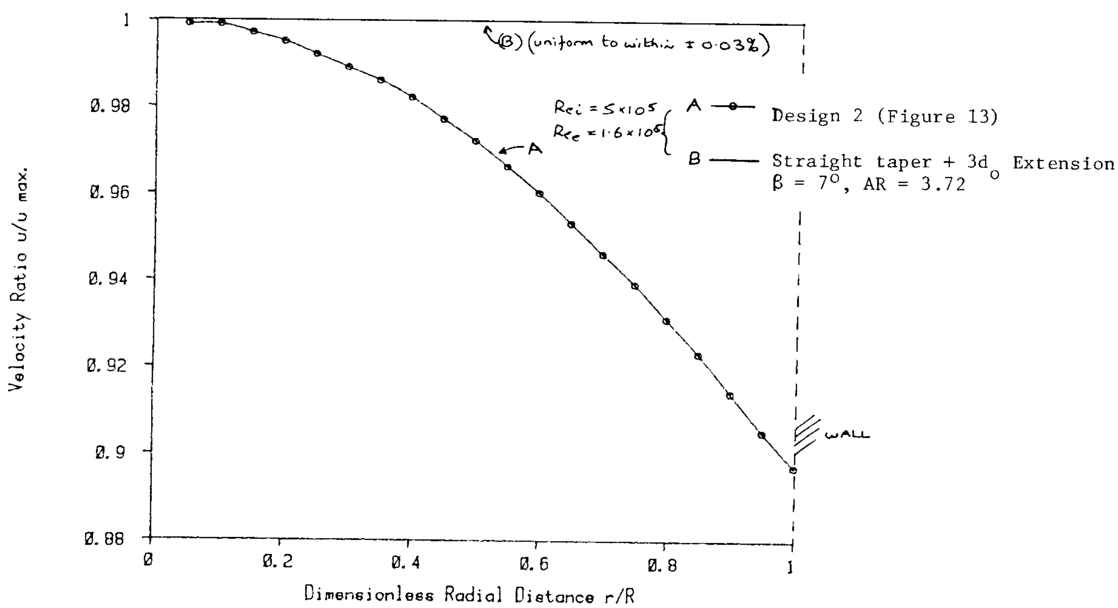


Figure 6. Nozzle Exit Radial velocity profiles

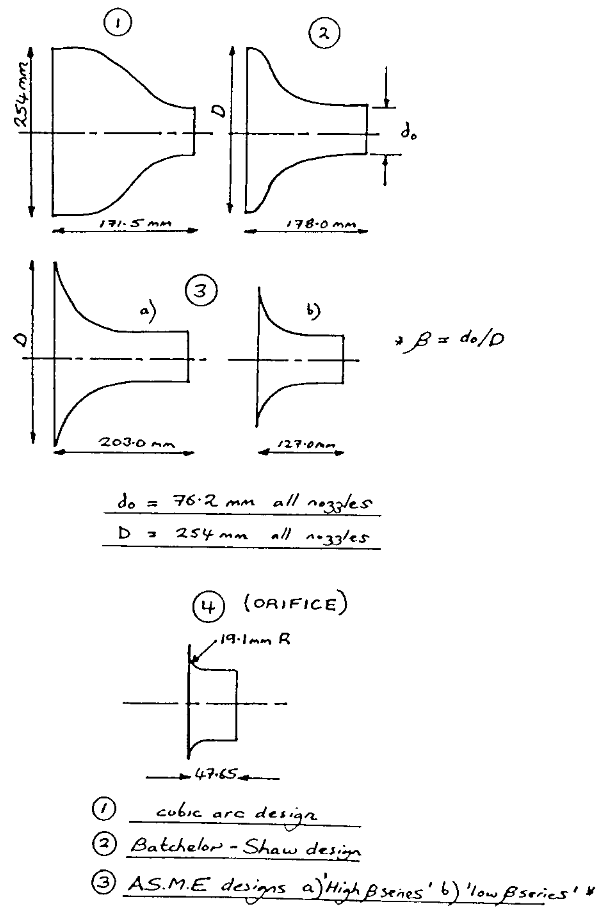


Figure 7. Nozzle Designs

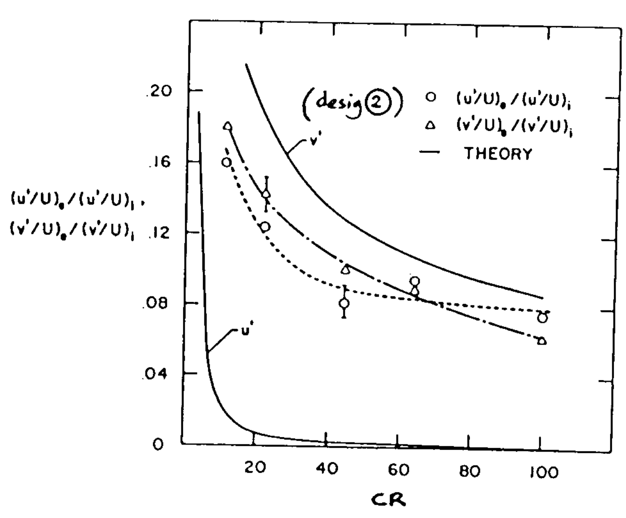


Figure 8. Effect of contraction ratio on turbulence intensity.

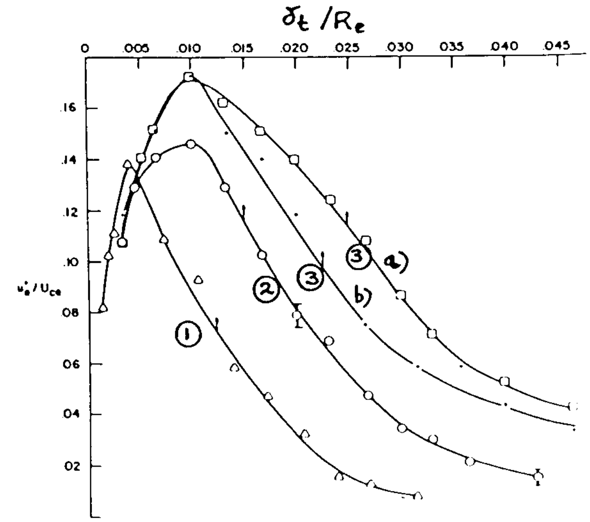


Figure 9. Effect of nozzle design on exit turbulence layer thickness.

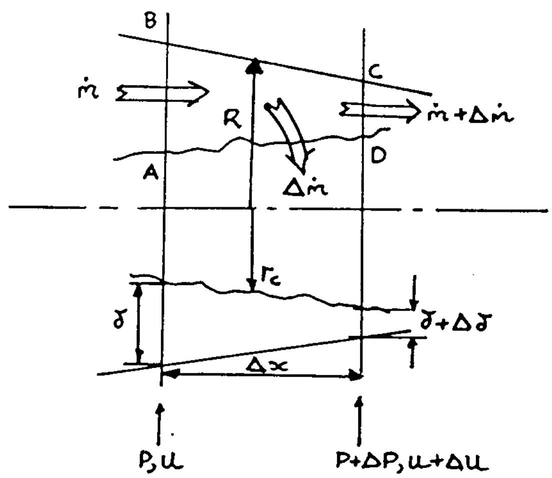


Figure 10. Flow element.

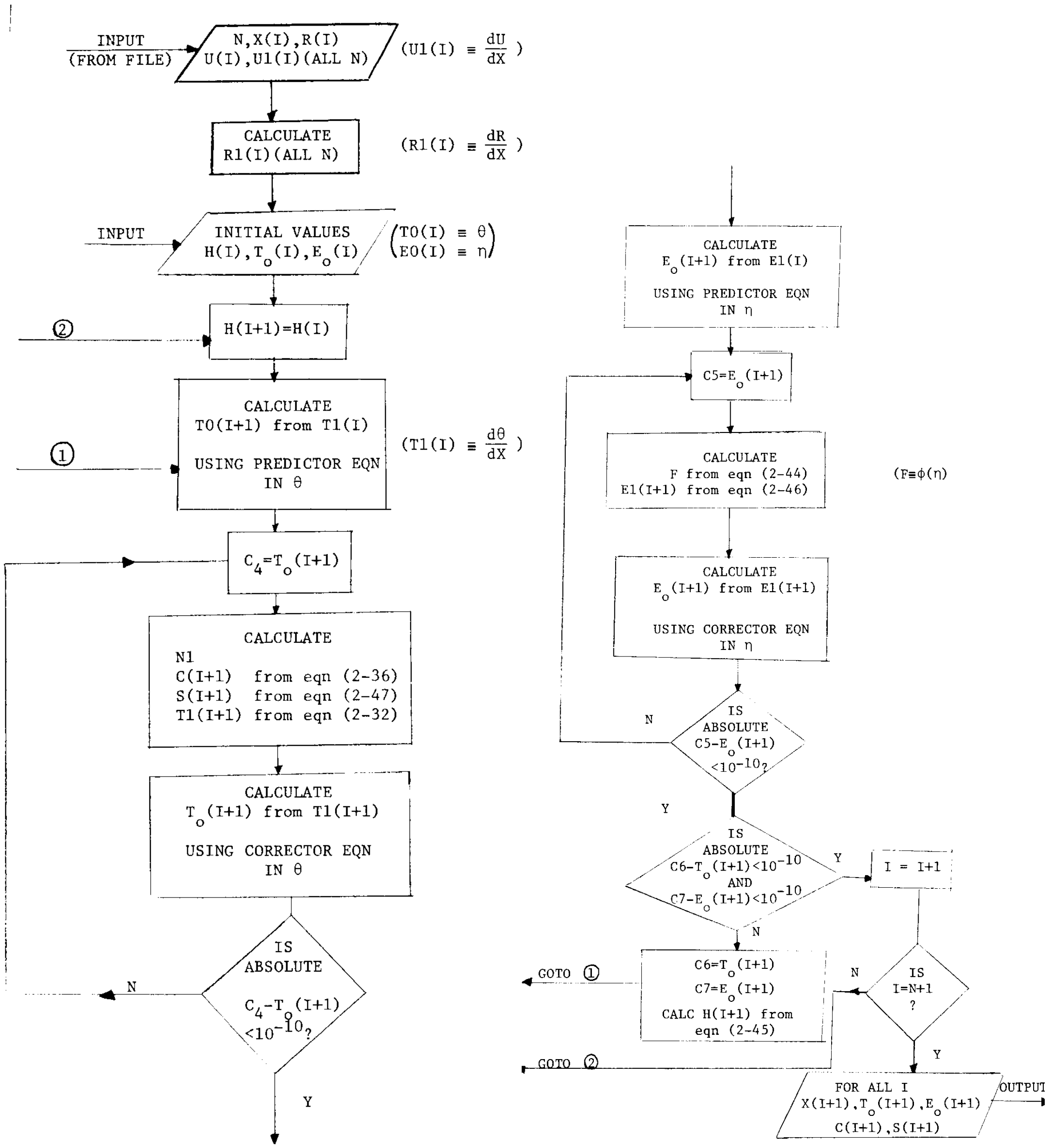


Figure 11. Boundary layer Solution scheme.

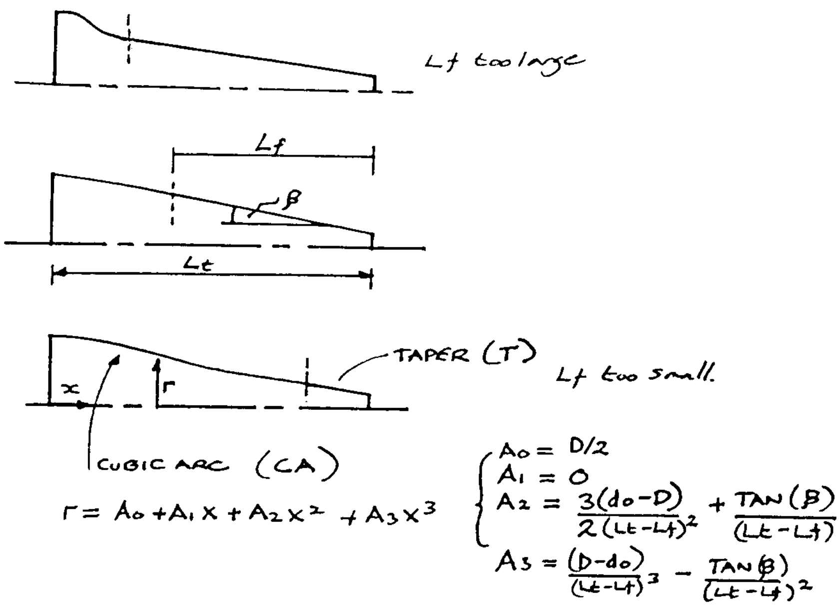
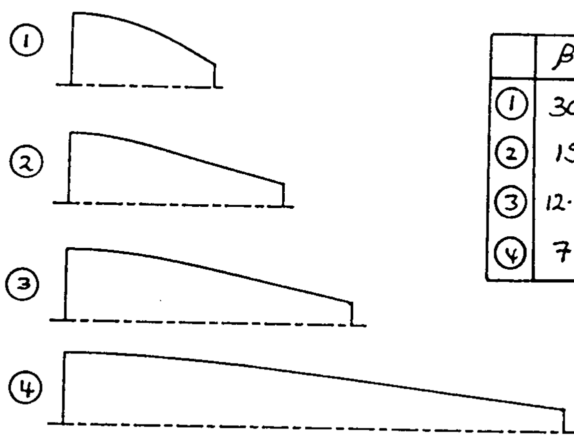


Figure 12. CA+T Design Philosophy.



	β	L_t (mm)	L_f (mm)	$AR = \frac{L_t}{D}$
1	30°	100	20	1.0
2	15°	150	100	1.5
3	12.5°	200	100	2.0
4	7°	350	150	3.5

Figure 13. Experimental Nozzle Designs.

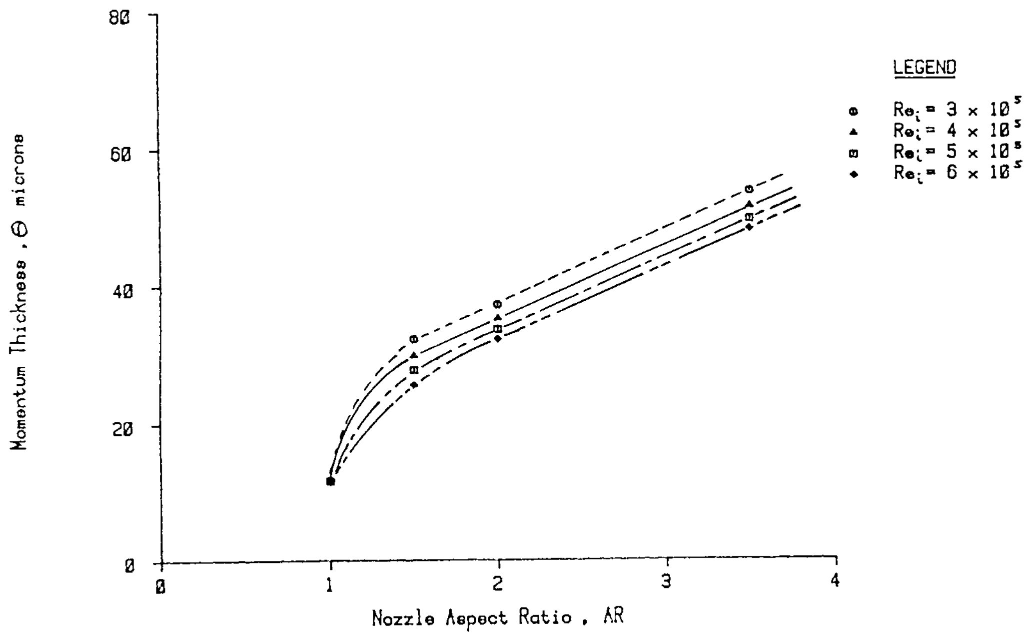


Figure 14. Nozzle exit momentum thickness.

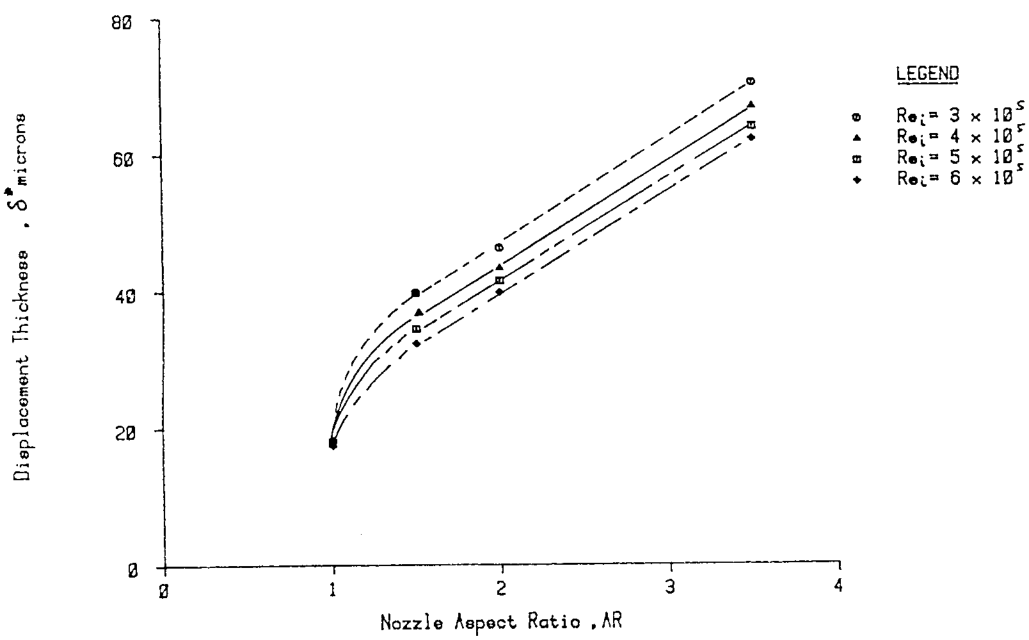


Figure 15. Nozzle exit displacement thickness.

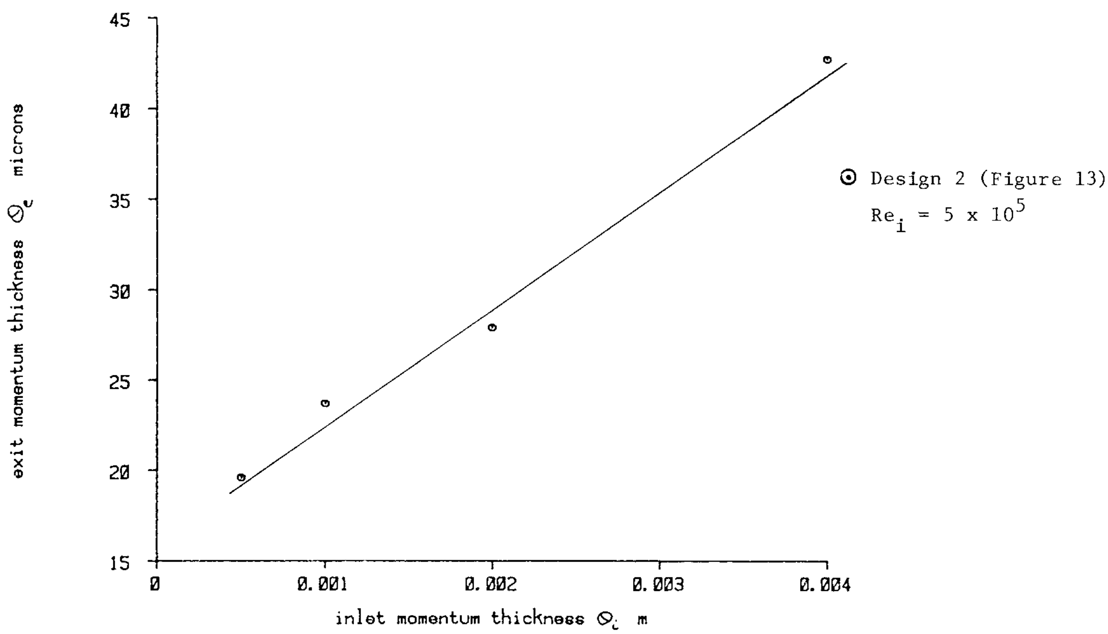


Figure 16. Effect of inlet b_1 conditions on exit b_1.

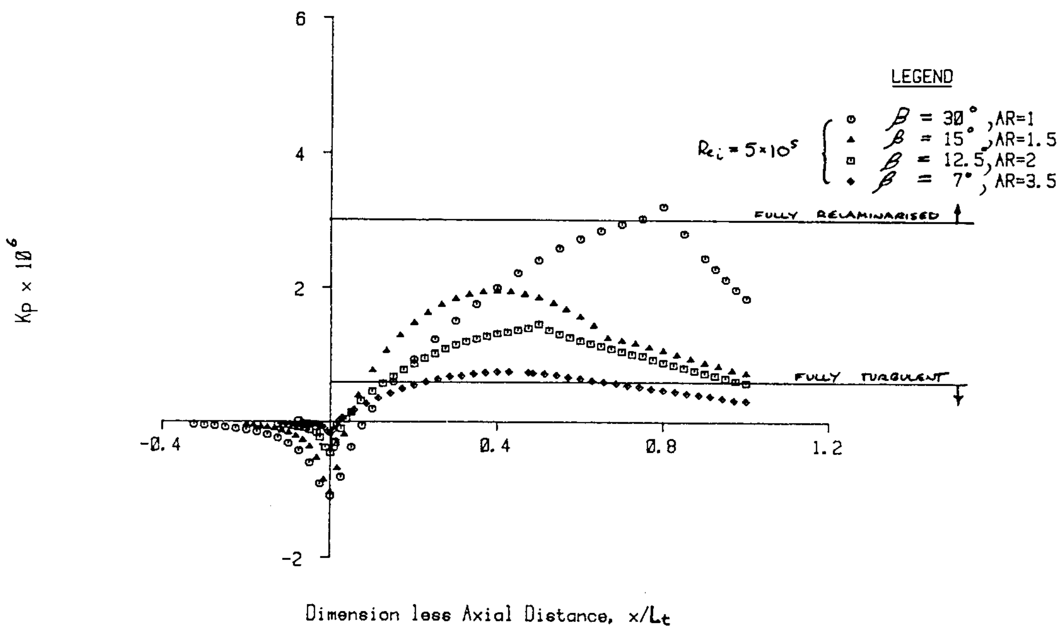


Figure 17. Axial distribution of K_p.

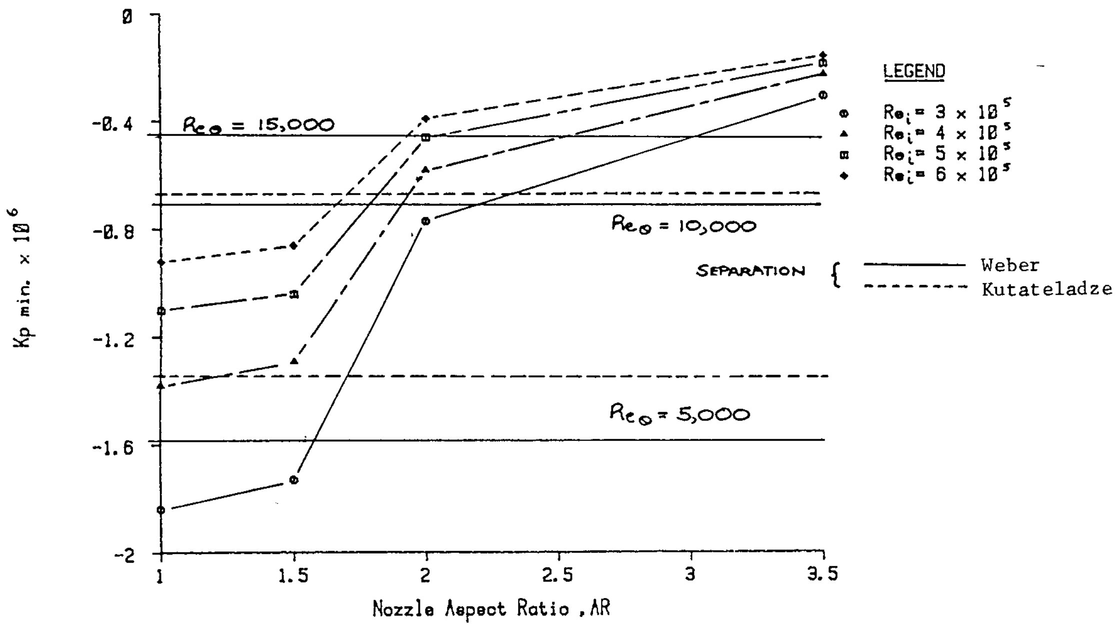


Figure 18. Examination of separation criteria.

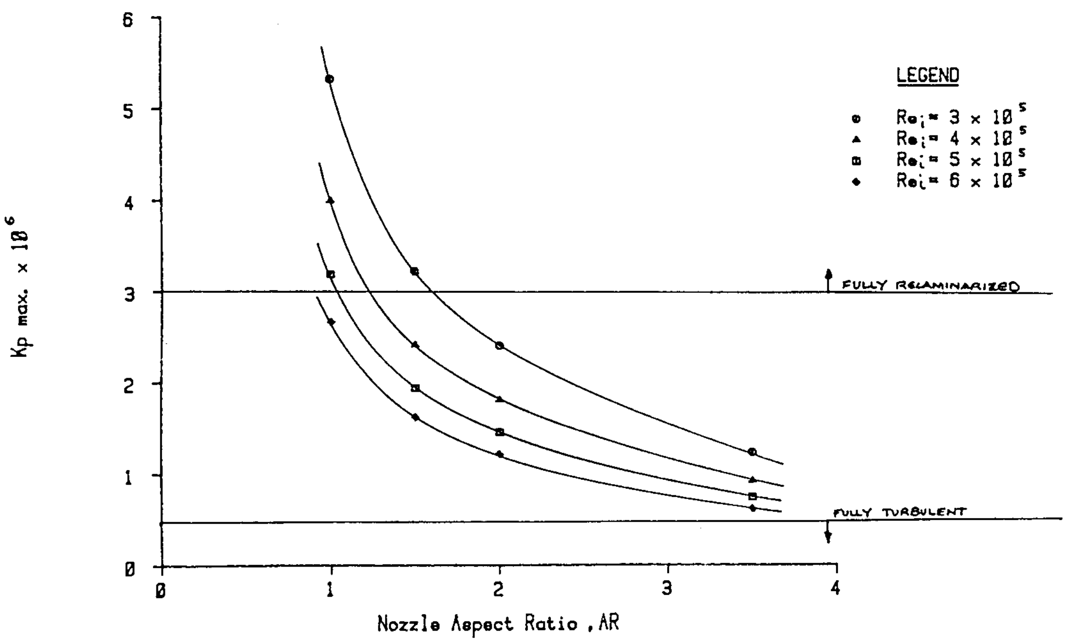


Figure 19. Relaminerization within nozzles.

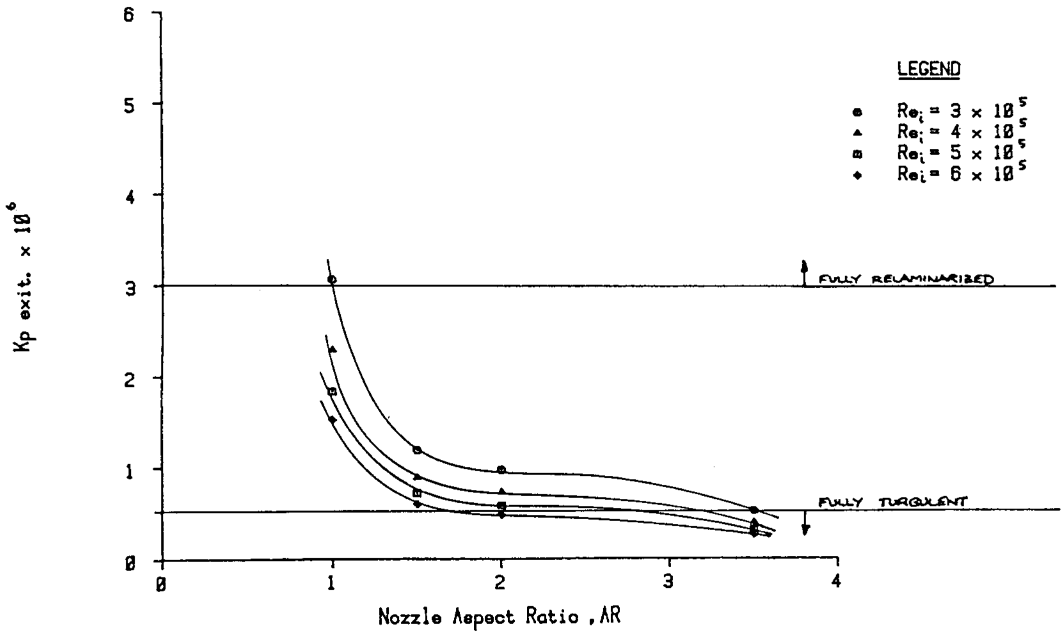


Figure 20. Relaminarization at nozzle exit.

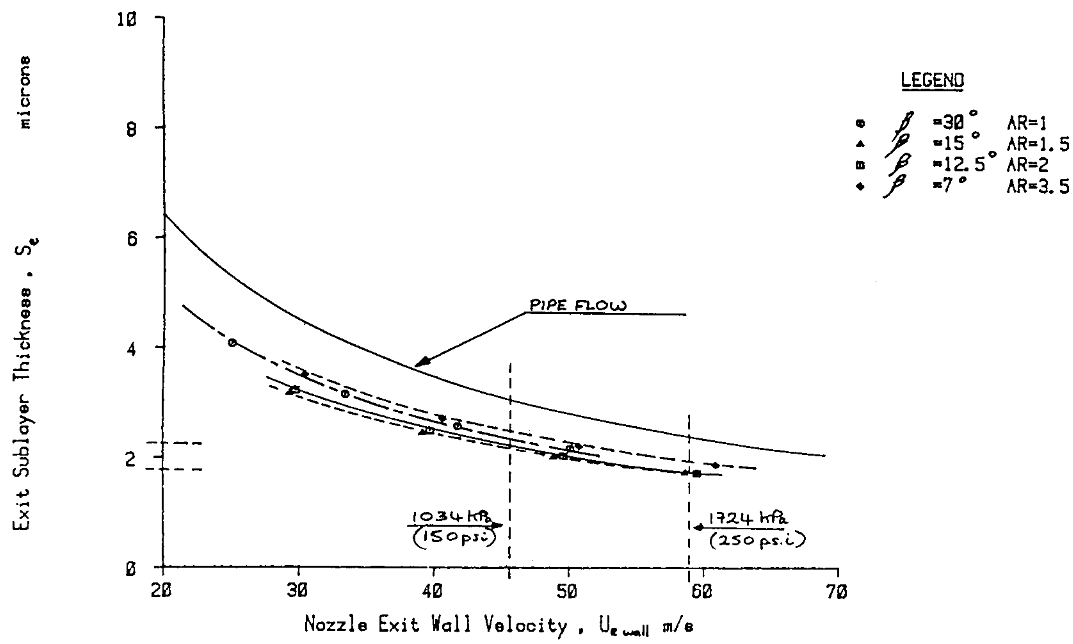


Figure 21. Nozzle exit sublayer thickness

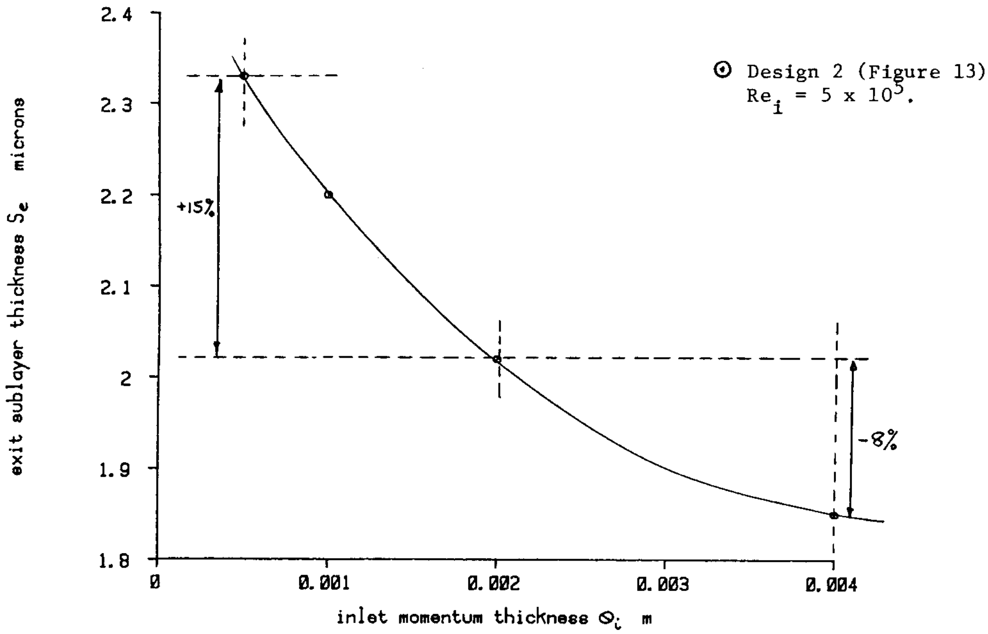


Figure 22. Effect of inlet b_1 conditions on exit sublayer.

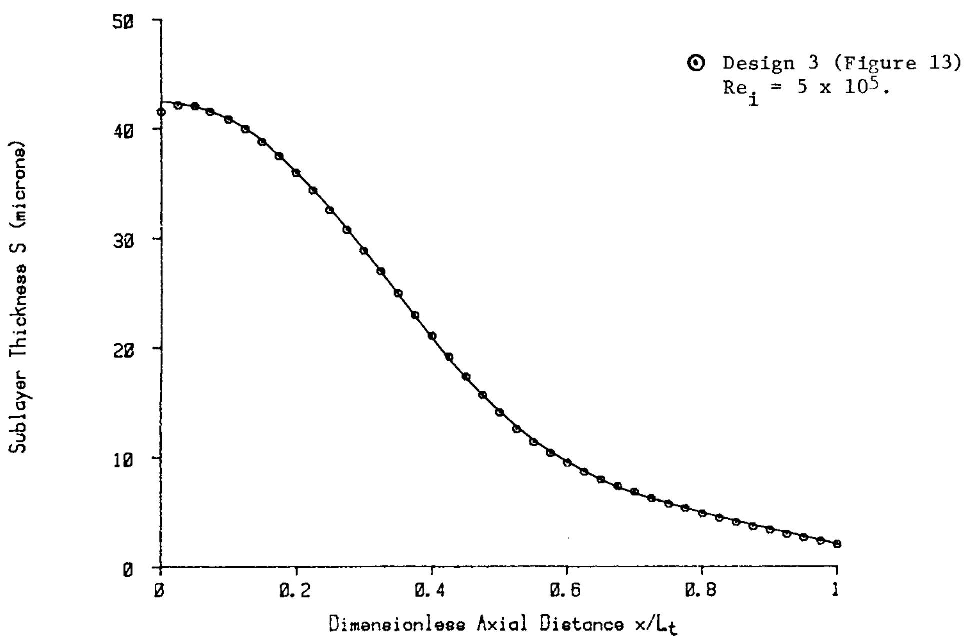


Figure 23. Sublayer profile for nozzle design 3 (Fig. 13)

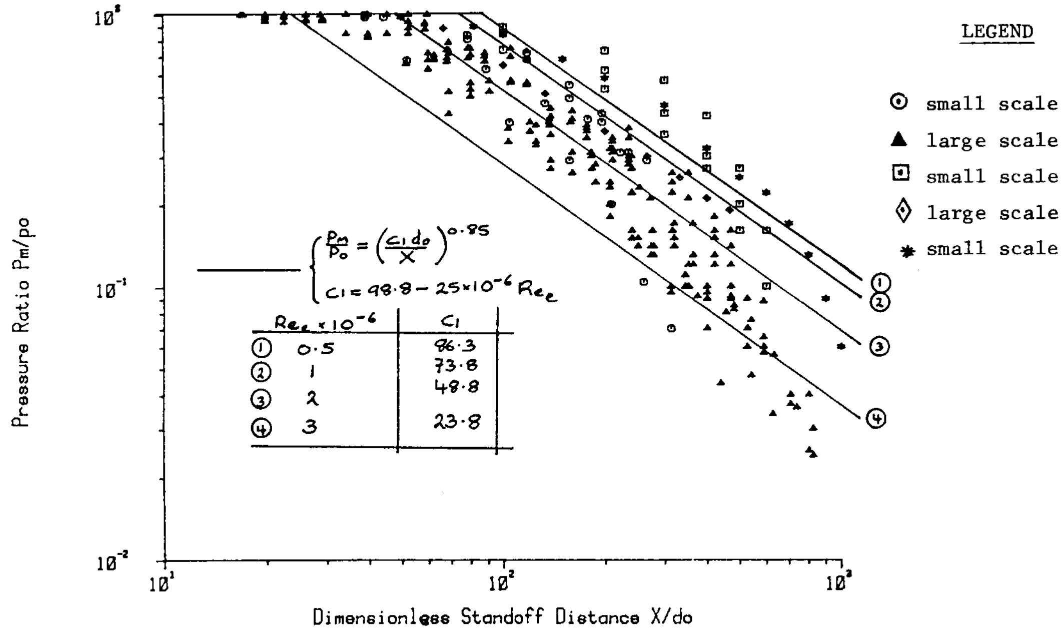


Figure 24. Pressure decay data.

DISCUSSION

NAME: Mohamed Hashish

COMPANY: Flow Industries, Kent WA

QUESTION:"The paper is quite thorough and needs a lot of time to digest; however, I have a quick question: Will this analysis be valid for high pressure small diameter jets? And if not, what additional factors need to be considered for small jets?"

ANSWER:The basic approach to the analysis of steady flow in nozzles is valid for any Newtonian system. The particular circumstances of a given nozzle/flow combination may produce domination of the analysis by a single parameter, rather than contributions from all the parameters discussed in the paper.

NAME: James L. Evers

COMPANY: Southern Illinois University at Carbondale

QUESTION:"Since the optimum nozzle shape requires a sharp edged nozzle exit, wear will be a problem. How does the nozzle performance degrade as you decrease the sharpness of the exit?"

ANSWER:Experiments show that the jet deterioraates catastrophically with loss of exit sharpness and uniformity.

NAME: David Eddingfield

COMPANY: Southern Illinois University

QUESTION: "My compliments on a good paper. Have you considered Peter Lohn's work?"

ANSWER: Our paper uses the work Of P. D. Lohn and D. A. Brent, U. S. Bureau of Mines, Report No. 28852-6005-TtJ-OO, as the basis for our own development.

VISUALIZATION OF THE CENTRAL CORE OF HIGH-SPEED WATER JETS
AN INFRARED TECHNIQUE

Eugene B. Nebeker, Ph.D., and John B. Cramer
Scientific Associates, Inc.
Santa Monica, California

ABSTRACT

Valuable information about structure, coherency, etc. of a high-speed water jet can be obtained by examining the central core. This problem is complicated because aerodynamic drag on the jet shears off the exterior surfaces of the core to form a mist which surrounds the jet. The presence of this mist often prevents direct observation of the core because the mist absorbs or scatters the light needed to visualize the core.

Many investigators have employed photographic methods to penetrate the surrounding mist for the purpose of observing the core. These earlier procedures did not produce photographs with sufficient contrast to allow the examination of fine details of the jet.

An infrared photography technique is presented which was developed to improve direct examination of a jet core. The procedure is inexpensive, relatively convenient, and adaptable to use in the field. The method uses light absorption and exploits the fact that water absorbs light in the infrared region of the spectrum much better than in the visible.

INTRODUCTION

Valuable information about structure, coherency, etc. of a high-speed water jet can be obtained by visualizing the central core. This problem is complicated because aerodynamic drag on the jet shears off the exterior surfaces of the core to form a mist which surrounds the jet. The presence of this mist often prevents direct observation of the core because the mist absorbs or scatters the light needed to visualize the core.

Many investigators have employed photographic methods to penetrate the surrounding mist for the purpose of observing the core (Brown). Unfortunately, this earlier work was not useful to us in our Percussive Jet development (Nebeker and Rodriguez, 1973, 1979) because we needed to visualize very minute details of the fluid mechanics of a bunching mechanism. This process was happening, for example, 20,000 times per second while the individual bunches were travelling at the bulk velocity of the jet at about 1000 feet/second.

Therefore, some process was needed to more effectively penetrate the mist surrounding the core of a high-speed jet. The technique should be capable of stopping the motion of the jet so details can be examined. The procedure should be inexpensive, relatively convenient, and adaptable to use in the field.

APPROACH

Most photographic techniques operate by measuring the scattering or absorption of light from a subject. In the case of water, scattering is enhanced by the presence of an uneven surface or the occurrence of droplets. Absorption seemed to be a more promising process because it increases with increasing density of the subject such as that associated with the core itself. The absorption spectrum of water (Fig. 1, Washburn, 1929) shows that water absorbs light in the infrared and ultraviolet regions of the spectrum very well. Visible light is absorbed quite poorly. For photographic purposes, the infrared region is often more convenient to use than the ultraviolet portion because photographic materials are more readily available and glass lenses, etc. do not absorb infrared light as easily as ultraviolet.

Over 200 years ago, Lambert (1760),discovered that when a parallel beam of monochromatic light of wavelength X and intensity l_o enters a homogeneous absorbing material, the light which is transmitted through a layer of thickness t will have the intensity

$$I(\lambda) = I_o(\lambda)e^{-k(\lambda)\ell} \qquad (1)$$

where k is a positive constant called the absorption coefficient of the material. Lambert's law can be derived by assuming that each infinitesimally thin layer of the absorbing material absorbs an amount of light which is proportional to the thickness of the layer and to the intensity of the monochromatic radiation reaching it. This relation is shown schematically (Fig. 2) for a typical jet.

Usually, strictly monochromatic light is not employed. However, equation 1 is also valid over a region of the spectrum when the absorption coefficient is a weak function of wavelength or in a reasonably small region of the spectrum for which the incident light is continuous (Nebeker, 1965). Problems sometimes arise if equation 1 is applied in regions where the absorption spectrum is discrete and not continuous. However, the experimental circumstances can often be adjusted so that this equation is an adequate representation of the observations.

Differentiating equation 1 with respect to thickness t (k and I_o are constants)

$$\frac{dI}{dl} = \frac{k}{I_o}e^{-kl} \qquad (2)$$

Hence, the rate of change of transmitted light intensity is proportional to the absorption coefficient k. Figure 1 shows that the absorption coefficient is two or three orders of magnitude greater in the infrared region of the spectrum than the visible. Therefore, equation 2 indicates that this infrared approach will yield results which are much more sensitive to variations in width and size of the central core of the jet than one would expect using light in the visible region of the spectrum.

DISCUSSION

Although employing the infrared region of the spectrum potentially offers great improvements over the commonly used visible region, the method of implementation must be discussed. Figure 3 shows a schematic drawing of the process. The light source emits radiation which passes through the Jet. The Jet absorbs the light according to its absorption spectrum and width l. The resulting radiation is passed through a filter to remove any ultraviolet or visible light. Upon impacting the film, the film will respond based upon its spectral sensitivity. If color film with a reversal film developing process is used, the spectral dye density curves will help to determine the density of the image. Naturally, as in all photography, the characteristic curves for the film show the effect of exposure time on the resulting image density.

In short, numerous considerations are involved in designing a photographic process of this type. Many photographic variables are available to the experimenter to manipulate so that a wide variety of results can be obtained. The following discussion represents a typical selection which we have found useful.

Source

The light source used was a General Radio Type 1538-A Strobotac and a 1539 Stroboslave. Flash duration was 3 microseconds. A typical spectral output from a xenon flashtube of this type is shown in Fig. 4. The data ranges from about 350 nm to 1100 nm in the spectrum. This spectral distribution can be altered by changing current density, pulse width, bore size, fill pressure, and envelope material. As an example, output in the near infrared region of 700 - 900 nm is maximized at a relatively low current density of 400 - 1500 amps/cm^2.

Absorption by Jet

The light with wavelengths from 350 nm to 1100 no passing through the Jet from the source will be absorbed by the Jet. Figure 1 shows relatively strong absorption between 350 and 400 nm and above 700 nm. Little absorption occurs between 400 and 700 nm.

Filter

A Kodak 87 filter was used to remove wavelengths below about 740 nm so that only infrared radiation remained. Figure 5 shows the characteristics of this filter.

Film

Color film, Kodak High-Speed Ektachrome Infrared, IE 135-20, was used. Color film is sometimes more useful than black-and-white film because color contrast is more apparent than black-and-white contrast alone. The spectral sensitivity of these layers is given in Figure 6. These curves show the film is not sensitive to radiation above 900 nm in wavelength. Any region of the spectrum to which photographic materials are sensitive can be recorded on a color film if the individual emulsion layer is sensitized accordingly. The color of the dye formed in a particular layer may be very different than the color of light to which the layer is sensitive.

Processing

Using reversal processing with a Kodak E-4 kit, various other colors of dye are formed in each layer. E-4 processing can be done readily in a simple photographic laboratory. Using the photographic materials previously discussed, jet photographs taken with light in the 740 to 900 nm region showed a predominantly red photograph. If another filter was used to move the lower wavelength down to 600 nm, so that light in the region 600 to 900 nm is allowed to strike the film, the developed image appears yellow. In regions where the central core of the jet absorbed large amounts of the infrared, the photograph was black. Figure 7 shows the spectral dye density curves for this film. The density in a developed film depends on the exposure which that part of the film received and the amount of development given the film. Figure 8 shows the variation in density resulting from various exposures under the same development conditions.

RESULTS

We found this infrared technique extremely useful and much more powerful than visible light photography. For any particular conditions, the experimenter should be aware of the considerations mentioned in Figure 3 before designing his photographic process. In this way, the relative light and dark areas in the Jet can be altered to bring out the specific detail that is desired. For example, Figure 9 is a photograph of a Percussive Jet travelling at 330 feet/second from left to right. Two bunches are shown which are forming in the Jet 300 times per second. Distances between the bunches is 12 inches. The central core of the Jet is 1/2inch in diameter, although the bunches have grown to 2 1/2 inches in diameter. Ligaments of water which have been sheared off from the extremities of the Jet can be clearly seen. Very little aerodynamic shear apparently affects the central core of the Jet because large scale irregularities seem to be moving with the Jet. Notice that vortexing or swirling cannot be seen behind the bunches.

ACKNOWLEDGEMENT

This activity was performed on U. S. Department of Energy Contract DE-AC01-79ET14280. Currently, the technical project officer is Mr. Robert J. Evans. Earlier, the technical project officers were Mr. Hamilton Reese, Jr. and Mr. Howard Parkinson assisted by Dr. Sedat Sami.

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7.

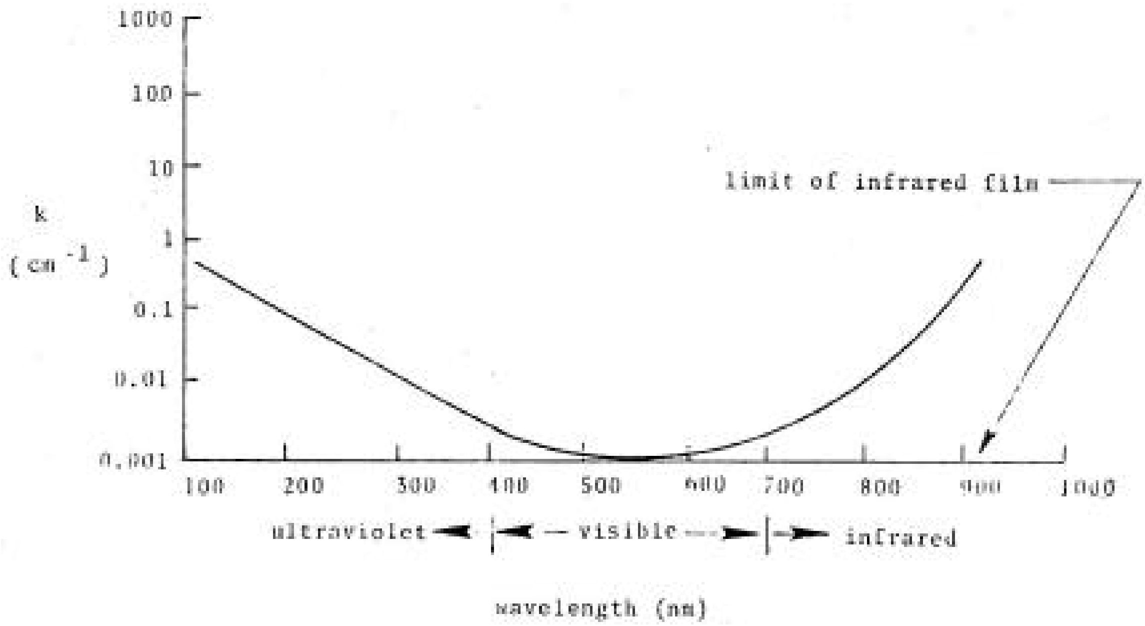

Figure 1. Absorption spectrum of water.

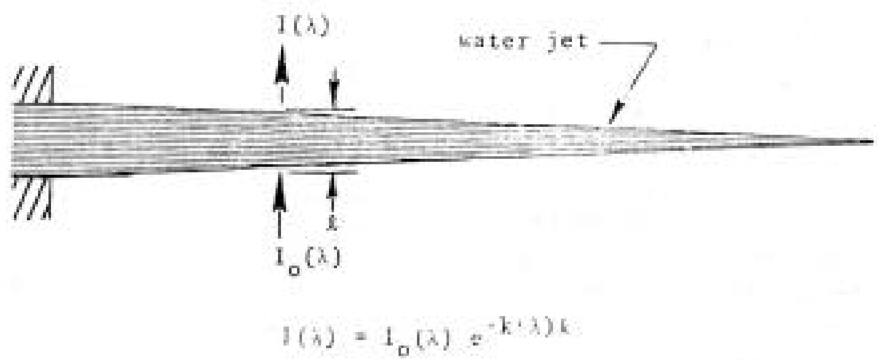

Figure 2. Representation of Lambert's Law.

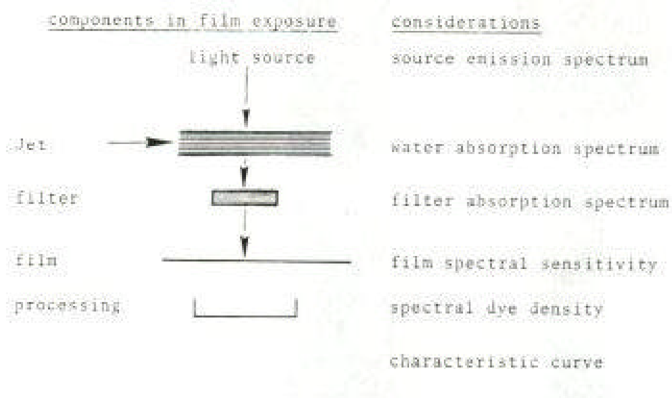

Figure 3. Process schematic

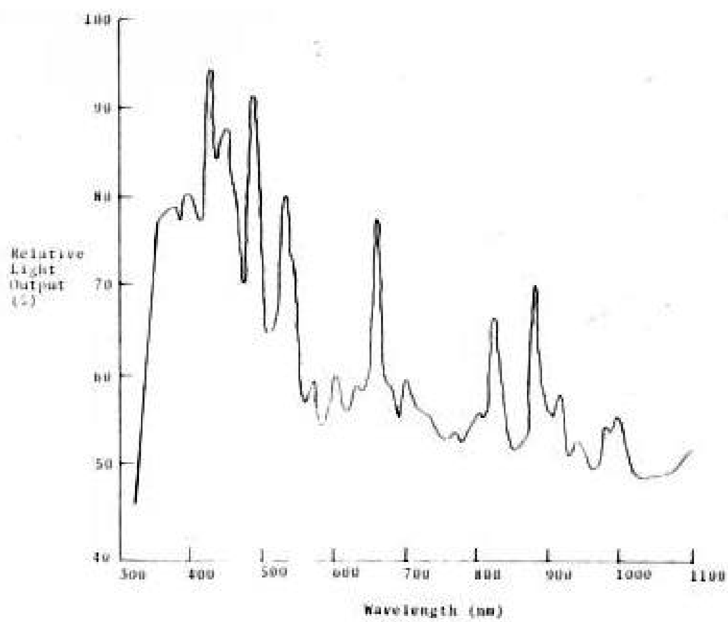

Figure 4. Emission spectrum of Xenon flash tube.

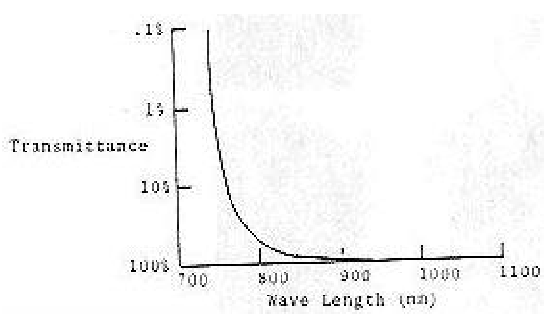

Figure 5. Absorption curve for type 87 filter.

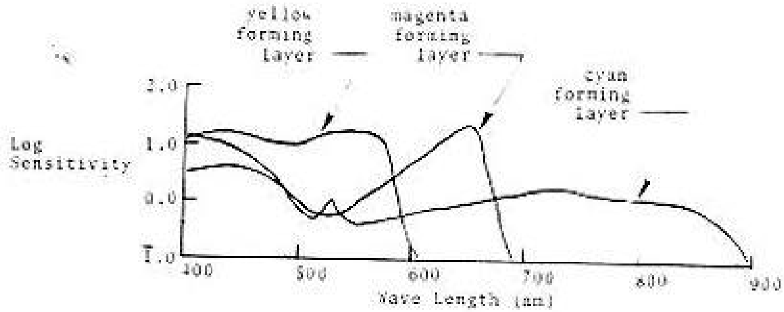

Figure 6. Spectral sensitivity curve.

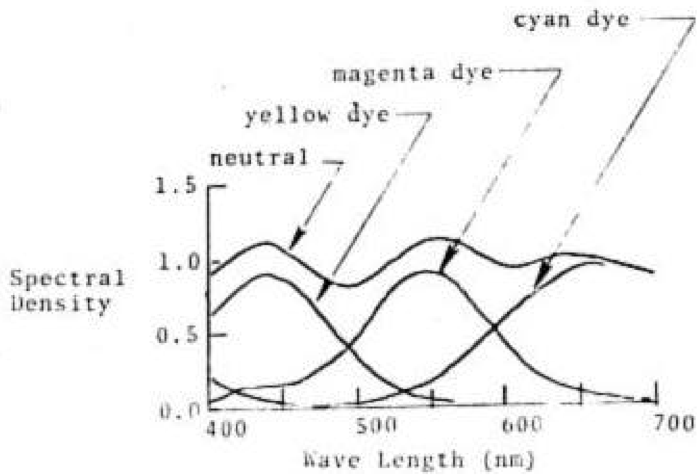

Figure 7. Spectral Dye density curves

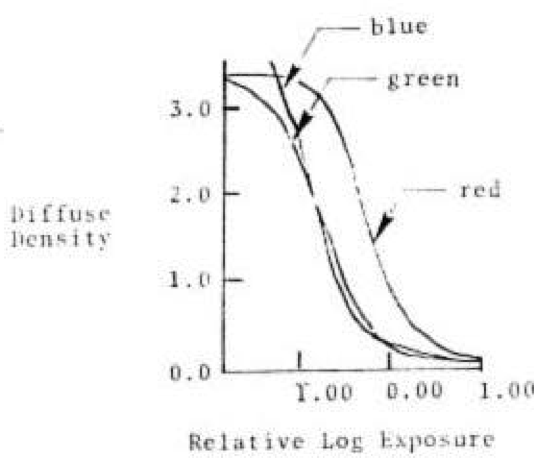

Figure 8. Characteristic curves

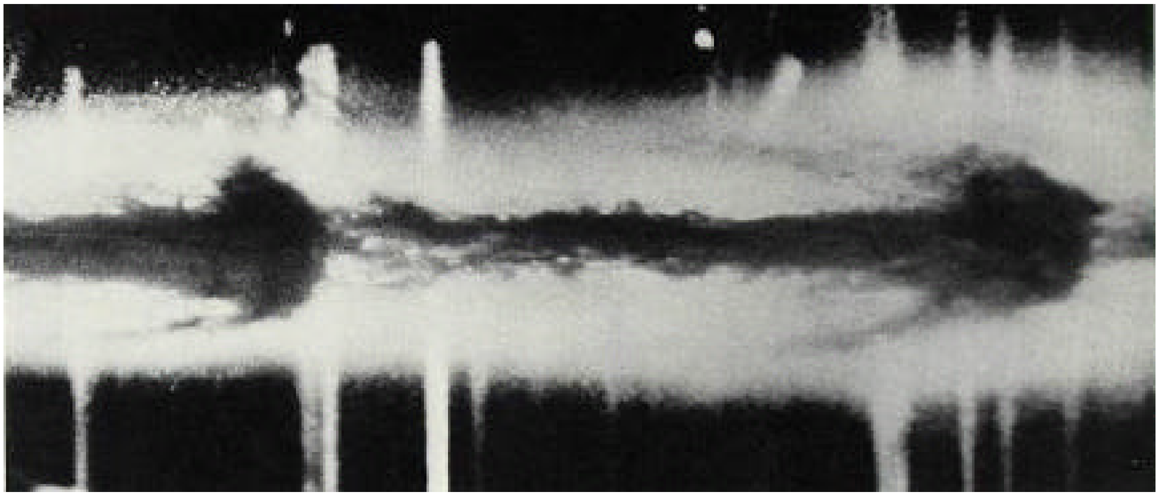

FIGURE 9. INNER CORE OF PERCUSSIVE JET DISCHARGED AT 500 PSIG. Using this infrared photography technique, the mist surrounding the central core of a high-speed jet can be penetrated. In this photograph, the bunches have formed and are about 2 1/2 times the diameter of the original jet (1/2-inch). Surrounding the central core is a sheath which is the mist which has been illuminated by scattered light. The Jet is travelling at about 330 feet/second and the modulation frequency is 300 cycles per second. Therefore the wavelength or distance between bunches in this photograph is about 12 inches. The exposure time was 3 microseconds.

DISCUSSION

NAME: Gerald Zink
COMPANY: Stoneage, Inc.

QUESTION: "The slide of a 'water pik' jet backlighted by 750 nm infrared light appeared to show absorption in the thinner edges of drops and transmission through the thicker center. Why is this?"

ANSWER: The "water pike slides were chosen to show how absorption in the main body of the water drop increased with increasing wavelength of light as predicted by the absorption spectrum of water. At the edges of the drop, some light is scattered or bent away from the camera and appears dark. As a result, using single drops to simulate a turbulent water jet has some limitations as you noted. However, the main point is well illustrated.

NAME: W.C. Cooley
COMPANY: Terraspace, Inc.

QUESTION:"Have you taken photographs of a percussive jet, and a steady jet impacting a target to observe any difference?"

ANSWER:We are currently attempting to do what you suggest; however, I cannot report any results at this time. I believe your point is excellent.

NAME: John R. Wolgamott
COMPANY: StoneAge, Inc.

QUESTION: "Could you say how well your nozzles rate in relation to the study of nozzle decay with distance, as just presented by Dr. Davies? I suspect you have a far from optimum design for the steady flow jet."

ANSWER:No, as shown in several references given in the papers (Nebeker and Rodriguez, 1973, 1976, 1979), we use Leach and Walker 3D-type nozzles which are acknowledged to have excellent performance for steady flow jets. We also do a certain amount of flow conditioning upstream of the nozzle to avoid precisely the additional problems Dr. Davies mentioned.

Although our nozzles are quite adequate, our improvement in standoff distance is due not to nozzle design but to changing the free stream fluid mechanics. Recent data shows that the standoff distance may be increased by a factor of two under certain conditions.

We feel that greater progress in making the jet more coherent can be accomplished in altering the free stream fluid mechanics rather than concentrating on nozzle design. However, I am pleased Dr. Davies presented some of this material.

AN EXTRUSION-TYPE PULSED JET DEVICE

Larry L. Pater, Ph.D. and Paul H. Borst
Advanced Technology Department
Dravo Corporation, Pittsburgh, Pennsylvania

ABSTRACT

An extrusion-type pulsed jet device was analyzed, designed, fabricated and tested. The device consists essentially of a cylinder into which a piston is pushed to extrude liquid through a nozzle located in the cylinder head. An incompressible fluid analytical model of the device is presented, results of which provided operational insight and design guidance. The design of the device, including the values of principal parameters, is described. Results of tests against concrete are presented and compared with other experimental data.

NOMENCLATURE

Parameters

- A area
- D diameter
- E pulse energy
- F the driving force
- H head loss
- K ratio of specific heats
- L length
- M mass
- P fluid pressure
- R nozzle area ratio, = A_p / A_j
- S power stroke length
- t time
- U velocity
- X piston face location, measured downstream from the original location of the piston face
- Y dummy variable of integration
- Z location coordinate, measured downstream from the nozzle entrance
- λ friction factor
- ρ fluid density (constant)

Subscripts

- j jet
- p piston
- n nozzle
- c chamber

Superscripts

- * dimensionless quantities
- \- average quantities

INTRODUCTION

Pulsed liquid jets show considerable potential for demolition and excavation of hard, brittle targets such as concrete and rock (Cooley, 1974; Young, 1978). Pulsed jet devices very rapidly release energy that was stored during the time between pulses. A repetitive device can thus achieve a large ratio of instantaneous jet power to connected input power. Such devices offer the possibility of large pulse energy and jet stagnation pressure, and thus good breakage performance, without consuming exorbitant amounts of water and input power.

Many of the pulsed jet devices that have been reported in the literature have utilized the so-called "cumulation" nozzle (Cooley, 1974; Edney, 1976; Young, 1978; Watson, et. al., 1982). This type of nozzle can yield a very brief transient jet with a peak velocity as large as several thousand meters per second. An important feature is that the jet peak stagnation pressure can be much higher than the peak internal pressure anywhere within the device (Ryhming, 1973; Glenn, 1975). Experimental results using single-shot laboratory apparatus have been very encouraging (Cooley, 1974; Young, 1978); however, no successful repetitive-firing cumulation nozzle device has been reported.

A simple extrusion device can also be operated as a pulsed jet device. Such a device consists essentially of a cylinder into which a piston is pushed to extrude liquid through a conventional nozzle located in the cylinder head, as shown schematically in Figure 1. The liquid flow is essentially quasi-steady, so the chamber pressure is always equal to or greater than the jet stagnation pressure. Chamber wall strength considerations thus limit practical attainable jet velocity. For a given pulse energy, the extrusion jet typically exhibits lower peak jet velocity but larger jet mass than a cumulation nozzle jet.

An attractive feature of the extrusion device is that only proven (albeit rather sophisticated) technology is required to build a repetitively firing machine. A large driving force, and quick delivery of stored energy, can be furnished by means of a hydraulically cocked gas spring of the type used in some mechanical breakers. The device can be made repetitive by refilling the chamber through a check valve as the piston is retracted in preparation for the subsequent pulse. Since there need be no metal impact, noise level and ground shock problems can be avoided and machine durability enhanced.

A few such machines have been reported in the literature (Voytsekhovskiy, et. al., 1964; Leach and Walker, 1965; Gnirk and Grams, 1972; Mohaupt, et. al., 1978; Yie, et. al., 1978), but performance data is sparse. The current paper reports an investigation of the performance of an extrusion pulsed jet against concrete targets. An analysis is also presented which provides operational insight and design guidance.

ANALYTICAL MODEL

The analytical model of the extrusion pulsed jet is based on quasi-one-dimensional flow of an incompressible fluid. Both transient and viscous effects are accounted for, the latter approximately by means of a constant friction factor. The objective of the analysis is information about the motion of the piston and of the fluid.

Both the piston and the fluid are initially stationary. The motion of the piston can be described by means of the energy equation as

$$\int_{y=0}^{y=X} F(y)dy - \int_{y=0}^{y=X} P_p A_p dy = \frac{M_p}{2} V_p^{\,2}(X) \qquad (1)$$

where y is a dummy variable of integration and friction has been ignored.

For the incompressible fluid, from continuity, the velocity at any station y within the nozzle at any time is related to the piston velocity according to

$$V(y,t) = \frac{V_p(t)A_p}{A(y)} \qquad (2)$$

Viscous effects can be accounted for approximately using a friction factor according to

$$dH = \lambda \frac{V^2}{2}\frac{dy}{D}$$

Viscous effects will be ignored in the chamber and the jet. Viewing the nozzle as an infinite number of constant area ducts of infinitesimal length, the head (energy per unit fluid mass) loss across the nozzle at any instant of time is found by spatial integration to be, assuming a constant friction factor,

$$H = \frac{\lambda V^2{}_p Q}{2D_p}$$

Here we have defined the parameter

$$Q = \frac{1}{D_p}\int_{y=0}^{y=L_n} [\frac{D_p}{D(y)}]^5 dy \qquad (3)$$

which is a function only of the nozzle shape, size and area ratio. The total viscous energy loss as a function of piston location is then

$$\int_{y=0}^{y=X} H\rho A_p dy = \frac{\lambda Q \rho A_p}{2}\int_{y=0}^{y=X} V_p^{\,2} dy \qquad (4)$$

An energy equation can then be written for the fluid, accounting separately for the kinetic energy of the fluid in the chamber, nozzle and jet, as

$$\int_{y=0}^{y=X} P_p A_p dy = \frac{\rho A_p}{2}(L_c - X)V_p^2(X) + \frac{\rho A_p \theta}{2} V_p^2(X) + \frac{\rho A_p}{2} R^2 \int_{y=0}^{y=X} V_p^2 dy + \frac{\lambda Q \rho A_p}{2} \int_{y=0}^{y=X} V$$

where we have defined another parameter

$$\theta = \int_{y=0}^{y=L_n} \frac{A_p}{A(y)} dy \qquad (6)$$

that depends only on nozzle shape, size and area ratio. Equations (1) and (5) can be differentiated and combined, eliminating the pressure P. to obtain the differential equation

$$[\frac{M_p}{2} + \frac{\rho A_p}{2}(\theta + L_c - X)] \frac{d}{dX}(V_p^2) + \frac{\rho A_p}{2}[\lambda Q + R^2 - 1] V_p^2 = F \qquad (7)$$

This equation can be non-dimensionalized by using the following characteristic parameters:

Force:

$$\bar{F} = \frac{E}{S} + \frac{1}{S}\int_{y=0}^{y=S} F dy = average\ force$$

Distance: S

Velocity: $(E/\rho S A_p)^{\frac{1}{2}}$

Pressure: $\dfrac{\bar{F}}{A_p}$

The differential equation then becomes, in terms of dimensionless variables,

$$\frac{d}{dx^2}(V_p^*) + \frac{C_2}{(C_1 - x^*)} V_p^* = \frac{2F^*}{(C_1 - X^*)} \qquad (8)$$

were we have defined the following parameter groups:

$$C_1 = \frac{M_p}{\rho S A_p} + \frac{\theta}{S} + \frac{L_c}{S} \tag{9}$$

$$C_2 = \lambda Q + R^2 - 1 \tag{10}$$

The solutions readily shown to be

$$V_p^{*2} = (C_1 - X^*)^{C_2}[\int \frac{2F^*}{(C_1 - X^*)^{C_2} + 1} dX^* + constant \tag{11}$$

The chamber pressure can be evaluated using equation (1), which can be differentiated and non-dimensionlized to obtain

$$P_p^* = F^* - C_3 \frac{d}{dx^*}(V_p^{*2}) \tag{12}$$

$$C_3 = \frac{M_p}{2\rho S A_p} \tag{13}$$

The next step is to specify the functional form of the driving force F. The assumption that Pu^k=constant for an ideal gas spring leads to the conclusion that the driving force is of the functional form (X + constant)$^{-k}$. We will approximate this form to facilitate the integration of equation (11). One possible approximation to simply assume a constant driving force equal to the average driving force. However, for the parameter ranges of interest in the present study, the driving force of an idealized gas spring can vary significantly, as shown in Figure 2. A linear model, however, is a quite good approximation and is readily integrable in equation (11). Thus we choose

$$F^* = \begin{cases} f + bX^* & 0 \le x^* \le 1 \\ 0 & x^* > 1 \end{cases}$$

Where f and b are constants chosen to yield the specified pulse energy and an appropriate value for dF*/dX*.

Using the force model (14) and the initial condition $V_p^*(X^* = 0) = 0$, equation (11) yields a closed-form expression for $V_p^*(X^*)$ for $0 \le X^* \le 1$:

$$V_p^{*2} = \frac{2(f + bX^*)}{C_2} - \frac{2b(C_1 - X^*)}{C_2(C_2 - 1)} - C_4[\frac{C_1 - X^*}{C_1}]^{C_2} \tag{15}$$

$$C_4 = \frac{2f}{C_2} - \frac{2bC_1}{C_2(C_2 - 1)} \tag{16}$$

Using equation (15), the value of X* at which the maximum velocity occurs can be shown to be

$$V^*(V_{MAX}{}^*) = C_1[1-[\frac{-bC_1}{f(C_2-1)+bC_1}]^{\frac{1}{C_2-1}}] \qquad (17)$$

At X*=1 the driving force drops to zero but motion continues due to residual kinetic energy of the piston and fluid. That solution can be obtained from F*=0 in equation (11), using the value of $V_p{}^*$ at X*=1 from equation (15) as an initial condition:

$$V_p{}^{*2} = k(C_1 - X^*)^{C_2}\,(X^* > 1) \qquad (18)$$

$$k = \frac{2(f+b)}{(C_1-1)^{C_2}C_2} - \frac{2b}{C_2(C_2-1)(C_1-1)^{C_2-1}} - C_4C_1{}^{-C_2} \qquad (19)$$

The expression for the chamber pressure, equation (12), can now be written as

$$P^* = F^* - C_3[\frac{2g}{(C_2-1)} + \frac{C_2C_4}{C_1}(\frac{C_1-X^*}{C_1})^{C_2-1}] \qquad (20)$$

for $0 \le X^* \le 1$, and

$$P^* = C_2C_3k(C_1 - X^*)^{C_2-1} \qquad (21)$$

for X*>1.

The cumulative jet energy can be evaluated according to

$$E_j = \int_0^{X(t)} \frac{1}{2}(RV_p)^2 \rho A_p dx \qquad (22)$$

or, in dimensionless form

$$E_j{}^* = \frac{E_j}{E} = \frac{R^2}{2}\int_0^{X^*} V_p{}^{*2} dy^* \qquad (23)$$

This integral can be evaluated either in closed form or numerically. Elapsed time is also of interest and can be evaluated according to

$$t(X) = \int_0^X \frac{dy}{V_p} \qquad (24)$$

or, in dimensionless form

$$t^*(X^*) = \int_0^{X^*} \frac{dy^*}{V_p^*} \qquad (25)$$

where time has been nondimensionalized by the parameter

$$[\frac{S^3 \rho A_p}{E}]^{\frac{1}{2}}$$

The elapsed time can be evaluated numerically more easily than in closed form.

ANALYTICAL RESULTS

The purpose of the analysis is to provide detailed understanding of the operation of a pulsed extrusion device. The analytical results are also very useful for designing the experimental apparatus. A typical design case of interest in the current project is as follows.

CASE I

E = 1000 ft-lbf = 1360 joules
M_p = 66.25 lbm = 30 kg
ρ=62.4 lbm/ft^3=1.0gm/cc
L_c= 2.5 in = 6.35 cm
L_n= 2.0 in = 5.08 cm
S= 0.793 in = 2.01 cm
D_p= 0.513 in = 1.30 cm
A_p= 0.207 in^2 = 1.34 cm^2
D_j= 0.0198 in = 0.5 mm
R= 670
λ= 0.02

The nozzle for this described by ($0 \leq Z \leq L_n$)

$$D(Z) = D_p esp[\frac{-Z \ln R}{2L_n}$$

and the fluid is water.

The piston mass is large to simulate the driver mass of a commercially available hydraulically-cocked gas spring. Analytical results for the chamber pressure at the piston face, the jet exit velocity, cumulative jet energy and elapsed time are shown in Figures 3 through 6. Inviscid results (λ = 0) for Case I are also presented in these figures. Quasi-steady flow has been achieved when the pressure history "tracks" the driving force per unit piston area (F/A_p). Figure 3 shows that quasi-steady flow is achieved for Case I at about X*=0.1.

Thus about the first 10% of the driver power stroke is required to accelerate the piston and water to quasi-steady velocity. A less massive piston would cause quasi-steady conditions to be achieved more quickly. Note the rapid decay of chamber pressure beginning at $X^{*}= 1$, which is the result of the small residual piston kinetic energy when the driving force drops to zero. Viscosity has a minimal effect on the chamber pressure, the most significant effect being attainment of quasi-steady conditions slightly more quickly. Jet exit velocity results are presented in Figure 4. The upper curve, labeled $\lambda =$ O. presents the inviscid flow results. The lower curve presents results which include an approximate model for viscous effects in the high speed flow through the nozzle. The effect of viscosity for this case is a reduction in jet exit velocity of about 5 6%. The consequence of this is that about 10% of the input pulse energy is dissipated by viscous effects, as is shown by the jet cumulative kinetic energy results presented in Figure 5. The elapsed time, presented in Figure 6, is about 14 milliseconds. The calculated effect of viscosity on elapsed time is an increase of about 6 % at $X^{*}= 1$.

Results of changing the nozzle area ratio are shown in Figure 7. All parameter values are the same as Case I except the following:

$S =$ 0.625 in = 1.59 cm
D_p = 0.591 in = 1.50 cm
A_p = 0.274 in^2 = 1.77 cm^2
D_j = O.020 or 0.060 in = 0.51 or 1.52 mm
R = 872 or 97

The results of Figure 7 illustrate that decreasing the nozzle area ratio, for example by increasing the nozzle exit diameter for a given piston diameter, delays achievement of quasi-steady conditions. In fact, for the R = 97 case, quasi-steady flow was not achieved before all of the fluid had been expended. Such a design is undesirable because the jet velocity is too low during much of the jet duration to effectively break strong targets. Also, the piston velocity, which is equal to V_j/R (see equation 2), increases significantly (e.g. ~7 m/sec vice ~1 m/sec), which may be deleterious to high pressure seal life.

The above results lead one to conclude that a well-designed extrusion-type pulsed jet device will operate under quasi-steady flow conditions for most of the power stroke. Thus some valuable insight can be obtained by examining the less complicated steady flow relation. If the entire stroke were to occur as steady flow, a valid energy relation would be (note V_j = constant)

$$E_j = \frac{\rho S A_p}{2} V_j^2$$

which can be rearranged to obtain

$$D_p = \sqrt{\frac{8E_j}{\pi\rho S}} \Big/ V_j$$

For a given driver (i.e. fixed E and S) and assuming pulse energy E ≈ Ej (which is tantamount to ignoring viscosity), we are left with

$$V_j = \frac{constant}{D_p} \qquad (26)$$

This means that the steady state jet velocity is independent of the nozzle area ratio. Equation (26) could also be obtained directly from the fact that chamber pressure is equal to jet stagnation pressure for steady inviscid flow. The duration of the stroke is given by, for steady flow,

$$t = \frac{S}{V_p} = \frac{SR}{V_j} = \frac{S}{V_j}[\frac{D_p}{D_j}]^2$$

Thus we see that the nozzle area ratio does affect jet duration.

Figure 8 shows calculated results for parameter values identical to Case I except for pulse energy E = 792 ft-lb = 1070 joules, S = 0.7 in = 17.8 mm, and as noted on the graph. These results show that equation (26) is indeed approximately valid for the quasi-steady flow for large values of the area ratio. We conclude that for quasi-steady flow and for a given driver, jet velocity depends more strongly upon piston diameter than upon nozzle area ratio. A ramification is that, for the experimental apparatus, jet velocity cannot be effectively varied by changing only the nozzle exit diameter.

EXPERIMENTAL APPARATUS

The pulsed extrusion apparatus was powered by the hydraulically-cocked gas spring of a boom-mounted mechanical breaker. A steady push, rather than an impact, was obtained by maintaining contact between the extrusion device plunger and the breaker ram throughout the power stroke. The values of the breaker pulse energy and power stroke length, required as input to follows ram was measured during the power stroke by means of a conventional accelerometer. Multiplying the acceleration by the mass of the breaker ram yielded a value for the net force exerted by the gas spring. Pencil lead breakwires wired into simple voltage divider circuits were used to obtain the values of elapsed time at two values of ram displacement. These data were sufficient to numerically integrate the measured a(t) curve twice to obtain V(t) and X(t). Using mean data values from ten repetitions of the experiment, the F(X) curve shown in Figure 9 was obtained by crossplotting. The area under this curve, 792 ft-lb (1070 joules), is the measured value of the pulse energy. The measured curve shows considerable departure from the ideal curve of Figure 2. For use in the design calculations the measured curve was approximated as shown in Figure 9, which conforms to the measured pulse energy and to the functional form of equation (14) with f = 1.231 and b = -0.462. The power stroke length was taken to be 0.7 in (17.8 mm) for purposes of the design calculations.

Many design cases, including those shown in Figure 8, were investigated. Design considerations included the size of suitable commercially available components such as seals and nozzles. The primary design objective was a jet exit velocity on the order of 3000 ft/sec (900 m/sec). A summary of the selected design calculation values of important parameters is listed below.

$E = 792$ ft-lb_F = 1070 joules
$M = 66.25$ lb_M = 30.0 kg (which includes the mass of the breaker ram)
$\rho = 62.4$ lbm/ft^3 = l gm/cc (water)
$L_c = 2.5$ in = 6.35 cm
$L_n = 2$ in = 5.08 cm
$S = 0.7$ in = 1.78 cm
$D_p = 0.534$ in = 1.36 cm
$A_p = 0.224$ in^2 = 1.45 cm^2
$D_j = 0.024$ in = 0.61 mm
$R = 495$
$\lambda = 0.02$

Results of the algorithm calculations for this case are shown in Figure 10. The calculated peak jet velocity is about 3050 ft/sec (930 m/sec), and the peak plunger velocity is thus about 6 ft/sec (2 m/sec). The peak chamber pressure is predicted to be about 70,000 psi (4.8 k-bar).

A schematic representation of the hardware design is initially in the retracted position and is in contact with the retracted plunger of the extrusion device. At this time water under pressure of a few bar is entering the chamber through the check valve and exiting through the nozzle. As the hammer starts its downward stroke the chamber pressure increases and the check valve at the water inlet closes, extruding the water through the nozzle. The device was operated only as a single-shot device, the objective being measurement of breakage performance under controlled conditions. However, the addition of a spring under the head of the plunger would serve to maintain the plunger in contact with the ram and thus enable the device to produce repetitive pulsed jets.

The extrusion device was designed to withstand the stall pressure, i.e. the pressure that would occur if the nozzle became plugged. The water chamber was constructed by pressing the sleeve into the body with a very heavy interference fit. This puts the inner surface of the chamber in an initial state of compression which results in an acceptable value of maximum tensile stress during operation. Although it was originally desired to provide for both a water inlet and a chamber pressure sensing port, initial calculations showed that piercing the body wall with a hole for either of these resulted in an unacceptable stress concentration. A two plunger design was adopted, one plunger with a water inlet and one plunger with a pressure transducer. When the plunger with the pressure transducer was used, the chamber had to be filled with water by hand before each shot. The nozzle used was a commercially available unit made of tungsten carbide. Tungsten carbide is very strong in compression and relatively weak in tension. The loading in this instance was internal pressure of a magnitude large enough to cause tensile failure. Again the part was put into an initial state of compression by pressing it into the nozzle holder. This provided an acceptable final tensile state during operation. The plunger seal was a commercial seal that was rated as marginally suitable for the pressure and surface speed.

Figure 12 shows the various parts of the extrusion device and Figure 13 shows the device assembled and ready to be mounted in the mechanical breaker. The large cylinder with three legs is a mounting bracket that allows the extrusion device to be mounted inside the mechanical breaker. The three legs are adjustable to allow variation of the standoff distance, i.e. the distance between the nozzle exit plane and the target surface. Figure 14 shows a typical test set-up, with the extrusion device mated to the boommounted mechanical breaker.

The target material for the jet breakage volume experiments was 10,000 psi (680 bar) ultimate compressive strength concrete. Target slabs were approximately 3 feet (l meter) square by 8 inches (20 cm) thick.

RESULTS

Chamber pressure measurements were made occasionally throughout the experiment; a typical pressure history is shown in Figure 15. The chamber pressure records were generally quite consistent throughout the test, which indicates that test conditions were reasonably uniform. The average peak pressure from the pressure records was 61,000 psi (4.2 k-bar); total variation of measured peak pressure was within $\pm$ 10% of this average value.

Jet velocity was not measured during the experiment, but can be estimated from the measured pressure data. The analytical results of Figure 10 provide a useful relation between chamber pressure and jet exit velocity; this relationship is quite straightforward for the quasi-steady portion of the flow. The analytical algorithm teaches that quasi-steady flow has been achieved when the pressure curve "tracks" the driving force curve, as shown in Figure 3. The analytical results for the chosen hardware configuration (Figures 9 and 10) thus show achievement of quasi-steady flow for this configuration at about $X^* = X/S = 0.2$ based on the idealized gas spring force model. The peak pressure (Figure 15) occurs much later in the stroke, and thus during quasi-steady flow. A peak chamber pressure of 61,000 psi (4.2 kbar) thus implies, from Figure 10, a peak jet velocity of about 2900 ft/sec (880 m/sec).

Breakage data for this pulsed extrusion device against 10,000 psi (680 bar) ultimate compressive strength concrete was measured for a total of 48 individual pulses. The breakage was of the type often referred to as "full face", as differentiated from breakage to a free edge. Typical target damage was a cylindrical hole about 0.08 in (2 mm) in diameter and about 1/4 in (6 mm) deep, with some degree of surface spalling around the hole. Crater volume was measured by weighing the amount of fine dry sand required to fill the crater. Detailed data for both crater volume and depth are presented in Table l. Three of the craters, one in each data set, were atypical in that a large shallow surface spall occurred, resulting in a shallow crater of comparatively large volume. These data were included in the statistical treatment; the resulting mean values, with and without the atypical data, are shown near the bottom of Table l. These data show only a moderate change in breakage performance for decreased standoff. It is also apparent that the check valve plunger yielded poorer breakage, possibly because of leakage past the check valve during extrusion. The overall mean crater volume, excluding the three atypical data, was 0.01 in (0.16 cc). The specific energy for face breakage was thus determined to be 79,000 ft-lb/in^3 (6700 joules/cc) at a nominal pulse energy of 792 ft-lb (1070 joules). At the same pulse energy, the semiempirical full face breakage model advanced by Grantmyre and Hawkes, 1975, for mechanical impact breakers indicates a specific energy of about 1300 ft-lb/in^3 (10^6 joules/cc) against rock. Yie, 1978, quotes specific energy for pneumatic hammers as 3000 to 19,000 ft-lb/in^3 (260 to 1600 joules/cc). This agrees with the Granymyre-Hawkes model at pulse energies typical of hand-held "jackhammer" type breakers, that is, on the order of 10 to 100 ft-lbs (13-130 joules). We thus conclude that the pulsed extrusion jet is at least 4 times less efficient in rock breaking than even the very smallest weight and pulse and pulse energy. It must be noted that the additional variable of target material is nested in these data. It is unlikely that this effect would be large enough to change the performance ratio significantly.

Shortly after the data of Table 1 had been obtained, the device ceased to operate properly. In particular, the check valve plunger no longer moved freely in the packing gland and was quite badly scored and galled. The remainder of the hardware, particularly the nozzle, barrel, 15-5 PH stainless steel transducer-plunger, and the seal, showed no evidence of damage.

CONCLUSIONS

The extrusion pulsed jet device used in the current investigation produced a peak jet velocity of about 2900 ft/sec (880 m/sec). The measured pulse energy was 792 ft-lb (1070 joules) and peak chamber pressure was about 61,000 psi (4.2 k-bar). The full face breakage performance of this device does not compare favorably with that of mechanical breakers of similar size, weight and pulse energy.

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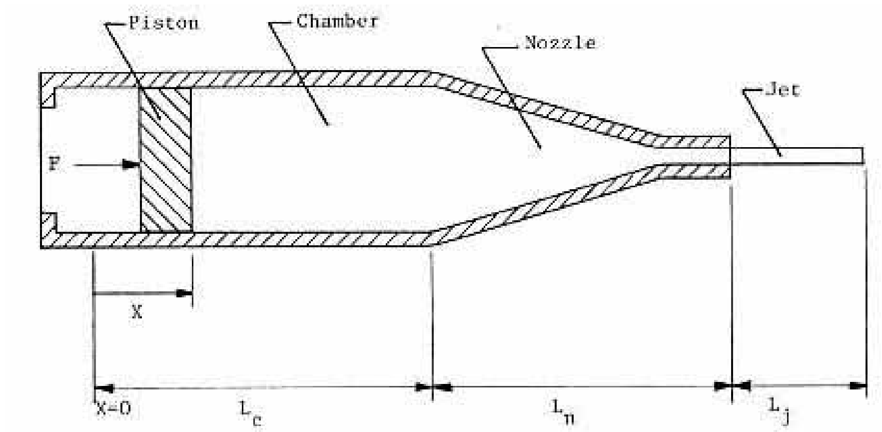

Figure 1. Extrusion Schematic.

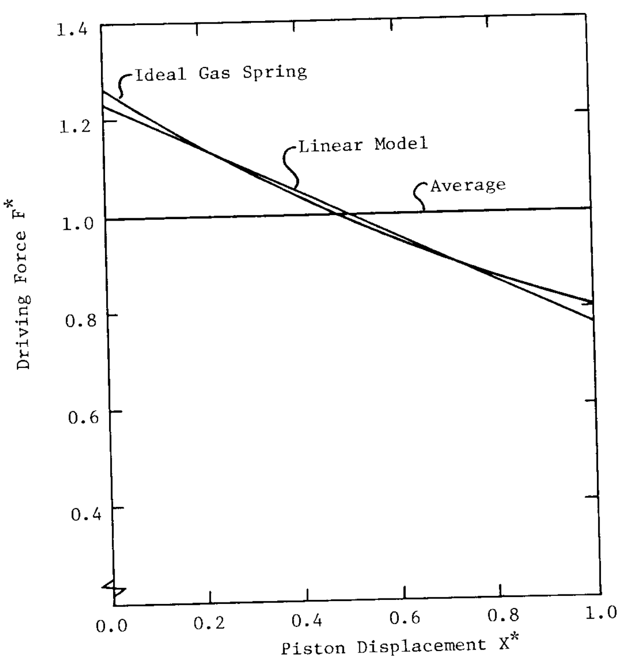

Figure 2. Driving Force for an ideal gas spring.

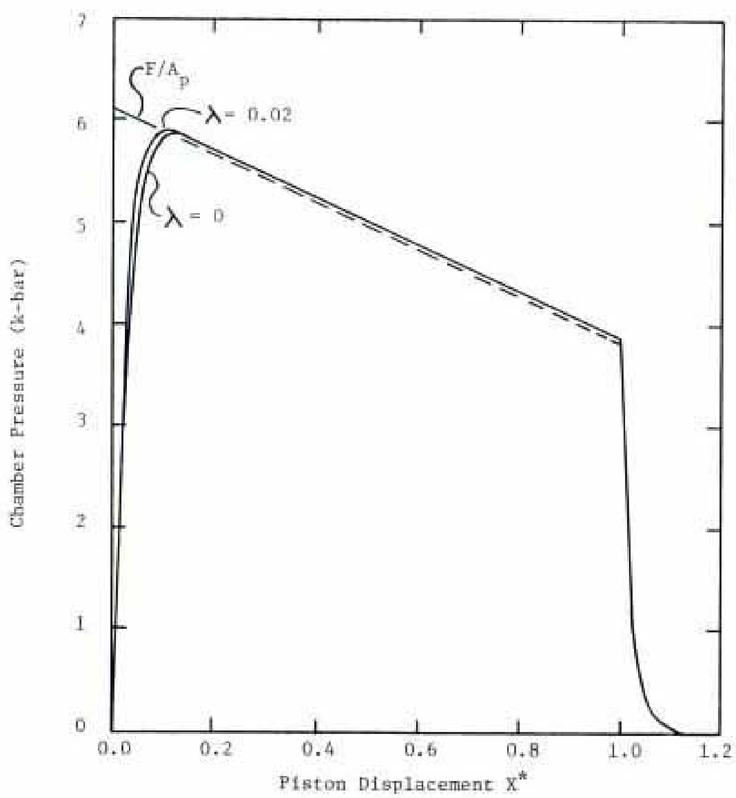

Figure 3. Chamber pressure at piston face.

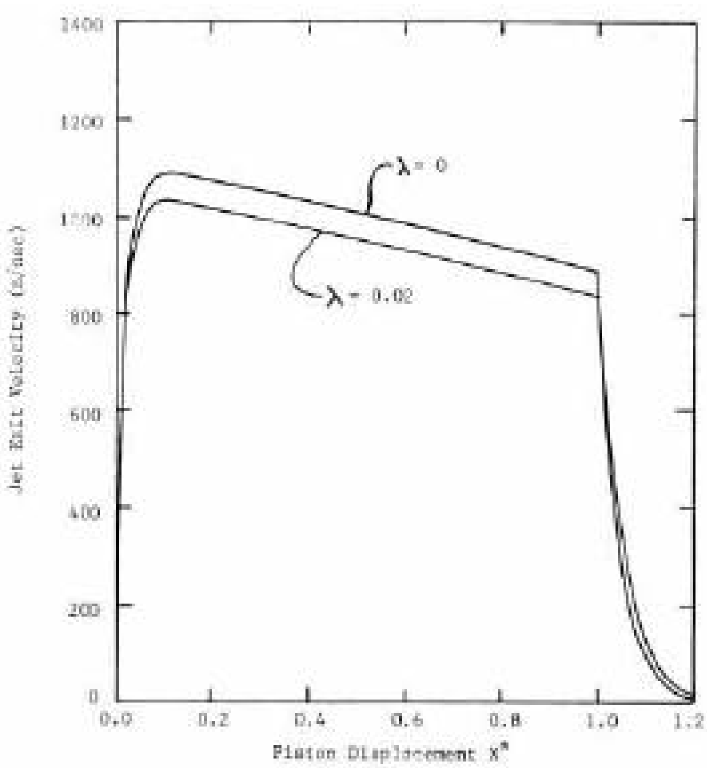

Figure 4. Exit jet velocity.

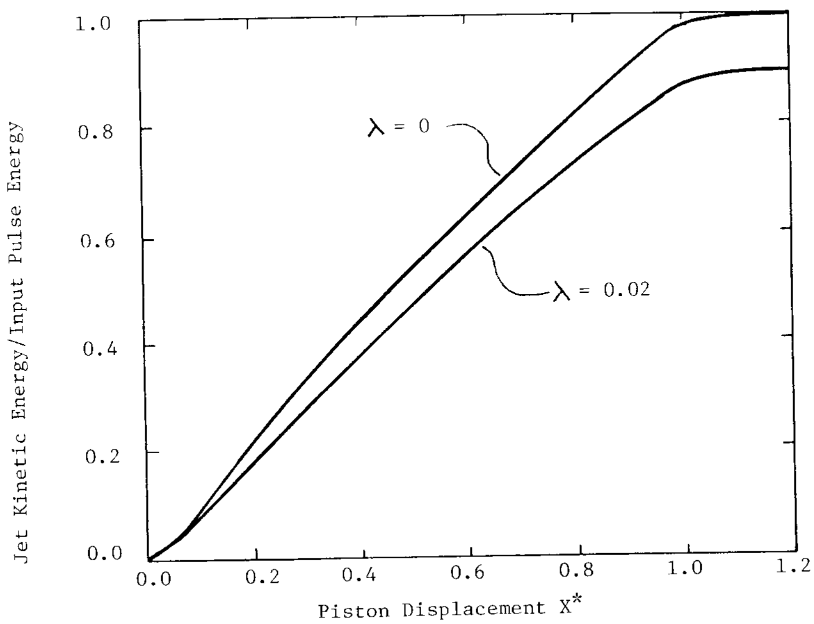

Figure 5. Jet cummulative kinetic energy

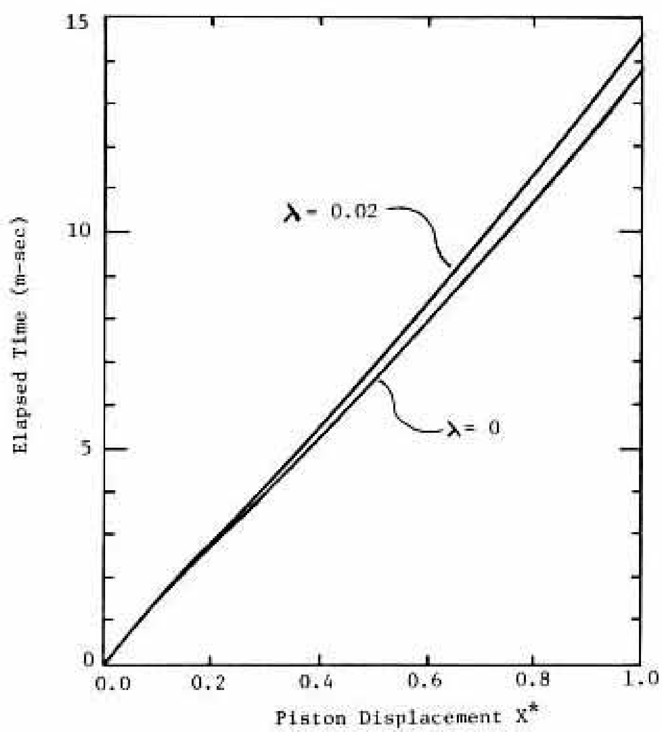

Figure 6. Elapsed time.

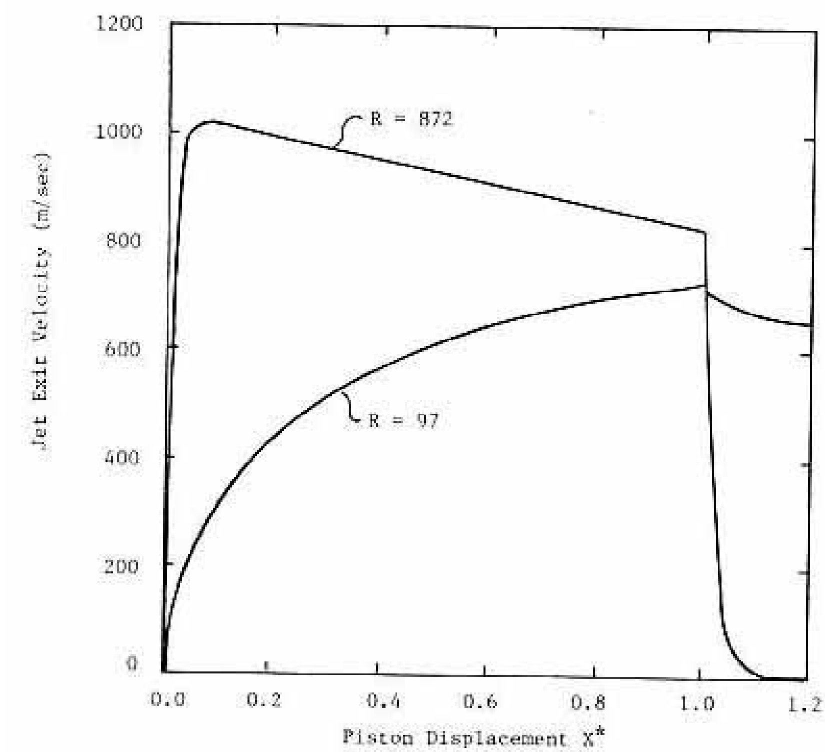

Figure 7. Effect of nozzle area ratio.

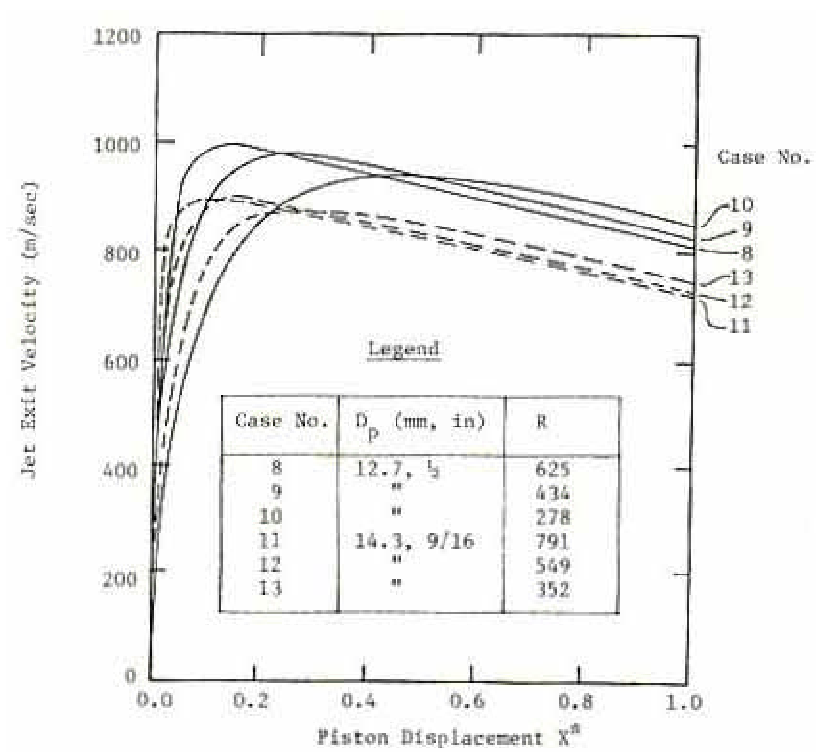

Figure 8. Effect of piston diameter and area ratio on jet velocity.

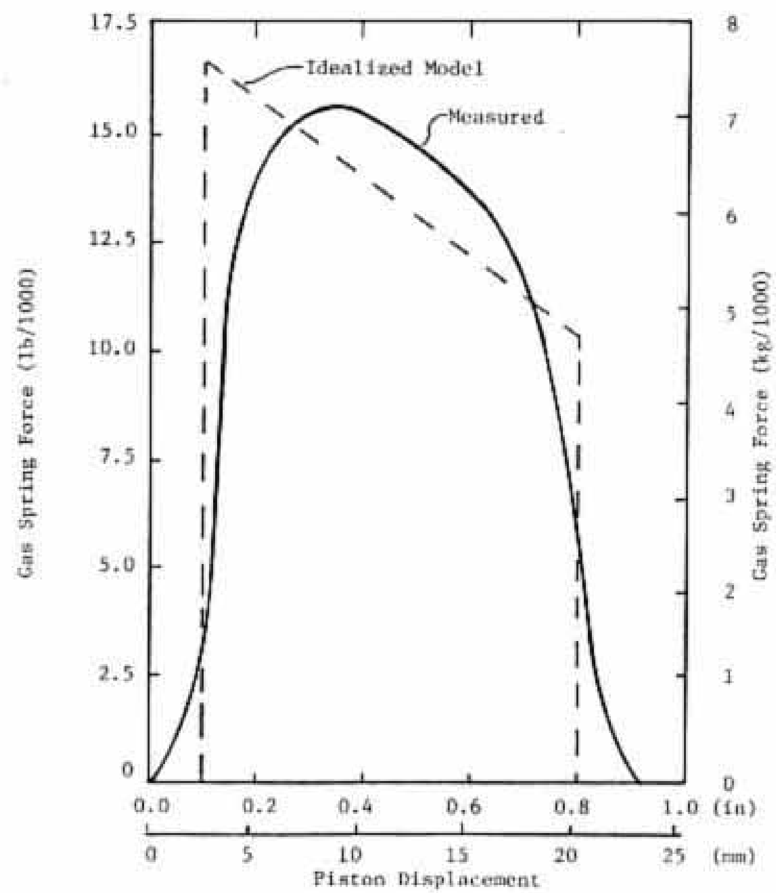

Figure 9. Gas spring force.

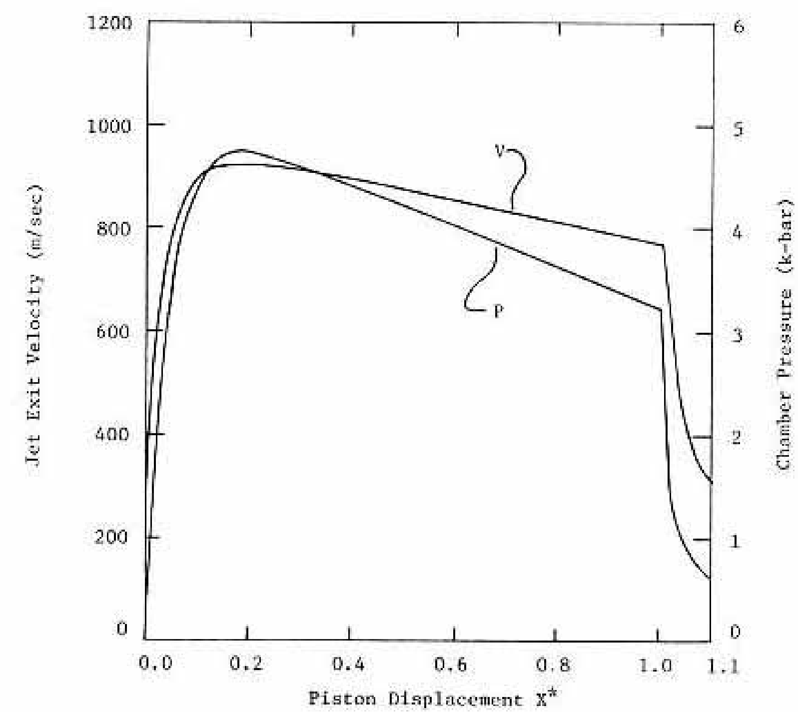

Figure 10. Calculated results for the selected design configuration

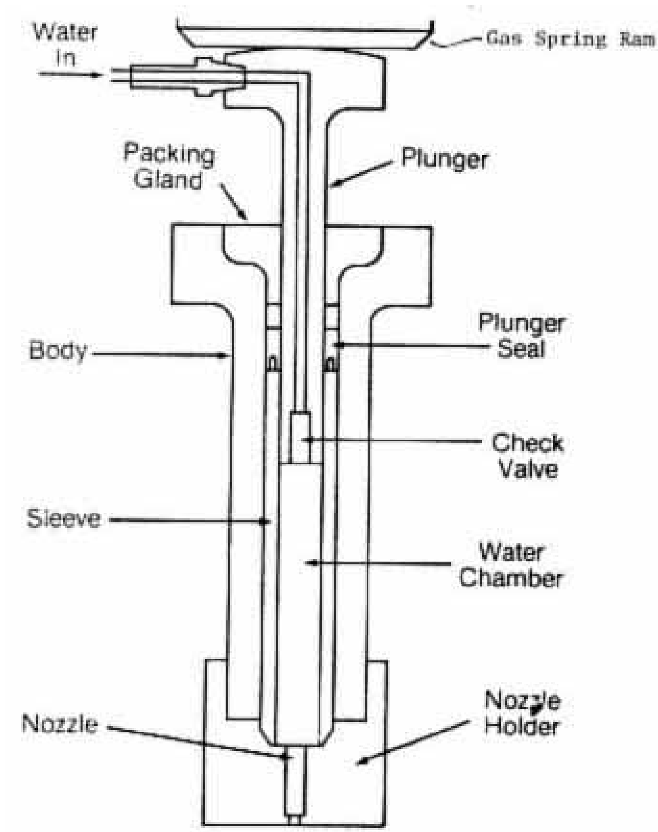

Figure 11. Extrusion device schematic

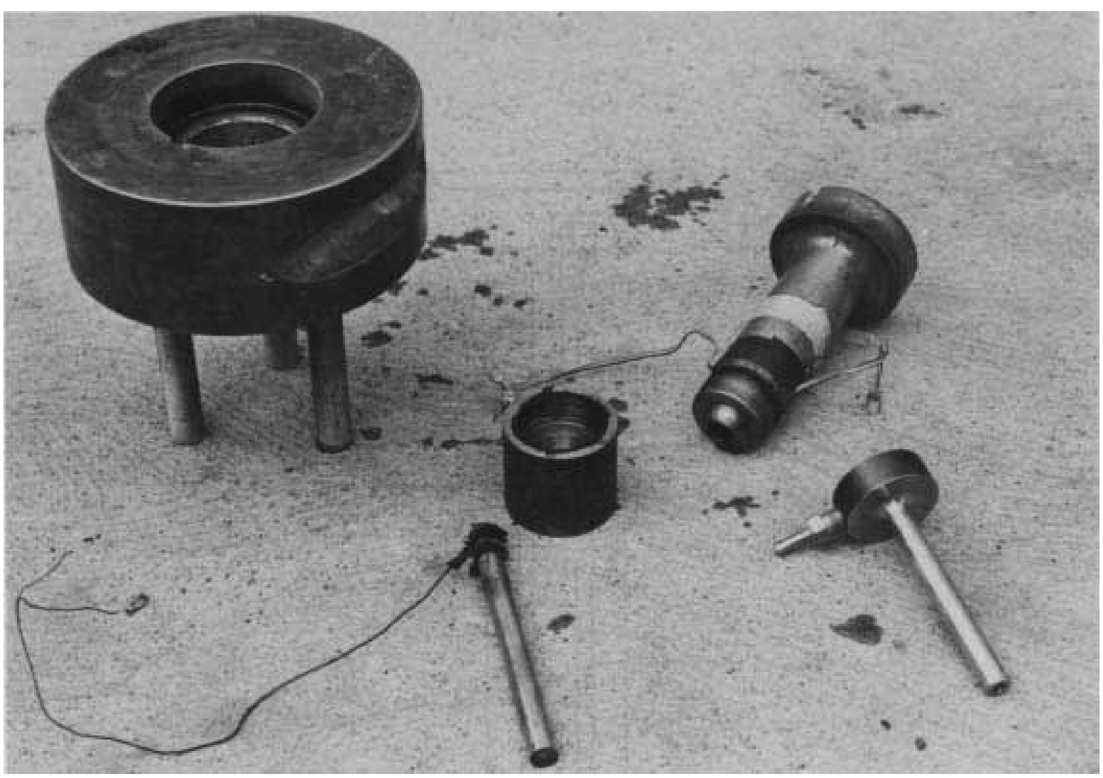
Figure 12. Extrusion Device Assembled

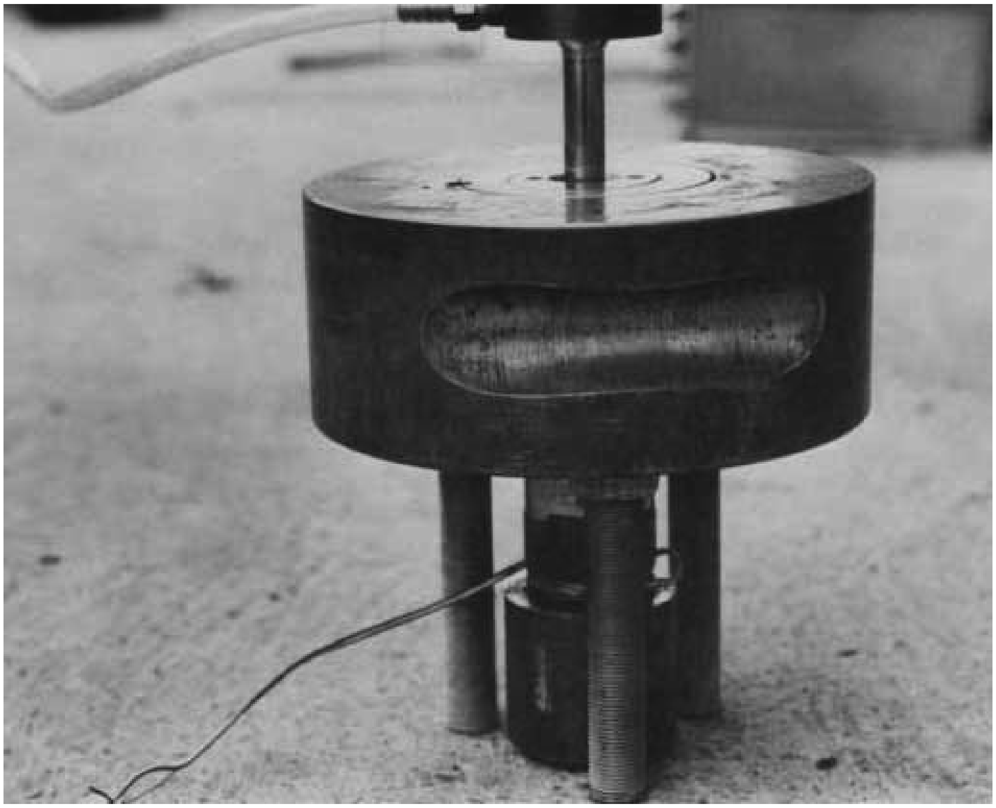
Figure 13.   Extrusion Device Assembled

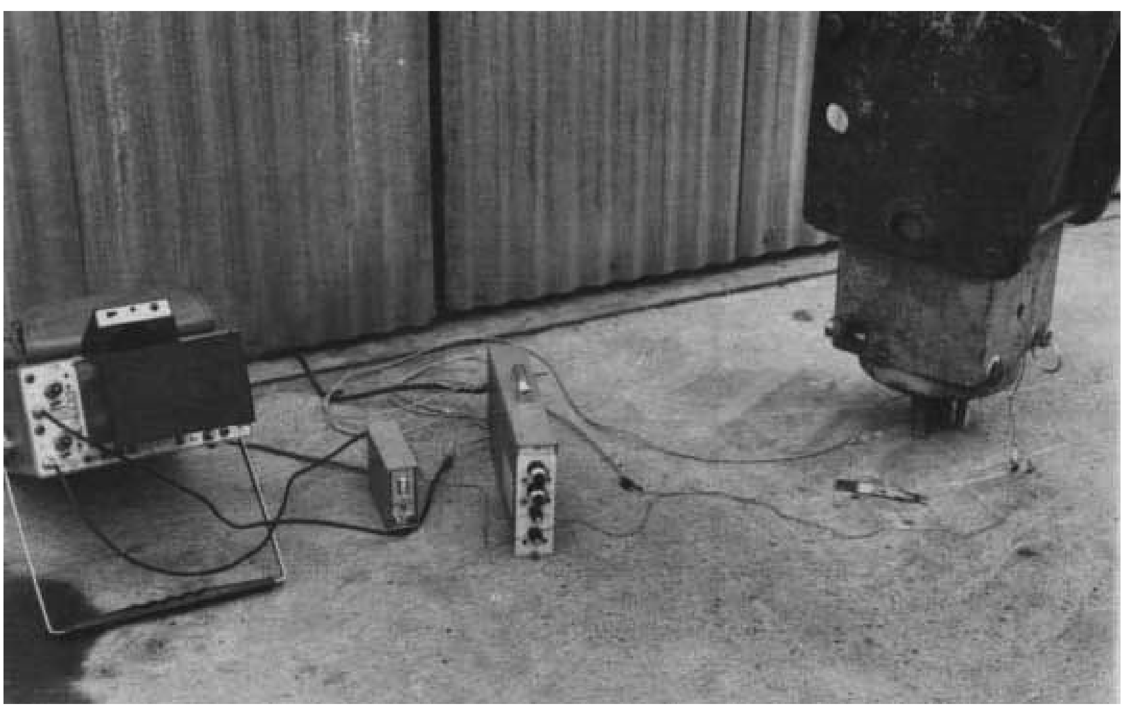
Figure 14. Test Set-up

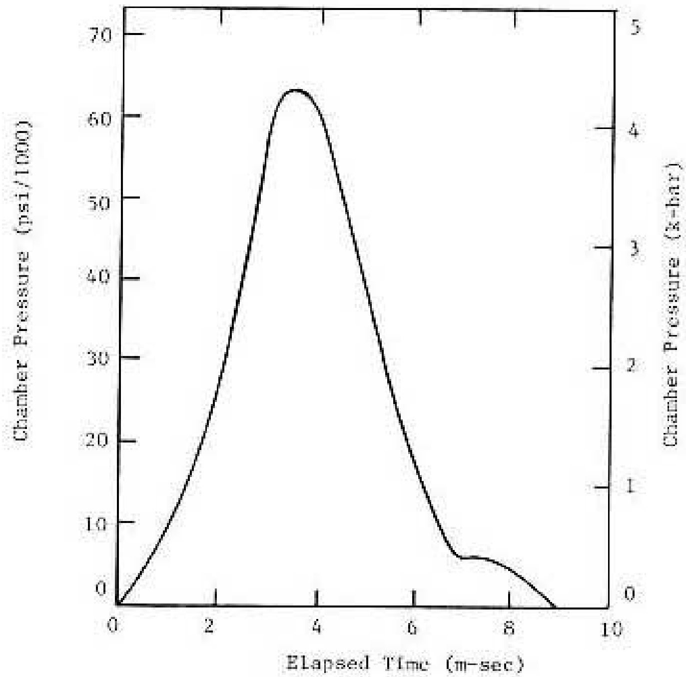

Figure 15. Typical chamber pressure data record.

Transducer Plunger 12.7 mm (½") Standoff		Transducer Plunger, 3.2 mm (1/8") Standoff		Check Valve Plunger, 3.2 mm (1/8") Standoff	
Volume(cc)	Depth(mm)	Volume(cc)	Depth(mm)	Volume(cc)	Depth(mm)
2.67	9.4	0.305	8.6	0.137	5.8
0.066	6.4	0.104	9.6	0.104	7.4
0.032	5.3	0.341	6.1	0.881	6.9
0.145	5.8	2.36	6.4	0.075	4.3
0.109	8.4	0.054	5.1	0.047	2.8
0.151	4.8	0.188	7.6	0.032	3.8
0.088	5.1	0.105	6.6	0.086	5.1
0.174	4.3	0.134	6.4	0.030	4.6
0.118	6.6	0.096	7.6	0.033	3.3
0.043	5.3	0.074	6.4	0.078	5.1
0.154	7.9	0.165	6.6	0.022	3.8
0.355	5.6	0.706	8.9		
0.230	8.1	0.042	5.6		
0.248	5.1	0.584	8.9		
0.433	5.6	0.420	9.1		
0.316	5.1	0.103	7.9		
		0.065	7.9		
		0.242	9.9		
		0.080	9.9		
		0.070	9.1		
		0.194	10.9		
0.334	6.2	0.306	7.9	0.139	4.8
0.178		0.203		0.064	

Table 1. Craters in high-strength concrete.

DISCUSSION

NAME: W. C. Cooley
COMPANY: Terraspace, Inc.

QUESTION: "I would like to comment that comparison of pulsed jet performance should take into account the length-to-diameter ratio of the slug of water, the number of jet pulses to one point, the pressure of a predrilled hole, and similar variables, in addition to pressure and energy of pulse, before drawing broad generalized conclusion regarding the specific energy for pulsed jets relative to mechanical breakage."

ANSWER: We, of course, agree that an oversimplified and/or overgeneralized presentation of performance data can be misleading. The conclusions of our paper are stated only for single-pulse face breakage by the extrusion pulsed jet device which we tested. We did not intend to state or imply any conclusion regarding the efficacy of pulsed jets in general.

NAME: Curtis Steele
COMPANY: Steele's

QUESTION: "What kept the pressure from extruding back through the seal instead of through the material, and what type of seal is used?"

ANSWER: The exit end of the nozzle was positioned a short distance (the "stand-off" distance) from the target. The jet thus travels through the atmosphere before it impacts the target material. The water in the chamber was prevented from escaping through the piston clearance by means of a commercially available high pressure reciprocating-type seal manufactured by the Bal Seal Engineering Company.

LABORATORY INVESTIGATION OF SOIL CUTTING WITH A WATER JET

by

Dimitrios K. Atmatzidis
Professor, Department of Civil Enqineerinq,
University of Patras, Greece

and

Frederick R. Ferrin
Captain. U.S. Army Corps of Engineers

ABSTRACT

Limited information is available regarding the effect that soil properties have on the penetration or excavation efficiency of high speed water jets. It may be inferred from research on materials such as coal and rock that material properties which could have an effect on the cutting of soils should include grain size, density, degree of saturation, permeability, and strength.

A continuous water jet with a driving pressure range up to 8,000 psi and a nozzle diameter of 1 mm was used to conduct a total of seventy six cutting tests on soil samples which were compacted in the laboratory. Four soils, ranging from a clean sand to a predominantly clayey soil, were used to provide an adequate variation of soil properties. The depth of penetration of the jet into the soil target was measured as a function of the time of exposure, the degree of saturation of the soils, the dry density of the soils, and the traversing velocity of the jet over the soil target

It is observed that the depth of penetration in a given soil is related exponentially to the corresponding time from initial impact and to the corresponding traversing velocity of the jet, and varies linearly with the coefficient of permeability of the soil. The degree of saturation and the dry density of the soil affect the depth of jet penetration in a soil since they affect the strength and the permeability of the soil. Increasing soil strength and decreasing permeability result in decreasing depth of penetration, Finally, the volume of soil affected by the action of the water let is larger than the actual hole or slot excavated and increases with increasing soil grain size.

INTRODUCTION

Liquid jets have been used for more than a century to excavate and remove large quantities of soil, utilizing rather low pressures and large volumes of water. It was not until recently that soil cutting with a water jet received more attention and was considered for application in specific field projects (Yahiro, Yoshida, and Nishi, 1974; Summers and Zakin, 1975; Shibazaki and Ohta, 1982). A wealth of information is available in the literature with respect to the design of a water jet nozzle, the factors affecting the structure, coherence, and impact pressure of a jet, and the methods to improve the efficiency of a jet. In contrast, extremely limited information is available with respect to soil properties which may have an effect on the soil cutting efficiency of water jets. However, research on the depth of jet penetration in permeable targets indicates that penetration is primarily affected by the strength and permeability characteristics of the material which are, in turn, affected by numerous other material properties such as grain size, porosity, density, and degree of saturation.

Accordingly, the investigation reported herein is focused toward the evaluation of the effects that specific soil properties have on the excavation efficiency of a water jet. Adequate fulfillment of this goal and development of the necessary information for use in the many, diverse field applications of soil cutting with a water jet, requires the performance of an extensive investigation which is beyond the scope of this work. It is intended that the results of this research serve only as a first step towards the comprehensive investigation of soil cutting with a high speed liquid jet.

EXPERIMENTAL PROCEDURES

The following restrictions were placed upon this experimental investigation

(a) a water jet emerging in ambient air was used;
(a) all variables associated with the water jet remained constant throughout the testing program so that any observed variations in soil cutting efficiency could be attributed to variations of soil properties;
(a) as an exception to the above restriction, the effect of nozzle pressure was investigated and tests were conducted with a stationary and a traversing jet; and
(a) four soil types ranging from a clean sand to a predominantly clayey soil, were used to provide an adequate variation of soil properties

The Water Jet System

The water jet was generated by a hydraulically driven. double acting Petrodyne Model 1780 intensifier. Water was supplied to the intensifier at a rate of 3 gpm and pressurized air was supplied at 60 psi. The jet pressure was registered by a gauge mounted between the nozzle and the flow junction which receives the pressurized water from the two chambers of the intensifier. Water was delivered to the nozzle at a pressure of up to 8,000 psi which remained constant throughout the stroke of the intensifier piston. However, at the termination of the stroke and during reversal of the piston there was a momentary fluctuation in jet pressure

A tungsten carbide nozzle with a 90^{o} internal angle of taper followed by a straight section with a length of 1.5 mm and a 1.0 mm orifice diameter was utilized. This was a readily available and economically feasible selection and was preferred since the objective of this investigation was to evaluate the effects of soil properties and not jet characteristics on the cutability of soils. Admittedly, best results can be obtained with a nozzle which has a 13^{o} internal angle of taper followed by a straight section with a length of about 2 5 nozzle diameters (Leach and Walker, 1966).

Regular tap water with no additives was used as the cutting fluid. The standoff distance (distance between the nozzle and the target surface) was held constant and equal to 150 times the nozzle diameter (six inches). This value was selected in order to achieve maximum impact pressures (Leach and Walker, 1966; Shavlovsky, 1972). The nozzle was fixed on a frame and provided a jet stream with a vertical, downward direction. An industrial lathe was utilized to provide support for the sail samples and to traverse them under the jet at a predetermined, constant velocity of up to 1.0 inch/second which was the maximum speed of the lathe.

Materials Tested

Four different soils with a substantial variation in engineering properties and behavior were subjected to the jet cutting experiments. The grain size distributions of these soils are shown in Figure 1.

Soil 1 is a readily available clean sand (Ottawa sand); soil 2 is a mixture by weight of 85% locally available sand (torpedo sand) and 15% natural silty soil (Vicksburg loess); soil 3 is a naturally occurring silty soil (Vicksburg loess); and soil 4 is a mixture by weight of 50% naturaliv occurring clayey soil (Chicaqo area clay) and 50% commercially available kaolinite. According to the Unified Soil Classification System these materials are classified as SP, SW, ML, and CL, respectively.

Sample Preparation and Testing Procedures

The following factors were taken into consideration for the selection of an adequate sample container:

(a) the container should be impervious as soil samples in a saturated and partially saturated state would be tested;
(a) the container should be large enough to avoid any adverse interference from the container walls while the sample was being exposed to the jet;
(a) the total weight of container and sample should be small enough to allow its convenient handlinq and transportation from an area of preparation to the location of the water jet system; and
(a) the container should be able to be placed upon the lathe which was utilized to traverse the samples.

Therefore, various lengths of six inch inside diameter PVC (plastic) pipe were chosen as appropriate sample containers This sample container also lended itself very nicely to the compaction of samples with a manually operated drop hammer.

To prepare a sample, dry pulverized soil was weighed and mixed with a predetermined amount of water to attain the desired degree of saturation. Frequent checks of moisture content were made to insure adequate control. Then a specific weight of the moist soil was placed in the container and was compacted (by vibration or using a drop hammer) to a predetermined layer thickness in order to attain the desired dry density. Enough layers were thus compacted to obtain the desired length of sample. Deviations from this procedure were necessary onlv when preparing samples of Soil 1. In this case, a sample of the desired length and dry density was first formed and was then saturated by allowing water to flow up through the sample by means of a valve at the base of the container; to obtain a partially saturated sample, some of the water was allowed to drain through the bottom of the container.

Soils 1 and 2 were placed in the containers in two inch lifts and between each lift was placed a four inch square paper marker which was numbered. The thickness of each layer was checked in order to insure adequate density control. The paper markers were used with these cohesionless soils since the holes or slots excavated by the water jet had a tendency to collapse. At the end of each test the depth of jet penetration could be established to within one inch by gradually emptying the container and inspecting the markers for signs of penetration by the jet. The diameter of the hole excavated was estimated as the average of the diameter of the holes present on the markers penetrated by the water jet. This method was not necessary for the cohesive soils 3 and 4 since the hole or slot excavated by the jet remained open for direct measurement of the depth of penetration and the diameter of the hole using a set of steel rods.

Variables Investigated

Available information on the cutting of rock using high speed water jets suggests that, other than grain size, perhaps the most important material properties which control the depth of cut are the void ratio or porosity, the permeability, and the strength of the target material. To provide control over the same properties for the soils tested during the course of this investigation, tests were conducted while carefully monitoring and varying the water content (or degree of saturation) and the dry unit weight of each soil.

The penetration of a material target by a water jet is not an instantaneous phenomenon; for stationary jet and target a certain amount of time is required to achieve the maximum depth of penetration which cannot be significantly increased by continued application of the water jet. Accordingly, the first series of tests for each soil type was conducted by maintaining constant all the other variables and allowing the time of application of the jet to vary so that a lower limit was obtained for the necessary time of application of the jet on each soil type to achieve maximum penetration.

Having established the necessary time for application of the water jet and selecting as a convenient jet pressure for conducting the tests the pressure of 1.000 psi two series of tests were conducted for each soil type with stationary jet and target. In one series only the degree of saturation was varied from as low as l0% to 15% (dry for soils 1 and 2) to almost 100% while dry density was maintained at a constant value. In the other series only the dry density was varied; moisture content remained constant. Finally, a series of tests was conducted for each soil type by maintaining constant all the other variables and allowing the target to traverse under the jet at velocities ranging up to 1.0 inch/ second. Thus, a total of seventy-six soil cutting tests were conducted.

RESULTS AND DISCUSSION

The effectiveness of a water jet in cutting a given material is usually evaluated in terms of the depth of jet penetration into the target material or in terms of energy required to excavate a unit volume of the target material (specific energy). For the purpose of this investigation, jet efficiency was assumed to be represented by the depth of penetration into the various soil targets, and appropriate graphs were prepared showing the variation in the depth of penetration as a function of jet duration, soil dry density, degree of soil saturation, and traversing velocity. The results obtained and the observations made during the experimental phase of this investigation are presented and discussed next.

Penetration as a Function of Time

For the purpose of this investigation, all samples of each of the soils tested were required to be prepared at the same dry density and degree of saturation. Due to difficulties in compacting fully saturated samples of the cohesive soils, all tests were conducted on partially saturated samples. Accordingly, the dry density had values of 102 lb/ft^3, 130 lb/ft^3, 109 lb/ft^3, and 102 lb/ft^3 for Soil 1,2,3 and 4, respectively; the degree of saturation had values of 50%, 53%,75%, and 62% for Soils 1,2,3 and 4,respectively.

When a continuous water jet impinges normally on a static target of homogeneous material, it begins to penetrate the target at a finite rate. As the jet penetrates into the target material, the impact pressure that it applies on a freshly exposed surface is reduced due to (a) the increasing distance from the nozzle orifice and (b) the turbulence generated by the counter flow of outqoing slurry. Accordingly, the rate of penetration should decline with time until a limiting penetration is eventually achieved. This trend was observed for all four soil types tested under a fixed driving pressure of 1,000 psi, and the results obtained are shown in Figure 2. For the specific materials tested, and for all practical purposes, an exposure time of between fifteen and twenty seconds would be sufficient to generate a maximum depth of penetration with a fixed driving pressure of 1,000 psi. It should be noted ,however, that the rate of penetration and the maximum depth of penetration may vary substantially with changes in the properties and/or composition of a specific soil, and such trends are discussed later in this presentation.

Although there is some question about the initial impact and its effects on initial rate of penetration, under idealized conditions penetration could be related exponentially to time by an expression of the general form proposed by Mellor (1972) and Sundaram and Liu (1978):

$$h = h_{\max}(1 - e^{-\frac{t}{\tau}}) \qquad (1)$$

where h is the penetration at time t after initial impact, h_{max} is the limiting penetration for infinite time, and τ is a constant (with units of time) which depends on jet and material properties. If h_{max} and τ are considered simply as curve fitting parameters, they can be determined from experimental data. Accordingly, the available data regarding depth of penetration as a function of time were fitted on the basis of Eouation 1 and the results are shown in Figure 2. It can be observed that, as a first approximation, the exponential formulation adequately correlates the time from initial impact and the corresponding depth of penetration.

<u>Penetration as a Function of Traversing Velocity</u>

The depth of penetration should decrease with increasing traversing velocity since the duration of jet application per unit length traversed is inversely proportional to the traversing velocity. Experiments were conducted on samples having the same material properties as the samples used to study jet penetration as a function of time and the results are shown in Figure 3. It can be observed that (a) the rate of decrease in penetration is high at the lower range of values of the traversing velocity and is gradually reduced as the traversing velocity increases, and (b) for traversing velocities larger than about 0.2 in/sec, the rate of change in the depth of penetration is higher for soils 1 and 2 (cohesionless) than for soils 3 and 4 (cohesive).

According to Mellor (1972) the depth of penetration could be related to the traversing velocity by a relationship of the form:

$$h = h_{\max}(1 - e^{-\frac{V^*}{V_T}}) \qquad (2)$$

where h is the depth of penetration for a traversing velocity V_T, h_{max} is the limiting penetration for traversing velocity approaching zero and can be set equal to the limiting penetration for infinite time and stationary jet, and V* is a constant (with units of velocity) which depends on jet and material properties. The available data were fitted according to Equation 2 and the results are shown in Figure 3. It can be observed that, as a first approximation, this formulation correlates the depth of penetration and the traversing velocity.

Sundaram and Liu (1978) proposed the following relationship which has the general form of Equation 2:

$$h = h_{\max}(1 - e^{-\frac{d}{V_T \tau}}) \qquad (3)$$

where τ is the time constant introduced in Equation 1, and d is the diameter of a zone of influence over which the "damage" mechanism of the water jet is applied. By assuming that the diameter of the influence zone, d, is equal to the diameter of a hole generated during a stationary mode of jet operation, Equation 3 was successfully used by Sundaram and Liu (1978) to predict the depth of penetration in coal for various traversing velocities. This same approach was used to predict the depth of penetration for the soils tested during this investigation; the depth of maximum penetration, h_{max}, and the time constant, τ , were obtained according to Equation l and had the values shown in Table l ; the diameter of the zone of influence was assumed to be equal to an average hole diameter observed during stationary tests on the same materials. The results are summarized in Table 1 and indicate poor predictive capabilities for Equation 3 when the forementioned assumption is made.

The failure of this method to predict the depth of penetration at various traversing velocities can be explained in terms of the assumption regarding the extent of a "zone of influence". An examination of Equations 2 and 3 indicates that, for the two expressions to be equivalent, the following relationship should hold true:

$$\frac{V^*}{V_T} = \frac{d}{V_T \tau} \qquad (4)$$

or

$$d = \tau V^* \qquad (5)$$

Since values for the time constant, τ, and thevelocity constant, V*, have been already estimated, the diameter, d, of an apparent zone of influence can be computed for each of the soils tested. These values are summarized in Table 2 and indicate that (a) the diameter of the zone of influence is substantially larger than the diameter of the hole generated by the water jet in any of the soils tested, and (b) the zone of influence becomes larger as effective qrain size increases

These observations indicate that, for soils, an impinging water jet affects a volume which is substantially larqer than the lateral extent of the jet stream and/or the hole that it generates. As the jet stream advances into the soil target, water is forced to permeate laterally into and through the soil voids, thus changing the degree of saturation of partially saturated soils. Furthermore, excess pore pressures can be generated in a large volume of the soil if continuity of pore water exists. Accordingly, the resistance of a soil to excavation by the water jet may be adversely affected .

This effect can be illustrated by the results shown in Figure 4 which were obtained by exposing dry and partially saturated Ottawa sand (Soil 1) to the action of the traversing water jet. It can be observed that

(a) at a stationary mode of operation, the depth of penetration in dry sand was only 44% of that in the partial ly saturated sand,
(a) the depth of penetration in the dry sand increased with traversing velocities up to about 0.2 in/sec and approached that of the partially saturated sand, and
(a) for larger traversing velocities the rate of change in the depth of penetration is similar for both sands but the actual penetration is larger for the partially saturated material .

This behavior can be explained by assuming that, at low traversing velocities, a front of partial saturation was advancing laterally through the dry sand at a speed larger than the traversing velocity. At higher traversing velocities, the relative movement between jet and target was faster than the rate at which the saturation front could advance.

Effect of Dry Density

The variation of the depth of penetration in each of the soils tested with changes in the dry density is depicted in Figure 5. Similar variations would be obtained if the depth of penetration was plotted versus void ratio or porosity. Variation of the dry density of Soil 1 (clean sand) resulted only in minimal differences for the observed depth of penetration which are within the experimental error inherent in the measurement technique used. A more pronounced effect was obtained for the predominantly sandy Soil 2. Changes in dry density had a dramatic effect on the depth of penetration in the two fine grained soils. For Soil 3 (silty soil), the depth of penetration was reduced from 30 inches to 9 inches by varying the dry density from 75 lb/cu.ft to 110 lb/cu.ft. Similarly for Soil 4 (clayey soil), the depth of penetration was reduced from 20 inches to 3 inches as the dry density was varied from 87 lb/cu.ft to 110 lb/cu.ft.

In general, it can be stated that

(a) the effect of changes in dry density on the depth of penetration is a function of soil type and becomes more pronounced as a representative grain size of the material decreases;
(a) the rate of change in the depth of penetration with changes in dry density is very small (if not negligible) for the clean sand but becomes very significant for the fine grained (silty and clayey) soils.

It should be noted that each soil type was tested at a constant water content and, consequently, at a variable degree of saturation (range of values are shown in Figure 5). Since the degree of saturation affects to a certain extent the permeability and strength of a soil and, thus, could affect the depth of penetration of a water jet into the soil the results obtained during this phase of the investigation may have been affected, to some extent, by the variations in degree of saturation.

Effect of Degree of Saturation

The results shown in Figure 6 were obtained by testing each soil at a constant dry density (or void ratio) and varying the degree of saturation over a range as wide as possible. It can be observed that variations of the degree of saturation similar to those involved in the evaluation of the effect of dry density result in changes of the depth of penetration which are negligible for the sandy soils (Soils 1 and 2) and relatively small for the silty and clayey soils (Soils 3 and 4). Accordingly, the variations shown in Figure 5 are considered realistic.

The results shown in Figure 6 indicate that the variation of the depth of penetration with large changes in the degree of saturation follows two, practically opposite, trends according to the soil type tested. For the clean sand (Soil 1) the maximum penetration was obtained at a degree of saturation of about 50%.; penetration decreased with either increasing or decreasing degree of saturation and reached minimum values for dry and fully saturated conditions. For the other three materials maximum penetration was achieved at very high degrees of saturation (about 95%); minimum penetration was reached as the decree of saturation was reduced to about 40% to 50%; further reduction in the degree of saturation resulted in increasing penetration. Finally, the chance in penetration as a function of degree of saturation was strikingly similar for both fine grained soils (Soils 3 and 4)

Effect of Permeability

The permeability of a soil depends on the characteristics of both the permeant and the soil, and various relationships have been developed which reflect the influence of these characteristics. Since the properties of the water used during this investigation can be assumed, for all practical purposes, to have remained constant, variations in the permeability of the soils tested can be attributed to changes in soil characteristics. In general, the soil characteristics which influence permeability are particle size, void ratio, composition, fabric, and degree of saturation (Lambe and Whitman, 1969). However, for each particular soil, particle size and mineralogical composition remained constant; variation is soil fabric were not evaluated and will be neglected in this discussion. Accordingly, any changes in permeability can be attributed to variations in void ratio and degree of saturation.

Various theoretical formulations and experimental observations have suggested that a slot of the coefficient of permeability versus some function of the void $e^3/[1+e]$, $e^2/[1+e]$. or e^2) should be a straight line (Lambe and Whitman, 1969). Although no information exists on the effects of permeability on the cutting of soils with a water jet, extensive investigations on other porous materials (such as rock and coal) indicate that the material permeability is one of the most critical factors affecting the depth of jet penetration (Nikonov and Goldin, 1972;Crow, Lade, and Hurlburt, 1974; Cooley, 1974). If soil permeability and depth of penetration are strongly correlated there should be strong correlations regarding the depth of penetration and the parameters which affect soil permeability.

The square of the void ratio was computed for each soil sample used to study the effect of dry density and was compared to the corresponding depth of penetration as shown in Figure 7. A linear correlation was obtained for each soil type tested.

Based on this information the following observations can be advanced:
(a) the depth of jet penetration appears to vary linearly with soil permeability;
(a) permeability changes have a negligible effect on the depth of penetration in the clean sand, (Soil 1),
(a) the effect of variations in permeability on the depth of penetration become more pronounced as the grain size of the material decreases.

Effect of Strength

The unconfined compressive strength of the target material has been recognized as having a great influence on the resistance of the target to penetration by a water jet and is routinely used to correlate cutting data (Cooley, 1974). However, it should be recognized that clean coarse-grained soils, such as Soil 1, exhibit no cohesion and, therefore, no unconfined compressive strength. On the other hand, soils which contain a substantial fraction of clay minerals exhibit true cohesion and their unconfined compressive strength may correlate with the depth of water jet penetration .

The strength of partially saturated soils is controlled by the effective stress within the soil. However, it is difficult to apply the effective stress principle to partially saturated soils because the relation between total and effective stresses involves the pressures in both the liquid and gas phases and a factor which is related to the degree of saturation. The best way to estimate strength is to conduct tests that duplicate the field conditions as closely as possible (Lambe and Whitman, 1969).

Soil 1 (Ottawa sand) used for this investigation derived its strength purely from inter-granular friction, and capillary tension provided practically no apparent cohesion. Due to the small depth of the soil layers tested and the lack of any overburden pressure or load at the surface of the samples, the normal stresses on any plane in the mass of the samples were very small and the corresponding shear strength can be considered negligible. This observation indicates that the response of this sand to the cutting action of the water jet should be practically independent of its shear strength and was primarily affected by permeation phenomena, and this conclusion is substantiated by the data shown in Figure 5.

Soil 2 (sand-silt mixture) exhibited very small true or apparent cohesion. Attempts to conduct unconfined compression tests on this material at different degrees of saturation were either unsuccessful of yielded small values (at a Y_{dry} of 130 lbs/cu.ft and a water content of 6% q_u was only 1000 lbs/sq.ft.). The information shown in Figures 5 and 6 indicates that strength increase due to densification or partial saturation could have contributed in the reduction of the depth of penetration. It should be noted, however that this behavior could have been primarily due to changes in permeability. A limited number of unconfined compression tests were conducted on samples of the fine grained Soils 3 and 4 and the resulting strength values are reported in Figures 5 and 6. It can be observed that the unconfined compressive strength of these soils was drastically increased by either densification or manipulation of the degree of saturation. Although the relationship between depth of penetration and unconfined compressive strength is not linear, it can be observed that the smallest depth of penetration in either soil corresponds to the conditions of highest unconfined compressive strength (highest dry density or a degree of saturation between 30% and 60%).

CONCLUSIONS

On the basis of the observations and discussions presented herein and within the limitations of this experimental investigation, the following conclusions can be advanced:

1. For all practical purposes, the time required for the jet to approach maximum penetration into the soil targets is on the order of tens of seconds and is less for granular, cohesionless materials than for cohesive soils; the depth of penetration appears to be exponentially related to the corresponding time from initial impact.
1. An exponential formulation appears to adequately correlate the depth of penetration with the corresponding traversing velocity; as the traversing velocity increases, the depth of jet penetration decreases.
1. The volume of soil affected by the action of the water jet is larger than the actual hole of slot excavated and increases with increasing soil grain size; this "zone of influence" could represent the volume of soil into which water permeates and/or excess pore pressures are generated under the action of the water jet.
1. Increasing dry density at a constant water content results in decreasing depth of jet penetration in a soil target; this can be attributed to the corresponding increase in strength and decrease in permeability of the soil. This effect is negligible for the clean sand tested but becomes very Pronounced for the fine grained soils.
1. The degree of saturation affects the depth of jet penetration in a soil; with the exception of the clean sand, maximum penetration is achieved at practically full saturation and minimum penetration at a degree of saturation of about 40% to 50%.
1. The depth of penetration in a given soil appears to vary linearly with the coefficient of permeability of the soil.
1. Increasing soil strength results in decreasing depth of penetration ;the unconfined compressive strength can be used as a strength index for fine grained cohesive soils.

ACKNOWLEDGMENTS

The experimental investigation reported herein was conducted in the laboratories of Northwestern University, Department of Civil Engineering. The Hayward Baker Company provided the water jet apparatus. The U.S. Army Corps of Engineers supported the graduate studies of F. R. Ferrin.

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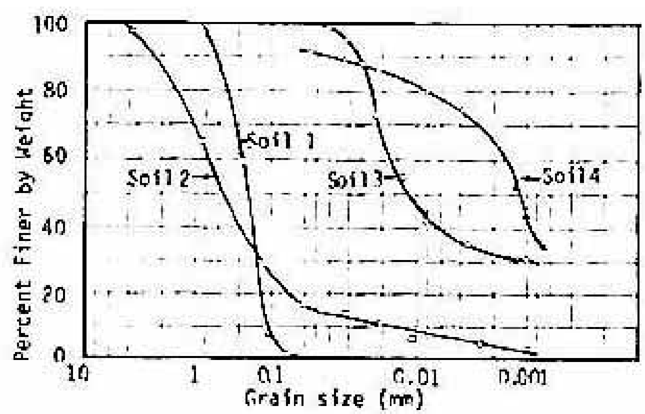

Figure 1. Grain Size Distribution Curves of the Soils Tested.

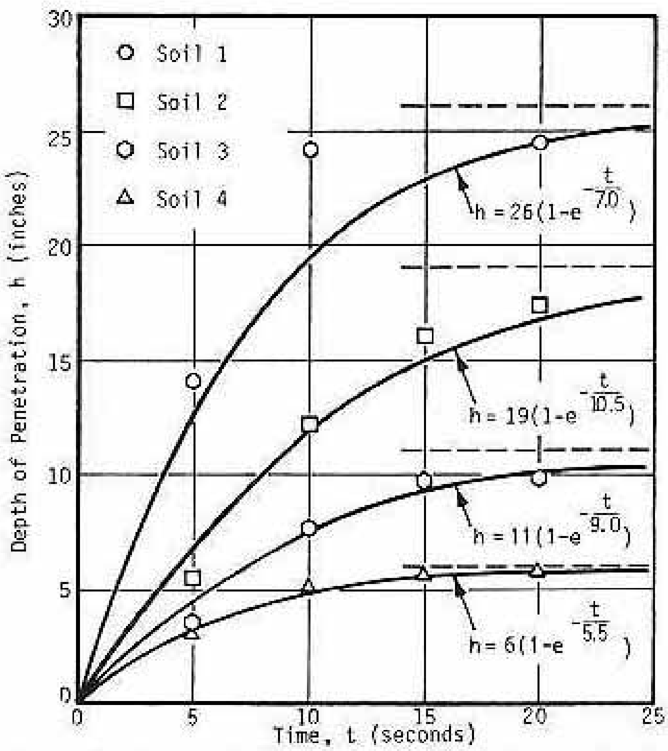

Figure 2. Idealized Relationships between Depth of Penetration and Time from Initial Impact

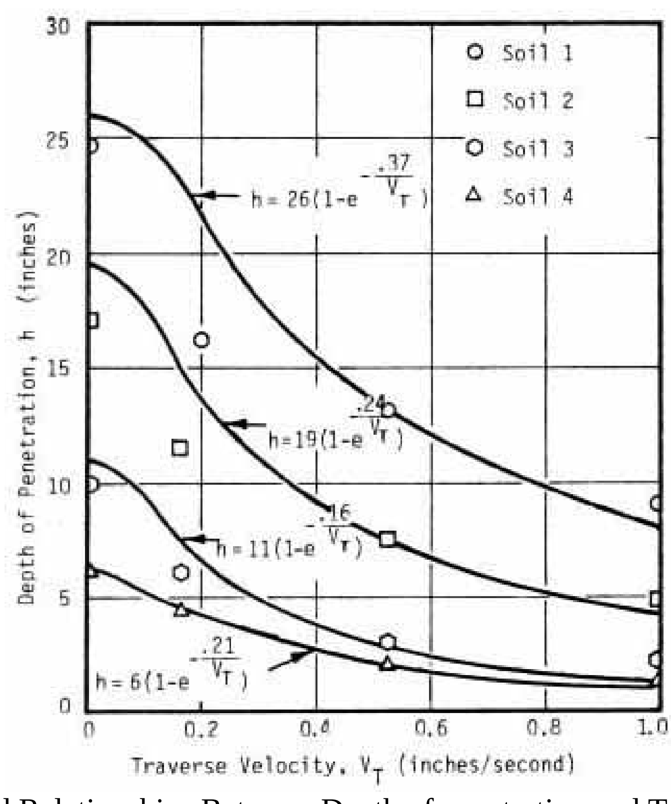

Figure 3. Idealized Relationships Between Depth of penetration and Traversing Velocity

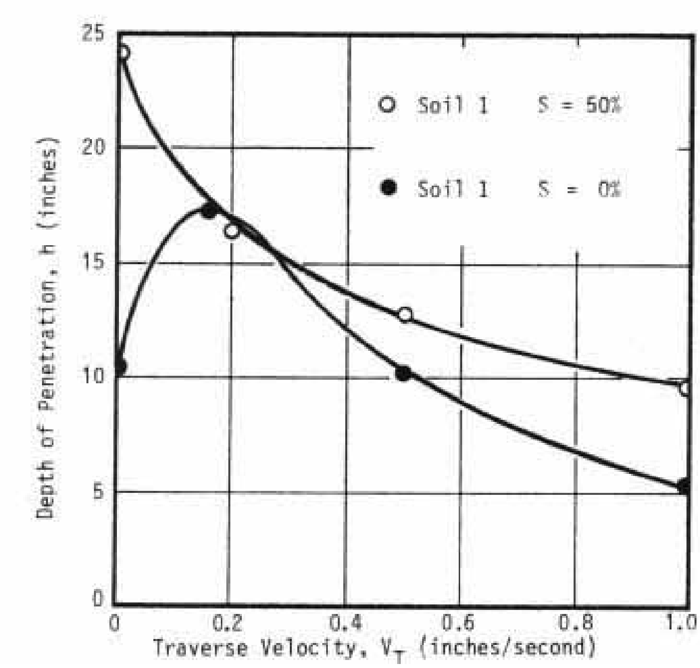

Figure 4. Effect of Saturation on the Depth of Penetration at Various Traversing Velocities

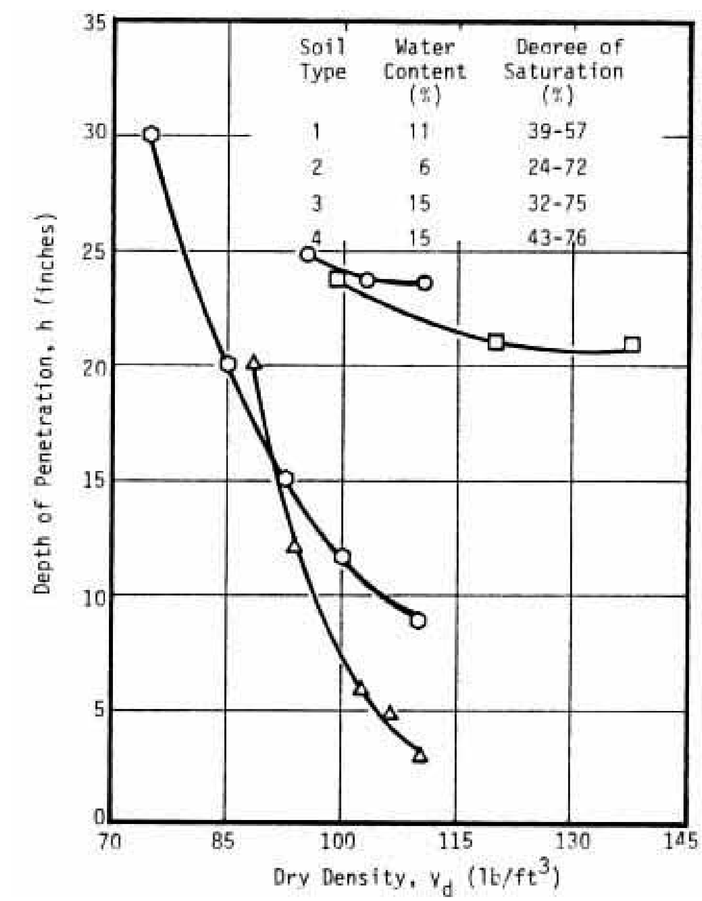

Figure 5. Depth of Penetration as a Function of Dry Density

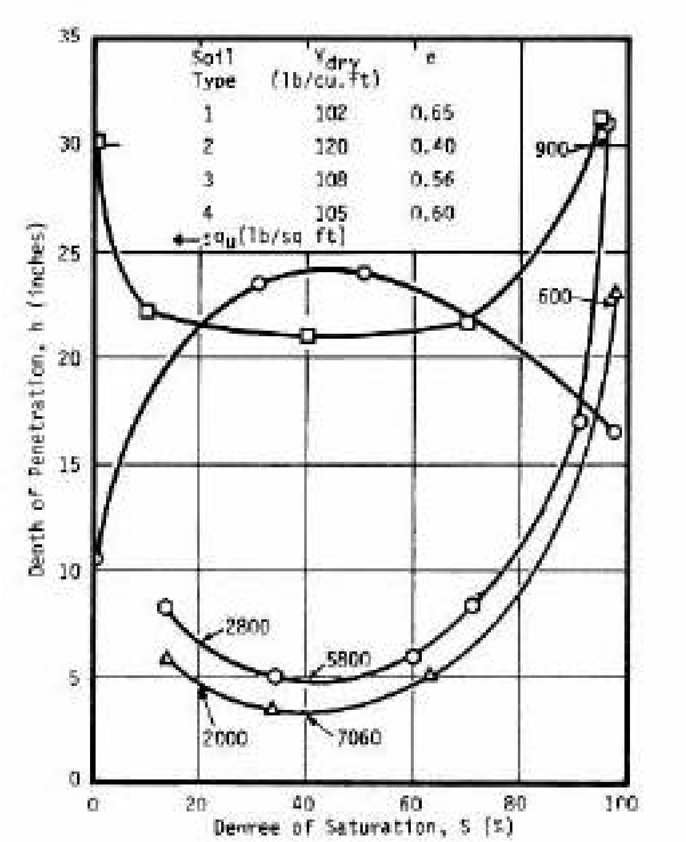

Figure 6. Penetration as a Function of Saturation

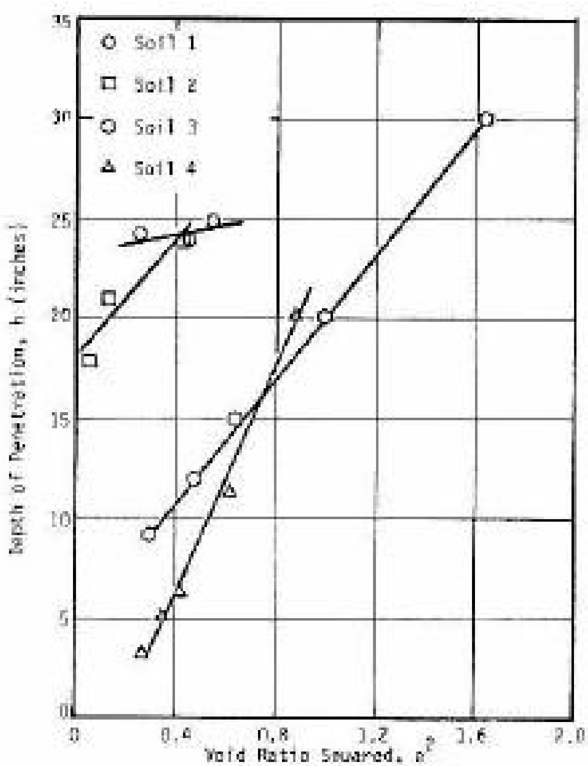

Figure 7. Depth of Penetration as a Function of Void Ratio

Soil Type and Parameters	Traversing Velocity (in/sec)	Penetration (in) Observed	Penetration (in) Predicted	Percent Under Predicted
Soil 1	.22	16.0	9.5	40
h_{max} = 26" d = 0.7"	.50	13.0	4.7	64
τ = 7.0 sec.	1.0	9.0	2.5	72
Soil 2	.17	11.5	5.4	53
h_{max} = 19" d = 0.6"	.50	7.5	2.0	73
τ = 10.5 sec.	1.0	4.5	1.1	75
Soil 3	.17	6.0	4.3	28
h_{max} = 11" d = 0.75"	.50	3.0	1.7	43
τ = 9.0 sec.	1.0	2.0	0.9	55
Soil 4	.17	4.3	3.0	30
h_{max} = 6" d = 0.65"	.50	2.0	1.2	40
τ = 5.6 sec.	1.0	1.3	0.7	46

Table 1. Predicted and Observed Depths of Penetration At Various Traverse Speeds

Soil	Time Constant (sec)	Velocity Constant (in/sec)	Zone of Influence Diameter (inches) Assumed *	Zone of Influence Diameter (inches) Computed
1	7.0	.37	.70	2.59
2	10.5	.24	.60	2.52
3	9.0	.16	.75	1.44
4	5.6	.21	.65	1.18

Table 2 Apparent Extent of the Zone of Influence

*According to Sundaram and Liu (1978)

DEVELOPMENT OF VARIABLE DELIVERY TRIPLE RECIPROCATING PLUNGER PUMP FOR WATER JET CUTTING

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ABSTRACT

With recent development of practicability of the water jet in various industries, advantages are being recognized in its special processing method. Particularly, it is most used for cutting of non-metallic materials and detergent operation, using its water hammer power and chemical solubility. Diversification in the range of applicability is requiring various specifications of high presure pump as a source of water jet. In cutting and detergency, it is necessary to control water jet pressure according to the object material in view of economy and efficiency. High operational efficiency is demanded of a pump by obtaining a delivery volume corresponding to the change of water jet flow. For these objectives, we have elaborated the development of a variable delivery triple reciprocating plunger pump, and, based on its design, succeeded in the manufacture of a high pressure pump. This report provides explanation of its structure and analysis of the results of tests. As a consequence, its high efficiency and small pulsation of water jet has assured the possibility of application in a number of industrial areas.

NOMENCLATURE

Symbol	Description
A1	Sectional area of piston (cm^2)
A2	Sectional area of plunger (cm^2)
N	Number of revolutions of crank shaft (rpm)
P1	Pressure of hydraulic cushion cylinder (MPa)
P2	Pressure of high pressure cylinder (MPa)
Q	Maximum delivery capacity of pump (l/min)
Q1	Volume of circulating oil inflow to the hydraulic cushion cylinder (l/min)
Q2	Volume of injection from the nozzle (l/min)
q1	Volume of a single inflow of oil to the hydraulic cushion cylinder (cm^3)
S	Piston stroke (cm)
S2	Plunger stroke in the Q2 delivery (cm)
W	Required power for totaldelivery (kW)
W1	Conductive power to crank shaft from piston (kW)
W2	Retentive power of Q2 (kW)
η	Nozzle efficiency

INTRODUCTION

An injection of high pressure water from a minute nozzle produces high density of energy increasingly utilized for new processing techniques in many industries. A waterjet of relatively low pressure and large flow is widely used in detergency. An efficient application represents the important station of an automobile assembly line to remove residual powder on mechanically processed engines and transmissions. Also in a chemical plant, a water jet is used for detergency of a polymer tank to assure safety of workers from poisonous gas. On the other hand, a water jet of relatively high pressure and small flow volume is seeking its application in the cutting operation, the objectives of which vary from paper diaper to compound materials for an aircraft to reflect the continuing expansion of applicability. With development of new materials, an effort to automate the cutting operation requires a wide range of applicable cutting pressure. And naturally, careful consideration should be given to maintain the life of machinery and a reasonable running cost. Generally, a water jet for the cutting operation needs a delivery pressure of more than 196 MPa and its maximum pressure is about 440 MPa. The high pressure pump used for this purpose is divided into two categories: a hydraulic booster pump and a crank structured closed coupled multiple plunger pump. These high-pressure pumps require:

i) Easiness of pressure control

ii) High efficiency

iii) Little pressure change

It goes without saying that the basic factors include a long life and easy maintenance. Practicability of the water jet for the cutting operation needs to be analyzed further. In the actual cutting operation, it is necessary to control the volume of water jet from the nozzle. A stop of the injection from the nozzle necessitates a total relief of the high-pressure water or prevention of occurrence of a high pressure. A decrease of nozzles in the cutting process causes a drop of delivery volume and consequently requires regulation of the high pressure water volume . For these purposes, many years have been spent for research and development in pursuit of an efficient variable delivery high pressure pump that permits easy control of pressure and wide flexibility of delivery volume. For the control of pressure, the closed coupled plunger pump has a relief valve on the high-pressure water side, and the hydraulic booster pump has a function of regulating the primary hydraulic pressure. For the flexibility of delivery volume, the eccentricity of the crank shaft or the number of revolutions is changed to increase efficiency of the closed coupled plunger pump, while the hydraulic booster pump is equipped with a variable delivery pump in the oil circuit. In these circumstances, we have developed a brand-new variable delivery triple reciprocating plunger pump (Sugino Aqua Jet Pump Mechanical Drive). This is a new type hybrid pump having a hydraulic cushion cylinder between the crank structure and the high pressure plunger that yields high efficiency and capacity of controlling the injection pressure of more than 196 MPa. Further, the resultant negligible pulsation permits the application in various industries especially for the water jet cutting pump. We will explain its structure and analyze the characteristics, hoping that they would constitute foundations that contribute to expand the range of applicability.

THE STRUCTURE AND FUNCTION OF THE PUMP

General Specifications

The specifications of our variable delivery triple reciprocating plunger pump are as follows:

Model:	Sugino Aqua Jet Pump AJM-4524
Delivery pressure:	Maximum 440MPa
Delivery volume:	Maximum 2.4 l/min
Plunger stroke:	Maximum 100 mm
Crank shaft revolution:	60 rpm
Crank shaft drive motor:	22 kW

Figure.1 shows Sugino Aqua Jet Pump AJM-4524.

Basic Circuits

The basic circuits belonging to the area of high pressure are illustrated in Figure 2. They are mainly composed of a hydraulic circuit, a feed water circuit and a high pressure water circuit. The pressure oil generated by the hydraulic unit is fed to the respective hydraulic cushion cylinder. The pressure in the hydraulic circuit is controlled by a hydraulic relief valve. In the high pressure cylinder, the water continuously sent from the water feed unit is pressurized by the reciprocating plunger. The high pressure water generated from the triple plunger passes through the high pressure filter and is injected from the nozzle. The high pressure filter prevents the nozzle from clogging by dust in the fed water and wear powder from the seal packing in the high pressure unit, and serves to restrain the pulsation of high pressure water circuit by means of the compressibility of the water. An On-Off valve is installed at the nozzle to stop the injection as necessary.

Principles of Operation

Figure 2 indicates the mechanism of the pump. The pressure oil generated from the small hydraulic unit is directed by the accumulator and the hydraulic cushion cylinder and fixed at P1 by the hydraulic relief valve. The motor revolves the crank shaft and reciprocates the piston, which then reciprocates the hydraulic cushion cylinder and the attached plunger. The piston can slide on the inside of the hydraulic cushion cylinder. The following equation is established between the pressure in the hydraulic cushion cylinder and that in the high pressure cylinder.

$$P1\ A1 = P2\ A2 \qquad (1)$$

The delivery capacity of the pump is maximum when there is no sliding motion of the piston in the hydraulic cushion cylinder. Assuming that the compressibility of water is 0,

$$Q = \frac{3\ A2\ S\ N}{1000} \qquad (2)$$

Neglecting the volumetric and mechanical efficiencies, the required power for the above delivery is:

$$W = \frac{P2\ Q}{60} = \frac{3\ P1\ A1\ S\ N}{60000} \qquad (3)$$

The injection volume from the nozzle is obtained by the following formula:

$$Q2 = 2.1 \quad D^2 \sqrt{P2}\eta \qquad (4)$$

(D is the nozzle diameter)

Generally,

$$O \leq Q2 \leq Q \qquad (5)$$

The retentive power of the Q2 is:

$$W2 = \frac{Q2 \quad P2}{60} \qquad (6)$$

When the Q2 is being injected from the nozzle, the plunger stroke is:

$$S2 = \frac{1000 \quad Q2}{3 \quad A2 \quad N} \qquad (7)$$

Each piston slides in the hydraulic cushion cylinder for a distance of (S - S2); when the piston moves forward, q1 is discharged to flow into other hydraulic cushion cylinders whose pistons are moving backward. This motion is repeated alternately and continuously in the three hydraulic cushion cylinders.

$$q1 = A1 (S - S2) \qquad (8)$$

Q1 circulating in the three hydraulic cushion cylinder is:

$$Q1 = \frac{3 \ q1 \ N}{1000} \qquad (9)$$

When q1 flows into the hydraulic cushion cylinder on the backward motion of the piston, the piston is pressed by the force equal to P1 A1 , which revolves the crank shaft. The total conductive power to the crank shaft from the three pistons is:

$$W1 = \frac{P1 \quad Q1}{60} = \frac{3 \quad P1 \quad A1 \quad (S - S2) \quad N}{60000} \qquad (10)$$

Therefore, the required power for the revolution of the crank shaft is:

$$W\text{-}W1 = \frac{3 \ P1 \ A1 \ S2 \ N}{60000} = \frac{Q2 \ P2}{60} = W2 \qquad (11)$$

As shown in the above formulas, the required power equals in effect to the retentive power of the Q2 ; if the injection volume from the nozzle decreases the required power for the crank shaft revolution also decreases.

The injection pressure can be controlled easily by changing the prescribed pressure in the hydraulic cushion cylinder with the relief valve in the hydraulic circuit.

CHARACTERISTICS OF THE PUMP

Characteristics of Power

In this section, the effect of power reduction of the hydraulic cushion cylinder is analyzed by means of calculation of torque characteristics. Figure 3 shows the structure of the crank. Provided that the force on the piston acted by the pressure in the hydraulic cushion cylinder is F and that on the crank shaft F1,

$$F1 = \frac{F}{\cos\phi} \tag{12}$$

Torque T is:

$$T = F1\ R\ \sin(\theta - \varphi) = F\ R\ \sin\frac{(\theta - \phi)}{\cos\phi} \tag{13}$$

where $\phi = \sin^{-1}(-\frac{R}{L}\sin\theta)$

The displacement of the plunger is:

$$X = R[(1+\cos\theta) - \frac{r}{4L}(1 - \cos 2\theta)] \tag{14}$$

Figures 4 - 7 show the rotation force on the crank shaft worked by each piston and the resultant power K at four different delivery rates: O %, 50 %, 75 %, 100 % (plunger stroke:0,S/2, 3S/4 , S). It is clear from these figures that the theoretical power acted by the hydraulic cushion cylinder decreases in proportion to the delivery rate; at the delivery rates of 0, 50 %, 75 %, the theoretical power corresponds at 0, 0.5, 0.75 .

The triple reciprocating plunger pump with a pressure control discharge valve in the high pressure water circuit has the same power characteristics as in Figure 7. However, if the delivery volume decreases, the valve prevents the power reduction of Figure 4 -6 and maintains the total delivery power.

Figure 8 shows the actual power consumed by the pump drive motor (actual power) and the retentive power of the total delivery from the nozzle (theoretical power), at a delivery pressure of 392MPa and variable delivery volume. The actual power also decreases in an approximate proportion to the delivery decrease.

Figure 9 shows efficiencies in four different cases: 1) at full operation of the hydraulic cushion cylinder, 2) actual efficiency of the present pump, 3) efficiency of the triple plunger pump installed with a pressure control valve in the high pressure water circuit, and 4) efficiency of a hydraulic booster pump using a variable delivery hydraulic pump (max. efficiency 90 %). From these figures, the following results are obtained: In the existing triple plunger pump, the efficiency is reduced to 43 % by the same rate of decrease of delivery volume, while in the present pump it decreases to only 79 %. Further, it is always higher than that of the hydraulic drive booster pump at any delivery volume. These results lead us to a conclusion that the present pump yields the highest flexibility and efficiency.

Analysis of the Pressure variation Characteristics

In any pump of the same crank structure as the present pump, the plunger speed varies with the change of revolution angle of the crank shaft, which induces a change in the discharge capacity and subsequent pulsation in the high pressure water circuit.

Figure 10 indicates theoretical characteristics of the discharge capacity at maximum delivery. According to this figure, the pressure variation $\alpha 1$ is 20 %, which is equal to that of the existing triple plunger pump. At a delivery volume of less than 80 %, the sliding piston in the hydraulic cushion cylinder acts on each plunger to discharge a necessary volume constantly, so that the total compound discharge offsets pressure variation theoretically.

Figure 11 shows the actual pressure variations in the high pressure water circuit at 100 % and 50 % of delivery capacity with 392 MPa of operational pressure.

While the anti-pulsation effect of the hydraulic cushion cylinder is relatively small in this test (α 2=12 % at 50 %delivery), the pulsation has been reduced considerably compared to the existing hydraulic drive booster pump; for the total absence of pulsation, the required accumulator volume of the present pump is only 70 % of that of a hydraulic drive booster pump (reciprocation rate: 20/min) of the same power.

DISCUSSION

Subsequent to the development of a variable delivery plunger, the analysis of its characteristics confirmed a considerable effect of power reduction of the hydraulic cushion cylinder. However, the present test shows a gradual decrease of efficiency with the decrease of delivery volume (Figure 9). If the hydraulic cushion cylinder functions perfectly, the efficiency must be 86 % at any delivery rate. Possible reasons for this deterioration of efficiency are friction resistance of the packing caused by sliding of the piston in the hydraulic cushion cylinder and fluid resistance in the cylinder due to the discharge and inflow of oil. For the same reasons, the pulsation did not reach O at the delivery rate of 50 %.

In Figure 5 (delivery volume at 50 %), our assumption provided that the pressing force of the piston at the time of oil discharge of the hydraulic cushion cylinder (F'') equals to that at the time of oil inflow (F"); -F' = F". However, with a modified formula of -F' = 1.1 F", the average compound power of each piston becomes about -0.52, and the efficiency rate thereof $\frac{-0.48}{-0.52}\,86 = 80$ equals to that in Figure 9.

The following conclusions have been obtained from various tests carried out on the present pump.

i) The power consumption can be controlled in proportion to the delivery volume to offer an efficient cutting operation.
ii) The injection pressure control is easier than the existing triple plunger pump.
iii) The pulsation is smaller than the existing hydraulic booster pump.

ACKNOWLEDGEMENTS
The authors gratefully acknowledge the contributions of Prof. O. Imanaka, Department of Technology, Kanazawa University.

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3. Nakaya M., Nishida N., 1980, The systems and water jet cutting data: Fifth International Symposium on Jet Cutting Technology., England, BHRA Fluid Engineering., Paper no. J2.
4.

Photo Figure 1. Sugino Aqua Jet pump AJM – 4524.

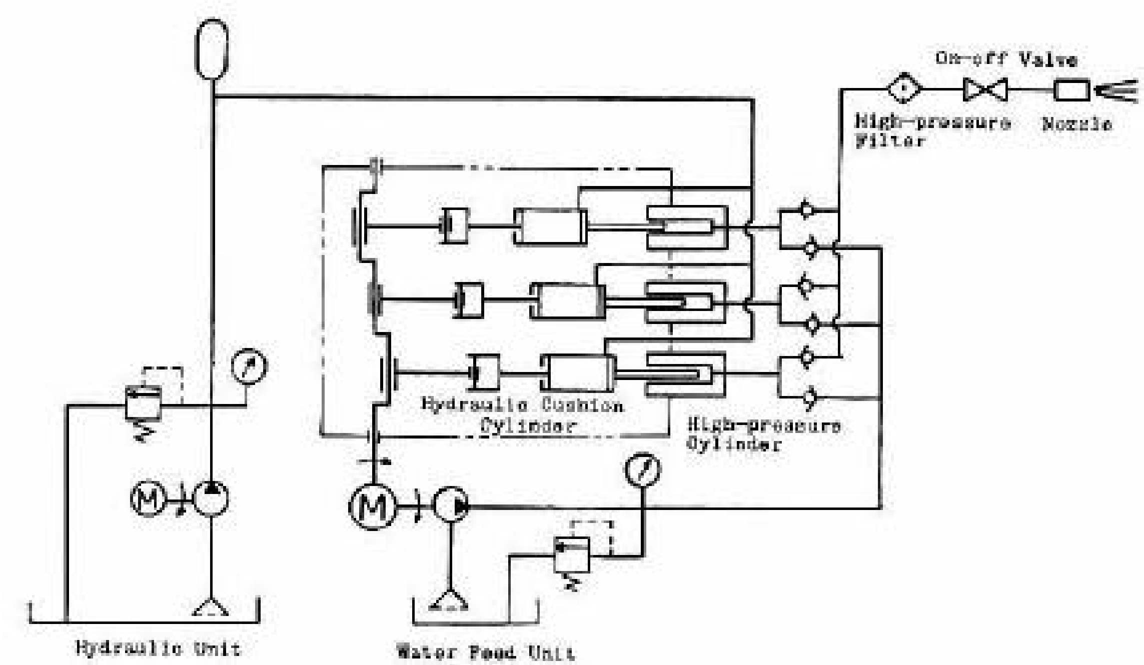

Figure 1. Hydraulic and water circuits.

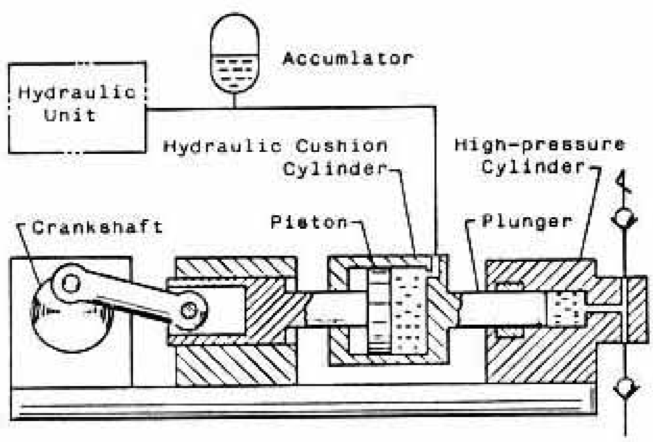

Figure 2. Mechanism of the pump.

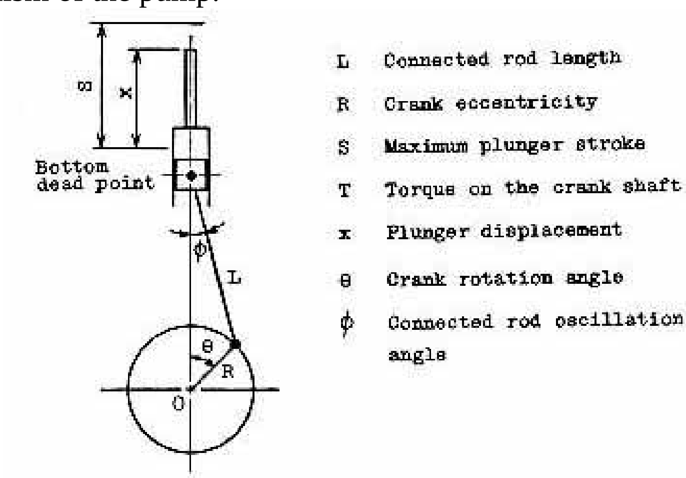

Figure 3. Structure of the crank.

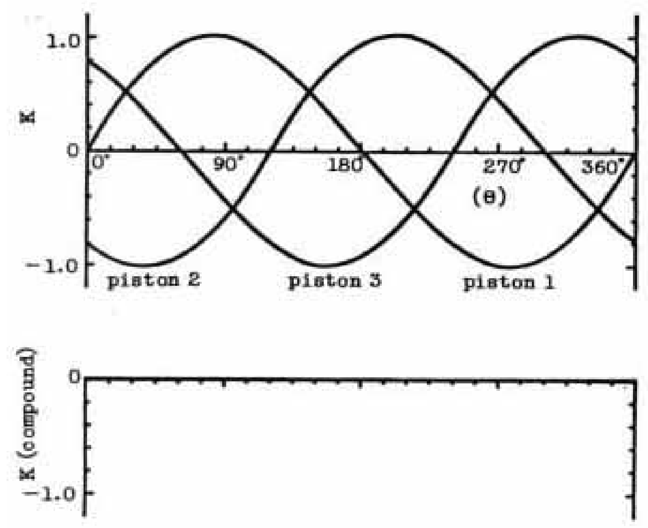

Figure 4. Theoretical required power K at Q2/Q = 0

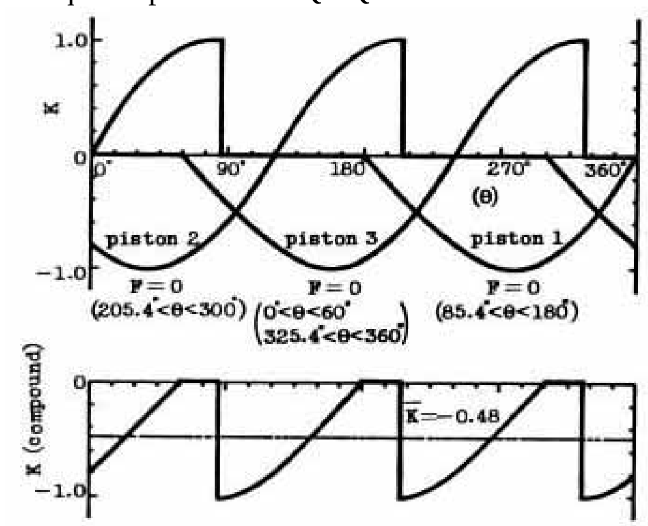

Figure 5. Theoretical required power K at Q2/Q = 0.5

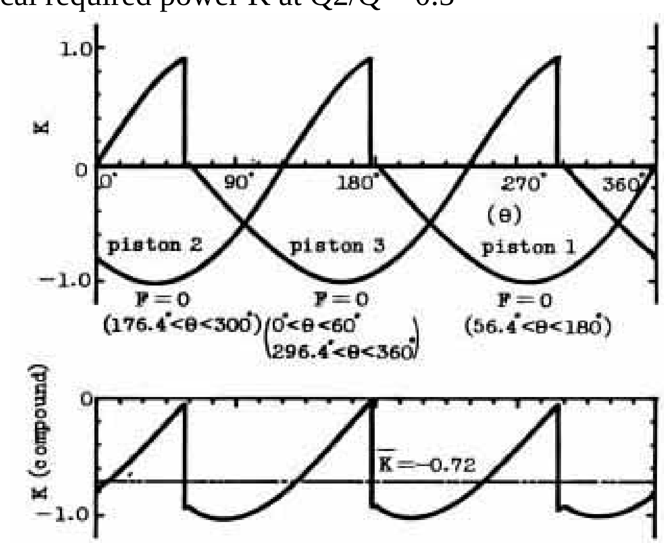

Figure 6. Theoretical required power K at Q2/Q = 0.75

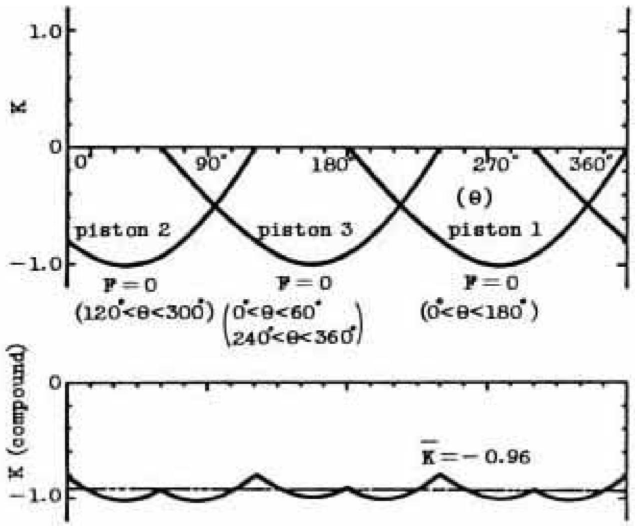

Figure 7. Theoretical required power K at Q2/Q = 1

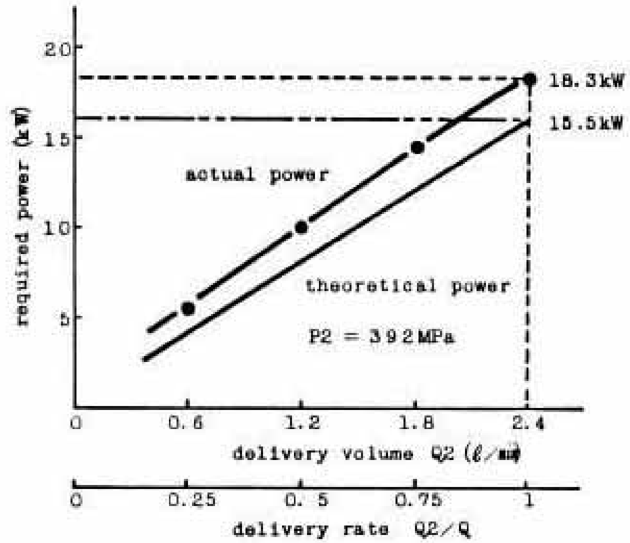

Figure 8. Variation of required power with delivery volume at constant pressure of 392 MPa.

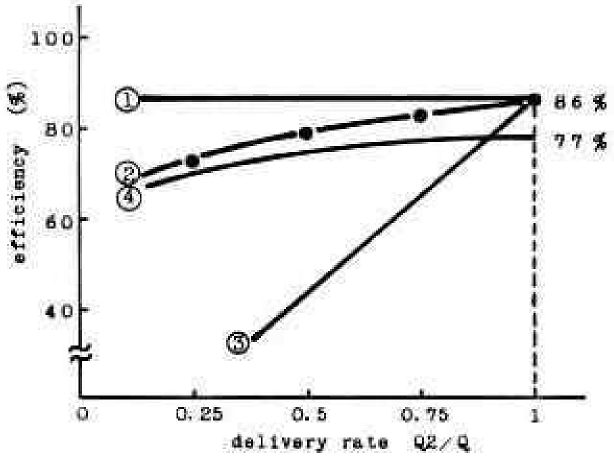

Figure 9. Variation of efficiency with delivery rate.

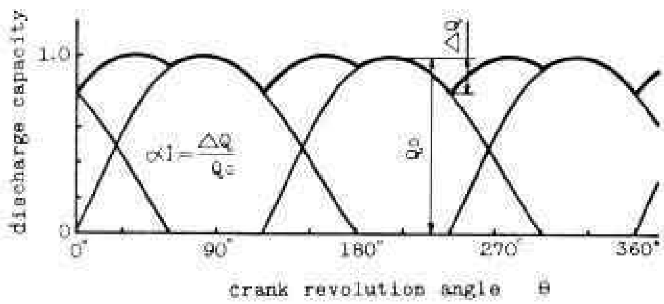

Figure 10. Theoretical discharge capacity at maximum delivery as a function of crank rotation angle.

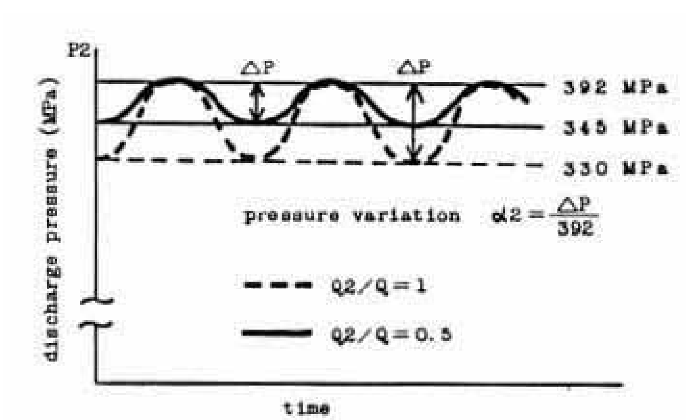

Figure 11. Variation of actual discharge pressure.

DISCUSSION

NAME: Mohamed Hashish
COMPANY: Flow Industries

QUESTION:1. "In your analysis you did not consider any dynamics that might effect the stability and response of the plunger to the rest of mechanical and hydraulic input signals. Is there any practical problems due to that?

ANSWER: We indicated in this paper that resistance of hydraulic circuit might influence pressure change and efficiency in operating hydraulic cushion cylinder. Additionally this resistance might effect slight dullness of response to the input signals. But it's no big problem in practical use.

QUESTION:2. "Why do you have to go to multiple pumping circuits and high pressure filters for a potentially limited dynamically stable pump with only a slight increase in efficiency over conventional intensifier pumps?"

ANSWER:We are thinking the rise in efficiency from 77% to 86% has significant meaning.

NAME: Dr. M. Vijay
COMPANYNRC

QUESTION: 1. "Material of fabrication of the pump?"

ANSWER: Precipitation hardening stainless steel is mainly used in high pressure parts.

QUESTION: "Cost of the pump?"

ANSWER: 2. "About $50,000."

THE "SKIPJACK" SEWER CLEANING NOZZLE

Michael H. Ewald, Manager, Marketing & Sales
WOMA Corporations 242 St. NichoIas Ave.
South Plainfield, New Jersey 07080

ABSTRACT

The most efficient use of high-pressure water nozzles in sewer cleaning occurs when the nozzle is constantly engaged well within the media to be cleaned. This media is generally most concentrated at the bottom of sewer lines .

When nozzle design or use allows the nozzle to float or roam freely through the sewer, cleaning efficiency is markedly reduced. More water is utilized, more power is consumed and more time is taken to accomplish a given job .

The new WOMA "Skipjack" was designed to assure that cleaning nozzles remain at or near the bottom area of sewer lines. This results in better and more efficient sewer cleaning practices .

INTRODUCTION

Larger I. D. sewer lines mostly have to be cleaned at the bottom only. This is where the unwanted buildup starts, and the sludge settles. Conventional cylindricl sewer cleaning nozzles and carrier heads spray symetrically which means sections of the inner sewer surface are covered that do not need any cleaning at all. Economically, this is a very questionable procedure.

Flat and very heavy nozzle carrier heads available on the market today are supposed to direct the jets towards the bottom of the sewer, however, obstructions and twisting of the hose can make effective bottom cleaning questionable particularly, due to the singularity or limitation of jet directions.

By design, WOMA "Skipjack" sewer cleaning nozzle maintains its position in the sewer at optimum locations for most efficient cleaning. The "Skipjack" with predetermined jet spraying angles optimizes cleaning and flushing results of sewer bottoms. A swivel joint within the hose also aids in maintaining the "Skipjack" in optimum cleaning positions. The "Skipjack" is most effective in sewers with diameters from 10 inches upwards.

The P I version (see chart) with jets set at 15 degrees are best used for sewers up to 25 inches I.D. and egg shaped profiles. For Larger I.D. sewers and all other profiles, the P II version with jets set at 30 degrees are best used .

TRACTION FORCE

Traction force requirements of the nozzles vary depending upon hose size and type, consistency of sewer, nature of debris and distances between manholes. For economic reasons it would be wrong to utilize maximum traction force performance if the job can be effectively accomplished with lesser traction force. The following table give you some typical examples on the performance (traction force) of both the P I and P II "Skipjack" based on certain pump performances.

TRACTION FORCE IN LBS FOR "SKIPJACK" WITH 7 NOZZLES

1503	Flow (GPM)	Pressure {PSI)	HP	Nozzle I.D (mm)	PI	PII
P45	56.5	1,900	75	2.2	121.5	114.5
P50	71.5	1,500	75	2.5	145.0	136.5
P50	71.5	2,200	110	2.4	167.0	158.0
P55	86.0	1,250	75	2.8	151.5	143.5
P55	86.0	1, 750	110	2.5	182.0	172.0
P55	86.0	2,500	150	2.4	212.5	201.0
P60	100.0	1,500	110	3.0	197.0	186.5
P60	100.0	2,000	150	2.9	227.5	215.0

NOZZLE SELECTION

The "Skipjack" can be equipped with either straight jet or fan jet nozzles. The choice depends mainly upon the task and there is no general rule. It is important, however, that fan jets are installed tangentially parallel .

At the front of the "Skipjack" a nozzle head with three bores is installed to clear the way and also reduce wear on the body of the "Skipjack" by creating a water cushion to travel on; thus, reducing friction to a minimum .

WEAR PARTS

The "Skipjack" is fitted with a replaceable wear plate made out of special wear resistant alloy steel to enable economical replacement of the principal wearing part.

AVAILABLE VERSIONS

For the U. S. and Canadian markets, both the P I and P II versions are available for typical sewer cleaner performance rates in the 60 to 70 HP range.

Important factors in considering the use of the "Skipjack" includes such data as size and profile of sewers, equipment being utilized, and the proposer cleaning methods.

When working with combination vehicles the P I version with the higher traction force and the higher production rate due to its 15 degree nozzle angle should be used. Flushing and vacuuming can be done simultaneously.

When working with a flushing and vacuwm (or suction) vehicle, the P II version with its 30 degree nozzle angle can prove advantageous .

Sludge and debris piles up under the manhole and can easily be vacuumed out.

HYDRO-BLASTING SAFETY

C.W. Adaway, Hydro-Services, Inc., Missouri City, Texas;
John F. Hinrichs, Hydro-Services, Inc., Missouri City, Texas;
J.D. Frye, Hydro-Services, Inc., Missouri City, Texas

INTRODUCTION

While this paper and supporting video presentation are most directly addressed to Hydro-Blast cleaning (use of high velocity water for removal of unwanted debris), there are many parallels that may be drawn between them and any other applications using high pressure fluids. It should also be noted at this point that this information is not meant to be, nor should it be, construed as an all-encompassing text of safety hazards. The intent is to point out the normal hazards involved in a basic hydro-blasting operation.

Anyone who has worked with high pressure Hydroblasting equipment knows of the tremendous force behind that stream of water.

GENERAL INFORMATION

The water pressure developed by these high pressure pumps can reach several thousand pounds per square inch. All this pressure is forced through very small openings in the tip end of the blasting gun. The resulting force is so tremendous it is strong enough to cut through almost anything in its path.

The information in this program should be of interest to anyone who will ever hold a Hydro-Blast gun in his hands or to anyone who will work close to this kind of machinery.

Remember, this stream of water can be moving in excess of 1,000 feet per second. At this point, it's not just water; it's a powerful lance which can penetrate almost anything and clean a surface down to bare metal in just seconds.

This is why Hydro-Blasting machine is so useful in industry. It is fast and thorough. To do by hand what the Hydro-Blaster does, would be extremely time consuming. But the machinery would be useless if it couldn't be operated safely.

SET-UP

The time you spend in set-up and safety inspection is time well spent. It is difficult to do a good job without good preparation. And in Hydro-Blasting, preparation begins with the location you choose for spotting your high pressure pump.

Select an area which will not be hit with spray from the blasting operation. Any other machinery or electrical utilities in the vicinity may require protection from the high pressure jet stream or its spray.

If engine driven equipment is parked inside a building or other enclosed area, exhaust ventilation to remove the carbon dioxide must be planned for. Also, the noise created by the engine may cause problems and hearing protection may be required.

A good water supply is needed for the pump. Fresh water is best. Brackish water or water with solid particles is hard on pumps and seals and will shorten their working life. Always check the strainer or the water intake. A clean and properly installed strainer will insure safe and proper operation of the machinery. Be sure that the strainer is clear and stays that way.

When setting up a Hydro-Blasting operation, the necessary running of hoses can create tripping hazards. Work to minimize the exposure of others to this problem. Where heavy machinery is likely to pass over the hoses, barricade the roadway to prevent this from happening. Do not allow the hose to be pinched off or ruptured, which could interrupt the water supply. Be sure all valves or electrical equipment that will affect the cleaning operation are tagged or locked out.

Barricading the area is extremely important when Hydro-Blasting machinery is in use. This may be the only protection another person has from the power of the blast gun. Use barricade tape and signs around the perimeter to warn people that the machinery is in use. But your responsibility does not stop there! Talk to the other people who will be in the work area and be certain they know the precautions for high pressure equipment. Discourage anyone from being too curious about the jet stream. No one wants to see someone "test" the water pressure by putting his hand in front of it.

Also, everyone should be told to stay clear of any spray which may contain deposits loosened in the scaling operation.

A worker involved in Hydro-Blasting should make a mental note of his surroundings. Keep in mind that there may be other activities going on in the work area and that your job is one part of the business at hand. An emergency situation could arise in the area which would suddenly involve you. So be familiar with the emergency signals where you are working. Learn the location of emergency or evacuation equipment that you may need and definitely learn the escape routes.

Another point to consider in your planning is the effect that water may have upon the material that is being removed. Check with your supervisor to be sure that this combination does not produce a hazardous material such as H_2S.

Since the blasting machinery generates many gallons per minute, adequate drainage is important. Plan for this runoff in advance by selecting the best drains to use for the disposal. If the drains in the area cannot handle the waste water, vacuum trucks or some acceptable other alternative will have to be found. Being considerate of the effect your work has on the environment is also part of working safely.

Remember, your Hydro-Blasting work must fit in with the rest of the work activity in the area. In general, it is best to do this when the least number of people are scheduled to be in the immediate vicinity. So, when the conditions permit, avoid the hours that are the busiest.

PRE-START INSPECTION

Before starting to use the Hydro-Blaster, check to see that the hoses and fittings are in good condition. Check for thread wear on the high pressure couplings and hose ends. The improper use of wrenches is the main cause of damage to hose ends. So take the time to use the proper tools.

The extremely high water pressure also erodes the rubber hose material. If the outer rubber is torn and the braided material is exposed, check carefully to be sure that the braids are not broken. Any high pressure hose that is in questionable condition should be taken out of service immediately. Do not try to repair a hose yourself. Leave that to the manufacturer.

Again, do not take a chance. It is dangerous to use equipment that is not in good condition. If you suspect that the equipment is faulty, replace it.

A safe Hydro-Blasting job can only be accomplished when the high pressure equipment is properly set up. You will need to clean and teflon tape all fittings and couplings before installing them in the system. When teflon tape is used, be sure that none is left in system, because it will plug nozzles or dump valves.

With the pump engine shut down, hook up the high pressure pipe, hose, fittings, and lances to the discharge manifold of the pump that will deliver the maximum hydraulic horsepower to the nozzle. Start the engine and set it at proper RPM. Flush the pump, hoses and lances by opening the dump valve with nozzle removed. This will insure that there are no heavy solids in the line before installing the jets, thus reducing the chance of the jets being plugged.

The dump valve is a safety device used to control the on-off of the high pressure water. The dump may be either a hand or foot operated type. In either case, it must have guards around the trigger to help prevent accidental activation.

PRE-START PRESSURE TEST

Now, restart the engine and pressure test the complete system as you have it hooked up. The test should be made up to one of the following limits:

1. Either the maximum rated working pressure of the pump head or the dump valve;
2. One and one-half times the rated working pressure of the hose, or
3. The highest pressure attainable with the hook-up you have, not to exceed 11/2 times the operating pressure you expect on the job.

Repair all leaks that you find, but never hammer or tighten any union or connection while the system is under pressure. Relieve the pressure first, tighten the leaking section, then repressurize to verify that the leak has been stopped.

Inspect the structures or area that you will be Hydro-Blasting for any loose objects which may become projectiles when hit with the high pressure spray.

There are a variety of pump and nozzle combinations that may be used depending upon the specific needs of the job. But in every case, the workers must be protected.

Figure 1 illustrates the normal protective clothing. The hard hat prevents head injuries. It is required in most industrial areas. He is wearing both a full face shield and safety glasses with side shields. If any water or flying debris gets under the face shield, the safety glasses should prevent it from entering his eyes. If necessary, he wears ear protection because of the high noise level. The rubber gloves protect his hands from both the spray and the equipment. The rubber boots he wears are extra thick and cannot readily be cut by the high pressure HydroBlast spray. All barrels on shotguns (Fig. 2) should be at least 48" long. This helps prevent foot injuries from the high pressure water. Special precautions should be used when the barrel is less than 48".

SHOT-GUNNING

Experience with Hydro-Blasting equipment is the best way to learn which jets and what pressure is the safest and most efficient for the work being done. If more than one jet is used, you must balance the force of the jets by arranging them so that the back pressure is equalized.

Watch for any indications that the nozzle orifices are becoming plugged. A plugged nozzle can offset the thrust so that the gun is forced strongly in one direction. This not only reduces the cleaning efficiency but can throw a worker off balance and cause him to lose his footing. If he loses control of the blast gun, an accident can result.

Remember, always operate within the rated limits of the weakest part of the system. The operating pressure should be capable of removing the deposit that is to be cleaned. A higher pressure should only be used where significant improvements can be realized, such as a faster or more thorough cleaning job. If a higher pressure is desirable, be sure the dump valve is rated at that higher pressure as well. If not, do not exceed the operating pressure of the valve.

The dump valve is your main safety device. Every hook-up must include one, and it must operate properly. The dump valve should always be controlled by the worker closest to the nozzle. If it fails for any reason, the work must be stopped immediately until the valve is repaired or replaced.

TUBE-LANCING

Figure 3 illustrates a common tube-lancing operation. The worker repeatedly applies high pressure water to clean the internal walls of tubes. Sometimes they are horizontal, sometimes vertical, and the tubes may be in any shape or size. This method requires one or more workers to handle the lances or hoses. Be cautious of plugged tubes which may cause a hydraulic back pressure since the water cannot escape through the end of the tube. If the pressure builds, the lance will tend to back out of the tube being jetted and you will know it is plugged. If so, relieve the pressure on the lance by pulling it back, hold it there until the pressure eases, then slowly move it forward. Before removing the lance from the tube, release the dump valve. This will allow the water to be diverted to waste. This makes moving the lance easier and eliminates anyone's exposure to the cutting action of the jets.

If using a flexible lance, a 2' stinger should be used. This will keep the worker from pulling the nozzle out under pressure. Once again, the worker operating the flexible lance should operate the dump valve.

If the opposite end of the tubes or lines are open, a cover or shield should be installed on the end to divert the jet spray. Barricade the open end to keep people out of this hazardous area. Again, protective clothing must be worn at all times.

LINE MOLEING

Figure 4 illustrates a different kind of HydroBlasting operation. This is called pipe line mole jetting. The jet mole is self-propelled through the line by the action of the high pressure water. Adequate precautions should be taken to prevent line mole reversal. Also, the first two feet must be cleaned by the shotgun. This will insure that line mole will not be pulled out under pressure. The worker doing line moleing should operate dump valve.

When the lines to be cleaned are below ground, consider the problem of toxic gases in the entrance tunnel. A customer representative should always confirm that the area is safe to enter. Also, remember that there must be some way to remove the waste water and deposits from the line.

SCAFFOLDING

There are occasions when it is necessary to HydroBlast above the ground level. In these cases, the work must be done from OSHA approved scaffolding only and no make-shift structure. This is not just an arbitrary rule designed to make things difficult. It is absolutely the only safe and dependable way to support workers using this kind of equipment.

With the back pressure produced by the shotgun, it would be foolish to attempt to Hydro-Blast while standing on a ladder. It would be far too easy to simply be blown over backwards. Use a ladder only to get to and from the scaffolding or platform from which the work will be done. Use scaffolding whenever the blasting is to be done above four feet from ground level. Above this point you will have to aim the shotgun slightly upward and that can cause trouble. So take the time to build a structure from which to work.

Let's talk about that back pressure again. That is the kick that you will feel from the blasting gun as you use it. Any experienced worker has a healthy respect for this back pressure. As the job progresses, this kick back can be tiring, so the sure footing of the worker is important. Prepare by removing any potential tripping or slipping hazards in the work area. To help eliminate the hazards of being thrown off balance by the back thrusts of the shotguns it may be possible to tie off the gun which helps to curb the back pressure.

Hydro-Blasting operations have added significantly to the safety and efficiency of industrial equipment maintenance. By using the proper machinery, common sense, and .a genuine concern for yourself and others, your job can be performed accident-free, and that's what we all want.

Figure 1. Safety Gear

Figure 2. Safety Gear

Figure 3. Tube Lancing.

Figure 4. Line Moleing.

DEVELOPMENTS IN CLEANING COKE OVEN DOORS

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SUMMARY

This paper will report on the work carried out over the last eight years on cleaning coke oven doors by using high pressure water jets.

On a coking plant the doors are regularly removed to enable the re-charging process to take place. The doors have to be cleaned within one and a half minutes during the "pushing cycle". Buildup of hard carbonaceous, bitumastic layers on the doors and seals prevent easy removal and replacement.

Additional benefits of clean door seals are the prevention of polluting the surrounding area, and a lessening of health hazard to the coke oven operators.

Investigation into optimizing the cleaning operation will be reported.

INTRODUCTION

Coke is used in the iron making processes and to make the coke, coal is baked in ovens for about 18 hours.

The gases released from the coal are used for firing the steel furnaces. There are many by-products, such as tar, benzine and the residue coke is discharged, and cooled, prior to being fed into the hungry blast furnaces, as required, for the steel making processes.

The ovens used for coke making are up to 10 meters high, 0.5 meters wide and 30 meters deep. In a large coking plant there are several batteries and each battery comprises 50 ovens (see figure 1).

The charge of coal is fed into the top of the ovens and after cooking for about 18 hours, vertical doors placed at each end of the oven are removed, the red hot coke is pushed through into specially built railway trucks. The coke is then cooled and then fed by conveyors directly to the blast furnaces. After the coke has been pushed through, the two doors are replaced. There has been an increasing problem with existing coking plants where the doors have suffered from fouling with tar deposits from the coal. During the 'cooking process', bitumen separates out mainly on the bottom of the oven and if there are any gaps in the door seal, coal tar oozes out of the door. One can imagine that with high narrow doors making a metal to metal seal is quite difficult if not impossible. Distortion of the doors is inevitable due to high temperatures in the oven.

The doors are held onto the oven with two latching catches; as these are turned from the vertical to the horizontal, the latch tightens onto the door jamb, pushing it towards the oven. At times, it is impossible to get the door back onto the oven because of a buildup of bitumen on the faces, thus a spare door has to be collected from the end of the battery and placed onto the open oven. This disruption wastes time and the target of pushing at least 6 ovens per hour falls quickly behind. On the side of the oven where the coke is deposited into the railway trucks, the door extracting machine is known as the 'Coke Guide' (see figure 7.) The coke guide removes the door, it moves along, positions the guide at the open oven and when the coke is pushed through, it prevents the red hot coke from falling onto the bench or walkway around the oven.

On the other side of the oven, the door extracting machine is very different and is known as a 'pusher' (see figure 1.) It is large enough in size to carry a large ram up to 30 meters long, the ram face being the size of the oven. The pusher removes the door and then lines up the ram which then pushes all red hot coal through the oven and out of the other side.

THE ENVIRONMENT

Leaky doors allow coal gas and sulphurous fumes to escape to the surrounding countryside. When coal gas escapes from a leaky door, usually the gases ignite, causing a flame on the outside of the oven. These flames allow local overheating of the outside skin of the door and door jamb and this uneven heating can cause distortion. So, in addition to the buildup of carbonaceous deposits on the door and the door jamb, a distorted door presents an even greater problem in making a good seal.

So the benefits of clean doors which are well sealed are many and are listed below, but not necessarily in order of their importance:

1. The operators have an improved working environment in that there is a reduction of obnoxious and potentially dangerous fumes.
2. The surrounding countryside is protected from pollution thus keeping EPA satisfied.
3. The doors can be removed and replaced as required and do not hold up the production of coke.
4. The doors are sealed against the ingress of air into the oven.
5. The oven temperatures can be stablized at the required temperature, therefore less energy is used.
6. A reduction of distortion due to flaming gases, which would overheat the doors and frames.
7. A reduction in the burning or wearing of the doors, seals and frames.

TASKS TO BE CARRIED OUT

The only time that is available for cleaning the coke oven doors is during the pushing cycle. The time allowed here is about 1 minute. Hitherto mechanical means have been tried by using hand-held scrapers, but with the size of the door and the time available, only one small part could easily be cleaned by an operator. This is sufficient in places such as India, where labor is cheap and their ancient coke ovens are only 3 meters high, but, of course, modern oven doors are up to 7 meters high and are impossible to hand scrape.

This led to the development of a tracing motion which contained a large number of spring-loaded scrapers. This tracing motion was positioned up to the door which had been taken off. Pressure was then applied to the scrapers and the scrapers were propelled around the door seal with a hydraulic chain drive conveyor. This system allowed buildup of tar on the scrapers; lumps of tar fell onto the chair drive, jamming up the system, causing chain breakages and the scraper jumped over hard deposits. This led the maintenance and operating staff to look for an alternative method of cleaning doors.

The first approach of the idea of using a water jet was rejected on the grounds that the water would cause thermal shock to the door which has a ceramic insulation cladding attached to it. It was also thought that the cold water jetting would cause further distortion to the metal seal. Early trials with hand-held lance showed that if the water was directed at the seal and not the door plug (the ceramic insulation), then thermal shock cracking did not take place. It was concluded that the amount of water used in high pressure water jetting was dispersed in evaporation before sufficient heat loss of the door plug occurred.

Trials were carried out to establish the cleaning rates and other information such as stand-off distance, water jet pressure, volume of water, effective spray patteren, velocity of traverse of water jet, etc. , etc. Initial ideas also included the use of a double water jet so that the door seal was cleaned twice in the one cycle. The object was to replace the scrapers with a lance carrier.

There was also the problem of hose pipe snagging. With high pressure water jetting, the hose pipe remains in a fairly still condition and therefore, if there was to be a rotation of the water jet lance around the door seal avoiding the sprocket wheels and chains, then hose pipe trials had to be carried out.

The possibility of a water jet lance with three nozzles attached to it was considered against a single jet of high pressure water with a greater stand-off distance. The next problem to tackle was the fact that it was found that a new or recently completely cleaned door could be kept clear if a water jet cycle was carried out at every door removal. Whereas doors which were already encrusted with hard bituminous deposits had to have a more intense cleaning operation carried out on them until bare metal was achieved. Always remembering that the time allowed for the door cleaning cycle had to not interfere with the 'pushing time'.

One idea was to water jet the seals at 10,150 PSI for one traverse of the periphery, knowing that it would not completely clean the door. After three cleaning operations, the door seal should then be down to bare metal. After this, the cleanliness could be maintained with a single pass carried out at a lower pressure - say 4,350 PSI.

The methods that were considered for changing the pressure of water jets are listed as follows:

1. The high pressure water is supplied by a triplex plunger pump, and the size of the plungers could be increased or decreased to give higher or lower volume of water by changing the size of the plungers.
2. The nozzle could be changed in the cleaning head to increase or decrease the orifice size.
3. One could vary the number of nozzles in the cleaning head.
4. The pressure could be changed by re-setting the unloading valve which has the effect of matching the jetting nozzles with the pump plungers used. The unloader is an adjustable spring-loaded annular orifice which can be increased or decreased to achieve any pressure desired. This could be wasteful of power in dumping water not

required, and although it is convenient in other water jetting situations, in a fixed installation this is not always found to be desirable.

5. The speed of the pump could be increased or decreased with a variable speed motor, thus increasing or decreasing the volume of water. Eventually, the last solution was used, but as only two pressures were required, then a double wound stator was used and it was found that without changing the nozzles, simply by changing a connection on the motor winding, that the motor would rotate at either 1450 rev/min or 960 rev/min. The pressure being approximately inversely proportional to the flow, therefore, the higher speed rotation of the pump gave 10,005 PSI and lower rotational speed of the pump gave a pressure of 6,525 PSI. This simplified the operation of the speed changes or pressure changes for the operator. He could merely move a lever on the starter which was indicated 'high pressure' or 'low pressure'.

So, as a door was removed, it was examined and the pump started at the required speed. Eventually, after the ovens had been used for about 3 times and all doors were kept clean by the lower pressure of 6,525 PSI, this saved pump power, use of water, pump wear, gland wear, nozzle wear and was generally thought to be the most efficient way of using the jet cleaning system.

DESIGN SOLUTIONS

The first cleaning machines were to be used on the coke side which generally has a more difficult problem with door sealing, as the hot coke is passed through this side of the oven. The coke guide machine is fairly small as it merely has the door extracting facility; a separate trolley car is towed behind the door extractor, which is quite a simple openwork structure. There was difficulty to find space to accommodate quite a large pump, 125 hp unit and a 400 gallon tank. The size of the tank was determined by the number of cleaning operations or pushing cycles required per shift and in order not to interrupt the flow of coke pushing, it was planned that the tank would be filled at the beginning of each shift. The tank was placed as high as possible in the coke guide machine to give a positive head to the triplex plunger pump, low level alarm and shutdown facility ensured that low level of the tank would either stop or inhibit the starting of the pump, this protected the pump from dry running.

High pressure water is fed through a pulsation damper and a high pressure hose to the nozzle cleaning lance. Special attention had to be paid to the gland of the pump to ensure that the coal dust and ash dust which penetrates every corner of the door extracting machines did not affect the plunger seals and crankcase oil seals (see figure 6.)

The tracing motion which originally carried the scrapers had to be redesigned to take the water jetting lance. The first design of the tracing motion carried two cleaning lances each on cleaning one half of the track around the periphery of the door seal. Eventually, this was replaced by one single lance, which rotated completely around the periphery of the door seal.

OPERATING EXPERIENCES & CHANGES TO DATE

Early use of the door cleaner in production had some problems with the lance becoming bent due to the buildup of tar on the door plug. This led to refinements on the hydraulic drive of the tracing motion. A relief valve was fitted so that if an obstruction was encountered, then the rotation would stop, giving a stall situation. After running the first machine for 15 months, it was then decided to install a water jet cleaner on the pushing side. The problems of installation were much simpler as the pushing machine is very large having to accommodate the hydraulic ram and there is much more space available for the water tank and the high pressure pump. Although this side of the coke ovens is a good deal cleaner than the coke guide side, provision was made for protecting the pump unit from coal and ash debris (see figure 6.)

In order to prevent the buildup of tar on the brick retaining surface (see figure 4), it was decided to attempt to clean from the knife edge seal back to the edge to the brick retaining piece. This necessitated a re-design of the nozzle lance assembly. It was decided in order to obtain maximum coverage and maintain effective cleaning to fit a 3 nozzle design lance with accurately spaced nozzles with a stand off of approximately 160 mm. This in practice worked extremely well, however, the tar deposits were still adhering to the refractory and causing the lance to bend. In order to overcome this problem, we decided to approach the coke oven engineers with a further design of lance with different nozzle orifice areas which would effectively lower the pump pressure in order that we might clean the refractory without damaging same. The refractory cleaning lance would be fitted to the carriage assembly when the buildup was becoming too great and endangering the main lance. The formation of tar deposit under normal operating conditions can take up to eight weeks to form so therefore this refractory cleaning lance could be scheduled into the planned maintenance requirements on a monthly basis. In practice, initial worries about cracking the brickwork were soon dispelled as the lance was found to work perfectly and completely eliminated the problem of the lances bending.

With the onset of automation in coke oven plants, it was requested that we design a fully automatic pushbutton system so that the door cleaning can be carried out completely from the coke car drive cabin. The only power sources available to us were electrical and hydraulic supplies. It was decided after due consideration and also the cost case to use hydraulics to actuate our diverter valve (see figure 10.) Variable orifice solenoid controlled hydraulic valves were already in existence on the coke cars since they were used for driving the old type coke oven door seal scraper unit but were made redundant when water jetting systems were fitted.

In order to eliminate the unloader valve, it was decided to incorporate in the body of the diverter valve a needle valve which would be used to regulate the pressure and compensate for any nozzle wear, etc. (see figure 10.) In doing this cost reductions were made to the overall system, simplicity was added and the operational problems of adjusting the unloader valve, etc. were removed. Also by fitting the hydraulically operated diverter valve, we, to a very large extent, eliminated the tampering and deliberate abuse of the manually operated dump valve which existed in the original system. As has been stated previously, the bottom sides of the door are a lot more contaminated with the tar deposits than the top end, therefore with an automatic sequence control unit we would be able to decide at the touch of a button whether a single pass at the bottom of the door was sufficient or if not a double pass could be instigated so that only the most heavily contaminated areas at the bottom of the door were cleaned twice. This particular development has greatly enhanced the overall efficiency of the total cleaning system.

Throughout the development of the systems, problems existed which were not immediately solvable since no equipment was available on the market. The problems were:

1. The hoses becoming caught in the cleaner frame.
2. The swivel couplings burst because of the high radial and axial loads they encountered and also dirt and acid ingress to the vital parts such as seals and bearing areas, etc.

A decision was made to design a swivel coupling which would overcome all these problems and would, through necessity, incorporate the following features:

1. Low breakaway torque, low dynamic friction.
2. Rugged design with robust bearings.
3. Would withstand the high operating pressure and intermittent high temperatures.
4. Very fast seal replacement in the vent of seal failures.
5. Would be resistant to acid attack and dirt ingress.

Prototypes of the swivel design were made and fitted to a coke oven plant on June 21st, 1982. So far these swivels have performed very satisfactorily and have had, to date, no seal failures or failures of any kind. They have completely eliminated the problem of the hoses kicking out sideways when the lance turns a corner, thus preventing the hoses from getting caught in the cleaner frame. The swivel couplings (see figure 9) are made of stainless steel and incorporate a large heavy duty roller bearing carrying the main shaft and a large heavy duty roller thrust race which carries thrusts developed by the high operating pressures. The swivel is fitted with a very high specification, low friction, pre-stressed seal, which is supported by an anti-extrusion ring. Many of these swivels have been made to date and so far not one has failed in any way. Since the swivel design has quite a large outside diameter, it was decided to bore out the carriage and fit clamps to accept the new design of swivel so that it was rigidly supported. Screwed directly into the swivel are the lances for cleaning the seals.

FINANCIAL ASSESSMENT

The installation of the water jet door cleaner will reduce the number of ovens lost due to the inability to latch doors after pushing because of their dirty condition. The number of ovens lost before the water jet cleaner became operational was 30 per month. This has now been reduced to 10. A saving of 20 ovens a month was made and this will increase the coke output by 4,080 tons per annum, assuming a wet coke output of 17 tons per oven. There are other expected savings, but these cannot be evaluated:

(A) Reduction in maintenance costs because of the reduced oven door jamb and refractory damage due to firing around ovens caused by overheating and distortion.

(B) Slight increase in gas (by-product) production because it is recovered and not burned around the door seals. Total savings are unaccountable here. But nonetheless, there are some savings.

20 ovens per month X 12 = 240 ovens per year
17 X 240=4,080 tons
4,080 tons X $75=$306,000 per year

(C) Capital Expenditure:

Material i.e. Pump unit	
Tracing Motion, tank etc.	$112,210
Installation	$ 32,060
Total	$144,270

The payback time is less than six months as this is the first machine on the battery. The second machine on the pusher or cooler side has a longer payback period, say 12 months as the problems of fouling are not so great and the saving returns are less.

CONCLUSIONS

High pressure water jetting has proved to be a practical facility which assists the production of coke. It improves the environment of the plant coke operators and reduces the atmospheric pollution.

The cleaning of doors is completed within the pushing cycle via a machine which is uncomplicated and reliable. Clean doors vastly improve door extraction and maintenance downtime on door machines has been reduced. Gas emissions have been cut down to a bare minimum, thus preventing open door fires which lead to oven door and jamb damage. There is more gas available for by-products recovery. Due to the elimination of door extraction and replacement problems, machine availability has been greatly improved. Also since the water jet cleaners have no mechanical contact with the door or plug, the machines do not cause any seal or plug damage; this has in turn reduced the maintenance requirement of the door seals. As can be seen, water jetting has assisted greatly in every coke oven manager's aims which are high production coupled with lowest possible running costs.

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2. Odds, D.J.H., 1982 Cleaning Coke Oven Doors 6th International Symposium on Jet Cutting Technology, University of Surrey, U.K.

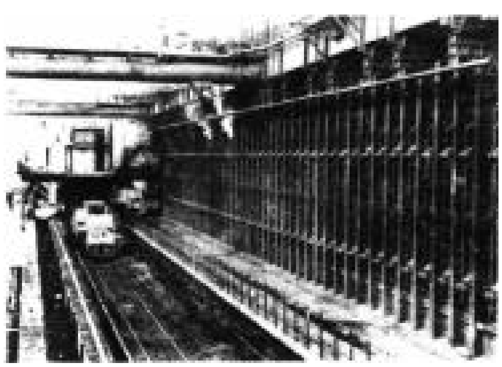

Figure 1. Coke oven battery seen from Pusher side.

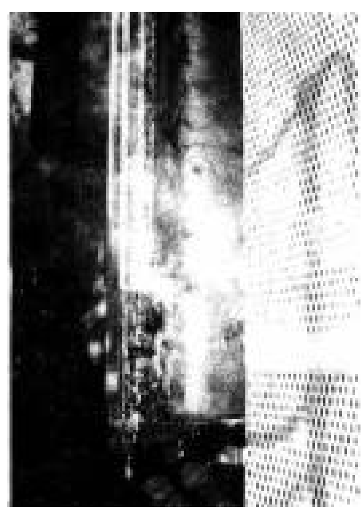

Figure 2. View of bottom of door with tar Deposit hanging from seal.

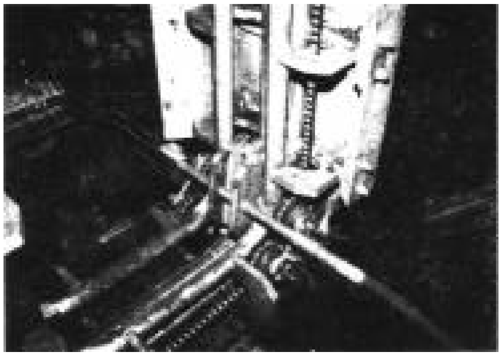

Figure 3. Tracing motion lance carrier.

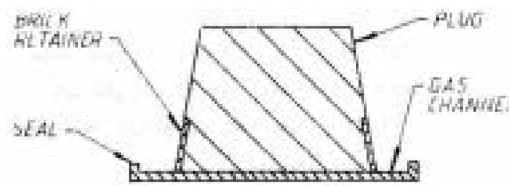

Figure 4. Section thru coke oven door.

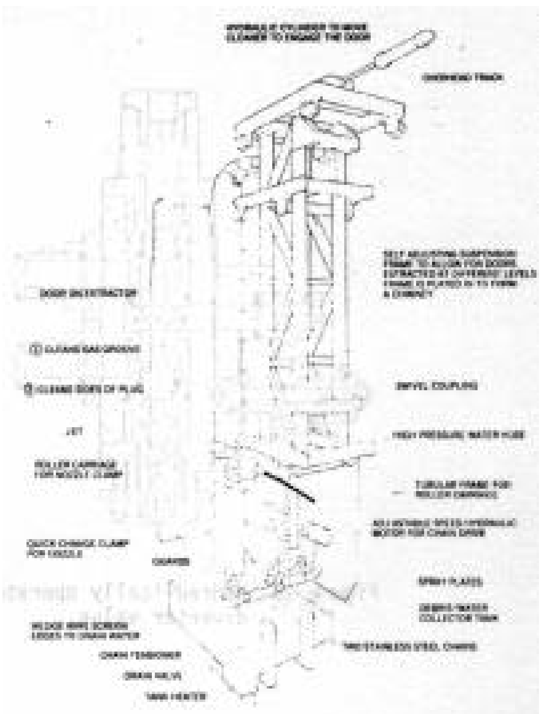

Figure 5. Coke oven door – tracing motion unit.

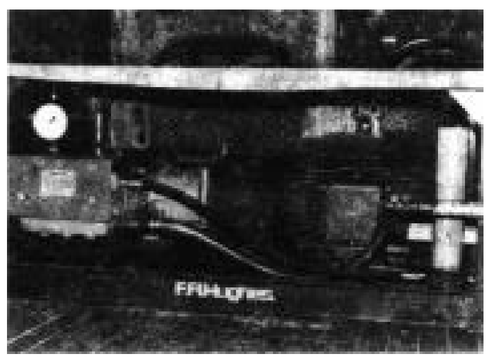

Figure 6. Water jetting unit in its normal environment.

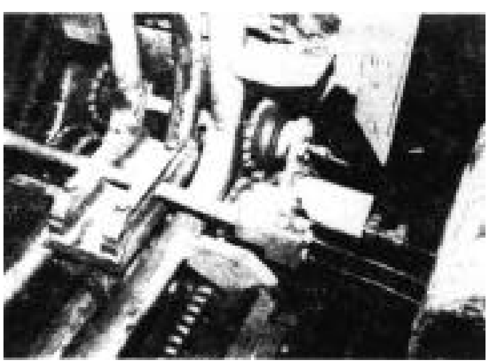

Figure 7. Jetting lance with single nozzle in operation.

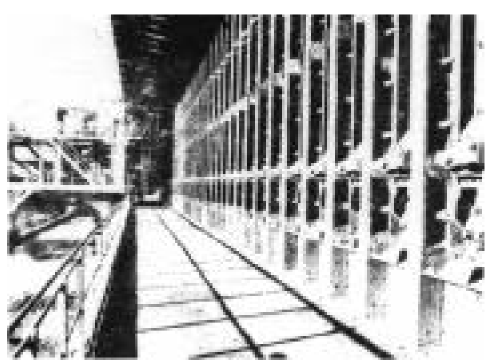

Figure 8. Cleaned up battery 12 months after installation of waterjetting equipment,

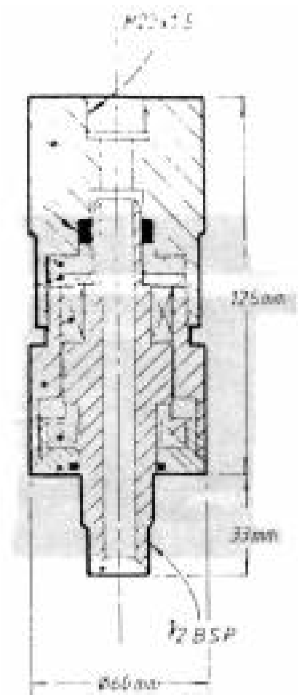

Figure 9. Swivel coupling

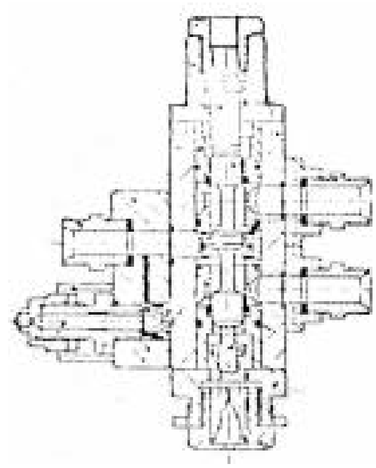

Figure 10. Hydraulically operated diverter valve.

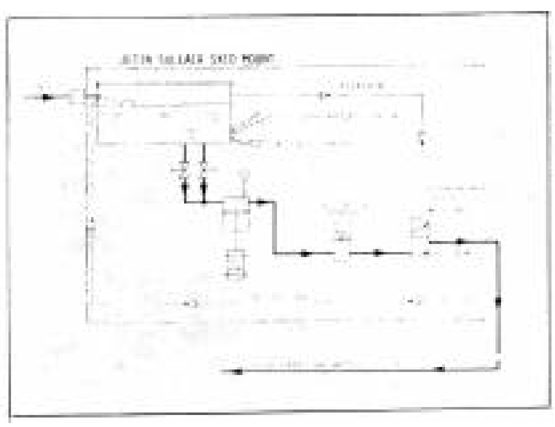

Figure 11. System layout.

DISCUSSION

NAME R. Pootmans
COMPANY: Indescor

QUESTI ON: "Is there not a problem in using multiple jets for cleaning the knife edge, the gas chamber, etc.? The reason for this is that Indescor holds USA patents for cleaning coke oven doors with multiple jets. British Steel gave us assurances that in trying to develop their own door cleaner, they would use single jets only."

ANSWER: I do not believe there to be any problem in using multiple jets. I am reliably advised that the Indescor U. S patent requires jets oriented in different directions. The British Steel Corporation arrangement provided most satisfactory operation, either with single jets or with a plurality of jets oriented in a common direction. I have no knowledge of the assurance mentioned.

OPTIMIZING JET CUTTING POWER FOR TUBE CLEANING

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ABSTRACT

A large variety of pump capacities, lance sizes and nozzle styles are used for waterblast cleaning the interior of pipes and tubes, The design and selection of equipment for each job has relied most heavily on personal experience. The trade-off between greater pressure at the jet tip or greater flow volume is critical to maximizing the cutting power. This is especially true for situations that require long lance rods of relatively small diameter.

The authors offer a method for calculating the optimum nozzle size to create the best flow and pressure for a given job. The flow restriction through various lances, hoses, and fittings is presented along with the formulas used to calculate the flow, pressure, and jet power. By arranging the waterblast cleaning system efficiently, the operator will maximize his potential to make a profit and keep a satisfied customer.

INTRODUCTION

Equipment with a wide range of capabilities is now available for waterblast cleaning. Selecting the best equipment for each cleaning job has become more complicated as the choice of components has multiplied.

Pump capacities now range from a few gallons per minute to several hundred gpm, and pressures of up to 20,000 psi. Pump capacity, hose and lance rod size, and nozzle designs, all have to be decided on.

The selection process has largely remained an art, dependent on the operator's past experience and onsite trial-and-error testing. Little has been published to help guide the operator in choosing which combination of components are best suited for his application. When a difficult tube cleaning job is encountered, just getting a bigger pump is not always the answer. In some cases using a smaller flow rate will result in greater cleaning capability.

The proper sizing of system components can mean more efficient and profitable operations. This paper will analyze the interaction of system components and propose a method for determining the best combination for each particular job.

CUTTING POWER

Nozzle power is a function of the jet volume times the square of its velocity. A very high jet velocity will have little cutting power if its mass volume is low. Conversely, a large amount of water moving at a relatively slow speed will also have little cutting or cleaning ability. Therefore, in any analysis of jet cutting performance, it is necessary to take both the jet velocity and volume into account.

Jet velocity is proportional to the square root of the nozzle pressure that produces it. Since it is easier to measure the nozzle pressure, it is used instead of the resulting jet velocity.

In much of the literature reporting jet cutting, the results are analyzed primarily as a function of the pressure at the pump. This is very misleading. If the nozzle size is constant and the pressure is increased, then the flow rate through the nozzle also increases. Therefore, the nozzle's power output has increased from both greater flow and higher pressure. In many of the cases it is the increase in total power that has improved the cutting performance. To get a true test of the effect of jet pressure, the nozzle size should be reduced accordingly to keep the nozzle power constant.

It is important to understand the distinction between nozzle pressure and jet power. Nozzle pressure is a measure of force per unit area. Most materials require a minimum nozzle pressure (threshold) for effective penetration. Jet power is a measure of the total amount of energy per unit of time which is being delivered by the jet. Large volume jets have a lot of power and can cut or clean many materials very effectively, even at low pressure.

It may be helpful to refer to Figure 1 which shows the relationships between, cutting effect and nozzle pressure for two sizes of nozzle.

Using the small nozzle and increasing the pressure from P1 to P2 will increase the cutting effect from E1 to E2. However, a larger nozzle could have achieved the same effect with more flow but at the lower pressure Pi. This is assuming that P1 and P2 are greater than the threshold pressure.

FRICTION LOSS

All of the piping elements in a waterblast system have some resistance to flow. Each of the elements contribute to the total pressure drop from the pump to the nozzle. The pressure drop depends on the flow rate and fluid viscosity. Unfortunately, the standard reference works on fluid flow typically deal with relatively slow velocities. They are of little value in analyzing waterblast systems. What is needed is a means of quantifying the potential for pressure drop in each element and a way to add them to give the total.

The authors have found it useful to employ the flow coefficient, Cv. This term is normally used to measure the flow restriction in control valves. The Cv for any device can be determined by measuring the flow rate and corresponding pressure drop through the device. Table 1 presents the measured Cv values for some commonly used elements in a waterblast system, and the resulting pressure loss from friction. In most tube cleaning jobs the lance rod is the major source of pressure loss due to its small diameter.

		PRESSURE DROP (psi)	
ELEMENT	Cv	@10 qpm	@20 qsm
QD Coupler 1/2"	8.5	1	5
Footdump w/f itt	5.0	4	25
Hose 1/2"x50 ft	1.32	60	230
Lance .3"x22 ft	.56	320	1300
Hose 1/4"X70 ft	.37	730	2900
Lance .18"x10 ft	.23	1900	7600

TABLE 1. THE FLOW COEFFICIENT, Cv, AND PRESSURE DROP FOR SEVERAL ELEMENTS.

One of the advantages in using Cv is the ability to combine several devices with different flow restrictions into a single quantity, independent of the flow rate. This allows entire plumbing systems to be compared for their pressure drop potential independent of the eventual flow and pressure conditions used. This is not possible with the Hazen-Williams or Darcy formulas

An empirical equation has been derived to predict the Cv for hose, pipe and tubing sizes. This eliminates the need to measure and tabulate the Cv for every size and length used. The equation predicts pressure drops that compare favorably with those given by the Hazen-Williams formula for pipes.

For a complete definition of Cv, and the formulas it is used in, please refer to Appendix A.

There is one important generalization that should be noted in regards to friction loss. Try to minimize the pressure losses through a system, this always improves the efficiency of your operation. This can be accomplished by: eliminating unnecessary or undersized connections, and using hoses and lances that are as short as possible and as large in diameter as possible.

OPTIMUM NOZZLE SIZE

Selecting the correct nozzle size to be used with a pumping system is critical to achieving high performance and efficiency. For some cleaning jobs the maximum cutting power is needed while on others a high nozzle pressure is necessary to cut the deposit. The nozzle size that can achieve either of these goals can be calculated once the pump, hoses, and lance size have been determined.

It is convenient to refer to a nozzle's size as if it were a single round orifice. Multi-jet nozzles have an equivalent single nozzle size that will pass the same flow of water at equal pressures. In general, a greater cutting effect will be achieved with nozzles that have few jets. This is because each jet is more powerful than if the same total power were divided among several smaller jets.

Maximum Power Nozzle

For any given plumbing system there is one nozzle size that will produce the maximum nozzle cutting power. This nozzle will create a flow rate and nozzle pressure that combine to yield optimum power output. Too large a nozzle allows a high flow rate that incurs excessive pressure losses due to friction through the system. Too small a nozzle causes a low flow rate, thus a low power outputs

In a well designed system, the pump will be capable of delivering sufficient flow and pressure to achieve the maximum nozzle power. It is useful to note in these cases the maximum nozzle power will be at a nozzle pressure which is 2/3 of the pump pressure. That is, if the pump pressure is 9,000 psi then using the optimum nozzle size will result in a nozzle pressure of 6,000 psi.

In some cases the calculated nozzle size will dictate a flow rate that is higher than the pump can produce. It is then necessary to resize the nozzle to use the maximum flow of the pump. The nozzle power then produced will be the maximum the pump can deliver through the plumbing system already defined The calculations necessary to determine this nozzle size are contained in Appendix A.

Maximum Pressure Nozzle

For cleaning very hard deposits it may be necessary to have as high a nozzle pressure as possible. We know the pressure drop through the piping system increases with higher flow rates due to friction losses. Therefore, a small nozzle size which restricts flow is desirable. However, enough flow must still be passed to yield a reasonable power output from the nozzle. A practical approach is to size the nozzle so as to achieve a nozzle pressure of approximately 90% of the pump pressure. This will yield a power output of about 75% of maximum. It is a simple process to calculate the size required once the piping system and pump are selected. Refer to Appendix A for details.

ECONOMIC BENEFITS

The proper selection of pump, plumbing and nozzle size is important to the success of every cleaning job. Differing conditions prevent any one system being effective on every job. Knowing exactly how the elements of a cleaning system interact to deliver cleaning power is very important in assembling an efficient cleaning system.

The ability to calculate the effect of different elements on the performance of the system reduces the amount of on-site testing required This can help avoid taking unnecessary equipment to the site and/or risking delay waiting for different equipment to be brought in.

An efficient cleaning system will have each of its parts sized so as to maximize the power output while minimizing the waste of over capacity. Using a pump that is larger than necessary costs more to operate, and wastes power and water. Unnecessarily long or small diameter hoses and lance rods cause pressure losses which reduce the cleaning rate. Undersized plumbing fittings or an excessive number of them have the same effect. Oversized nozzles allow excessive flow, low nozzle pressures and low power output. Undersized nozzles restrict the flow rate resulting in less than maximum power output.

When an operator is using the most efficient waterblast cleaning system available, he will maximize his potential to both: make a profit and keep a satisfied customer.

COMPUTER PROGRAM

The concepts and relationships presented in this paper have been organized into a computer program to enhance their useability. The computations involved in analyzing a typical cleaning system are not very complicated. However, they become very repetitive and involved during the course of a normal selection process. The computer frees the user from worrying about calculating correctly so he can concentrate on designing the best cleaning system.

The program is arranged into two parts. The first part concerns the overall system: pump, hoses, lance rods, etc. The second part deals with nozzle design: number, angle and size of jets. This division is a natural extension of the way a cleaning system is normally assembled. The two parts of the program are capable of being used independently offering greater flexibility. They share basic system information for easier and faster repetitions.

Part one, "THE SYSTEM", requires the user to define the elements in a cleaning system, then the program will display the resulting operating conditions: pump pressure, flow rate, and the dimensions of the plumbing elements or their flow coefficient values. All of the values are displayed and the user can repeatedly change the parameter of interest. This part will also calculate the nozzle size that produces the maximum power output for the defined system, and display the nozzle's diameter, pressure, and power output. The program will also calculate and display a nozzle size that produces a nozzle pressure of 90% the pump pressure.

<u>Example</u>

The following represents the display of information using the computer program

SYSTEM DESIGN

PUMP PRESSURE: 10,000 psi
No. of Elements: 4

ELEMENT	ID	LENGTH	Cv
1	.5	150	.79
2	0	0	5.0
3	.5	50	1.37
4	.21	10	.35

TOTAL SYSTEM Cv = .31

NOZZLE TYPE	DIAM (in.)	FLOW (gpm)	PRESS (psi)	POWER (hp)
Max Hp	.085	17.9	6667	70
Max Press	.058	9.8	9000	51
Your Choice	.065	12.0	8500	59

Part two, "THE NOZZLE", allows the user to design a multi-orifice nozzle by entering the number, size, and angle of orifices. The program will calculate and display: the flow rate, power output, pressure and thrust, for individual jets and for the total nozzle. By adjusting the design of the nozzle in this way the resulting change in performance can be immediately understood.

<u>Example</u>

NOZZLE DESIGN
NO. OF JETS: 4

JET No.	DIAM (in.)	ANG (deg)	FLOW (gpm)	POWER (hp)	THRUST (lbs)
1	.040	45	4.4	21.8	21.3
2	.040	45	4.4	21.8	21.3
3	.023	-45	1.5	7.2	-7.1
4	.023	-45	1.5	7.2	-7.1
TOTAL	.065		11.8	58	20.0

APPENDIX A

The following is a compilation of the formulas used in waterblast calculations. It is not the purpose of this paper to perform a rigorous theoretical study. Rather it is an attempt to provide a simple means of dealing with the principles involved in high velocity fluid flow.

<u>Nomenclature</u>
HP - Hydraulic power
P-pressure
P - Change in pressure
D- Diameter
Q-Flow rate
T-Thrust
L-Length
Cv- Flow coefficient
K- A constant
U- Velocity of jet
ρ- Density

Subscripts
n - Nozzle
p - Pump
t - Total

Hydraulic Power

$$HP = K1 * P * Q \qquad (1)$$

When HP is in horsepower, P is in psi, and Q is in gpm, this becomes,

$$HP = (P*Q)/1714 \qquad (2)$$

Definition of Cv

Cv is defined as the flow of fluid in gpm through a device with a 1 psi pressure drop across it.

$$Cv = Q*(\rho/62.4*\Delta p)\text{^}1/2 \qquad (3)$$

For water at approximately 70 F this can be rewritten,

$$P = (Q/Cv)\text{^}2 \qquad (4)$$

The total Cv for a system composed of several elements is,

$$(Cv\ total)\text{^-}2 = (Cvl)\text{^-}2 + (Cv2)\text{^-}2 + \ldots (Cvn)\text{^-}2 \qquad (5)$$

Calculatinq Cv

An equation has been derived empirically from test measurements by the authors, to predict the Cv for tube, pipe and hose.

$$Cv=53 * D\text{^}2.5/L\text{^}1/2 \qquad (6)$$

This equation yields Cv's that are within 5% of the measured values for a large range of plumbing elements commonly used in waterblast cleaning. Water temperatures of + 30° from 70°F caused only a + 2% change in measured Cv. When testing relatively unrestrictive devices it is necessary to use only the pressure drop resulting from friction loss and not the velocity increase.

Optimum Flow

For any given pump pressure and system Cv, there exists an optimum flow rate that will produce the maximum nozzle power. This value of Q can be solved for directly. From earlier discussion we know that the nozzle

$$HP_n=K_1*P_n*Q \qquad (1)$$

and the pressure loss through the plumbing system is;

$$\Delta P = (Q/Cv)^2 \qquad (4)$$

By definition the nozzle pressure is,

$$P_n=P_p-\Delta P \qquad (7)$$

Substituting eqn. 4 and 7 into 1 gives;

$$HP_n = K_1*Q*P_p - K_l*Q^3*Cv^{-2} \qquad (8)$$

Taking the first derivative with respect to Q and setting equal to zero to find an inflection yields:

$$O = K_1*P_p — K1*Q^3 *Cv^{-2} \qquad (9)$$

Solving for optimum Q;

$$Q = (P_p/3)^{1/2} * Cv \text{ (for max. HP)} \qquad (10)$$

Furthermore, substituting eqn 4 into eqn 9 and solving for the pressure drop gives,

$$\Delta P = P_p/3 \text{ (for max. HP)} \qquad (11)$$

<u>Jet Velocity</u>
For U in feet/sec and P in psi;

$$U = 12.186 * P_n^{1/2} \qquad (12)$$

<u>Flow Rate</u>
For orifice diameter D in inches, P in psi, and Q in gpm;

$$Q = 29.91 * D^2 * p^{l/2} \qquad (13)$$

<u>Jet Thrust</u>
For T in lbs-force, D in inches and P in psi;

$$T=(\pi/2)*D^2*P \qquad (14)$$

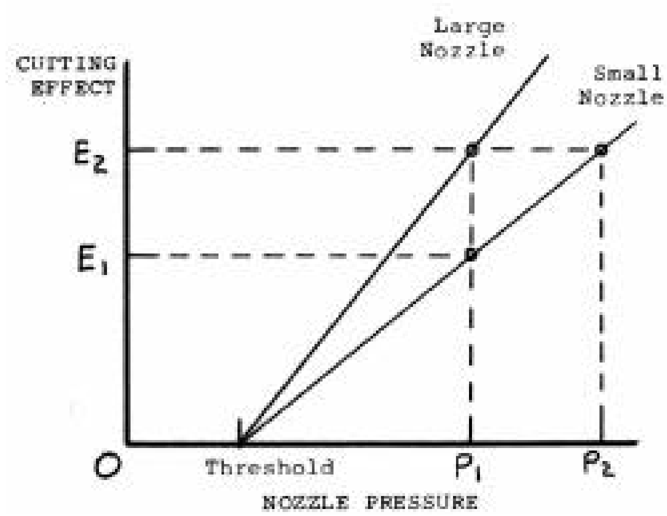

Figure 1. Cutting effect as a function of nozzle size and pressure.

DISCUSSION

NAME D. Eddingfield
COMPANY: Southern Illinois University

QUESTION: "Why have you chosen to ignore two hundred years of work on internal flows? Namely, the work of Moody on pipe friction and the work on minor losses sequential in the form:

$$KV^2/2g?$$

ANSWER: What we are concerned with are not minor losses. Some waterblast cleaning systems loose over 50% of their energy to friction losses in the plumbing system between pump and nozzle

We prefer using Cv values from empirical equations for several reasons.

1. They eliminate the guessing of what friction factor to select;
2. Cv values can be obtained by simple flow tests for any plumbing element, not just tubes or pipes;
3. For several plumbing elements in a series their Cv's can be combined and the total system flow potential can be easily computed - independent of the flow rate; and
4. The use of Cv's allows equations to be developed and solved for optimization of the system performance.

NAME: Mohamed Hashish
COMPANY: Flow Industries

QUESTION: "Do you have results that substantiate the statement that greater cutting effect will be achieved with nozzles that have fewer jets? I think there should be an optimum number of jets for a given power that will produce a maximum cutting effectiveness.

ANSWER: Yes, we've been involved in numerous projects where nozzles designed with fewer jets have exhibited superior performance. Theoretically, there should not be any difference due to the number of jets -- cutting effect is simply proportional to the power applied. However, in practice there are several reasons why a few large jets are preferable over many small jets (1) Small jets deteriorate quickly with standoff distance. Given the same nozzle pressure and flow rate more power will reach the target material if concentrated in a single jet rather than divided into several small jets. (2) Larger jets are easier to fabricate of higher quality. Small nozzles generally have lower discharge coefficients than larger nozzles made using the same technology. Wall roughness and entrance irregularities induce turbulence into the jet stream proportionately more in small nozzles than large ones. (3) Larger jets are more reliable because they are less susceptible to plugging and damage by particles in the flow stream.

CONSIDERATIONS IN THE COMPARISON OF CAVITATING AND PLAIN WATER JETS

David A. Summers,
Curators' Professor of Mining Engineering; Director, Rock Mechanics and Explosives Research, University of Missouri-Rolla, Rock Mechanics and Explosives Research Center, Rolla, Missouri.

ABSTRACT

Water jet cleaning of surfaces can be achieved be several different methods. A brief discussion is given on the advantages and disadvantages of plain, fan, spray, shrouded, and core cavitation systems.

INTRODUCTION

High pressure water jets are used for cleaning on one of two basic configurations. Conventionally a round jet issues from the normal nozzle (Fig. 1) or, by slicing a V or U shaped notch across the face of the throat section, a thin fan-shaped spray of water, referred to as a spray jet will be created (Fig. 2).

In similar manner, the use of cavitation has, over the last ten years, become a practical way of utilizing the energy contained within a stream of water. To induce cavitation, two different types of system can be envisioned. In the first, a water jet is simply placed under water and cavitation bubbles are induced by the shear motion of the high speed jet relative to the ambient water surrounding it. The cavitation bubbles are subsequently entrained in the high speed jet flow, and collapse on the target surface (Fig. 3). In contrast it is also possible by placing a flat eroded probe in the central section of the nozzle with the flat ended tip just past the point of maximum convergence of the acceleration section of the nozzle, to induce cavitation within the central core of the ensuing water jet (Fig. 4). Thus, four different conditions can be considered in comparing the mechanisms for water jet cleaning. It is the purpose of this paper to give some of the benefits and disadvantages of each of the different mechanisms.

COMPARATIVE SYSTEMS

The use of the different techniques for water jet cleaning are in part constrained by the conditions under which the surface to be cleaned exists. For example, if the target surface lies under water, for example, the oil rig structures in the Gulf or the hulls of ships, then which ever method of water jet generation is used, the high speed action of the water will induce cavitation around the jet stream regardless as to whether or not it is deliberate. Within this constraint, one can however, admittedly define a certain number of parameters. For example, Dr. Conn has indicated that a Leach & Walker nozzle is one of the most difficult to cavitate under water. The reason for this might be that the Leach & Walker nozzle gives a better jet structure. In the early phases of the passage of the jet from the nozzle, the pressure gradient across the water jet is very steep and exists in a very narrow acceleration band on the edge of the jet, while within the central section of the jet the velocity is relatively constant. Thus the zone of the water jet in which cavitation can be induced is minimal.

The water jet which issues from other nozzle designs will have a constantly changing velocity profile across the jet. It is likely therefore that, underwater bubbles will be formed in much the same way as, in a conventional water jet system in free air, water droplets are formed. The resulting bubbles have some of the attributes of a water jet droplet. This is, the collapse of a bubble on a target surface can give rise to very high impact pressures on the order of those which are evident when a droplet hits a flat surface. The structure of the target surface is however different, the structure of the damage pattern is different and the level of pressure induced is also not equal. Nevertheless, the two phenomena have a certain initial similarity given that a very small, highly intense pressure pulse is directed at the rock surface. In both cases the energy contained in the droplet or within the jet itself, cavitation bubble is sufficiently small that only a surficial crack is generated without any great damage in the short term beyond this surface. Where the target is however flexible or fluid, then in either case the ability of the target to deform under impact makes the surface more resilient and thereby less damaged by the intensity of the original impact phenomena.

For example, if one is cleaning a greasy surface then neither cavitation nor droplet impact will do an efficient cleaning of the surface. Either system will rather tend to move the material around. In order to remove material completely from the surface, it is necessary that sufficient energy enter the flaw created by the impacting short duration event, such that the flaw will grow and if this flaw penetrates to the boundary layer between dirt and the underlying substrate, then the sequential water jet pressure, behind the initial impulse will be sufficient to grow the split and peel off material.

In long term cavitation erosion, where one is eroding the target at a very low traverse velocity, the migration of cavitation bubbles to a pre-existing flaw, and its exploitation, to the point where large material removal occurs, has been demonstrated. But on a smaller scale, this event will also occur but must, in order to occur in an acceptable time frame, be accelerated by the pressure of the water behind the initial droplet impact or cavitation bubble impact (Fig. 5). Thus flaws in the surface cover are exploited more rapidly and in an acceptable time frame rather than if the cleaning were done by impact pressure alone. The reason that this amount of emphasis on the mechanics of water jet cleaning is given is because, as has been discussed previously, water jet droplets which rapidly disintegrate into a fine spray do not have this highly contained back up energy to exploit any flaws which they generate in the material. In contrast a conventional round jet, because it has very little pressure differential across its width may not have sufficient energy to create the flaws in the target surface which are a pre-requisite for effective material removal.

Under these circumstances, cavitation induced with a conventional round jet can create the very small flaws in the target surface, which, when combined with the major jet energy of the jet stream, are sufficient to rapidly accelerate the erosion process. One example of this can be taken from a different area. Specifically, if one is cutting granite that will, under normal circumstances be effectively cut at a pressure of 15 ksi with a conventional jet, but when cavitation bubbles are produced within the jet stream this pressure can be lowered to perhaps 11,000 psi. This is because there is now a greater flaw density in the target material and the jet can grow these flaws faster at a lower pressure, removing more material.

If one accepts this idea and moves toward the design of an effective cavitation system, certain additional features need to be considered. It has been found (Fig. 6) that the width of application of a cavitating jet is approximately 3 times that which would occur where cavitation was not present. Thus, as well as being able to reduce the required jet pressure, cavitation can extend the cleaning width of the jet beyond that which otherwise can be reached. Such energy gains are not, however, without cost, a point not always fairly appreciated.

When one generates cavitation by driving the water jet through an existing fluid, one is immediately reducing the effective range of that jet system to an extent, dependent on the pressure in the ambient fluid. At the same time when cavitation occurs by submerging a water jet, one is subjecting the operator of the cavitating lance to a dangerous environment. Specifically the frequency of noise generated by a cavitating water jet when submerged is of a high frequency. This can have extremely dangerous side effects on a diver. The cavitation system also has the same type of problem that the water droplet system has, only to a more exaggerated degree. The cavitation bubble only exists in a region of low pressure and as the water jet flows from the nozzle it pulls the bubble from a low pressure zone into a zone of ambient or higher pressure. The bubble then collapses. This means, in realistic terms, that the zone of influence of a cavitating jet is extremely short. In research which was carried out at Rolla, the maximum intensity zone for a cavitating system has been measured at approximately 1/2 in. (Fig. 7). The total zone of impact damage for a nozzle, 0.04 in. diameter, may stretch to between 1 and 2 inches and, in soft material can extend 3 inches, or some 75 nozzle diameters. The depth, however, at which the cloud does damage, and this mainly refers to submerged jets, is tightly controlled by jet pressure, fluid pressure, nozzle diameter and time of impact.

There is, in this regard, a contrast between the different approaches, the submergence of the cavitating jet under water and the induction of cavitation in the central sector of the jet. For several reasons, the latter would appear to be the more promising technique. The reason for this is that the bubbles in the former case are created by the shear on the outer surfaces of the moving water jet in ambient water. Therefore the surrounding pressure is likely to be somewhat less than is the case where the cavitating bubbles are generated in the center of the jet and thereby exposed to the greater impact pressure where that jet is decelerated upon the target surface. Secondly, the flow of the jet, itself upon impact will protect the surface from cavitation bubbles on the outer edge but will confine and direct internal bubble collapse onto the rock surface. The bubbles will collapse in the middle of the zone of high pressure water, which can thus exploit the cracks in contrast to cavitation on the outer edges of the jet where the flow of water across the surface will to a degree, protect the surface from exposure to the cavitating bubbles. This has not been given a great deal of attention. Further, since the cavitation bubbles are carried in the middle of the stream and carried at the maximum jet velocity rather than those on the outer edges which are carried at the much lower velocity, the range of cutting ability will be greater. Holes deeper than 6 inches have been cut with a cavitating water jet system where cavitation has been induced in the center of the water jet, rather than the inch or so created, in a longer time, with submerged cavitation.

CONCLUSIONS

Water jet cleaning systems require a high energy transfer from the pump, through the water, to the target surface. In conventional water systems a round jet is capable of carrying energy to a much greater distance than a fan or spray jet.

Cavitating systems may be of advantage close to the nozzle, where the collapse of bubbles may create sufficient weakness planes in the target that lower pressures can be used, than would otherwise be necessary. The limitations on their use are, however, similar to those of the fan jet, in that, at greater distances from the nozzle their effectiveness decays very rapidly.

ACKNOWLEDGEMENTS

This paper is built around results from a number of research projects carried out at the University of Missouri-Rolla. I am particularly indebted to Dr. Marian Mazurkiewicz, Dr. Paul Worsey, Mr. J. Blaine, and Mr. L.J. Tyler who helped in this work, and to Mr. B. Hale, recently of the University staff who built our cavitation cells. I would also like to thank Mrs. Vicki Snelson for typing this paper.

REFERENCE:

1. Summers, D.A. and J. Peters, "The Effect of Rock Anisotrophy on the Excavation Rate in Barre Granite," Second International Symposium on Jet Cutting Technology, Cambridge, U.K., paper H5, April 1974.

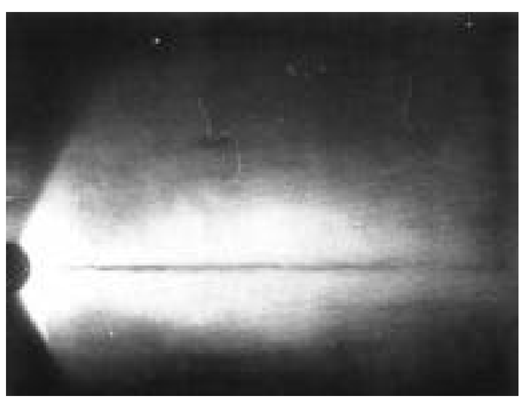

Figure 1: Backlit photograph of a 20,000 psi water jet issuing from a 0.010 inch diameter sapphire nozzle.

Figure 2: Fan jet issuing from a nozzle at l,000 psi, taken by a backlit flash technique.

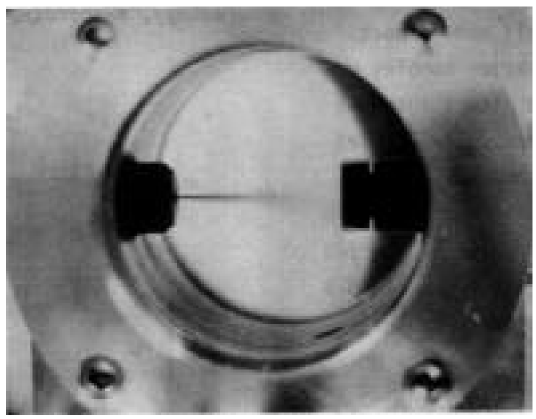

Figure 3: Cavitating jet issuing into the flooded, and pressurized chamber of a Lichtarowicz cell.

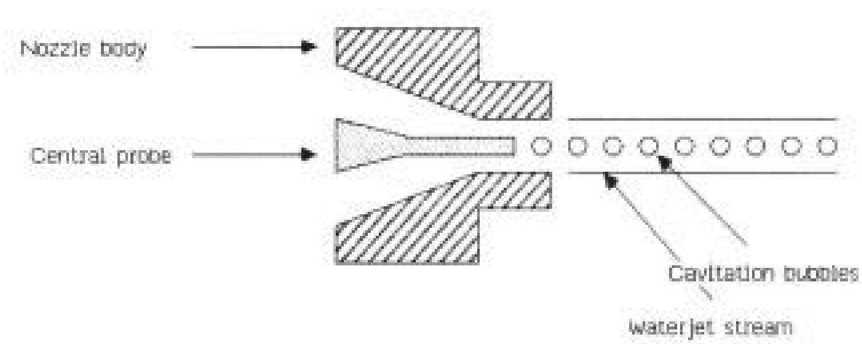

Figure 4: Stylized version of a central-probe nozzle, with ensuing centrally generated cavitation bubbles.

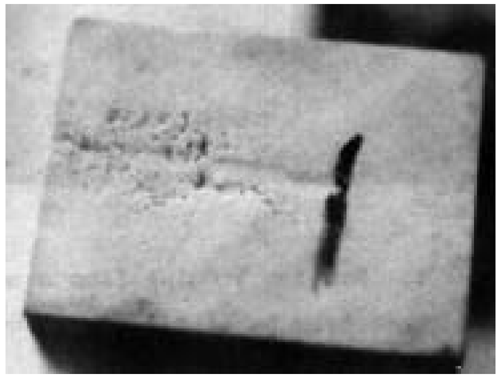

Figure 5: Path of a cavitating jet across d dolomite sample, the deeper narrower jet path can be contrasted with the wide pitted zone created by the cavitation attack.

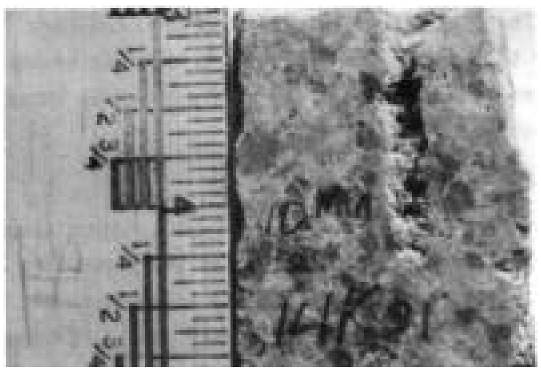

Figure 6: Sample of granite attacked by a transverse cavitating water jet, at]4,000 psi pressure and 0.005 cavitation number, showing the small and narrow zone of maximum cavitation damage.

DISCUSSION

NAME: John Griffiths
COMPANY: Sheldon Birmingham England

QUESTION: "Do you agree that in part the relative inefficiency of a fan jet, when compared with a round jet is due to the form and method of production of conventional fanjets? With fan jets of a more exact form, where care is taken, on the shape and surface finish, destructive power is exhibited at distances up to 4 times shown in your paper."

ANSWER: While it is true that considerable improvement in jet performance can be achieved where care is taken in the construction of the nozzle, it must be recognized that the basic form of the fan issuing from a spray nozzle is such that as the sheet leaving the nozzle moves further away, it will, at the same time get thinner. Thus a point is reached, as a function of the flow rate through the nozzle, at which the sheet is no longer thick enough to resist the surface tension forces and it will therefore disintegrate into droplets. This phenomenon is unlikely to be much affected by changes in the nozzle geometry, beyond a certain point, although it will of course be markedly affected by the angle at which the jet is directed away from the nozzle.

NAME: Larry Pater
COMPANY: Dravo

QUESTION: "Does the shape of the tip of center body probe strongly affect the generation of cavitation bubbles?"

ANSWER: The shape of the centerbody has a substantial affect on the induction of cavitation bubbles into the jet stream. From our experience a flat ended shape has been the best, although our study has been by no means exhaustive. I would suggest that there may be better shapes, information on these might well be obtained from studies of cavitation flows induced in conventional fluid streams.

WATER JET CLEANING SPEEDS - THEORETICAL DETERMINATIONS

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ABSTRACT

In an effort to improve state-of-theart water jet cleaning techniques as used in oilwells, a study was undertaken in 1980, to evaluate certain known parameter and their relationships. Using theoretical, observed, and empirical field data, a statement based on rotary applications was developed. Cleaning speeds and effectiveness were increased, and job results, in economic terms, were significantly improved in most cases. This paper deals not so much with the results of that development, as it does with the suggested use of those relationships to a linear application.

BACKGROUND

Oilwell water jet cleaning is a patented process (Hutchison, 1973 & Zublin, 1982) whereby directionally controlled hydronic energy is applied through very small jet nozzles to clean liner slots or perforations, and well screens that are partially or completely plugged.

During normal operations the jets are rotated, as well as pulled up or lowered, through the interval to be cleaned. The resulting overlapping spiral track created by the jet streams removes most of the plugging material contacted.

DEVELOPMENT

In early 1971 tank tests were performed (Hutchison, 1973) that clearly demonstrated the effective cleaning width of a water jet with respect to nozzle size ("D") and standoff distance ("X"). It was shown that for a range of short dwell times or single pass operations, effective cleaning was not accomplished for "X/D" ratios in excess of 10. Limited jet sizes and narrow ranges of pressure drops across the jets were explored. Track cleaning speeds (the reciprocal of dwell time) covered a goodly range and were not considered critical.

In utilizing this information and other related data, this author explored the results and concluded that dwell time needed to be evaluated further. Additionally it was expected that dwell time for effective cleaning, with respect to differing deposited plugging materials, would be different. The following analysis furnished an understanding of travel rate requirements for effective cleaning under varying field conditions.

Fluid Rate and Velocity

Early workers (Pittman, Harriman and St. John, 1961) developed a relationship between the flow rate, jet diameter pressure drop and fluid density as follows:

$$Q = 69\ D^2(P/e)^{1/2} \qquad (1)$$

where

Q = Flow rate - gpm
D = Jet Orifice diameter - inches
P = Pressure drop - psi
e = Fluid density - lbs/gal.

It was noted that "e" is not limited to Newtonian fluids. The investigation was limited to water without solids ranging from 8.3 to 8.7 pounds/gallon.

Converting to velocity of fluid through the jet from gallons per minute yields:

$$Q=V \quad D^2/.408 \qquad (2)$$

and solving for V

$$V = .408\ Q/D \text{ - ft/sec } (3)$$

then combining Equation (1) with Equation (3) yields:

$$V = 28\ (P/e)^{1/2} \text{ - ft/sec } (4)$$

<u>Impact</u>

Assuming that the Impact of a fluid stream is equivalent to Kinetic Energy per unit time, one obtains:

$$\text{Impact} = MV^2/2 = WV^2/2g \qquad (5)$$

where

$$M = \text{mass} = W/g$$

W = Weight of fluid used in one (1) second
g = Gravity (constant = 32.2 ft/sec2)

Then by substitution from Equation (4) one obtains:

$$\text{Impact} = 14.1\ D\ P^{3/2}/e^{1/2} \qquad (6)$$

<u>Track Speed</u>

The Track Speed of the jet track for oilwell applications is expressed as follows:

$$V_t=((V_{th})^2 + (V_{tv})^2)^{.5} \qquad (7)$$

where

V_t = Track Speed - ips
V_{th} = Horizontal component of V_t - ips
V_{tv} = Vertical component of V - ips

Substituting field terms: (Zublin, 1982)

$$Vt = ((RDh/19.09)^2 + (Vv/5)^2)^{.5} \qquad (8)$$

where

R = Rotational rate - rpm
Dh = Internal diameter of the pipe to be cleaned - inches
Vv = Vertical Travel rate – fpm

<u>"N" Factor</u>

In order to obtain Cleaning Energy Flux ("CE"), 'Vt" must be related to impact. Combining Equations (6) and (7) will result in a dimensional inaccuracy.

Testing shows that an average width cleaning path for a jet whose nozzle diameter "D" is about 0.030" and whose "X/D" ratio is between 6 and 10, is 1/ 4 inch. Further because of the imposed interlocking relationships (Zublin, 1982) between the horizontal and vertical components, the jet track center to center distance was held at 1/8", thereby yielding an Overlap Factor of 2.

In the mathematical analog the term "N" is used as follows (Zublin, 1982):

$$N = \frac{\text{No. of Cleaning Tracks} \times \text{Overlap Factor}}{\text{Linear Inch}}$$

which is dimensionally equal to:

$$N = 0f/Tw \qquad (9)$$

where

Of = Overlap Factor (if no Overlap = 1)

Tw = Cleaning Track width - inches (see Fiqure No. 1)

As the applications discussed herein are visible and operator or machine controlled, Of = 1, then N becomes the reciprocal of Tw.

<u>Power Efficiency</u>

The relationship between Power and "X/D" is expressed as follows (see (Chart No. 1):

$$Px=Po213/(x/D)^3 \text{ and } Po \geq Px \qquad (10)$$

where

Px = Power at Target
Po = Power at Jet
X = Standoff Distances - inches
D = Jet Diameter at Origin – inches

Again, not limited to Newtonian fluids. (Brown & Loper, 1961 and Forstal & Gaylord, 1955).

Bernouli's laws define the lower limits of "X/D". There appear to be no upper limits to this relationship, other than those governed by economics.

Polymer

Subsequent experimentation by this author and others (Private Communication, 1980) demonstrated that by the addition of long chain high molecular weight polymers (such as "SUPER WATER") to the fluid, the jet stream could be made to maintain its departure cross section for a greater distance, as well as increase target impact (NPRA 1978 and Private Communication 1980).

With the addition of polymers a statement similar to Equation (10) was suggested from preliminary test data as follows (see Chart No. 2):

$$P_x = P_o \, 2130/\left(\frac{X}{D}\right)^3 \qquad \textit{and } P_x \le P_o \qquad (11)$$

Clean ng Energy

Combining Equations (6), (7) and (9),

CE = 14.1 D^2 P^.75 N/e^.5 Vt (12)

where

CE = Cleaning Energy Flux (Lb - Ft/in .)

Equation (11) (Zublin, 1982) represents the "CE" values developed at the jet. In order to determine the value of "CE" at the target, power efficiencies must be considered. Using Equations (10) or (11) and rewriting we have:

Px/Po = C/ (X/D)^3 = E_f (13)

Where

C = a constant (213 or 2130).

E_f = system efficiency in %age.

Operating Speed

Now rewriting equation (12) to solve for Vt and combining with Equation (13) yields:

Vt=14.1*D^2 P^3/2 N*Ef/e^.5 CE (14)

as the linear expression for jet cleaning travel rate in inches per second (Zublin, 1982).

DISCUSSION

While equation (14) relates all of the variables and is dimensionally correct, there are some parameters that need to be explored.

"CE" Values

Numerous observations have been made on a variety of different deposited materials, from the standpoints of surface bonding, internal bonding, material strength, hardness, lamination susceptibility, and impact failure. To the best of this author's knowledge, none of these data have been refined or observed under controlled conditions. Nevertheless, certain obvious relationships appear to hold. For example, barium sulfate is obviously harder and more tenacious than calcium carbonate; and calcium carbonate is more tenacious and harder than a complex deposit consisting of hydrocarbons, fine silicates, water scales, and so on. Therefore based on more than casual observation and data accumulated for over a decade, a Table showing Estimated Hydronic Energy values ("CE") was developed.

Table No. 1 shows these values and their relationship to plugging materials commonly found in oilwells. The numbers shown have been in use for about 2-1/ 2 years and have generated successful operating rates. Specific evidence for the correctness of these values is not available and needs to be developed.

Efficiency

The "E_f" ratios discussed are for true water-like fluids with an "e" value ranging from 8.3 to 8.7 with no suspended solids. It is obvious from inspection of Equation (5) that any increase in "W" may result in an increase in Impact values. Increasing "W” by means of suspended solids will result in different "E_f" values for similar operating conditions. Here again, thorough exploration has not yet been conducted, at least within this author's knowledge.

"N" Values

As shown in Table No. 2, the values for "N" are known for a very narrow jet diameter ("D") range; 0.029" to 0.033". The size restriction was because of the planned use of multiple jets and the economic requirements of horsepower. Work with additional jet sizes would increase the useful value of the data presented herein.

CONCLUSIONS

Assuming that Equation (14) is reasonably accurate, Table No. 2 was developed to suggest how the linear travel rate could be applied to given situations. Review of Table No. 2 shows that certain values listed yield large equipment sizes and costs that may be uneconomic. It is suggested that the information contained herein be utilized to further investigate the theories presented and verify the relationships.

ACKNOWLEDGMENTS

This author was employed by DownHole Services, Incorporated during the time some of the relationships were developed. Some of the concepts were conceived through the joint efforts of the author and technical staff at DownHole. This author is indebted to the management of DownHole's successor company for making certain data available. This author is also indebted to Dr. Glenn Howells, without whose encouragement and aid this paper might not have been written.

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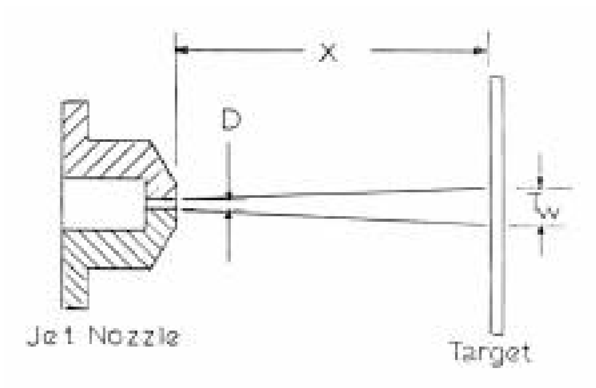
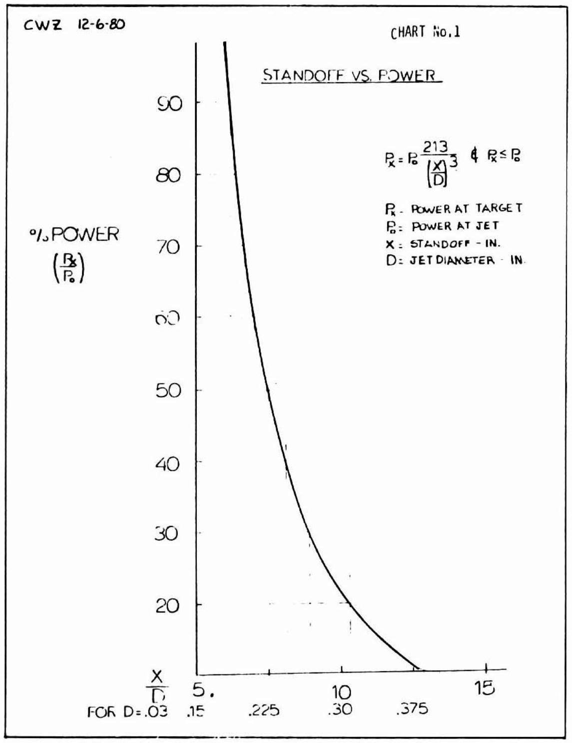


Figure 1. Nozzle and target (CWZ 15 Mar ’83)



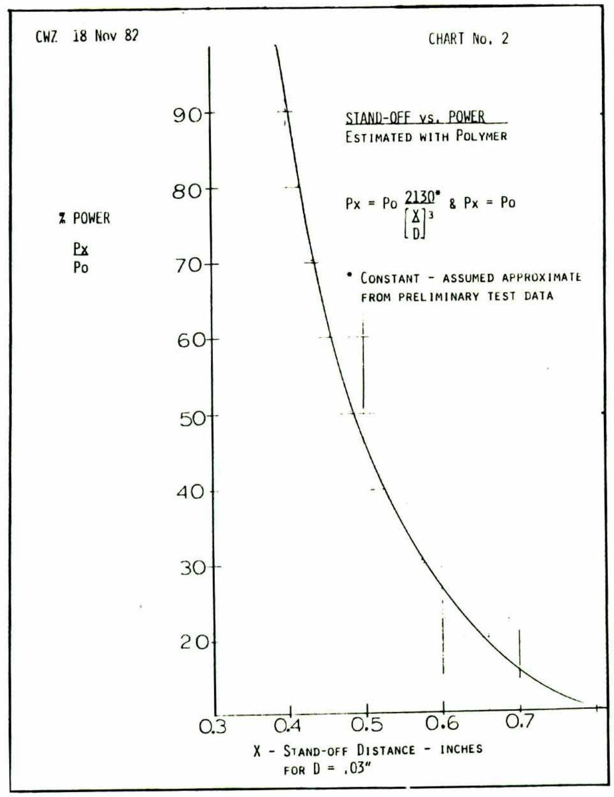


Chart 1. Standoff vs Power

Chart 2. Standoff vs Power (with polymer)

MATERIALS	ESTIMATED ENERGY FLUX (TE/IN^2)	
BARIUM SULFATE	7000	
SILICATES	6000	
CALCIUM CARBONATE	5500	
CALCIUM SULFATE	4500	RANGE
CARBONATE - SULFATE - SILICA COMPLEXES	3800	
WATER SCALES AND HYDROCARBON COMPLEXES	3200	
COAL TAR	3000	
COKE WITH OR WITHOUT COMPLEXES	2500	RANGE
WAX WITH OR WITHOUT COMPLEXES	2000	
PARAFFINS	1200	
SLUDGES	1000	
THIXOTROPIC MATERIALS - MUD, JEL	800	RANGE
NON THIXOTROPIC MATERIALS	500	

Table 1. Estimated hydronic energy values needed to clean or remove various deposited materials in oil wells. (CSZ 18 Nov '82)

JET CLEANING SPEED AND OTHER DATA

IN. D	N	IN/SEC V_T	IN. X	FT^2/MIN	MIN/FT^2	GPM Q	HHP
.033	4.0	1.258	.25	0.131	7.60	2.25	9.80
.039	4.4	1.758	.29	0.183	5.46	3.15	13.78
.047	3.2	2.043	.30	0.266	3.75	4.58	20.04
.055	2.6	2.273	.41	0.364	2.74	6.27	27.44
.061	2.6	2.796	.46	0.448	2.23	7.72	33.78
.067	2.3	2.984	.50	0.541	1.85	9.31	40.74
.078	2.3	4.044	.59	0.733	1.36	12.62	55.22
.086	2.0	4.275	.69	0.891	1.12	15.34	67.12
.107	2.0	6.617	.80	1.378	0.72	23.75	103.92
.125	1.7	7.676	.94	1.881	0.53	32.41	141.82

WHERE: D IS TAKEN AT STANDARD AVAILABLE SIZES

N IS ESTIMATED EXCEPT FOR D = .033

X IS SOLVED FOR X/D = 7.5 (SEE CHART NO. 1)

FT^2/MIN = V_T/N (.4166)

HHP = HYDRAULIC HORSEPOWER = QP/1714

AND

P = 7500 PSI , E = 8.3 LBS/GAL , E_c = .5 FOR X/D = 7.5

CE = 5500, AND NO POLYMER IS USED.

AND THE USE OF EQUATIONS (1), (9) AND (14).

Table 2. Jet Cleaning Speeds and other data.

DISCUSSION

NAME: W. G. Howells
COMPANY: Berkeley Chemical Research

QUESTION: "Mr. Zublin, what methods are used to determine the effectiveness of your downhole cleaning procedure?"

ANSWER: There were two methods used to determine downhole effectiveness. One was visual, the other was economic.

Firstly, the relationships between rotational speed and vertical speed that were used, previously recommended by the orginators of the process, were detailed then, based on the analysis made, the change in these relationships were detailed and reviewed. Then both the new and old systems were tried on a liner *in-situ*.

The liner was retrieved from the hole and visually inspected. It was easy to establish that the new method, although consuming less time, was 3 to 4 times more effective than the previously used technique.

The economic method was a bookkeeping job. We kept track of the number of jobs done throughout a given period and for whom. We then recontacted the clients to determine what increase in production, if any, and compared the net cost recovery to the cost of the job. This then gave us an average time to cost recovery. Using the data that three months cost recovery was effective economics for any oil well, we then divided the work up between wells with cost recovery of 90 days or less and 90 days or more.

History showed us that when utilizing this approach the older technique showed 58% of the work done to have cost recovery in less than 90 days. With the new technique the data showed that the cost recovery in less than 90 days was better than 83% of all wells done.

One of the fallacies in the above technique is that it is entirely possible that the selection criteria utilized by our customers prior to using our new services improved with time. Therefore, we were used under conditions in which the service was more effective. Ignoring that, it would seem that the newer methods were more effective. If I were to hazard a guess, I would say that the effectiveness would be reduced to somewhere in the ballpark of 75%, allowing for more efficient selection criteria.

NAME: Mohamed Hashish
COMPANY: Flow Industries

QUESTION: "The impact force of a jet (i.e., quasi-static force) is not kinetic energy per unit time as equation (5) gives in conjunction with Figure (1). The force is the rate of change of momentum = WV where W is the fluid flow rate and V is its impact velocity. This might be reason for the statement you made in the conclusion.

ANSWER: The utilization of the statement WV in lieu of the statement WV /2g yields a difference in dimensions. WV gives a dimension in pounds which is Force. WV2/2g gives a dimension in Energy. If the Force statement is to be used then the relationship shown in equation (14) is incorrect, because P instead of being related to the Vt by the three halves power, is then related directly as P.

Prior use of the Energy Statement (Equation (14)) in subsurface cleaning applications has yielded good economic results. This is not to say that changing the relationship in accordance with your recommendations would yield equally good results. It is only that it has not been analyzed from the Force standpoint.

Based on the observations we have made, and operating within the very narrow operating framework of pressures, jet diameter sizes, and stand-off distances, submerged on in air, there is good reason to assume the relationship between the Vt and P is that of the Energy relationship.

I appreciate very much your suggestion and hope that there will be the opportunity to determine which of the two statements would be most accurate.

CLEANING AND CUTTING WITH SELF-RESONATING PULSED WATER JETS

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Virgil E. Johnson, Jr. Gary S. Frederick
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ABSTRACT

A new type of pulsed water jet, which uses principles of self-resonance to create pressure fluctuations, is now being developed. The advantages of pulsing to improve the erosive action of a jet—by interrupting the flow, hence using the water hammer impact stress created by each individual slug—have long been appreciated by the users of water jets for cutting and cleaning applications. Of several self-resonating pulsed jet, or "SERVOJET", concepts, a design incorporating a Helmholtz resonating chamber, tuned to drive an organ-pipe segment, has proven feasible for in-air interruption of a water jet. Experimental results for several cleaning and cutting applications are described.

INTRODUCTION

<u>Motivation</u>

The increased erosivity afforded by causing a water jet to break up into a series of water slugs has long been recognized. Such interrupted or pulsed jets offer the following advantages, relative to steady-flowing jets, for either cleaning a substance from a substrate or cutting into a bulk material such as a rock:

1. Larger impact stresses, due to the water hammer pressure, which enhance the local erosive intensity,
2. Larger outflow velocities across the surface being cleaned (or cut) thus providing an increased washing action, (or opening cracks and flaws in a bulk material for cutting applications),
3. Greater ratio of impacted area per volume of jetted water, thus exposing larger areas of the surface to the water hammer pressure,
4. Cycling of loading; this promotes unloading stresses which may enhance the process of de-bonding the substance from the substrate, or fracturing the bulk material being cut, and
5. Short duration loadings, which tend to minimize energy losses within either the substrate being cleaned or the bulk material being cut, and hence increase the material removed per input energy.

A variety of mechanical techniques have been used to interrupt water jets. There have been external, rotating disks containing slots, holes, or sprockets (Summers (1975), Lichtarowicz and Nwachukwu (1978), Erdmann-Jesnitzer, et al. (1980)); an internal, spinning slotted rotor (Nebeker (1981); and an internal, piezoelectric transducer (Danel and Guilloud (1974)). Although each of these methods produced a series of water slugs—and improved erosion relative to steady jets was consistently observed—the frequency of interruption was always well below optimum as discussed below. Also, the complexity and short component life-times associated with these mechanical methods has prevented the development of a practical, high-pressure interrupted water jet system. The possibility of achieving very high frequencies of jet pulsation—with no moving parts—motivated the ongoing effort to learn how to create self-resonating nozzle systems, as described in this paper.

Background

The original impetus for development of a self-resonating nozzle arose from the need to create improved submerged cavitating jets to augment the action of deep-hole drill bits. Although it had been shown that 'conventional (i.e., non-resonating) cavitating jets could provide improved rates of drilling (Conn, et al., (1981))the cavitation inception limit of such jets (about 1,220 m (4,000 ft) depth with a nozzle pressure drop of 13.8 MPa (2,000 psi)) led to a search for some means to cause jet cavitation at greater depths. Several self-resonating nozzle design concepts were developed, which proved capable of achieving the goal of enhanced cavitation at greater depths (Johnson, et al., (1982a, b, and c)). Labeled "STRATOJET" (STRuctured Acoustically Tuned Oscillating JET), these new nozzle systems caused cavitation to be attained to depths two to six times deeper than conventional drill bit nozzles.

Three self-resonating nozzle system concepts are shown schematically in Figure 1; detailed descriptions of the performance of each of these concepts can be found in the references cited above. As described in the next section, the "PULSER-FED" self-resonating nozzle system (Figure 1c) has proven to be an effective design for producing high frequency interruptions of a water jet operated in air. This design, to be hereafter termed the "SERVOJET" (SelfExcited Ring-Vortex, Organized JET), was used in the experiments to be discussed, which have shown that high frequency jet pulsations can indeed improve the performance of an in-air cleaning or cutting water jet.

A non-dimensional parameter which defines the periodic characteristics of any oscillating system is the Strouhal number, Sd. For jets:

$$Sd = fd/V,$$

where:

f is the resonating frequency;
d and V are, respectively, the jet diameter and velocity.

Preliminary analyses of the characteristics of in-air pulsed jets (Chahine, et al. (1982a, 1982b, and 1983)) have shown that optimum performance of such jets should occur for Strouhal numbers in the range: $0\ 3 < Sd \leq 1.2$. The criteria evaluated were: (1) relaxation of the impulsive stress created by each slug, (2) cushioning of the stress by a liquid layer from the previous slug, and (3) aerodynamic effects causing slug disintegration. Although the optimum Sd-range can readily be achieved within a passive self-resonating system, all of the mechanical jet-interruption systems cited previously were operated at frequencies which produced Sd values well below the optimum range. In each of these mechanical system studies, where frequency variations were made—increases in the interruption rate always produced improvements in erosivity.

This trend of increased pulsed jet erosiveness with frequencies that tended to approach the optimum range was an additional encouragement for our attempt to adapt the submerged self-resonating ideas to the interruption of a water jet operated in air.

SELF-RESONATING JET CONCEPTS

The "PULSER-FED" self-resonating jet (SERVOJET) shown in Figure 1c, was the concept found to be most readily adaptable to creating an in-air pulsed jet. It can be seen from the several configurations in Figure 1 that the PULSER-FED concept is a combination of the I^PULSERII (Figure la) and "ORGAN PIPE" (Figure lb) configurations. A tandem-orifice Helmholtz resonating chamber (diameter, dT, in Figure 1c), is tuned so as to excite a standing wave within the organ-pipe section (length, Lp, Figure 1c). Peak resonance in this system occurs when the frequencies of the Helmholtz chamber and the organ-pipe wave are matched to a preferred jet structuring frequency for the exit orifice (de in Figure 1c). By varying the several dimensions of this system and the operating pressure, first, second, or third mode resonances can be selected. See Chahine, et al. (1982a, 1982b, and 1983) for further details on the performance of these sel fresonating systems.

EXPERIMENTAL PROCEDURES

To learn how to create a self-resonating pulsed jet, it was first necessary to develop ways of quantifying their performance. Several techniques were used, as indicated in Figure 2, to measure the pressure and flow fluctuations in these jets. There were: (1) pressure transducers in the organ-pipe segment of the nozzle and (2) in a target plate which could be located at various standoff distances, X, away from the nozzle, (3) a laser beam, shone through the jet so as to impinge upon a photo multiplier tube, and (4) single flash and high-speed photography.

Good agreement was observed amongst the frequency spectra measured with each of these techniques. Typical correlations are seen in Figure 3, where the pressure fluctuations in the organ-pipe section of the SERVOJET nozzle (lowest trace) are compared with target plate pressures measured at several standoff distances. These pressure fluctuations, p', have been normalized by the pressure drop across the nozzle, Δp. The principal frequency for this test occurred at f 5.5 kHz (Sd 0.38), with a subharmonic at f = 3.1 kHz (Sd = 0.20), and second and third harmonics at about 11 and 16.5 kHz. These spectra are seen, with varying amplitudes of the fluctuations, at each measuring location. Note, for instance, that the subharmonic - which is we11 defined inside the pipe - is very weakly seen in the pulsed jet as it strikes the target. Conversely, the second and third harmonics - quite small in the pipe - are strongly amplified in the jet, particulary at X = 17.8 cm (7 in.) (X/d - 38), the optimum standoff distance for this jet at this velocity. At X = 17.8 cm, the target plate fluctuations were 10 percent, for in pipe fluctuations of only 2.5 percent. Similar correlations were also observed between the laser beam interruption measurements and the pressure fluctuations.

Several high-speed movies were taken, using a HYCAM camera. The best photographic results, however, were achieved by single flash lighting, using a MAMIYA camera and Kodax 4X black and white film. Several typical photographs are seen in

Figure 4, for ap values ranging from about 3.4 to 4.5 MPa (500 to 650 psi). Although such photographs were successfully taken for pressures up to about 6.2 MPa (900 psi), it was necessary to rely on the pressure fluctuation measurements for higher pressures. Tests up to 62 MPa (9,000 psi) have been performed with SERVOJET nozzles; with nozzle orifice diameters ranging from 2.2 to 4.7 mm (0.085 to 0.185 in.).

EXPERIMENTAL RESULTS

Since the SERVOJET nozzle concept is actively under development, extensive evaluations are still to be performed. Several preliminary and feasibility studies have been conducted, however, and some of these results will be described in the following sections. In each case, tests were conducted by causing the jet to pass in a straight line, at a constant velocity, across the target.

Paint Removal

During a feasibility study supported by the Office of Naval Research[1], Chahine, et al. (1982b) showed that a passively interrupted jet could provide enhanced rates of removal of paint layers from both aluminum and graphite reinforced epoxy panels. Comparative runs were made between SERVOJET nozzles and a Leach and Walker (1966) nozzle, to determine whether the pulsed jet could be used to remove successive paint layers without damaging the substrate, thus providing an alternative to the chemical stripping methods now used on aircraft. Also, since chemicals cannot be used on the composite segments which are being installed at an accelerated pace on these airplanes - some sort of mechanical paint removal alternative is now being urgently sought.

The preliminary results from this study were encouraging, but it must be emphasized that considerable work remains to be done before a viable pulsed jet aircraft paint stripping system is available. In comparison to the Leach and Walker (L&W) nozzle the SERVOJET created a consistently wider, more uniform cleared path. Some typical results are shown in Figure 5 for the removal of various paint layers from an aluminum aircraft panel. This panel had three MILSPEC polyurethane-based coats above a yellow epoxy primer. It should be noted that the SERVOJET nozzle system was operated with a total pressure drop of 37.2 MPa (5,400 psi); however, due to the internal pressure drop at the Helmholtz chamber, the exit orifice pressure drop was about 32.7 MPa (4,750 psi). Thus, the jet velocity for the SERVOJET was about 256 m/s (840 ft/s), or slightly lower than the 265 m/s (870 ft/s) for the L&W nozzle operated at 35.2 MPa (5,100 psi). The total power absorbed by this SERVOJET system was about 34.7 kw (46.6 hp) versus about 32.4 kw (43.4 hp) for the L&W nozzle. Comparable results were obtained on the graphite reinforce epoxy panels, as typified by Figure 6.

Rock Cutting

A feasibility study[2], comparable to the paint removal program described above, was undertaken, with the objective of determining whether self-resonance could improve the rock cutting capability of low velocity water jets (Chahine, et al. (1982a)). Additional

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[1] Contract No. NOOO14-82-C-0143.
[2] NSF contract no CEE-8114063.

high velocity tests have been run recently, and results from each of these test series will be described.

The importance of the self-induced pressure fluctuation amplitude is demonstrated by the results summarized in Figure 7, for slot cutting trials on samples of Berea sandstone. In the upper portion of this figure it is seen that the optimum pressure fluctuation (of about 8 percent) inside the nozzle occurred at a jet velocity of 91 m/s (300 ft/s). The lower portion of Figure 7 shows that the maximum improvement of the SERVOJET nozzle, relative to the L&W nozzle, occurred at the velocity corresponding to maximum fluctuations in the self-resonating system. At the 91 m/s (300 ft/s) jet velocity, the STRATOJET nozzle removed about 33 cm^3/s (2 $in.^3/s$), absorbed a hydraulic power of 6.9 kw (9.3 hp), and hence the volume removal effectiveness was 17 x 103 cm^3/kw-hr (0.45 ft^3/hp-hr). The corresponding values for the L&W nozzle were: 3.3 cm^3/s (0.2 $in.^3/s$), 4.3 kw (5.8 hp), and 2.8 x 103 cm^3/kw-hr (0.07 ft^3/hp-hr). Therefore, the STRATOJET nozzle was over six times more effective, when operated at its optimum resonating condition.

The results of some higher pressure slot cutting tests on Indiana limestone are summarized in Table 1. Contrasted are the performance of an L&W nozzle, and a SERVOJET nozzle, each having smaller diameters than the nozzles used for the lower pressure tests on the sandstone.

Table 1. Slot cutting tests on Indiana Limestone
(standoff distance 28.7 cm (11.3 inches) Translation velocity 10.2 cm/sec (4 in/sec))

Nozzle:	Leach and Walker	SERVOJET
Diameter, mm (in.) :	3.9 (0.085)	2.6 (0.101)
Total Δp, MPa (ksi) :	37.2 (5.4)	44.8 (6.5)
Nozzle Δp, MPa (ksi) :	37.2 (5.4)	39.3 (5.7)
Power, kw (hp) :	35.3 (47.3)	45.8 (61.4)
Slot width, mm (in.) :	8.1 (0.32)	15.7 (0.62)
Slot depth, mm (in.) :	9.9 (0.39)	9.7 (0.38)
Volume removal rate, cm^3/s (ft^3/hr) :	8.2 (1.04)	15.4 (1.96)
Effectiveness, $\frac{cm^3}{kw\text{-}hr}$ $\left(\frac{ft^3}{hp\text{-}hr}\right)$:	8.4 x 10^2 (0.022)	12.2 x 10^2 (0.032)

It is seen that the main difference between the slots cut by these two nozzles lies in the slot width—almost twice as wide for the SERVOJET. The resulting effectiveness, despite the larger hydraulic power used, is almost fifty-percent greater for the resonating nozzle system.

Decontamination

Methods for rapidly removing toxic contaminants from military vehicles are now being sought by the U. S. Army. As part of an ongoing program[3] involving the development of equipment to evaluate water jets for this cleaning problem, a series of cleaning trials was conducted on painted panels containing deposits of a simulated contaminant. The importance of resonance is clearly indicated by the data shown in Table 2. Here, test results using the same exit nozzle orifice, with and without the use of the Helmholtz resonating chamber, are contrasted. As in the rock cutting tests discussed above, the self-resonance induced water slugs virtually doubled the path width cleaned.

Table 2. Cleaning of simulated contaminant
Exit Nozzle: oval, fan type 4.7 mm (0.185 in) equivalent diameter
Translation velocity: 61 cm/s (24 in/s)
Standoff distance: 30.5 cm (12 in)

Nozzle:	Leach and Walker	SERVOJET
Diameter, mm (in.) :	3.9 (0.085)	2.6 (0.101)
Total Δp, MPa (ksi) :	37.2 (5.4)	44.8 (6.5)
Nozzle Δp, MPa (ksi) :	37.2 (5.4)	39.3 (5.7)
Power, kw (hp) :	35.3 (47.3)	45.8 (61.4)
Slot width, mm (in.) :	8.1 (0.32)	15.7 (0.62)
Slot depth, mm (in.) :	9.9 (0.39)	9.7 (0.38)
Volume removal rate, cm^3/s (ft^3/hr) :	8.2 (1.04)	15.4 (1.96)
Effectiveness, $\frac{cm^3}{kw\text{-}hr}$ $\left(\frac{ft^3}{hp\text{-}hr}\right)$:	8.4×10^2 (0.022)	12.2×10^2 (0.032)

CONCLUSIONS

From the results achieved as of this time, in an ongoing effort to develop erosive water jets which have passive, internal pressure fluctuations, and hence deliver a series of water slugs, the following conclusions may be drawn:

1. A tuned, self-resonating nozzle system, consisting of a Helmholtz resonance chamber, followed by an organ-pipe segment, can be tuned to the intrinsic frequency of a water jet nozzle, for a wide range of jet velocities (nozzle pressure drops).
2. Pressure fluctuations inside the nozzle system correlate with target plate pressure fluctuations and optical measurements of the water slugs.
3. Self-resonating pulsed jets have produced substantially more effective cleaning and cutting results in comparison to comparable non-pulsed jets.

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[3] Under MRC Corporation Purchase Order No. 65390, a subcontract for ARRADCOM, CSL Contract No. DAAK1182-C-0150

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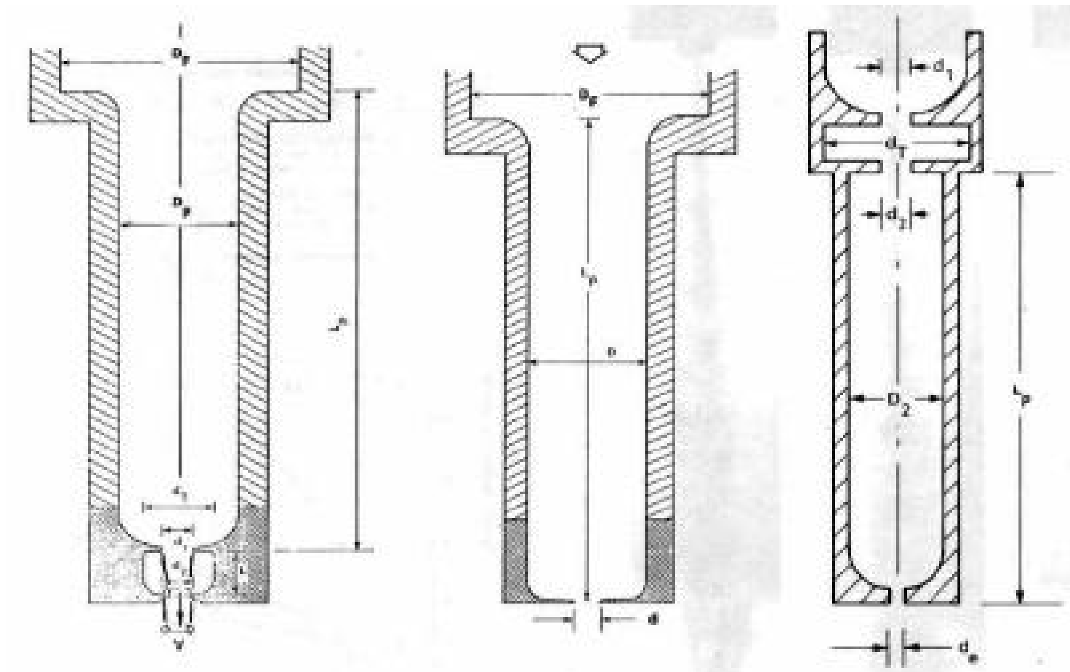


a. “PULSER” b.”ORGAN-PIPE” c.”PULSER-FED”

Figure 1. Self-resonating nozzle system concepts

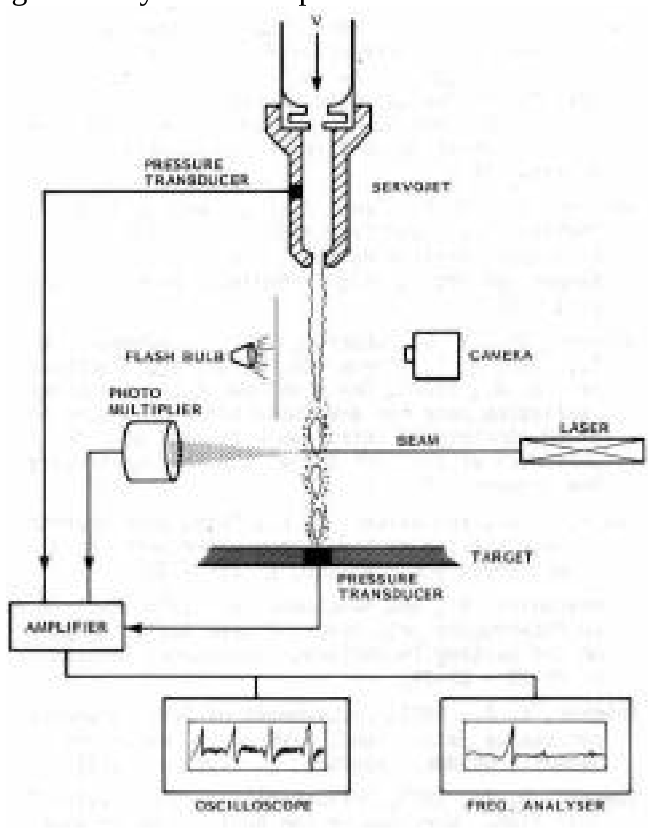


Figure 2. Experimental set-up.

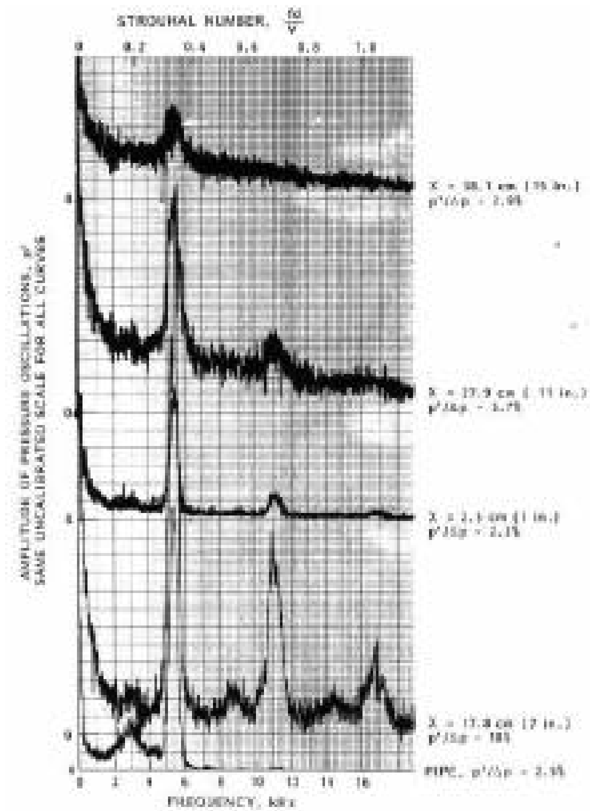


Figure 3. Pressure fluctuations in pipe and on target transducer at several standoff distances. V 65.5 m/s, (215 ft/s), d = 4.7 mm (0.185 in).

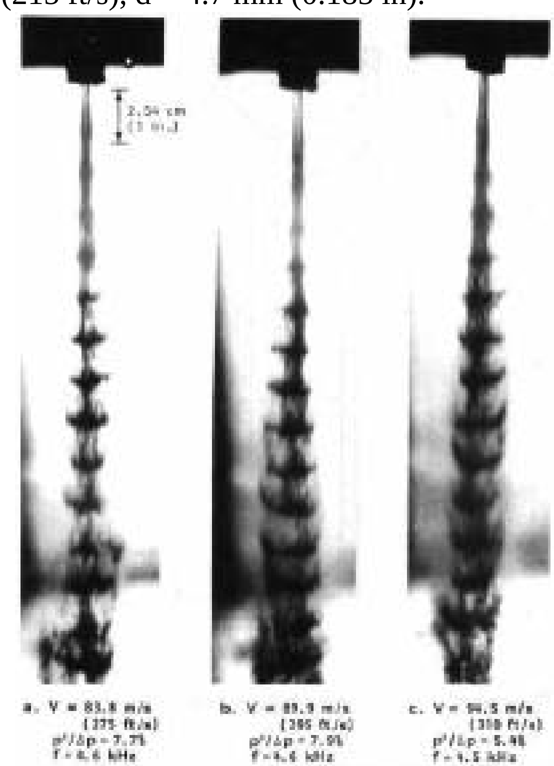


Figure 4. SERVOJET appearance in the maximum oscillation region, d = 4.7mm(0.185 in).

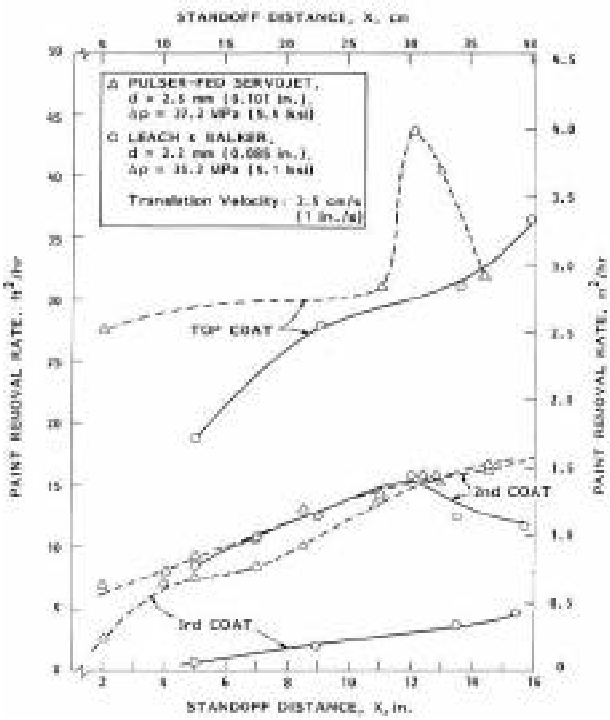
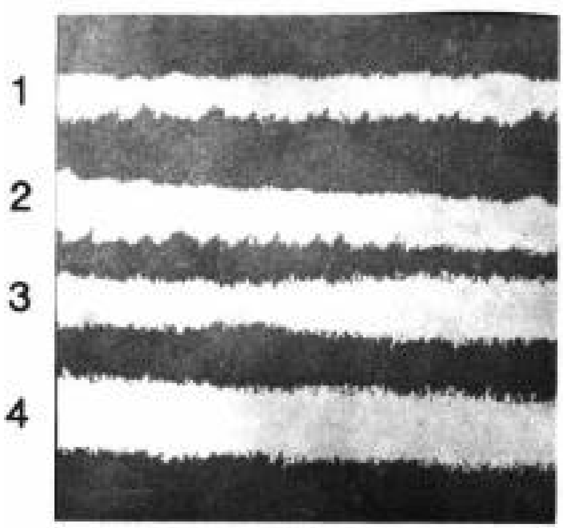


Figure 5 – Removal rate for various paint coatings on aluminum aircraft panel.

GRAPHITE REINFORCED PLASTIC



RUNS 1 & 2: LEACH & WALKER NOZZLE
d = 2.2 mm (0.085 in.)
Δp = 35.2MPa (5.1 ksi)

RUNS 3 & 4: SERVOJET NOZZLE
d = 2.6 mm (0.101 in.)
Δp = 37.2 MPa (5.4 ksi)

Run:	One Pass		Two Passes	
	1	2	3	4
Paint Removal Rate: m^2/hr (ft^2/hr)	1.16 (12.5)	1.45 (15.6)	0.72 (7.8)	0.95 (10.2)
Effectiveness: m^2/kw-hr (ft^2/hp-hr)	0.036 (0.29)	0.041 (0.33)	0.022 (0.18)	0.027 (0.22)

Figure 6. Comparative Paint removal trials, gray areas are polyurethane top coat, white areas are epoxy primer.

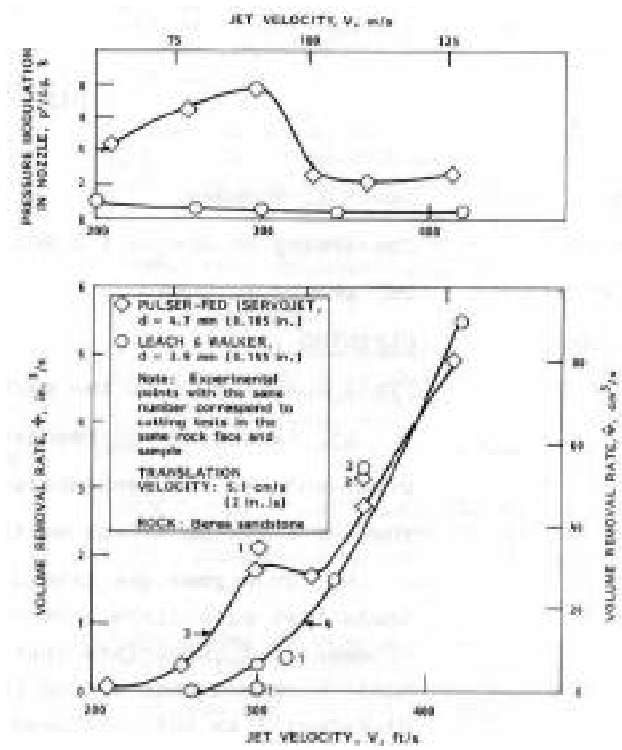


Figure 7. Pressure fluctuations in nozzle tube and rock cutting results for pulsed and non-pulsed nozzles.

DISCUSSION

NAME: David A. Summers
COMPANY: University of Missouri - Rolla

QUESTION: This question is in two parts:
a) Is the use of resonatingly pulsated jets limited in any way by pressure? We can use cavitation, for example, to lower the operating pressure of a system - can we do this with a pulsating system?"
b) At higher jet pressures, above the rock threshold pressure, is there that much difference between a Leach and Walker and a resonating jet? (Comment: I appreciate that you take good care to make a good resonating nozzle - given that a good Leach and Walker nozzle will cut 2,000 nozzle diameter; I am not convinced that you take as good care to test against comparable quality Leach and Walker designs.)

NAME: Bruce Williams
COMPANY: Liquid Lasers, Inc.

QUESTION: "At what pressures has the self-resonating pulsed jets been used?"

NAME: Mohamed Hashish
COMPANY: Flow Industries

QUESTION: "The concept of the 'servojet' is quite intriguing; however, for a valid comparison of its cleaning rates, or erosivity, with conventional steady water jets, the Leach and Walker nozzle is not the most effective cleaning tool. If you distribute the power used of 32 Kw approximately over many smaller steady jets, the cleaning effectiveness will exceed that of Leach and Walker nozzle, and probably that of the 'servojet'. I suggest you make this comparison with the best steady jet arrangement, also with the best "servojet" arrangement

ANSWER: Two of the questions (from Dr. Summers and Mr. Williams) ask about pressure usage with self-resonating pulsed jets. To date, this type of nozzle system has been operated over a range of from about 500 to 10,000 psi. Although we do not envision any intrinsic limitations to the phenomenon, the pump capacity in our laboratory has restrained us to the 10,000 psi value. In each case which we have examined the SERVOJET nozzle systems served to provide the same advantages previously seen for CAVIJET nozzles, namely, the ability to provide enhanced erosivity at the same nozzle pressure, or comparable results at a lowered value of nozzle pressure, relative to conventional, steady water jet systems.

Drs. Summers and Hashish question the differences in performance we have reported using as a base-line the so-called Leach and Walker nozzle for the non-resonating nozzle. All of our nozzles received the same degree of care in manufacturing--none, of course, of the "jeweler's quality" that Dr. Summers has achieved with electroplating techniques. At any pressure, if we have good resonance, the

SERVOJET nozzles were seen to out perform a comparable Leach and Walker nozzle. By comparable, we mean a nozzle delivering essentially the same flow and pressure. Comparisons of performance are then made by normalizing the cutting or cleaning results by the hydraulic power delivered through each nozzle. This is certainly a valid way to compare two nozzle systems; however, as Dr. Hashish has suggested, this does not claim to be the optimum case for either the resonating or the non-resonating system. Indeed, we too have found situations where a plurality of smaller nozzles will out perform one larger nozzle which delivers the same hydraulic power. We have found, however, that the opposite case can occur for some cleaning or cutting situations. Optimization should not be confused with one-to-one comparisons of two similarly sized, single-orifice nozzle systems, wherein the normalization described above will certainly tell you whether you've improved things or made them worsen

NAME: James Evers
COMPANY: Southern Illinois University - Carbondale

QUESTION: "Did stand-off distances increase in your cutting tests as they did in Gene Nebeker's results?"

ANSWER: Dr. Evers asks whether stand-off distances increase in our tests as they did for Gene Nebeker's tests. First, it should be emphasized that the optimum stand-off distance, X, for our self-resonating jets is a function of Strouhal number, Sd, and percentage of modulation: V/V, where V is the amplitude of jet velocity modulation and V is the mean jet velocity. Athough the main objective of our work to date has been to optimize jet erosivity and not to maximize the stand-off distance, by controlling these parameters, within a range we have yet to fully explore, we can vary X in accordance with the needs of a given cleaning or cutting application. In comparison to a comparable cavitating jet, we have found that a SERVOJET nozzle can be made to operate at greater stand-off distances. Also, if desired, we can reduce the stand-off distance by proper selection of Sd and V/V.

DRILLING BORE HOLES IN COAL MINES USING H.P. WATER

Assessor des Bergfachs H. Becker
Steinkohlenbergbauverein, Essen, West Germany

Dipl.-Ing. B.H. Schmidt
Woma-Apparatebau, Duisburg, West Germany

ABSTRACT

In the coal mining industry of West Germany, coal face infusion is used for longwall mining. Bore holes are drilled using drilling mounts and drilling rods with drill cutter heads. The infusion equipment usually includes a high pressure water pump which created the idea to drill these infusion holes by using high pressure water. The development of a special fast drilling turbo jet and the gained knowledge concerning the drilling of infusion holes in general is described in the following report.

INTRODUCTION

In West Germany coal faces water infusion is usually used in longwall mining. Depending on the set-up of the coal mine itself, two different methods can be put to work. Water infusion can be done in those entries which are prepared before mining. They can be continuously infused, because there is no time limitation caused by the actual production. For the infusion, holes with a 2 in. diameter and a length of 65 ft to 260 ft are drilled.

If there are no entries available and the infusion has to be made from the face, the drill and infusion work can only take place during the non-production time. Within a few hours the water has to be infused in large amounts per time unit. Hereby, the infusion holes are normally 20 ft long and have a 2 in. diameter. The actual drill work can only be carried out by hand drills. This is a very slow and exhausting process, especially when drilling very hard coal.

To speed up the drilling work, investigations were made to use the high pressure water technique for drilling bore holes, because high pressure water systems are used for infusion work anyway. Using pressures of up to 5,000 psi at 20 gpm for this kind of work, the units can be converted to 7,000 psi at40 gpm without any problem.

The bore hole seal elements which are normally used for the infusion require a bore hole diameter, not bigger than 2 in. Only by increasing the diameter of the infusion head (seal element), bigger bore hole diameters can be used. Therefore, the following parameters were taken for the development of high pressure water coal drilling turbo jet.

<u>Bore hole diameter:</u> 2 in. to 2.2 in

<u>Bore hole length</u>:*When infusing out of the face: up to 30 ft
*When infusing out of the entry: 65 ft to 260 ft

High pressure pump:

*Drive 120 to 130 hp
*Pressure: 6,000 to 7,000 psi
*Flow: 25 gpm

For safety reasons, the operating pressure should be kept as low as possible. The turbo jet should be operated by water which is fed-in by a high pressure hose, 3/4 in., connected to the pump.

Turbo jet guide:

*The jet should only be guided by a specially formed slide.

Jet design:

*The construction of the nozzle should be simple and practical, with a long life time and a low price.

To realize this development plan, tests with non-rotating nozzles were performed. After a short period of time, it was found that this set-up was not suitable to drill bore holes.

Because of the given development goals - bore hole diameter and nozzle guidance - a rotating nozzle with separate drive could not be considered. Therefore, a drill nozzle with a high pressure water drive was developed. As shown in picture (1), the high pressure water drill head and the driving motor are mounted on a free-floating shaft. With this turbo jet which works at 5,600 psi and 10,000 rpm, the first drill tests were made, examining especially the positioning of the jets in the drill head. Some good results were made, concerning drill speed, bore hole length and bore hole diameter. But it was realized that this nozzle did not start properly. drill head spins around a fixed shaft as shown in picture (2). The drill head equipped with a protecting cage is mounted on a guiding sled which holds additional backfire nozzles and the high pressure hose.

With this nozzle the following tests were performed:

First tests were made on the surface, using coal chunks that were cast into concrete blocks measuring 10 in. x 10 in. x 10 in. The results were not satisfying, because the coal chunks were too small and their solidity did not meet the underground conditions.

Afterwards, surface tests were performed using an artificial coal pile. This material matches the coal underground, concerning tensile and crushing strength. But, the artificial coal pile does not have cleavages which are found in deposits underground. That makes it difficult to study the characteristic reaction attitude of coal when it is hit by a high pressure water stream.

In connection with the given hardness, the shakiness of the coal has crucial influence on the drill behavior, consequently, the results achieved with the artificial coal pile could not be judged as representative. Therefore, the next tests were scheduled to be performed underground with the nozzle shown in picture (3).

The coal drilling nozzle was connected to the high pressure pump by a high pressure hose and a quick-acting foot valve. For piercing, the nozzle was guided by a protective pipe which was attached to a prop. This was done to assure that the drill direction could be easily adjusted in accordance with the position of the coal layer.

The tests were performed at a mine of the Ruhrkohle AG, Bergbau AG Niederrhein. At pressures of 5,600 psi and a drilling speed of 13 ft to 16 ft per minute, bore hole depths of 100 ft were reached.

The tests showed that the following deposit conditions have a big influence on the drill behavior: size of deposit, loosening of coal in the drill area, kind and direction of the cleavages.

To conceive all the data is very difficult, because exact physical measurements of the facts are very time consuming and expensive. The optimization of this drilling technique needs a lot more research to achieve deeper drill holes on a sure base.

Looking at long bore holes there are also difficulties caused by the increasing weight of the high pressure hose and the flushing of the bore material.

The further optimization of this drill device was accompanied by surface tests on foamed concrete blocks and transparent thermoplastic pipes which were filled with a gypsum-concrete mixture. The tests on the foamed concrete blocks were made to optimize the cutting attitude of the drill head using different feed rates and bore angles on the nozzles.

The tests on the Plexiglas pipes were made to examine the nozzle feed and the removal of the bore material. It was found that drill speed, the nozzle feed and the transport of the bore material by the back flowing water are strongly influenced by the water flow in the bore hole. It was also realized that a depression in front of the bore hole supports the forward movement of the drill nozzle. But, if the depression grows, the transport of bore material gets worse which is caused by a whirling motion of the water in the bore hole. A high forward feed strength and constant material transports, however, are necessary to achieve large bore hole depths. Therefore, a systematic research of the bore material will give some more clues on problems that arise from drilling long holes.

For the drilling of short infusion holes, a couple of coal drilling nozzles are tested. Final conclusions cannot be made right now, since a lot of additional test drillings are needed.

This research program of the Steinkohlenbergbauverein, Essen, Bergbau AG Niederrhein and WOMAAPPARATEBAU, Duisburg, is running under a research program of the European Community for the development of water infusion techniques for the sake of dust control.

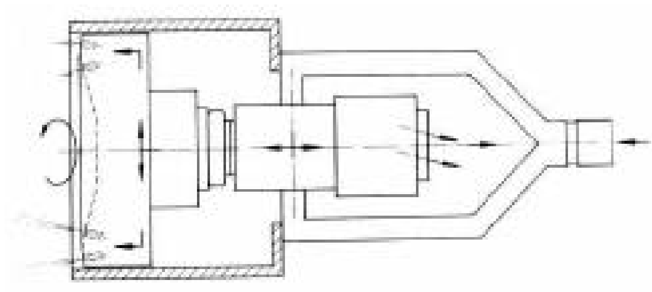


Figure 1.

Figure 2.

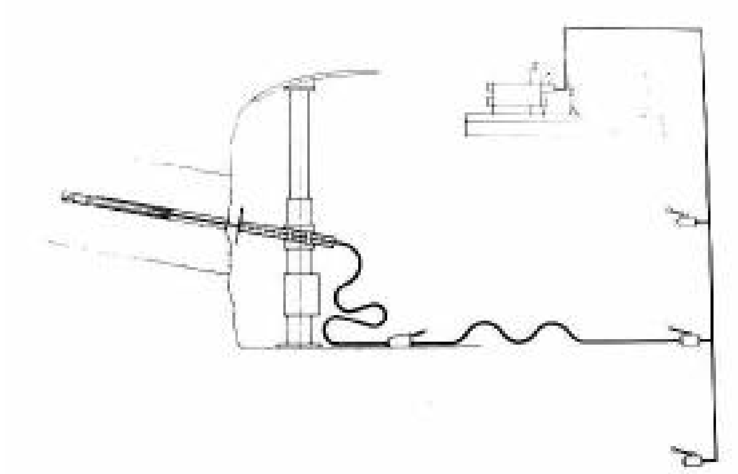


Figure 3.

DISCUSSION

NAME: David Summers
COMPANY: Universitv of Missouri - Rolla

QUESTION: "What are the nozzle diameters?"

ANSWER: The rotating drilling head has 4 nozzles altogether; 2 nozzles are of 1.5 mm diameter, and are jetting in forward direction, and the other 2 nozzles of 1.2 mm diameter are jetting backwards. In addition 2 nozzles of 1.2 mm diameter are incorporated in the slide as backfire nozzles. In total: 6 nozzles.

NAME: W. C. Cooley
COMPANY: Terraspace, Inc.

QUESTION: "If a drill were trapped in the coal, and later mined through, would it damage the longwall miner?"

ANSWER:Most probably; however, it is a minor damage, since only the drill head is of hardened material.

NAME: Michael Hood
COMPANY: University of California, Berkeley

QUESTION: 1. "Is this a production unit or is it still in prototype testing?

ANSWER: A series of 10 pieces has been manufactured and delivered. At the same time, continuous tests are being made in order to optimize the product.

QUESTION 2. "How much does the unit cost?"

ANSWER: <u>COAL DRILLING NOZZLE</u>: With short guide block and drilling head Type IV, suitable for a working pressure of 400 bars and a water capacity of approx. 100 l/min., complete with reaction jets. Nozzle is ready to be connected to VHP hose ID 13:
<u>Price:</u> DM 2,682.--

<u>Accessories</u>:

<u>Protective</u> <u>Tube</u> equipped with socket piece for diversion of water ID 50:

	PRICE
one splash guard on the drill hole, and one on the hose entrance:	DM 802 80
ID 13 - 15 m long, 420 bars, mining design DM 256. -	

<u>Foot</u> <u>Valve</u>

complete, 2 x M 22 x 1 5	DM 1,594 70

<u>Union Stem</u>

M 22 x1.5	DM 27.80

A STATUS REPORT ON THE CONCEPTUAL DESIGN OF A SEMIAUTONOMOUS MINING SYSTEM

Mukund D. Gangal
Lon Isenberg
Elisabeth J. Dutzi
Jet Propulsion Laboratory, Pasadena, CA

ABSTRACT

The concept of a semiautonomous mining system (SAMS), developed by a design team at JPL, holds the promise of increasing the productivity of underground coal mining by 400% while reducing serious injuries by at least 50%. The concept envisions the use of multiple, semiautonomous, room and pillar mining units working in each section, operated by a crew of the same size as contemporary systems[1].

The excavator module in the system is based upon oscillating water-jets technology developed by D. Summers, University of Missouri-Rolla, and others. The jets are embedded into two U-shaped wedges at the end of a convergent structure, which is mounted on advancing chocks. Coal is broken by the combined action of the jet kerfing and wedging action of the excavator. It is then sized and conveyed by a central auger into a mixing chamber, where it is mixed with water to form a slurry. The slurry is transported via an extensible armored hose carrier to the main slurry pump, and injected into hard slurry lines for hoisting to the surface.

This document presents the rationale and the tradeoffs behind the concept development and then describes the key elements in detail.

[1] Copies of final report #DOE/ET-l2548-l7, Distribution Category UC-88 may be obtained from Jet Propulsion Laboratory, California Institute of Technology, Pasadena, California. This final report was prepared by the United States Department of Energy through an Agreement with the National Aeronautics and Space Administration.

HYDRAULIC MINING EXPERIMENTS IN AN UNDERGROUND MINE IN THE ST. PETER SANDSTONE, CLAYTON, IOWA

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U.S. Department of the Interior, Bureau of Mines
Minneapolis, Minnesota 55417

ABSTRACT

This paper describes the results of in-mine tests of two high-volume (400gpm), low-pressure (1,000-psi) mobile monitor water-jet mining systems. These systems consist of water-jet cutting and slurry pumping equipment. The former employs a monitor mounted on the articulated arm of a vehicle and connected by hoses to the output of a boiler feed pump driven by a 300-hp electric motor. The slurry pumping system consists of a suction box incorporating a simple jet pump.

These tests showed that the jet dissaggregates the St. Peter Sandstone at the rate of 25 to 50 tph and that the disaggregated sandstone can be pumped as a slurry away from the face at 30 to 40 tph. Operational problems were encountered in advancing the face and in sheltering the operator from the backwash. Recommendations are made relating to the design of a remotely controlled monitor which would overcome the difficulties encountered in operating the manned monitor.

INTRODUCTION

Water-jet mining and slurry transport are attractive alternatives for the mining industry. Coal is fragmented by water-jet impact and transported as a slurry in the U.S.S.R. (0khrimenko et al., 1974), Japan (Otsuka et al., 1981), and Canada (Parks and Grimley et al., 1975). In the United States the phosphate mining industry uses hydromonitors and slurry pumps for flurrying and transporting phosphate ore (Cooley et al., 1976).

This paper describes a testing program to use hydraulic mining and transport technology in underground mines or tunnels in sandstone. Many sandstones, such as the St. Peter, are ideal for water-jet fragmentation and transport because they disaggregate readily to individual sand grains. Many of these sandstones are economically valuable because the sand can be used for glass manufacture oor foundry sand. Other sandstones contain economic concentrations of mineral commodities such as uranium or copper. While these tests were in a foundry sand mine, the results will be equally applicable to underground mining of uraniferous sandstone.

TEST SITE

The work described here was conducted in a Clayton, Iowa, sandstone mine, formerly operated by Martin Marietta Aggregates. This mine is on the bank of the Mississippi River in northeastern Iowa. The mine, no longer in operation, was developed in the St. Peter Sandstone through an adit in the bluff along the river. The sandstone was mined using a room-and-pillar system. Traditional practice is to drill 53 holes 8 ft deep into a 50- by 50-ft face. These holes were loaded with ANFO, and the face was blasted.

Front-end loaders loaded the broken sandstone onto 20-ton trucks, which hauled it to a crusher. The sandstone was progressively crushed to smaller sizes until the individual quartz grains were liberated. The sand was washed with water, the fines were removed in a screw classifier, and the remainder was dried and shipped. Historically, because of the low unit value of sand, sandstone mining operations have been hindered by the cost of repeated handling of the rock, abrasive wear on machine parts, and dust control.

These problems can be overcome by a hydraulic mining system where water cuts, liberates, and transports sand. The unit operations of drilling blasting, loading, hauling, and crushing can be replaced by water-jet cutting integrated with slurry transport.

PHYSICAL PROPERTIES OF THE ST. PETER SANDSTONE

Samples of St. Peter Sandstone from the working face in the Clayton mine were subjected to mechanical property testing. First, the samples were cut into rectangular blocks approximately 2 in high by 1 in square. They were cut without water or oil to leave the samples in their original moisture conditions because the strength of the sandstone is moisture dependent. These samples were air-dried for 1 week, and their density was measured. The mean value was 129 lb/ft^3 with a standard deviation of 1.12 lb/ft^3 for 97 specimens. The specific weights facilitated the calculation of tons mined from measured volume of rock removed.

Uniaxial compression tests were conducted on three groups of samples taken from the 97 specimens in order to relate rock strength to the rate of rock disaggregation under water jet impact. One group was simply air-dried, while two other groups of five were used to test the effects of water content on strength. The data are given in Tables 1-3. Note that the water-saturated samples were significantly weaker than the dry samples. This corroborates observations made during the mining that the rock disaggregates readily when it is saturated by the cutting jet. In addition, it has been noted that roof control can be enhanced in underground uraniferous sandstone mines by dewatering.

EQUIPMENT

A 1,000-psi, 400-gpm water jet was used to fragment the St. Peter Sandstone at the intersection of rooms O and PP in the north section of the mine (Fig. 1). The jet issued from 0.62-in-diam nozzle housed in a monitor, which was a 4.5-ft length of steel tubing containing a flow straightener upstream of the nozzle. The flow straightener was a single strip of metal (25 in long) fitted into the monitor's interior. The monitor is mounted on an articulated arm of a mobile holder, two of which were employed during these tests. One was a Gardner Denver[1] (GD) model MEHTTOO1 Air-Trac drill carrier. This drill carrier was driven by compressed air from a Joy 600QP or a GD model 340

The other holder used was an International Harvester (IH) model 3616 tractor. The monitor was mounted at the end of a 20-ft articulated arm designed for a backhoe.

[1] Reference to specific manufacturers or trade names does not imply endorsement by the Bureau of Mines.

The monitor was fed pressurized water through 50 ft of hose (3.5-in inside diam) from an Ingersoll Rand model 2MTA, seven-stage centrifugal pump driven by a 300-hp, 3,600-rpm Toshiba electric motor. A reduced-voltage starter started the motor, and it was powered by a Caterpillar 250-kw generator driven by a diesel engine. The pump suction was fed by the output of a Homelite model 610 pump, pumping water from an impoundment area normally used to recycle water from a washing plant.

OPERATING PROCEDURE

A miner manipulated the controls of the mobile monitor carrier, thereby directing the jet onto the face (Fig. 2). Sand liberated from the rock was deposited in a sump 50 ft west of the face. A suction box (Fig. 3), containing a jet pump, was buried in this sump. Sand from the face was drawn into the suction box and pumped through a slurry hose away from the face.

CUTTING TESTS WITH MONITOR M0UNTED ON A GARDNER-DENVER DRILL CARRIER

Two series of cutting tests were performed to measure the ability of the jet to fragment the sandstone and to determine the most advantageous operating procedure. These tests differed only in the vehicle used to carry the hydraulic monitor. The first tests used the converted GD Air-Trac drill carrier as the holder (Fig. 4). Table 4 summarizes the results.

In the initial tests performed on March 3, 1981, single horizontal slots were cut for 1 min with the nozzle-face distance ranging from 9 to 18 in. The jet cut a 10 to 17-in-wide kerf with a traverse velocity of 15 ft per min. Total production was estimated by measuring the cavity produced: 3- by 3- by 6-ft was cut in 4 min. resulting in an estimated production rate of 52 tph.

The operator had extreme difficulty seeing the face because of backwash of water from the face. The cutting mode was changed on March 4 to keep the operator out of the backwash. Vertical slots were cut in a new area of the face with the configuration as shown in Fig. 1.

Eight vertical kerfs were cut, the first three by traversing the jet three times, 8.5 ft up-and-down on the same path, for a total of nine traverses in each kerf. A 6-in interval was left between them. The next three were cut with two up-and-down traverses, and last two kerfs were cut with a single up-and-down traverse. Each up-and-down traverse lasted 1 min 39 sec (I min 2 sec up; 37 sec down). The nozzle-face distance was 20 in. The kerfs were 9.5-ft deep but the ribs failed only within 2 ft of the face.

In the next experiment single up-and-down traverses were cut with the nozzle 7 in from the face. This experiment lasted for 16 min. and 10 kerfs were cut 5 to 6 ft deep. These two experiments indicated that the vertical kerfs should be spaced 6 in apart and that it is not necessary to make more than one traverse for rib removal.

The next experiment involved cutting horizontal kerfs over the same face in which vertical kerfs had been cut. The cutting lasted 12 min 15 sec with the nozzle 12 to 20 in from the face. A 6- by 6.5- by 8.5-ft cavity was produced.

The monitor was then moved forward, and horizontal cutting resumed for 20 min. At the end of testing an 8- by 9- by 9-ft cavity had been excavated; 317 cu ft were cut during the last 20 min for an incremental production rate of 62 tph. The average production rate for the day's cutting was 34 tph.

Testing on March 5 consisted of 50 min of cutting designed to merge the faces cut during March 3 and 4. During this cutting a 16 - by l0 - by 9.5 - ft cavity yielded an incremental production rate of 64 tph, which brought the cumulative production rate to 46 tph

Testing on March 10 consisted of 70 min of cutting. The operator placed the jet where he thought it would be most effective. The first 40 min were performed with the monitor transverse to the drill carrier boom as shown in Fig. 4. During the second 30 min the monitor was longitudinal to the boom so that the drill carrier was perpendicular to the face. This experiment resulted in an incremental production rate of 27 tph and a cumulative production rate of 39 tph.

Clay was seen flowing in the backwash and segregated in the settling pond. This gives rise to the speculation that the clay is separated from the sand during cutting and could easily be eliminated from the pipeline feed by passing the slurry through a cyclone.

Testing on March 11 consisted of 60 min of cutting in two equal segments. In the first, the monitor was longitudinal to the boom, and the face was mined head on. During this test the cavity depth was increased to 15 ft. and an incremental production rate of 36 tph was achieved. The monitor was placed transverse to the boom for the second segment, and the cavity was widened to 21 ft. Sand was mined at a rate of 48 tph.

A final 30 min of testing was performed on March 12; 22 tons were produced at the rate of 36 tph. During this test, as for the entire series, the operator’s vision was obscured by backwash.

CUTTING TEST WITH MONITOR MOUNTED ON AN INTERNATIONAL HARVESTER BACKHOE

The March experiments pointed up the need for a more effective mobile monitor. The one used in March was inadequate because:

1. Excessive backwash from the face totally obscured the operator's vision.
2. It required the use of a large air compressor.

The GD Air-Trac drill carrier used in March was replaced with the IH model 3616 tractor (Figs. 5-6); the monitor was mounted at the end of a 20-ft articulated arm designed for a backhoe.

The IH rig was considered to be superior to the GD drill carrier because:

The articulated arm was longer than the GD boom, thereby putting the operator farther from the point of impact of the jet.

1. It is driven with a diesel engine, while the GD rig required compressed air.
2. It has a canopy over the operator's station, affording protection from roof falls.
3. It had more traction and was more maneuverable in soft sand.
4. It could be used to dig trenches and move equipment when it wasn't in use as a monitor carrier.

During the March experiments it was found that the most effective method of mining was to cut parallel kerfs so spaced that the ribs failed by gravity. However, the Air-Trac had to be trammed forward each time to index to the next kerf.

To overcome this inconvenience, the IH rig was fitted with a mechanism to move the monitor 24-in along the length of the articulated arm without moving the tractor. The hydraulic cylinder normally used to drive the digging tool of the backhoe slid the monitor along a rail mounted at the end of the articulated arm The monitor was configured as shown in Fig. 5 and is shown at the face in Fig. 6.

During the shakedown tests it was discovered that the jet reaction force created a moment acting on the backhoe arm which rendered one of its mechanical movements inoperable. Although the backhoe boom has free motion in both the XY and YZ planes (Fig. 5), the thrust of the 1,000-psi, 400-gpm jet (571 lb) reacting along the +XY axis overcame the hydraulic cylinder responsible for swinging the boom in the -X direction. As soon as the jet was turned on, this moment caused the jet to become non-normal to the face, and the arm could not be moved back along the -X direction.

To maintain the perpendicularity of the jet and to take the strain off the boom, steel plates (6- by 18- by 0.5 in) were welded at the base of the backhoe arm. This eliminated arm movement along the +X axis so that the monitor lost some of its maneuverability.

During the March tests the operator's vision was obscured continually by backwash from the face. In the December tests the operator could stay out of the backwash if a rib of material were kept in place between him and the point of jet impact. However, the operator still could not avoid the backwash if the rib were absent or if cutting of the top of the cavity was attempted. The longer arm of the IH rig was helpful in reducing the time that the operator was in the backwash, but it did not eliminate the problem.

Cutting tests were performed (Table 5) to determine the rate at which kerfs could be cut into the sandstone and the rate at which sand could be produced through a combination of kerf cutting and the failure of the ribs between the kerfs. The rock between the kerfs becomes disaggregated by saturation in water. The experiments showed that sand was mined at a top rate of 19 tph during the kerf-cutting phase. Higher production rates (20 to 90 tph) were realized in other phases, indicating that most of the mining occurs by the failure between kerfs. This rock has the fortunate characteristic that it disaggregates in water so that the blocks of failed rock break into individual grains in the backwash.

The optimum kerf cutting rate occurs at a traverse rate of 3.24 ips (16.2 fpm) in the range of speeds available with the IH backhoe with a 1 ,000 psi, 400-gpm jet.

The cumulative production rate of 25 tph achieved with the IH monitor holder was lower than the 40 tph achieved in March with the GD holder. The lower rate was a result of increased standoff distance required with this rig because of the lost maneuverability of the arm. It thus appears that neither rig, as presently constituted, can be used as a mobile monitor holder in a production setting. A workable rig might be made by building a remote station from which the GD rig could be controlled, thereby keeping the operator away from the face.

HYDRAULIC TRANSPORTATION EXPERIMENTS

Slurry pumping tests (Table 6) were conducted using a suction box as illustrated in Figs. 1, 3, and 7.

This system was operated to determine if waterjet cutting and jet-pump slurry pumping could be integrated. Pumping rates of 30 to 40 tph were realized, indicating that this simple setup could pick up the slurry generated by the cutting process.

CONCLUSIONS

The St. Peter Sandstone can be disaggregated and pumping away from a face in an underground mine with a 1,000 psi , 400-gpm water jet housed on a mobile monitor holder and working in tandem with a jet slurry pump. These experiments also showed that the optimum traverse rate of the jet across this sandstone is 16 fpm and that most of the mining occurs as a result of the disaggregation of ribs between kerfs.

Neither mobile monitor described in this paper is practical for commercial use at present. The GD system could be practical if it were fitted with a remote operator's station. Its boom is sturdier than that of the IH rig and was not affected by the reaction thrust. With a remote operator's station about 50 ft from the face, the operator would have a clear view of the face and would be out of the backwash. The face could then be attacked head-on, and the rig could follow into the cut, keeping the cutting distance to a minimum while advancing the face.

ACKNOWLEDGMENT

The author gratefully acknowledges the assistance provided by Martin Marietta Aggregates, especially R. M. Huftal, Plant Manager of the Clayton, Iowa, Plant. Thanks are also offered to Walter Krawza, Stephen Connors, Eugene Anderson, Dennis Barber, and Bror Haynes who conducted the field tests for the Bureau of Mines.

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3. Parks, D. M., and A. W. Grimley, 1975, Hydraulic Mining of Coal, Presented at Coal Convention of the Am. Min. Cong., Pittsburgh Pa.

Table 1. Uniaxial compression tests on air dried samples.

Sample number	Uniaxial compressive strength (psi)	Young's modulus (psi)		Strain at failure $\times 10^{-3}$	Specific weight (lb/ft^3)
		Tangent	Secant		
1	602	131,950	146,015	5.19	131
2	544	143,350	178,350	4.47	131
3	477	184,150	53,650	5.90	129
4	652	227,650	392,250	5.19	129
5	596	152,250	182,700	5.66	129
Mean	573	168,200	190,593	5.28	130
Std. Dev.	67	39,150	12,325	0.55	1

Table 2. Uniaxial compression tests on dry samples saturated with distilled water.

Sample number	Uniaxial compressive strength (psi)	Young's modulus (psi)		Strain at failure $\times 10^{-3}$	Specific weight (lb/ft^3)	
		Tangent	Secant		Dry	Wet
12	62	18,850	8,700	5.67	130	140
13	55	26,100	4,350	4.49	128	139
15	139	102,950	78,300	3.38	129	139
16	117	55,100	78,300	3.11	130	139
17	113	46,400	113,100	3.69	129	139
Mean	97	49,300	56,550	4.07	129	139
Std. Dev.	36	33,350	47,870	1.03	1	1

Table 3. Uniaxial compression tests on specimens saturated with mine water.

Sample number	Uniaxial compressive strength (psi)	Young's modulus (psi)		Strain at failure $\times 10^{-3}$	Specific weight (lb/ft^3)	
		Tangent	Secant		Dry	Wet
6	131	107,300	71,050	1.81	130	140
7	136	87,000	143,550	2.71	130	140
8	129	87,400	146,450	2.59	130	141
9	96	34,800	36,250	4.09	130	140
10	75	43,500	11,600	4.76	128	139
11	133	88,450	36,250	4.32	129	139
Mean	117	73,950	73,950	3.38	129	140
Std. Dev.	25	29,000	58,000	1.17	1	1

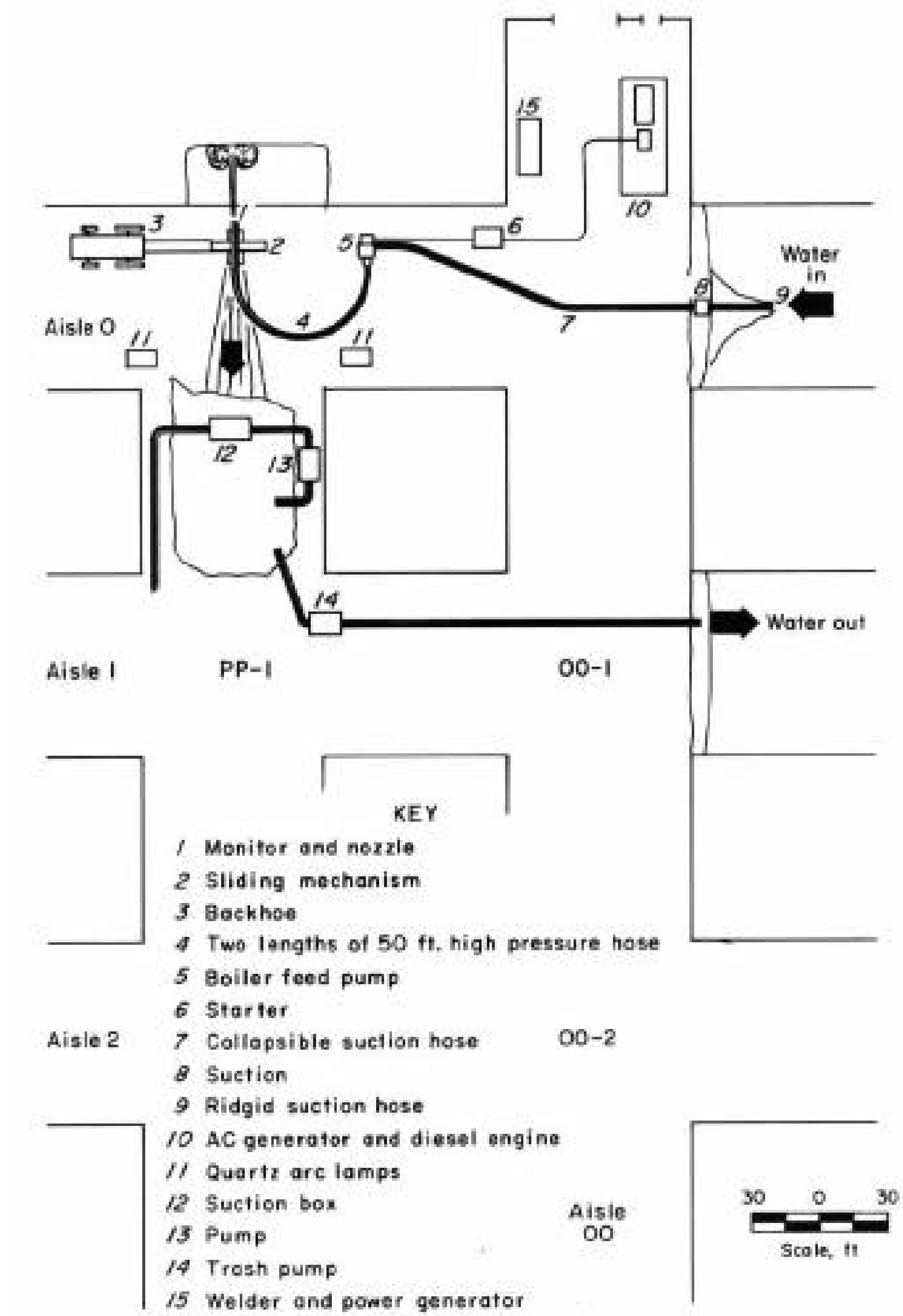

Figure 1. Schematic diagram showing plan view of the water jet mining system.

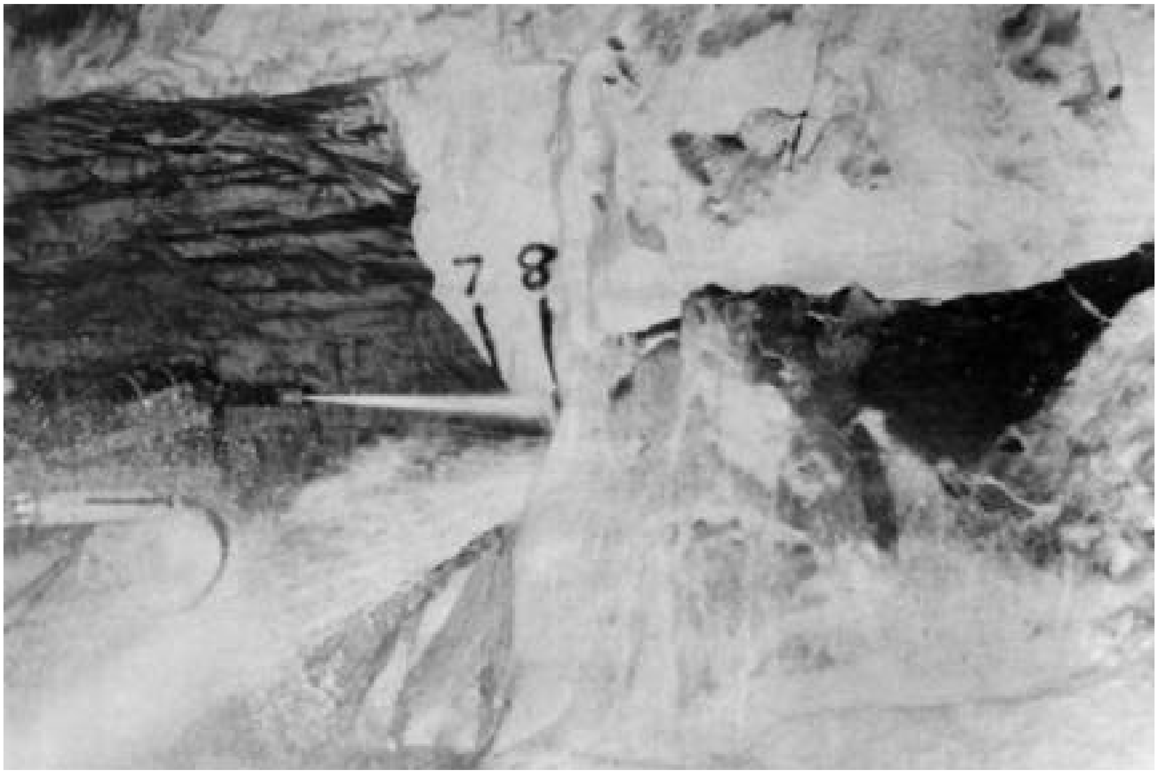

Figure 2. Jet cutting sandstone at the face.

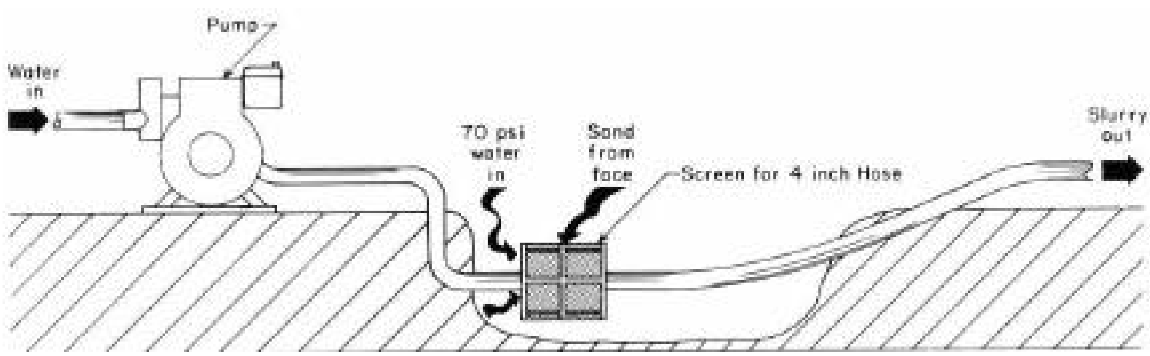

Figure. 3. Slurry pick-up apparatus.

Table 4. Summary of cutting tests using G-D mobile monitor.

Date	Test	Description	Dimensions of cavity length (ft)	Cavity volume (ft^3)	Volume change (ft^3)	Time (min)	Cumulative time (min)	Production rate[1] (tons/hr)	Cumulative production rate (tons/hr)
03/03/81	1	Horizontal slots				1			
	2	"				1			
	3	"	3- by 3- by 6	54	54	2	4	52	52
03/04/81	1	Vertical slots				27	31		
	2	"				16	47		
	3	Horizontal slots	6.5- by 6- by 8.5	331		12.25	59	23	
	4	"	8- by 9- by 9	648	317	20	79	62	35
03/05/81	1	Cutting at discretion of operator	10- by 16- by 9.5	1,520	818	50	129	64	44
03/10/81	1	Cutting at discretion of operator	10- by 16- by 12.5	2,000	480	70	199	27	39
03/11/81	1	"	16- by 15- by 9.5	2,280	280	30	229	36	39
	2	Vertical slots	5- by 7.5- by 9.8	2,647	367	30	259	48	40
03/12/81	1	"	6.5- by 7- by 7.5	2,977	330	30	289	43	40

[1]Production rate calculated as follows: $\frac{54\ ft^3}{4\ min} \times \frac{130\ lb}{ft^3} \times \frac{T}{2,000\ lb} \times \frac{60\ min}{hr} = \frac{52\ tons}{hour}$

Figure 4. Gardner-Denver mobile monitor.

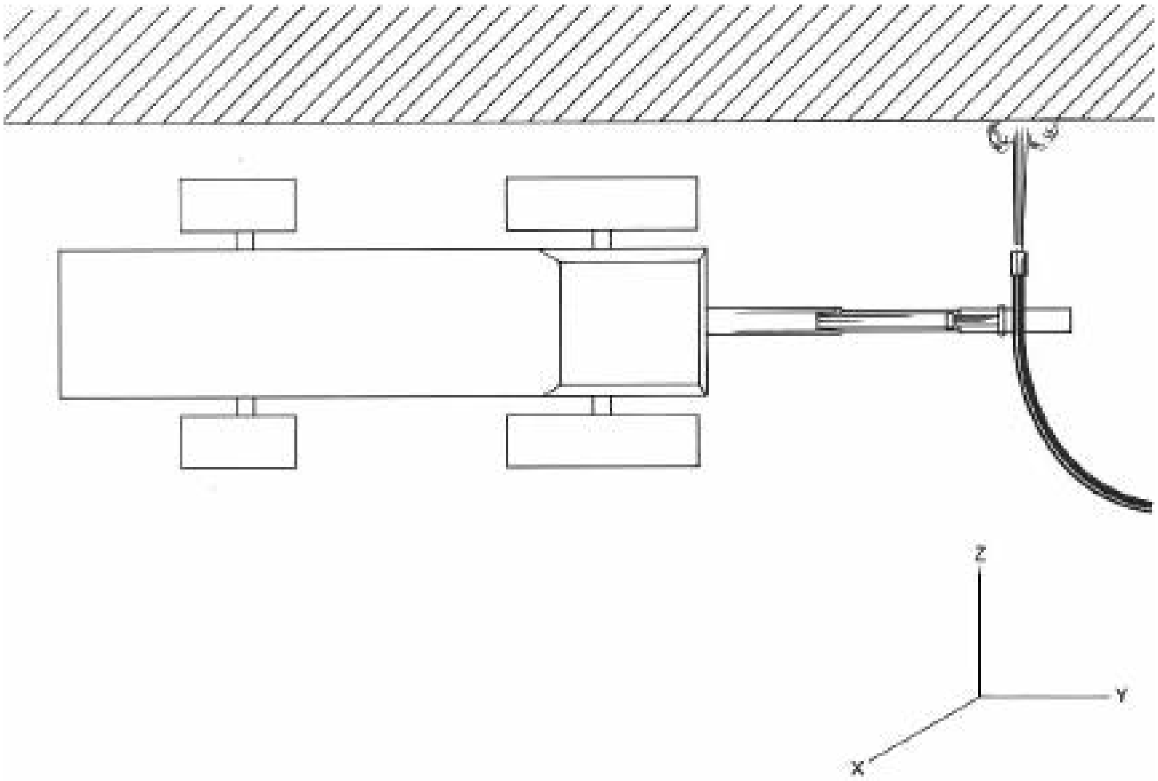

Figure 5. Schematic plan view of International Harvester mobile monitor.

Figure. 6. Backhoe with monitor at the face.

Table 5. Cutting tests using mobile monitor.

Date	Test	Description	Total volume of cut (ft^3)	New volume cut lxwxh (ft)	Time (min)	Total time (min)	Production rate (tph)	Cumulative production rate (tph)	
12/04/81	1	Shakedown							
12/04/81	2	Test arm-production	150	3- by 10- by 5	6	6	97	97	
12/04/81	3	Production	270	3- by 10- by 4	25	31	19	4	
12/05/81	1	"	271	1- by 1- by 1	4	35	1	30	
12/05/81	2	"	391	3- by 10- by 4	7	42	67	36	
12/05/81	3	"	691	3- by 10- by 10	20	62	58	43	
12/05/81	4	Roof cutting	696	1- by 1- by 5	10	72	20	37	
12/07/81	1	"	708	3- by 2- by 2	5	75	9	36	
12/07/81	2	"	732	2- by 3- by 4	35	112	3	25	
12/07/81	3	"	737	1- by 1- by 5	20	132	1	22	
12/08/81	1	Production	1,108	11- by 9- by 3.75	40	172	36	25	
12/09/81	1	Kerf cutting experiment	1,110	3.3- by 0.08- by 7.50	0.57	173	14	25	1.18 ips traverse
12/09/81	2	"	1,111	3.3- by .08- by 3.73	.17	174	19	25	3.24 ips traverse
12/09/81	3	"	1,112	3.9- by .08- by 2.5	.12	174	4	25	6.78 ips traverse
12/09/81	4	Production	1,287	175	38	212	12	24	
12/10/81	1	"	1,537	250	24	236	41	25	

Table 6. Pumping Tests

Date	Test	Pumping time (min)	Volume pumped (ft^3)	Pumping rate (tph)	Comments
12/07/81	1	30	100	13	Delta built up in PP-1
12/08/81	1	5	4.4	3.5	Hose severely crimped pumping uphill
12/08/81	2	3	25.7	33	30 feet of hose emptying into bucket of a backhoe buried in the floor
12/09/81	1	38	412	42	Delta built up in PP-0

Figure 7. Suction Box.

DISCUSSION

NAME: Gerald Zink
COMPANY: StoneAge, Inc.

QUESTION: "If long-term, full scale hydraulic mining were carried out in sandstone, the rock mass around the face might become saturated with water. Would the reduction of compressive strength cause ground control problems in a 50' x 50' unreinforced heading?"

ANSWER: That is a reasonable assumption but it was not borne out by observations. We noted no ground control problems because the sandstone drained so quickly.

NAME: David Eddingfield
COMPANY: SIU

QUESTION : "Does product of water jets mining need crushing/grinding after extraction?"

ANSWER : No. The quartz grains are completely liberated from the sandstone so that the slurry coming off the face contains individual quartz grains.

NAME: Michael Hood
COMPANY: University of California, Berkeley

QUESTION: "The mining rate (average) achieved with a monitor was, if I recall correctly, between 25 and 40 tons/hr. What was the mining rate with conventional mining techniques?"

ANSWER: I do not know the production rate achieved in the Clayton mine. An average figure for similar mines would be about 30 TPH.

USE OF HIGH PRESSURE WATER JETS FOR CUTTING GRANITE

Roger J. Raether[1], Ronald G. Robison[2], and David A. Summers[3]

ABSTRACT

Conventional methods of quarrying granite are becoming increasingly expensive in real and environmental terms. The existing flame torch is, for example, a slow, destructive and noisy method of creating slots. In order to overcome the problems of current equipment, a high pressure water jet slotting device has been developed, field evaluated and is now in commercial production.

Water jet cutting has also advantage in secondary preparation of monument stone, where the jets can create straight surfaces and right angled cuts to an adequate level for final surface acceptance. The productivity and methodology for this is discussed.

INTRODUCTION

The practical use of granite as facing for buildings and as one of the more common monument materials is widespread, within the United States and abroad. Granite outcrops in many states of the Union and can be found in a range of colors from black to mahogany to pink to almost white. This diversity has enhanced its usefulness. In recent years, however, the costs of quarrying the rock has increased at a rate much above that of current inflation. This, together with the strength of foreign competition, has threatened the vitality of the industry. A method is here described which will improve the economics of quarrying, but first it is necessary to describe the current way in which granite is prepared.

CONVENTIONAL EXCAVATION TECHNOLOGY

In its original form granite is found in large intrusions, the most common of which is a relatively large dome shaped structure, which can extend over several miles. The first stage in quarrying is to isolate a large block, typically on the order of 40 to 60 ft long, 12 ft wide, and 10 ft deep by means of long vertical cuts. The block can then be freed from its base and slit into smaller pieces for removal to the finishing plant. The most conventional method for creating the original slots is to use a flame burner to spall out the cut. A flame burner consists of a fuel oil:air mixture which is ignited in a combustion chamber attached to the end of a long pipe, much akin in concept to that found in a small jet engine. The flame that comes out is sufficiently hot, when played upon the granite that it will heat the surface rapidly to temperatures in excess of 1400^{o} C. This causes a change in the quartz structure which in turn creates a rapid expansion of the quartz, causing a spelling of material from the surface. The temperature gradients are so high that the spalls are very thin and thus the debris consists of a very fine particular matter and carries with it adjacent particles of feldspar and mica.

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[1] President, Hi-Tec, Inc., Milbank, South, Dakota
[2] Sales & Service Manager, Hi-Tec, Inc., Milbank, South Dakota,
[3] Curators' Professor of Mining Engineering and Director, Rock Mechanics & Explosives Research Center, University of Missouri-Rolla, Rolla, Missouri.

Disadvantages of the technique begin with the high cost of fuel, a typical burner will use between 10 to 15 gallons of fuel per hour. This system is generally hand-operated, so that the cutting rate, around 8 sq ft/hr, depends on the operator skill, burning fine particulate silica dust which is a potential health hazard. Most obviously, however, when one visits a quarry, burning creates a noise level on the order of 140 dbA. A flame torch has difficulty cutting through ground which is weathered, or where large inclusions of quartz or mica occur. The flame will melt the offending mineral into a difficult-to-remove blob of molten rock. It remains, however, the most prevalent method for creating initial blocks.

The base of the block will be broken by drilling holes along the bottom of the rock with a conventional drill. A fracture plane can either be made by firing small amounts of black powder in these holes or by driving a series of wedges into the holes and gradually increasing the distance that each wedge is driven into the face by manually driving each wedge in turn. This feather and wedge technique, whereby the wedge is driven down between two metal plates or feathers pre-inserted into the hole, is also used as the most common method of breaking the rock down into maneuverable sized pieces for lifting out of the quarry.

FINISHING PROCEDURE

The blocks are taken, from the quarry to the finishing plant. Here the first object is to reduce the blocks down to an acceptable size. This is done by slicing the large block into slabs typically by using a wire saw. The wire saw is a long, perhaps one-half mile long, length of wire which is used to draw carbide particles across the surface of the rock thereby slowly cutting a path down through it. The production rate of such a wire saw is on the order of 6 ft^2/hr. In the course of the last 10 years, the price of the carbide has gone from $7.00/bag to $58.00/bag with typical use in a quarrying operation being on the order of 20 bags/week.

When the blocks are sawn to the right geometry, it is necessary to do additional trimming, since the use of the flame jet will heat-weaken the rock behind the surface some 2 to 4 in. of rock, which must be cut from the block before good quality rock is exposed. These slabs are then broken into the required shape using a hydraulic splitter or cut with diamond saws to the required contour. The final surface is polished prior to having the figurines, carvings, and inscriptions cut into the face using a dry sandblasting technique.

The major problems of this technology are several-fold. The most obvious, apart from the environmental problems, is an economic one. Not only do the flame torch and wire saw have high operating costs, they also have subsidiary costs which multiply this figure. With the flame torch, particularly, large volumes of the rock must be wasted. This may seem to be a relatively small point, however, it is not widely recognized that the vast majority of the rock in a quarry is permeated with thin streamers of quartz and inclusions of mica and other (colors) which detract from the even grain structure required for first class quality rock. High quality rock is thus relatively difficult to find. Thus, the use of a flame torch to cut this block, reduces the volume of high grade material which is available to the quarry owner.

PREVIOUS WORK

The increasing cost of the quarrying and its environmental problems led to a research project being funded in cooperation between the National Science Foundation and the Elberton Granite Association in 1978. The research, to find a better way of quarrying, demonstrated that high pressure water jet cutting had considerable potential as the most likely candidate to solve many of the problems which existed in the granite industry. Following a demonstration for practicality of this project, the Dakota Granite Company of Milbank, South Dakota, recognized the potential of the system and began to develop this technology for use in its own quarries.

Early in the development of the equipment it was found that the development of a high pressure water jet system for cutting granite required a reliable field operational coupling. This should be operable, on a continuous basis, at flow rates of 20 gpm at pressures of 20,000 psi. After examining the state-of-the-art of commercial couplings, at the time that this work was undertaken, at the end of the 1970's, the company concluded that no satisfactory swivel existed that would meet the demand to be placed upon it. Accordingly a design program for such a coupling was initiated. Concurrently, an improved high pressure circuit was designed, which would allow consistent and automatic operation of the equipment. Over three years the system has been re-designed, re-engineered, and re-structured, so that it is now possible to pump water at 24,000 psi, through a swivel capable of achieving 600 rpm while passing flow rates of up to 110 gpm. Consequent to this, HiTec Corporation was founded to manufacture and market high pressure water jet, granite cutting equipment (Fig. 1).

The equipment developed has advanced beyond manual operation to encompass automatic operation where the unit may be left to function automatically while it cuts down through the rock creating a slot normally 60 ft long by 12 ft deep. The potential is, however, much greater and currently slots are being created up to 175 ft long and 16 ft deep.

During the course of the development of this program a number of operational parameters has been identified. Very obviously the water jet system must be designed to fit each individual rock which is being cut. However, for the purposes of illustration, the capabilities of the water jet as they have been demonstrated in cutting granite from Elberton, Georgia are cited. This granite is a 26,000 psi compressive strength granite.

It has been found by experiment that it is possible to cut this granite at water jet pressures of between 15-16,000 psi. In order however to cut a slot of the required depth, the water jet lance must be in perfect balance. This is because the long, high pressure tubing cannot be supported very easily within the approximately 2 in. wide slot which it cuts over the full depth of the cut. Accordingly it has been found that if 2 equal diameter nozzles are located diametrically opposing one another within a nozzle holder, that the water jet forces are balanced and a slot of the required depth can be achieved of an adequate straightness (Fig. 2). The water jet nozzles, for a flow rate issuing from a pump sized to a 150 hp pump, are on the order of 0.047 in. diameter. The exact size of the nozzle is a function of the motor and pump unit which are used and the length of hose from the nozzle to the pump.

The slot which is cut must be cut wide enough for the nozzle to pass through the rock. In the trials and examples which are here described, the rock is presumed to have been freed from its initial stress condition. In many instances however, when a new cut is made in a quarry to a deeper level than previously existing, high stresses which exist within the rock surface can close the crack down. This initial slot closure can seize the blade of a diamond saw, if this is used for the slotting purpose, and is one of the reasons why this technique is not that prevalent. Another is the high cost and difficulty in operating diamond saw blades in excess of 8 ft in diameter, which would only give a cut of some 3 ft due to the arbor.

In order to cut a slot wide enough for the water jet lance and the nozzle to freely enter, a slot approximately 1/2 in. wider than the nozzle holder is required. A balance must be achieved between the direction of the jet to cut forward into the rock while maintaining sufficient clearance to allow the nozzle to advance. The greater the angle that the jet makes to the line of advance, then the greater the number of passes the jet will need to make in order to cut through the material. On the other hand should the jet be inclined too far forward and at an insufficient angle to the line of advance, three different problems arise. The first of these is that the jet will tend to incline inward toward the center of the slot. This is particularly true if the inclination angle falls to less than 11^o to the angle of advance (an included angle between the two orifices of some 22^o) (Fig. 3). The second problem that arises is that relatively large steps in the advance rate will give a more jagged edge to the cut which can be a disadvantage if the surface being cut is required as a finished face. Thirdly, the slot may become too narrow to allow the passage of the nozzle, unless the cutting takes place at a large high standoff distance from the nozzle body. Where the latter is the case, then the efficiency of cutting is reduced. It has been our experience that an included angle of between 30^o and 40^o is optimal to this cutting ability.

Concurrent with the need for a water jet system to cut an adequate path for itself, it is also important that the water jet removes all the material in front of the nozzle. This means that adjacent passes of the water jet over the slot surface must be sufficiently close that they either overlap or are sufficiently close that the small rib of material between adjacent passes will be removed either by the water jet force or by a very slight pressure from the nozzle (Fig. 4). In order to insure that this overlapping occurs, it is necessary to relate the advance rate of the nozzle down the slot to the rotational speed of the nozzle itself. In the work carried out at the University of Missouri-Rolla, this is achieved by passing the flow from the hydraulic motor, advancing the head down the face, through the hydraulic motor, rotating the nozzle, and by a suitable of gearing, ensuring that a pass frequency of approximately 6 passes/inch occur as the nozzle rotates. Given that this requires 3 rotations of the nozzle/ inch, this can be tied into the rotation to the advance rate of the cutting head which is on the order of 10 ft/min, to give a rotational speed on the order of 360 rpm. Under these circumstances, a water jet slot approximately 1/3 in. deep will be cut upon each pass of the lance, which is therefore advanced at that rate sequentially down the slot. By placing solenoid valves at each end of the cut, it is possible, as mentioned earlier, to automate the cutting sequence, and achieve a slotting rate of around 16 sq ft/hr. The generator which powers the entire unit will consume around 5 gallons of fuel oil per hour.

FINAL SURFACE PREPARATION

The problem with flame jet cutting, as described earlier, is that it leaves a heat damaged layer of rock between 2 and 4 in. thick behind the cut surface. High pressure water jets do not have this disadvantage and as Mr. Swann describes in his paper elsewhere in the Symposium, water jets do not do significant damage to the residual surface. Thus the surface which is left by a water jet is an adequate final surface for monument structures where these are required (Fig. 5). This gives the advantage to the technique, of increasing the amount of available rock which can be quarried from a given volume of the original massif. It is possible where the water jet sequence of cuts are made in the right order to also generate very accurate corner cutting with the jet system. This however requires a recognition that there exists within the rock structure a preferential breakage pattern identified as the rift and bedding of the rock and this should be recognized in the layout of the original cuts in the granite surface (Fig. 6).

CONCLUSIONS

The water jet system which has been briefly described in this paper has been found sufficiently promising that its development in the field has now reached a commercial proposition. Not only can the water jets be used as an original cutting tool in the quarry to isolate individual blocks of granite which can then be removed for final finishing, but the quality of the cut which the water jet makes across the surface is of an adequate quality that it can be left as a finished surface in structural usage. This therefore increases the quantity of rock which is available from a given rock while concurrently lowering the purchase and operational price of the quarrying equipment and reducing the environmental hazards of existing quarrying devices. The considerable benefit which can be achieved from this equipment is likely therefore to lead to its dominance in the market place within the next decade.

ACKNOWLEDGEMENTS

This work was carried out largely at the quarries of the Dakota Granite Company, the authors are grateful to Mr. Stengel, the owner of the quarry for his assistance in this work. We also gratefully acknowledge the assistance to the work carried out at the University of Missouri-Rolla, Rock Mechanics and Explosives Research Center where Dr. Mazurkiewicz, Mr. John Tyler and Mr. Jim Blaine were largely responsible for the construction of the equipment. Dr. Clark Barker assisted in some of the early research carried out in this program. This paper was typed by Mrs. Vicki Snelson, to whom we also extend our grateful thanks.

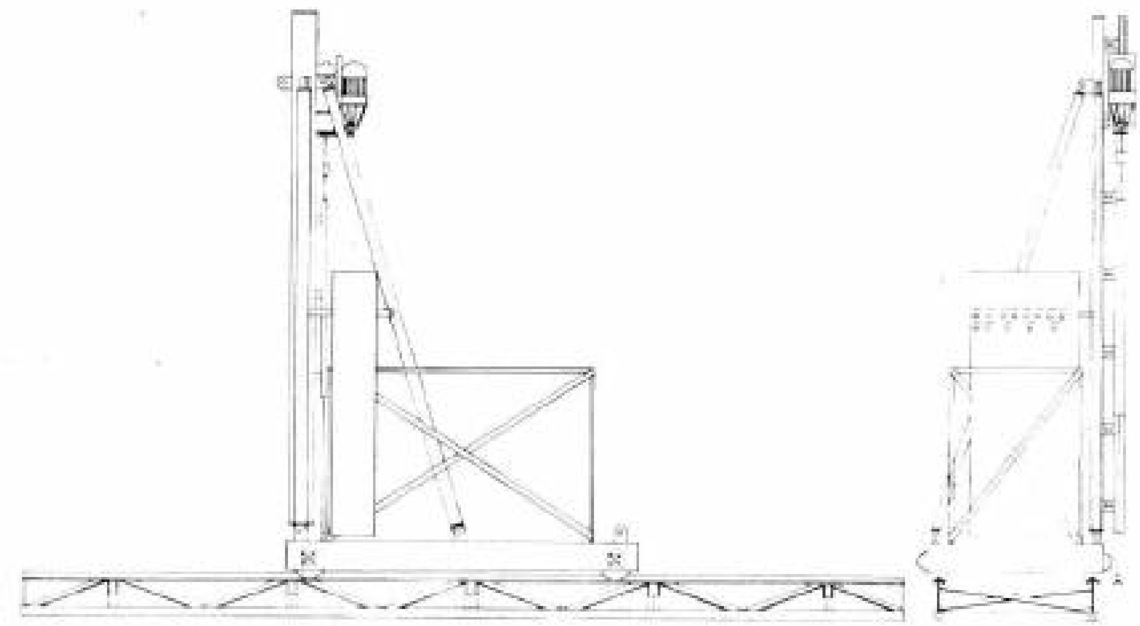


Figure 1. Schematic of water jet quarrying equipment (courtesy of Hi-Tec Inc).

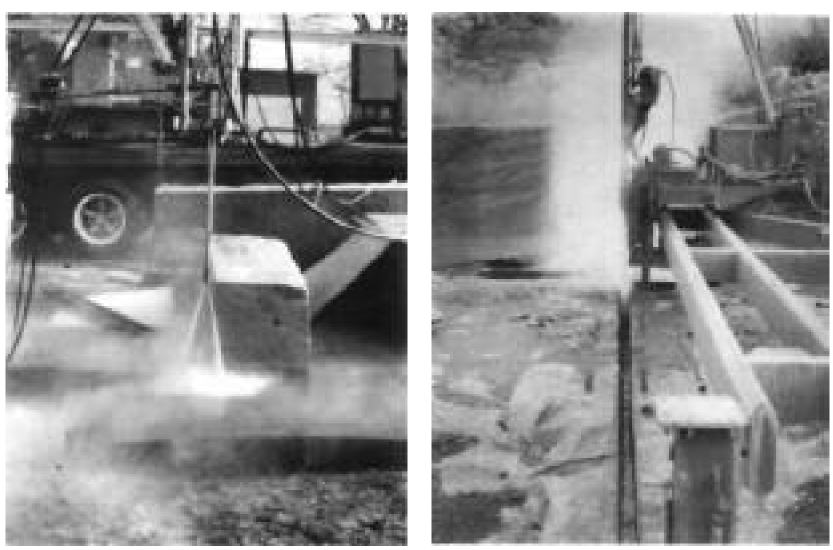


Figure 2. a) dual orifice balanced jets
b) Quarrying unit in operation showing straightness of cut.

Figure 3. Slot cut by jet at 11° showing included angle showing taper as hole deepens.

Figure 4. Jet cutting rock edge, the water has been turned off to show nozzle geometry, surface quality, and ribbing of the surface being cut.

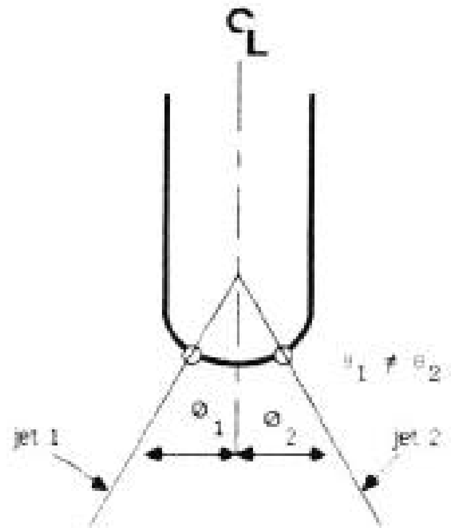
Figure 5. Final surface cut left by water jet.

DISCUSSION

NAME: Jerry Hagers

COMPANY: Spraying Systems

QUESTION: "Are the spray orifices balanced with respect to angle or orifice (mass flow) sizing in order to hydraulically balance the bit? Dr. Summers' design for the spray head did not use identical angles off the spray head C_L, i.e. , if I recall correctly."



ANSWER: The orifices are balanced (on the work we have done to date). That is, that the angle and the flow are the same for each nozzle. This is not to say that another system may or may not be better.

NAME: Michael Hood
COMPANY: University of California

QUESTION: (A) "What is the pressure and flow rate through your nozzle?"

ANSWER: The pressure is 19,000 psi and the flow is 24 gpm for the two nozzles.

QUESTION: (B) Also is the nozzle self-rotating or is it driven?"

ANSWER: The lance (nozzle) is driven.

JET-MINER SURFACE AND IN-SEAM TRIALS RUN BY BERGBAU-FORSCHUNG GMBH, GERMANY

Dr. -Ing. E. H. Henkel
Department And Test Side Coal Winning, Conveying, Face-End-Systems
Bergbau-Forschung GmbH, Federal Republic of Germany

ABSTRACT

Following to a detailed discussion of the underground trials run with the experimental version of the Jet-Miner, the development of the prototype is delineated.

The prototype design, the drive units for high pressure water supply and haulage, the array of the high pressure water cutting heads, the water and energy supply system as well as integrated operation with the other coal face equipment are explained. Subsequently, initial results of surface trial are discussed, the preparations rid lay out with respect to underground operation of the Jet-Miner are described.

Underground trials under production conditions at Lohberg colliery are undertaken.

The measured results relative to technical data, winning performance and quantity of dust, particle-size distribution and moisture content of the coal are described.

INTRODUCTION

The average of the worked seam thicknesses in German hard coal mining which today is 1.9 m will be reduced in the future to approximately 1.5 m. This requires the development of heavy duty coal winning machinery specially designed for use in seam thickness ranging between 1.0 m and 1.5 m. The coal ploughs used today in seams of such thickness do not meet the requirements is all respects, and consequently new developments, especially on the basis of plough technology, are pushed ahead more vigorously.

The following targets are set for winning machinery to be used in hard coal and in seams of 1.0 to 1.5 m thickness:

- high winning performance in hard coal
- large-sized lumps (improved quality)
- cutting into adjacent rock
- less dust.

The development of the Jet-Miner, a winning machinery using high-pressure waterjets for cutting, constitutes an important step towards the above-mentioned targets. The co-operation between M.A.N.-GHH Sterkrade, Ruhrkohle AG, and Bergbau-Forschung GmbH resulted first in an experimental unit by which the basic suitability of the techniques implied were proven (Refs. 1, 2, 3).

Based on knowledge gained throughout the trials a first Jet-Miner prototype was constructed by M.A.N.-GHH-Sterkrade. This prototype is supposed to meet the requirements defined to a very great extent. The following chapters will describe the development of this prototype, the surface trials run and the underground operation.

UNDERGROUND TRIALS RUN WITH THE EXPERIMENTAL VERSION

Trial Face

Following to the good results of the surface trials run with the experimental version, an experimental coal face was established at Lohberg colliery in seam N (4). The experimental coal face was of 50 m length and of 1.6 m of seam thickness. According to its strength the coal was categorized as "difficult to plough" (Fig. 1). In comparison to the configuration run in the surface trials, the machine was slightly modified; the key data reads:

Machine under tests:	
Operation voltage	U = 1000 V
Installed power for the pumps	P = 300 kW
Flow (max.)	V = 226 l/min
Water pressure	p = 70 MN/mZ
Total mass	m = 23.000 kg
Hydrostatic haulage drive	
Installed power	p = 3 x 80 kW

The measuring program covered the following parameters:

- Cutting force of a cutting F_s
- Pulling force (chain) F_k
- Pressure of the hydro-motor P_H
- Travelling speed

In addition coal samples for determination of particle-size distribution and moisture content were taken, and furthermore dust and noise measurements were carried out.

Test Results

From the coal-winning point of view the experimental version worked quite satisfactorily from the very beginning. The coal face was undercut smoothly by the oscillating high-pressure waterjets and subsequently broken off by the cutting heads. The coal face did not exhibit any cutting or grinding traces caused by the mechanical tools. This meant that the coal, as envisaged, was cut exclusively by the waterjets and then broken off by the cutting heads.

A further proof for this phenomenon is the fact that after the tests the cutting heads exhibited no wear and in particular no wear imputable to mechanical cutting work.

The hydraulic drive of the haulage system was adjusted to a value of F_K = 370 kN for the pulling force acting on the chain. The average value as recorded by measurement was of approximately F_K = 340 kN.

The force analysis leads to the result, that the power requirement of the high pressure pumps could be reduced from P = 300 kW to P = 220 kW and the water flow could be reduced from V = 226 l/min to V = 170 l/min (Fig. 3).

A main target of the underground trials was to obtain a maximum incremental area of face advance (web x travelling speed). The chain traction effort and the web s = 0.4 m were kept constant for the experimental version. The travelling speed V_M, however, was adapted to the prevailing coal hardness. Fig. 4 shows the optimum working configuration. At a web of s = 0.4 m a travelling speed of V_M = 0.4 m/s was obtained. This corresponds to an incremental area of face advance of 9 6 m^2/min.

The corresponding values of the most efficient conventional winning machines (ploughs, shearer loaders) read 7.5 and 6.0 m^2/min respectively for the same coal hardness.

Besides these high winning performances also a considerably better particle-size distribution of the raw coal was achieved (Fig. 5).

The operation mode of the machine enabled a very careful handling of the raw coal which eventually resulted not only in a better particle-size distribution but also in definitely less dust production. Comparative measurements have shown that the quantity of dust released corresponds to only approximately 30% of the dust produced by a shearer loader in the same seam.

Surprisingly, the moisture content of the coal was only half as high as the one recorded for comparison in a shearer-loader face. Consequently, the moisture content of the raw coal did not cause any problem to experimental operation.

In trial operation problems were encountered above all with respect to loading. The experimental unit left only little clearance underneath the machine frame so that the broken off coal could not be removed from underneath the machine. Furthermore the electrical energy supply as well as the low pressure water supply recorded frequent trouble. In this configuration the supply system a modified version of the supply system used for shearer loaders - turned out to be ill-suited.

DEVELOPMENT OF THE JET-MINER

Concept

After successful trials run with the experimental version the development of the first Jet-Miner prototype was started in cooperation of Ruhrkohle AG, M.A.N.-GHH Sterkrade, and Bergbau-Forschung GmbH.

Since the basic suitability of the oscillating waterjet system was proved throughout the trials, the original concept and design features of the cutting heads were retained to a forgoing extent. As mentioned in the introduction the Jet-Miner was supposed to be used in seam thicknesses ranging between 1.0 m and 1.5 m. Accordingly, a machine body of 1000 mm height was designed. According to an array comprising two or three cutting heads, a cutting height from 1.0 to 1.5 m can be catered for (Fig.6) . Since during the underground trials clearing problems arose, the total machine body of the Jet-Miner was situated in the space between the conveyor and the coal face. The conveyor is just straddled by a portal structure so that a good clearance profile is assured (Fig. 7) .

In order to eliminate problems of energy and water supply a special cable guidance system was developed which is supposed to allow trouble-free supply via cables and hoses even at travelling speeds of VM = 0-5 m/s.

The prototype built by M.A.N.-GHH-Sterkrade subsequent to elaboration of a joint concept exhibits the following design data (Fig. 8):

Length: 8.56 m
Height (fin): 0.98 m
Pump drives: 215 kW
Water flow: 162 l/min
Water pressure: 70 MN/mg
Total mass: 18.000 kg

Surface Trials

The Jet-Miner prototype is designed for coal winning underground. For checking the overall function, a surface testing program was carried out. Except for the length, the surface test corresponds for all machinery components to the underground installation. In the testing facilities for coal winning and coal clearance techniques of Bergbau-Forschung GmbH, a 35 m long rig was set up. The rig implies a mockup coal face of 13 m length, 1.5 m height and 6 m depth. This mock-up coal face, as to its strength, corresponds to the respective values of coal prevailing in the production face envisaged (Fig. 9).

The surface as well as the underground test array comprises the following components:

- Guidance of the Jet-Miner: Halbach & Braun Plough haulage system with chain 34 x 126 mm
- Haulage drive: 2 hydrostatic drives of 143 kW each Dusterloh
- Conveyor: EKF 3 (Halbach & Braun)
- Conveyor drive: 60 kW (surface)
- Cable/supply system: M.A.N.-GHH-Sterkrade
- Advance system with Control bracket (Halbach & controls: Braun)
- Electrical control: Audio-frequency system (Siemens)

The surface trials had have the objective of testing the haulage system, the hydraulic drive system, the electric control installation and of conducting output tests. After extensive adusting work, cutting and loading trials could be carried out. The cutting test resulted in a winning performance 50% higher than with the first test rig. The loading capacity could also be improved. When passing horizontal and vertical undulations, the unit showed good running properties.

UNDERGROUND OPERATION

For the last tests only components cleared for underground operations are used. The hydrostatic drive system e.g. is run on non-inflammable fluids only. Furthermore for control of the drives and the Jet-Miner an audio-frequency system supplied by Siemens is used by which all plant components (conveyor, Jet-Miner, machine haulage, and high-pressure pumps) are controlled centrally from a control con-sole.

Furthermore measuring equipment of Bergbau-Forschung is used which are cleared for underground operation. The following parameters are measured and recorded by UV-recorders and magnet-tape systems:

- hydraulic pressure of the hydrostatic drives (measure for chain traction effort)
- speed of the drives (travelling speed).

At Lohberg colliery the seam R_1 is prepared for the test under production conditions. The existing plough system is replaced by a Jet-miner.

The coal face is 260 m long and the seam thickness is of 1.45 m (Fig. 10) . The Jet-Miner is designed in a way that it can run on the existing plough guides. Only the special guidance system for energy and water supply needs to be fitted. At this time the trials are carried out, results will be given on the conference.

SUMMARY

After the very successful underground trials run with the experimental version coal-winning performances were achieved which are hardly achieved by conventional coal-winning methods, the prototype version was jointly designed by M.A.N.-GHH-Sterkrade, Ruhrkohle AG, and Bergbau-Forschung GmbH and constructed by M.A.N.-GHH-Sterkrade.

This Jet-Miner is suited for operation in seam thicknesses ranging between 1.0 and 1.6 m. At a travelling speed of V_M = 0.4 m/s webs of s = 0.4 m are scheduled to be cut. This requires a water pressure of p = 70 MN/m^2 at a flow rate of V = 162 l/min. The pumps on the machine require 215 kW of installed power. The Jet-Miner hauled by outside drives undergoes intensive testing on mock-up coal faces on the Bergbau-Forschung test rigs before being transferred to trial operation underground. After these surface trials, operation under production conditions started in seam R_1 of Lohberg colliery. Results will be described.

ACKNOWLEDGEMENT

The research and development work was sponsored by the "Bundesminister fur Forschung und Technologie (Federal ministry for research and technology)" as well as by the "Minister fur Wirtschaft, Mittelstand und Verkehr (Ministry for economics and transport)" of North Rhine-Westfalia.

The project was set up jointly by M.A.N.-GHH Sterkrade, Ruhrkohle AG, and Bergbau-Forschung GmbH.

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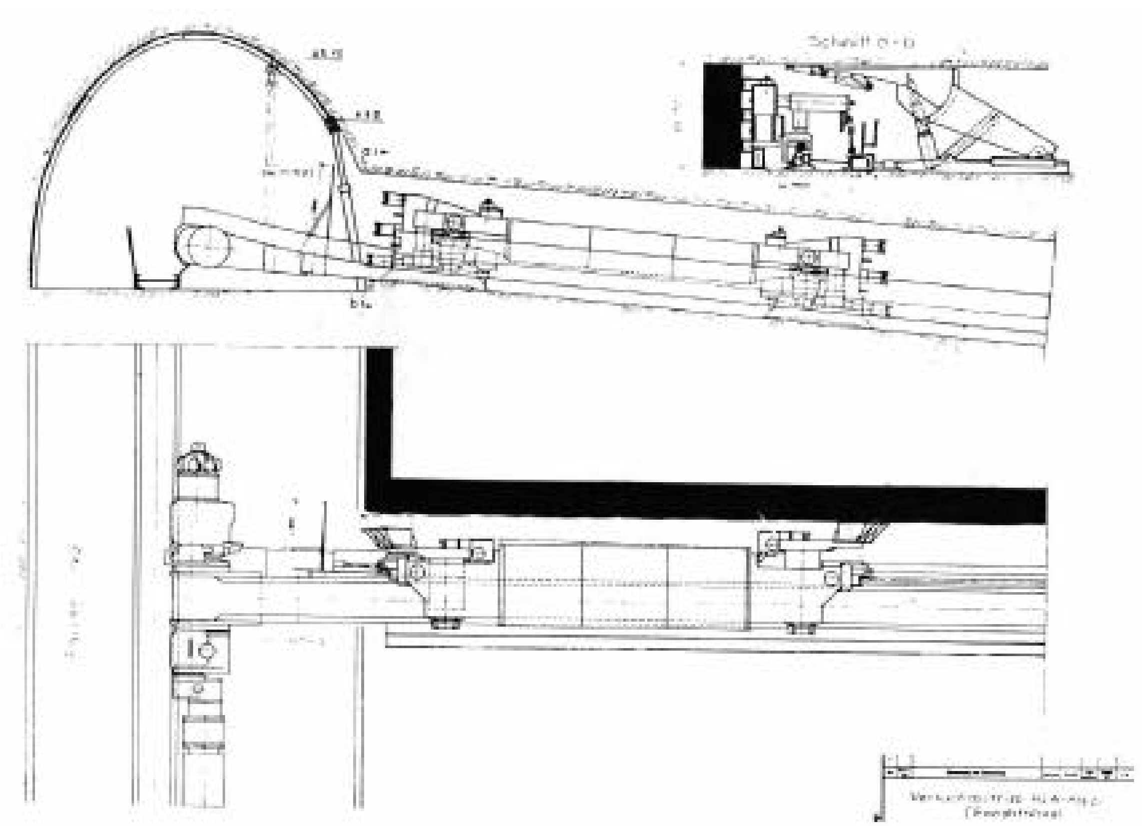

Figure 1. Experimental coal face, Lohberg colliery

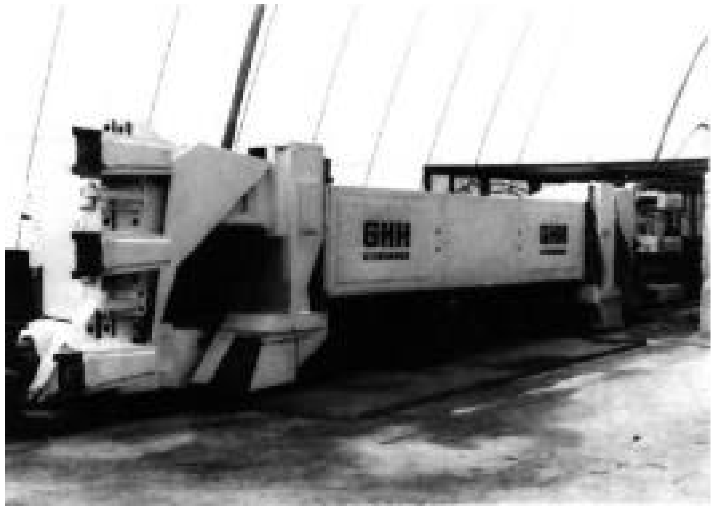

Figure 2. Experimental version.

Figure 3. Experimental version at the face.

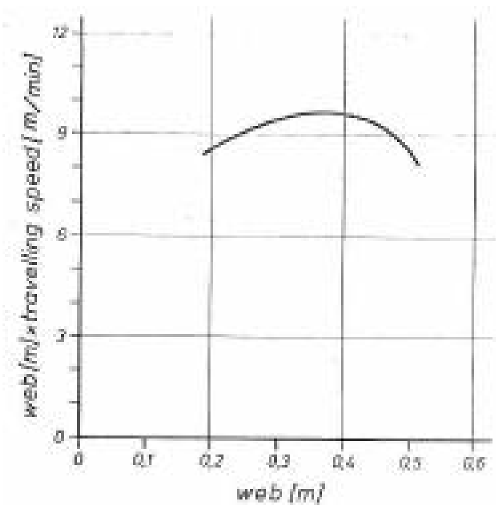

Figure 4. Product of web and traverse speed vs web.

Figure 5. Coarsely sized coal.

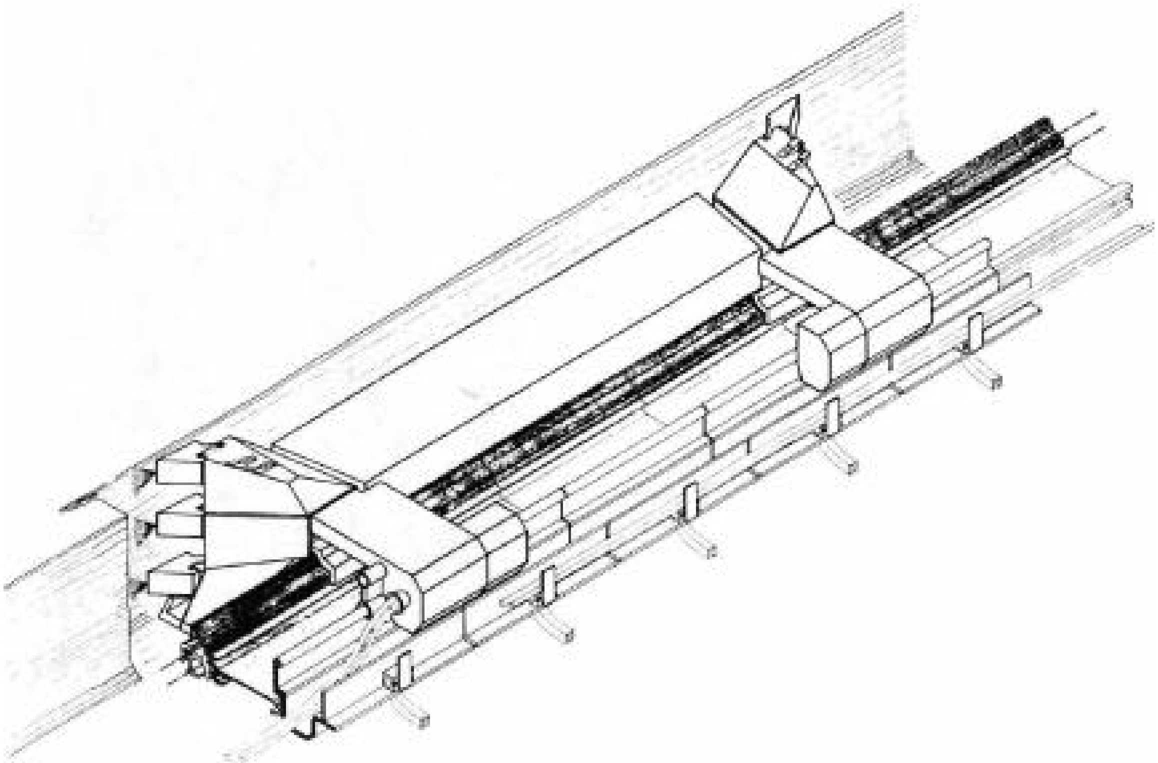

Figure 6. Jet-Miner prototype.

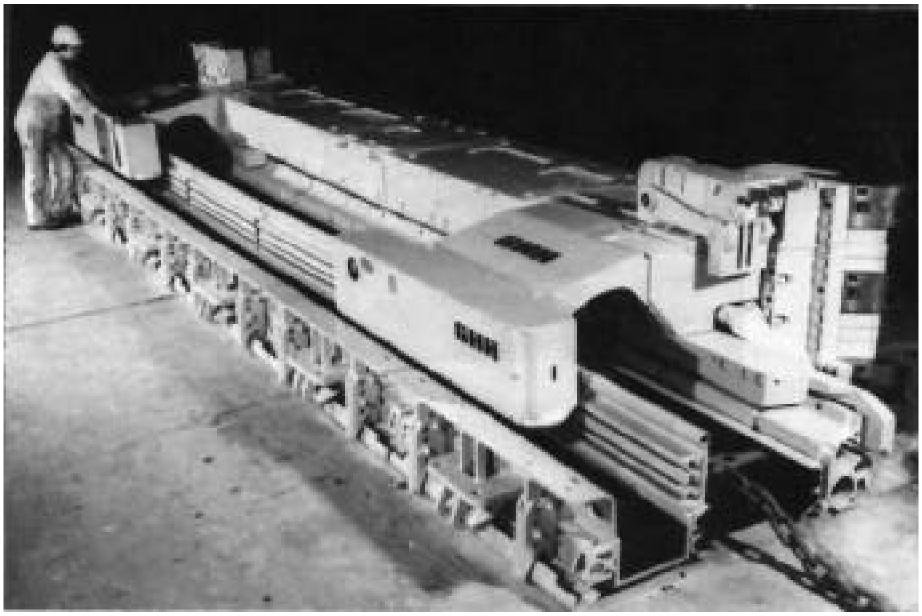

Figure. 7 Jet-Miner prototype

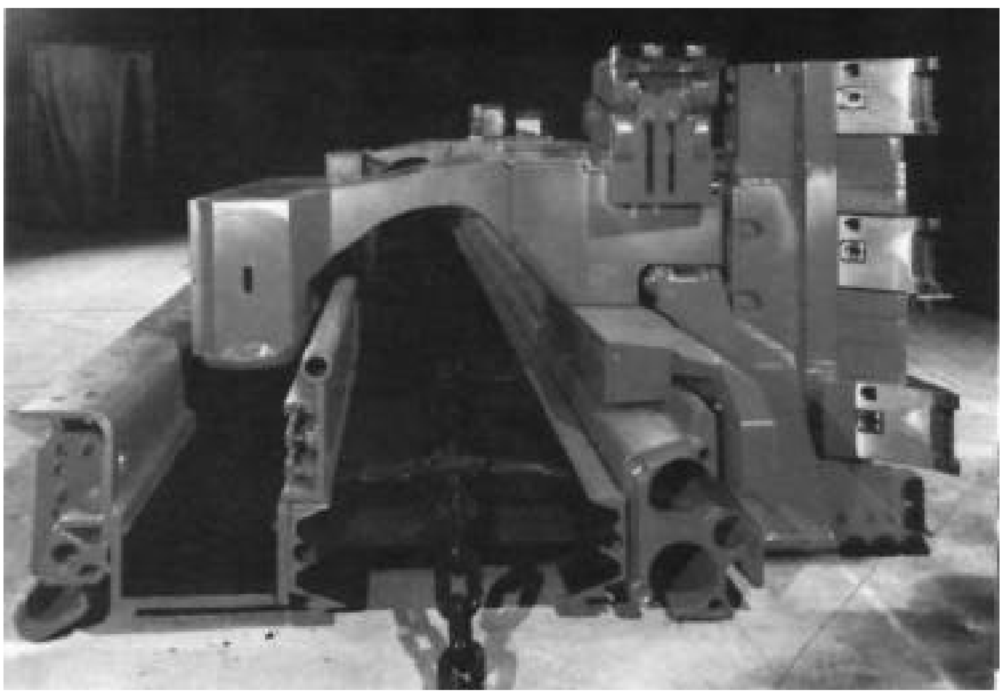

Figure. 8 Jet-Miner on the conveyor

Figure. 9 Test array at Bergbau-Forschung GmbH

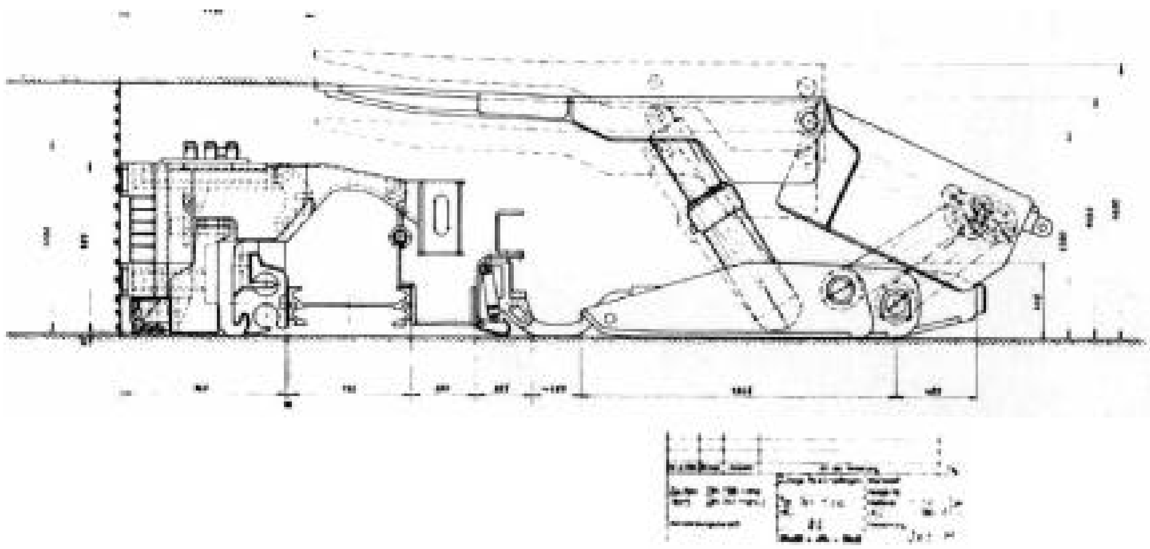

Figure 10. Jet-Miner at Lohberg Colliery

SOME PATTERNS OF TECHNOLOGY TRANSFER AND UTILIZATION IN WATER JET CUTTING

William E. Souder[1] and R.J. Evans**[2]

ABSTRACT

From time to time, major technological innovations arise that alter the time-honored ways of doing things. Such innovations can dramatically benefit society in terms of new products, increased outputs, reduced costs and general modernizations. However, the advent of these types of technologies is likely to be resisted because of their uncertain potentials, and because they disrupt established routines and industry patterns. This paper reviews the case history of water jet coal cutting and assesses its potential to become a major innovation within the next decade.

INTRODUCTION

Innovations in coal mining typically occur very slowly. Recent major innovations like the continuous miner, the articulated conveyor and the longwall are fundamentally very old technologies that have gradually evolved into their modern forms. Moreover, the evolution of these technologies has followed a circuitous route of sporadic efforts, with alternating periods of interest and disinterest among the developers and users (Souder and Evans, 1982; Souder and Palowitch, 1981).

Water jet assisted coal cutting is another older technology that has recently evolved into a new form. This new form has the potential to increase the health, safety, productivity and profitability of underground coal mining. If one considers that one-third to one half of today's underground mines could potentially be using jet assisted cutting, these advantages translate into very significant socio-economic benefits (Souder and Evans, 1981; Souder and Palowitch, 1981).

This paper reviews the technology transfer and evolutionary patterns of water jet cutting. An analysis of the results reveals many lessons about technology transfer and the management of new technologies. Based on these findings, prognoses are made for the future of this technology.

WATER JET MINING TECHNOLOGIES

Low Pressure Technologies

The use of natural flowing water to mine various minerals is an old technology. The Egyptians and Romans used inclined sluices and reservoirs for mining gold. This art was copied and modified by Welsh and Polish miners during the 17th Century. These practices subsequently found their way to California, Idaho, Alaska,

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Montana and Brazil during the gold rush of the 1800's, where they evolved and became known as "booming'. By 1852 this technology had been upgraded into a pump-hose-nozzle affair. The device was various called a "Giant" or a "monitor", and it had a water velocity up to 4 ft/sec. However, applications were limited by a court decision, the availability of near-surface deposits and the demise of the gold rush (Souder and Evans, 1982; Souder and Palowitch, 1981).

By 1867, this technology had found its way back to Europe. By 1918 one-third of Russia's industrial peat was being mined by mechanically assisted booming. By the early 1920's, giants and monitors were being used in Wales, Europe and Asia (Souder and Evans, 1982).

As Figure 1 shows, this early era was characterized by a criss-cross evolutionary pattern of the technology. With each new application, the user made some changes to adapt the technology to his particular use. This further facilitated some subsequent new application. Substantial technological progress was made by augmenting this basic technology with new technologies that appeared during this era. The ancient Egvptians and Romans were limited by the water velocities of naturally occurring streams and waterfalls, and by the state-of-the-art in gravity-fed technologies. The introduction of manual hydraulic pumps during the early 1800's permitted the Welsh and European miners to improve on these practices. The subsequent invention of the steam engine and the advent of steam power made booming more effective.

Medium Pressure Technologies

From 1935 to 1939, the Leningrad Institute developed a series of water cannons and monitors with swivelling and pulsating nozzles that sliced the coal from the face, and significantly increased productivity. Interrupted by the war, research resumed in 1946 with the construction of a special hydraulic mining institute at Stalinsk. By 1952, outputs of 600 tons/shift were being achieved with swivelling-pulsating monitors in Poland (Kramer, 1980; Souder and Evans, 1982).

In 1949 the American Gilsonite Company unsuccessfully attempted to duplicate the Russian experiences which they found described in the literature. However, it wasn't until 1954, when higher pressure pumps became available, that the company was able to develop a 2,300 psi system. This system was effective for extracting gilsonite, a unique material that couldn't be safely and economically mined by any other known methods (Frank, Fogelson, and Chester, 1972; Souder and Evans, 19B2).

As Figure 2 shows, this era was characterized by two paths of effort. In one path, the Polish and Russian research institutes engaged in basic research to develop some high productivity technologies. In the other path, Western Europe continued to reduce the existing technologies to practice. The first major U.S. underground mining application by American Gilsonite drew heavily on these basic experiences.

Development of High Pressure_Technologies

The U.S., Polish and Russian successes rekindled world-wide interest in water jet technologies. During the 1960's, West German, Russian, Polish, British Chinese, and U.S. coal mining engineers attempted to apply 2000 psi jet systems in a variety of seams with mixed results. In 1964 and 1965, Polish and Russian engineers conducted underground tests in longwall operations using a chock-mounted monitor.

In the fall of 1959, a Polish engineer conceived a system of high pressure water jets to augment a coal plow. In the early 1950's, Consolidation Coal Company carried out a series of field trials in the U.S. using 15,000 psi water jets. Though these trials were plagued by water handling and geologic problems, they established the superior dust suppressing characteristics of water jet technologies (Frank, Foqelson, and Chester, 1972; Kramer, 1980, Souder and Evans, 1982; Summers, 1979).

Because these approaches used large volumes of water (700-800 gal/min) that limited their coal applications, U.S. attentions shifted toward the industrial potentials for high pressure technologies. During the 1970-1981 period there was considerable activity in the use of 15,000-20,000 psi percussion and cavitation jets to cut, extract, drill and clean a wide variety of materials. Seeded by federal monies from several sources (U.S. Bureau of Mines, NSF, ERDA, and NASA), by the end of the 1970's these endeavors had spawned new equipment and practices in drilling, cutting and cleaning in several U.S. industries (Frank, Foqelson, and Chester, 1972; Kramer, 1980; Souder and Evans, 1982; Summers, 1979).

Meanwhile, European efforts continued their focus on coal. By the early 1960's, Polish mining enqineers had developed 8000 psi pulsating water jets capable of extracting 88 tons of coal per hour using only two men. Early in 1964, a Polish patent was filed describing a water jet assisted longwall plow concept. This system operated at 6000 psi, using a single oscillating jet to fracture the coal ahead of the plow. Demonstrated at the Typer Colliery (Katowice) early in 1965, this equipment extracted as much as 250 tons of coal per hour. In 1964, the Soviet Union developed the MVU, a machine somewhat like the Polish equipment, except that it undercuts and top-cuts the coal before wedging it off into a conveyor. In 1968, the West German government funded several consortia of equipment makers and mine operators to develop and test water jet equipment. By 1977, a consortium of GHH-Sterkrade, Ruhrkohle AG and Bergbau-Forschung GmbH had successfully proven two experimental units. These units, named the 'Hydrohobel" and the "Jet Miner", were based on water jet experiments that the government had previously funded at various research institutes. From 1978 to date, a series of comprehensive in-mine trials were run on these units, and further modifications were made to them (Kramer, 1980; Schwarting, Goris, Kramer, and Rifle, 1981; Souder and Evans, 1982; Summers, 1979).

As Figure 3 shows, during the 1960 to 1980 period, water jet research and developments in Europe focused on coal while the U.S. concentrated on non-coal applications. These different orientations reflected differences in national priorities, industry structures, private sector economics and perceived needs. For example, coal-dependent Poland sought a higher-output mining alternative for their thin, undulating seams that were not amenable to conventional hydraulic methods. West Germany sought a high output methodology that would be needed to handle the challenges of mining harder coals and thinner seams, as the thicker seams became worked-out.

Growth of High Pressure Coal Technologies

In West Germany, Westphalia's fixed-jet "Hydrohobel" and GHH-Sterkrade's oscillating-jet "Jet Miner" underwent full scale tests during 1980 and 1981. Using 48 gal/min of water at 10,000 psi, a traveling speed of 65 ft/min and 19 in. web, outputs up to 21 ton/min were recorded in West German mines. Improved coal sizes, low dust (70% less than shearer-loader), lower moisture content (50% less than shearer-loaded coal) and high outputs were reported for coals that were "problem areas" and "difficult to plow" (Frank, Fogelson, and Chester, 1972; Kramer, 1980; Schwarting, Goris, Kramer, and Wille, 1981; Souder and Evans, 1982).

By the end of 1981, the idea of using high pressure/low volume water jets to augment the mechanical winning of coal had become relatively well established in Europe. Meanwhile, the many burgeoning industrial applications of high-pressure water jets had effectively moved this technology from the laboratory to the field in the U.S. During the early 1970's, the U.S. Bureau of Mines funded several investigations related to the use of the water jet assisted mechanical coal cutting devices. As a result of this work, the Bureau funded three parallel conceptual approaches to the application of water jets in coal. In one of these approaches, a modified hydraulic mining system was designed. In another, a 50,000 psi jet-assisted continuous mining device was specified. The third approach, dubbed the "Hydrominer", was based on the use of 10,000 psi water jets to cantilever cut the coal ahead of a plow. Based on the results from these studies, in 1975 the Hydrominer design was selected for further development (Kramer, 1980; Schwarting, Goris, Kramer, and Wille, 1981; Souder and Evans, 1982.

Meanwhile, European progress continued. In 1972, the Soviet Union reported outputs of over 600 tons/ shift with 87% to 90% recovery in the Uklony district with MVU-type equipment. This was two to three times the productivity of other equipment, and up to 10% greater recovery. By the end of 1974, the Carl Funke and Hansa Collieries in West Germany were reporting average daily outputs of 1,700 tons to 1,900 tons with Hydrohobels and Jet Miners (Kramer, 1980; Schwarting, Goris, Kramer, and Wille, 1981; Souder and Evans, 1982).

In 1975, a Polish mine at Katowice set a new output record using water jets. In 1976, a West German mine set an output record with a Hydrohobel. By the end of 1977, the development of a very high pressure jet MVU was completed at the Leningrad Mining Institute. In 1980, new output records were established in a German mine using the Hydrohobel (Schwarting, Goris, Kramer, and Wille, 1981; Souder and Evans, 1982).

Early in 1974 the U.S. Bureau of Mines funded construction and development of Hydrominer I. This work involved the development and fabrication of effective nozzle configurations, the design of an optimum cutting head, laboratory testing, field tests of the entire device in a mock mine face, and the construction of a prototype for actual deep mine testing. In the summer of 1976, the feasibility of Hydrominer I was demonstrated above-ground. By the summer of 1977, a redesigned version called the Hydrominer II had been developed. However, tests to date have been inconclusive and the question of the viability of water jet cutting in U.S. coals remains unanswered at this time (Souder and Evans, 1982).

Fiqure 4 chronicles the events and accomplishments during the 1970-1980 period. During this era, significant advances were made in the application of water let technologies to coal mining in Europe. But the gap between the U.S. and European states-of-art continued.

AN ANALYSIS

A review of Figures 1 through 4 provides an overall picture of the birth and maturation of water jet cutting. Spurred by the need for higher productivities, the technology arose rather naturally from crude ancient practices. Several processes transitioned the technology from birth to maturity, as summarized in Table 1. During the low pressure era (Fig. 1), cross-cultural transfers of technology and sporadic adaptations from other uses were the predominant processes. A kind of criss-cross evolutionary pattern predominated, as the technology found its way out of its crude origins. By contrast, the medium pressure era (Fig. 2) exhibited a more focused, linear sequence of events. Here, private and public sector research funds were concentrated on the embryonic results from the low-pressure era, to elaborate the technology and push it to a higher status. During the high-pressure era (Fig. 3), efforts focused on the development of the basic technology, which then became applied and implemented during the coal era (Fig. 4).

Throughout its evolution, water jet cutting has been aided by the appearance of several inventions and facilitating technologies. As depicted in Figure 5, the invention and development of hydraulic pumps, steam engines, high pressure orifices, high pressure pumps and pulsating jets pushed the technology through sequentially higher thresholds. As Figure 6 shows, substantial technological achievements were made during the 1960 to 1980 period. At the start of this era, 3000 to 4000 psi jets were considered a threshold level of common know-how. At the end of the era, 20,000 to 50,000 psi was a common working state-of-art level in the U.S. In Europe, the potential productivity of water jet coal cutting doubled during this period. This dramatic achievement was accomplished by integrating water jet and coal plow know-hows.

CURRENT STATUS OF WATER JET COAL CUTTING

Technological Capabilities

Higher productivity and reduced costs have been demonstrated in European underground coal mines for water jet assisted coal mining. The Hydrominer I trials have indicated that similar performances may be possible in U.S. coals. Recent analyses have indicated that the productivity of U.S. shearer longwalls could be doubled with little or no increase in haulage and energy requirements (Schwarting, Goris, Kramer, and Wille, 1981). The future potentials for water jet technologies are thus substantial.

Perceived Advantages and Disadvantages

Tables 2, 3 and 4 present some of the advantages, disadvantages and barriers cited by potential users of water jet cutting. Water jet assisted coal cutting techniques are essentially the same as those employed today with unassisted plows and shearers (Schwarting, Goris, Kramer, and Wille, 1981; Souder and Evans, 1982). In fact, mine planning and development practices, production operations, face equipment (roof supports, winning equipment and face conveyors), haulage and logistics are essentially unchanged when jet assists are added Moreover, today's water jet technologies use no more water than conventional dust suppressing shearer sprays. These considerations facilitate the adoption and use of jet assists in underground mining. On the other hand, the mechanism of coal failure under water jet attack is considered to be a controversial issue by most U.S. mining authorities. Performance seems to vary somewhat unpredictably with variations in nozzle characteristics and coal structures. Relatively little is known about the relationships between various jet cutting variables, jet cutting efficiencies and U.S. coal parameters (Souder and Evans, 1982). These considerations work against the rapid spread of water jet assists in U.S. underground coal mining operations.

A sample of mine operators were asked to rate their performance needs, and a sample of water jet cutting experts were similarly asked to rate the capabilities of the technology (Souder and Palowitch, 1981). According to the results, as shown in Table 5, water jet coal cutting technologies may be deficient (Degree of Capabilities is less than the Perceived Importance) in four areas: capital costs, operating costs, flexibility of operation and face recovery. The widespread adoption and use of water jet technologies may not be anticipated until these deficiencies are eliminated. It must be noted that Table 5 shows another interesting result. Lighter weight equipment, less complex equipment and superior recovery of fines-three aspects of water jet cutting that have been touted--turned out to be relatively unimportant.

SUMMARY AND CONCLUSIONS

Figure 7 summarizes the key events in the evolution of water jet coal cutting technologies. Water jet methods have evolved over a very long time period: nearly 3,000 years. The technology has evolved through five major technology transfer stages: adaption, research and extrapolation, improvement, modification and implementation. Beginning with the crude practices of the ancient Egyptians and Romans, the initial embodiment of the technology in mechanical devices like water cannons and monitors was "pulled" or fostered by the need for greater productivity in mining precious metals. Man's quest for new knowledge, the inherent human need to respond to new technological capabilities and the U.S. industrialization movement of the 1800's and 1900's "pushed" technology to higher plateaus. Facilitated by the invention and development of the steam engine, the hydraulic pump and the jet nozzle, the potentials of the technology began to attract the attention of new users. Intrigued with the possible capabilities of this technology, serious R & D efforts were mounted by both the private and public sectors. After a haitus in non-coal applications, water jet cutting has now been returned to the coal mining field by various entrepreneurial developers, and it has begun to spread among a population of users. These patterns are typical of coal mining innovations (Souder and Evans, 1982; Souder and Palowitch, 1981).

What do we learn from this case? Table 6 lists ten key factors that have influenced the evolution of water jet cutting technologies. These factors reflect important considerations for anyone desiring to develop or use a new technology. First of all, a pool of disembodied know-hows must be present. The crude 16th and 17th centuries practices represented a pool of disembodied knowledge-- a potential technology waiting to happen. Second, what makes a technology "happen" is a need or a technology pull: a problem that can be solved if the know-hows can be harnessed and embodied in some device. Timing is all-important: the technical capability and the need must occur in the same time frame. For example, water cannons were devised in response to the needs of the gold rush. Human behaviors are essential to this process. People must have sufficient desires for new knowledge and the urge to respond to the situation (factors 3 and 4 in Table 6). Facilitating technologies (factor 5) are also essential. For instance, water jet cutting could not have developed to its current state without the development of jet nozzles. Research and development funding, and the desire to have better equipment and advanced capabilities (factors 6 and 8 in Table 6) are also important. Note that water jet <u>coal</u> cutting did not become a prominent technology until water jets had become established and been proven in industrial cutting and cleaning applications. Water jet cutting took a long excursion into peripheral fields before it <u>returned</u> to its coal origins (factor 7). This is a typical occurrence with coal mining technologies (Souder and Evans, 1982; Souder and Palowitch, 1981). Finally we must note that a real application need must exist (factor 9) and many users must adopt the technology before it becomes established. A kind of "neighborhood effect" often occurs, as new users pick up the technology in emulation of others who are using it (factor 10). Thus, the technology spreads or diffuses through the entire population of users.

What does this case tell us about the types of policies that can be used to encourage technology transfer and utilization? Table 7 lists the actions that are suggested by the water jet cutting case. Aside from the traditional advertising and promotional activities (items 1, 2, 3 and 5 in Table 7) that are used by equipment makers, four other actions are important. Research into facilitating technologies and the encouragement of their development is essential to the speed evolution of the technology. And any activities that generally spur the user to appreciate new methods or encourage a desire for new knowledge are important. For instance, American Gilsonite's use of water jets was spurred not only by their unique needs, but by their corporate culture of high openness and receptivity to new things (Schwarting, Goris, Kramer, Wille, 1981). Finally, it is often important to purposely trial a fledgling technology in a peripheral, less technically demanding field. This provides vital pilot experience and an opportunity to explore other capabilities of the technology, while avoiding the risk of loss of the major market in the event of a failure.

Is water jet cutting the next major innovation in mining? The last major breakthrough in coal mine cutting technology was in the 1940's when the introduction of tungsten carbide significantly increased the life and cutting ability of mechanical tools. Recent studies with water jet cutting indicate that significant improvements can be made to the cutting process when using this technology (Kramer, 1980; Souder and Evans, 1982).

Using conventional cutting technology, today's mining machines such as the continuous miner, longwall shearer, tunnel boring machine, and roadheader have generally been optimized with respect to their cutting ability in relation to their size and weight. All of these machines use thrust and torque, which is a function of the machine's weight and tractive effort, to provide the forces required for cutting. Most attempts to increase thrust and torque have resulted in increased machine weight and size thereby decreasing the machine's maneuverability and productivity. What is needed is improved cutting technology to substantially increase machine productivity.

The potential for a major advance in cutting technology has been demonstrated using water jets or water jet assisted cutting systems. Experiments have been conducted in a wide variety of rock types and coal, both in the laboratory and underground, that have demonstrated that not only can substantial improvements in cutting performance be achieved but also substantial improvements in health and safety. These improvements include: considerable reductions in dust during the cutting operation; reduction in fines; elimination of frictional sparking and ignitions; and in the case of water jet assist cutting, significant reductions in pick forces and increased pick life (Souder and Evans, 1982).

One can clearly see the potentials of water jet cutting in U.S. coals. The time is ripe for an enterprising equipment maker and an entrepreneurial coal company to collaborte in a successful field trial that ignites a bandwagon of water jet coal cutting applications.

ACKNOWLEDGEMENTS

The authors are indebted to Mr. Ernest A. Curth and Mr. John Kovac, U.S. Bureau of Mines, for their comments on an earlier draft and to Dr. D.A. Summers, University of Missouri-Rolla and Dr. E.R. Palowitch, consultant, for many helpful inputs.

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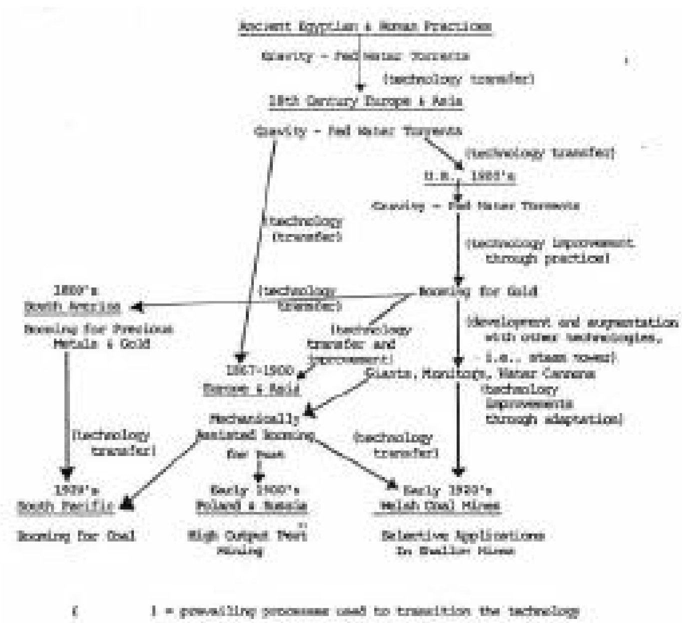


Figure 1. Technology evolution during the low pressure era.

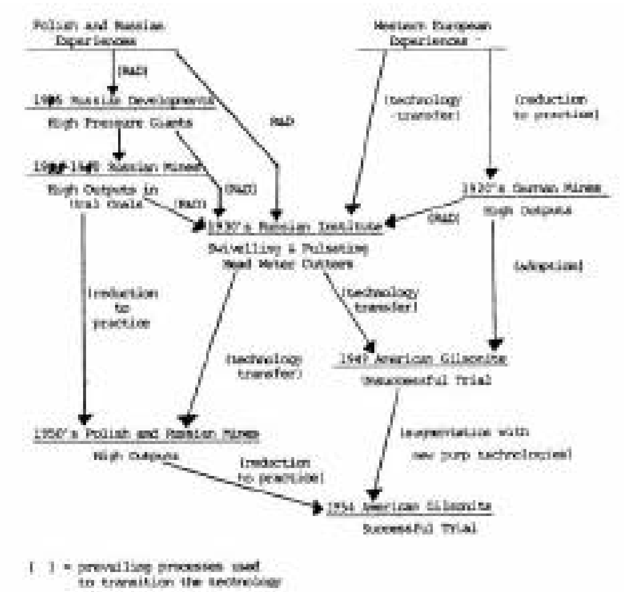


Figure 2. Evolution of medium pressure water jet technologies

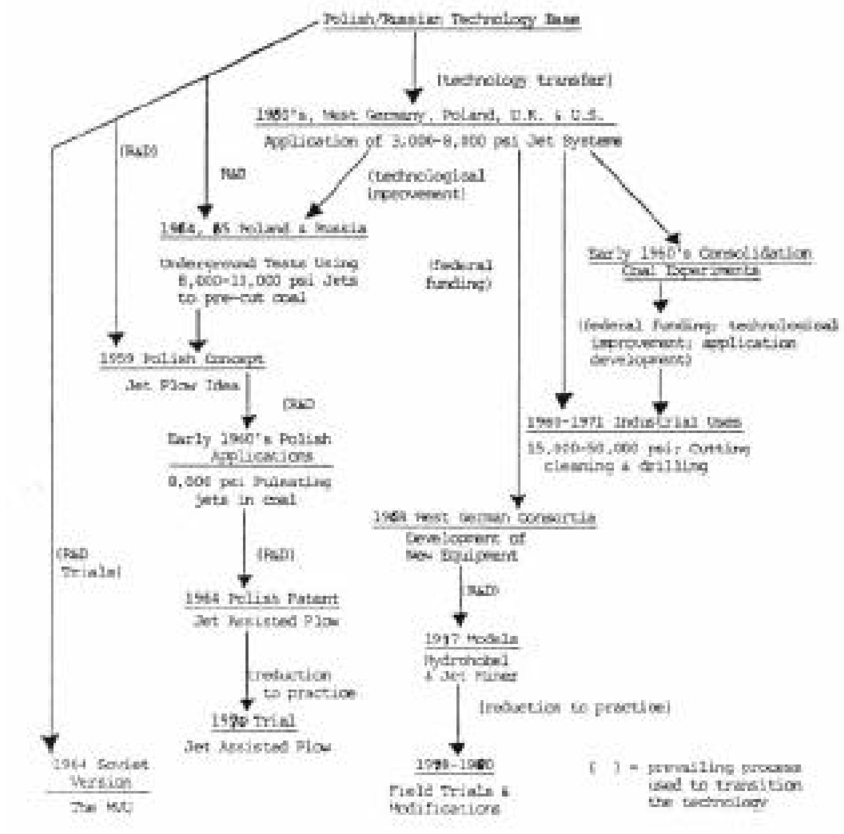


Figure 3. Emergence of high pressure water jet technologies

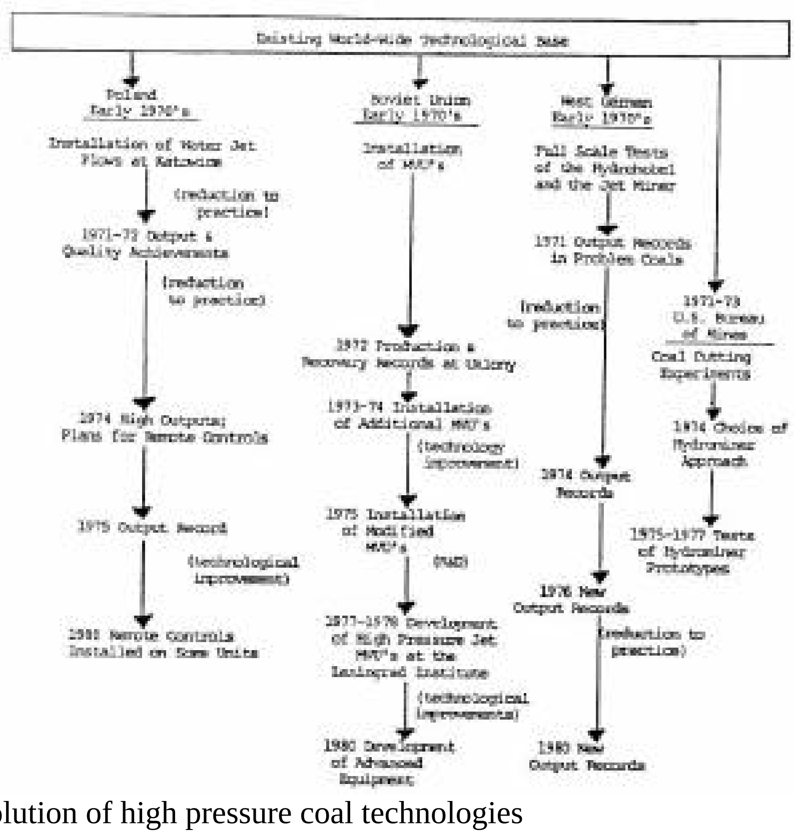


Figure 4. Evolution of high pressure coal technologies

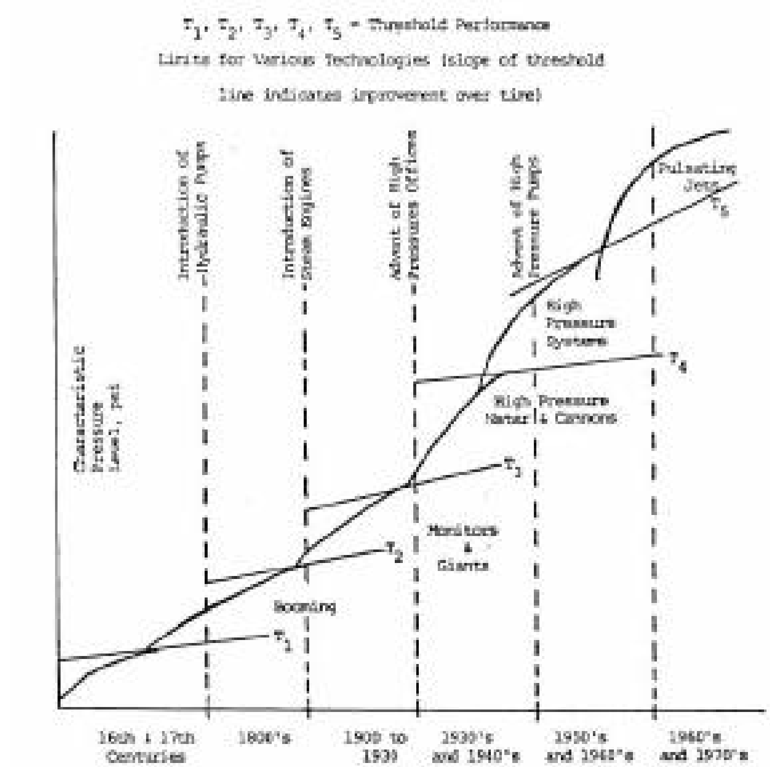


Figure 5. Technological achievements and improvements.

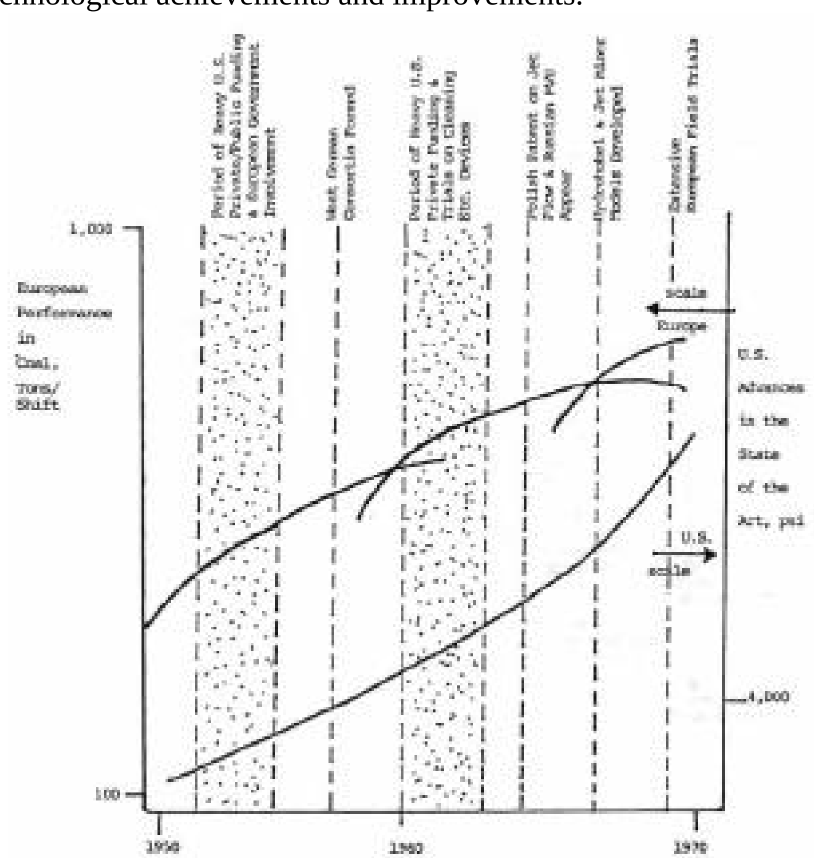


Figure 6. Performance advances over time.

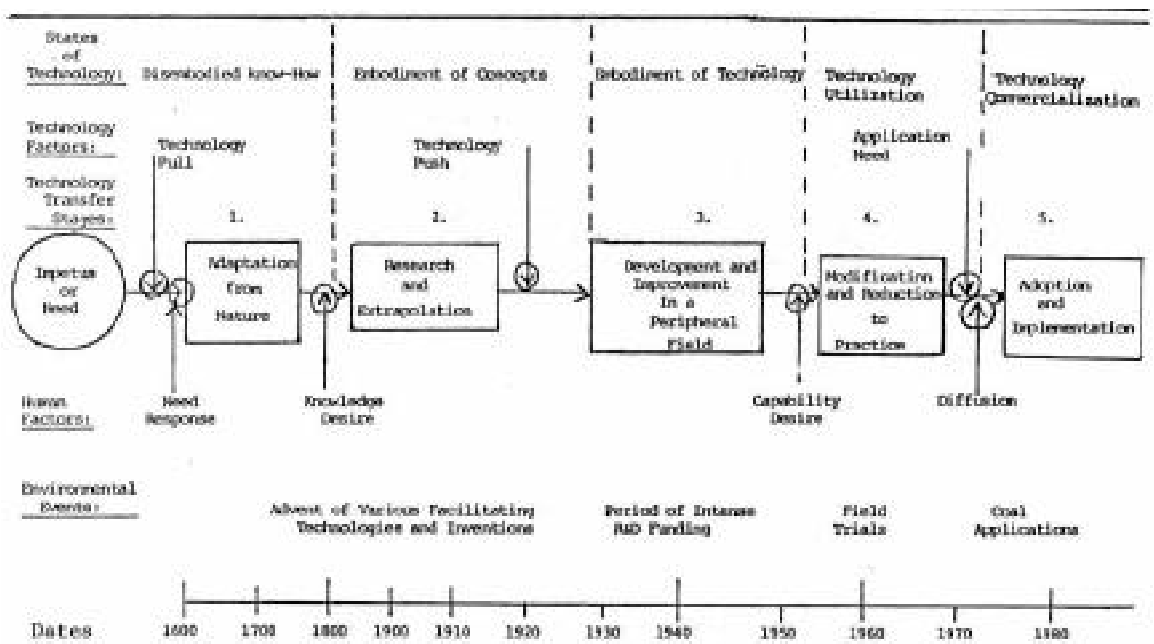


figure 7. Technology evolution model

<u>Table 1. Summary of Eras.</u>

Era	Primary Process
Low Pressure (Figure 1)	Cross-cultural Technology Transfer Adaptation from Other Uses
Medium Pressure (Figure 2)	Basic Research Elaboration of Earlier Research Results
High Pressure (Figure 3)	Research and Development Technological Improvement
High Pressure Coal (Figure 4)	Reduction to Practice Implementation

<u>Table 2. Perceived advantages of water jet assists.</u>

1. Intrinsic safety from sparking and methane
2. Healthier, dust-free environment
3. Longer life of cutting tools4
4. Lover maintenance costs and downtimes
5. Large-sized coal with fewer fines and better clean-up
6. Reduced coal washing costs
7. Capability to cut larger webs with less horsepower
8. Ability to mine faulty seams
9. Reduced haulage forces and vibrations
10. Generally higher productivity
11. Lower energy requirements
12. Fewer men required underground

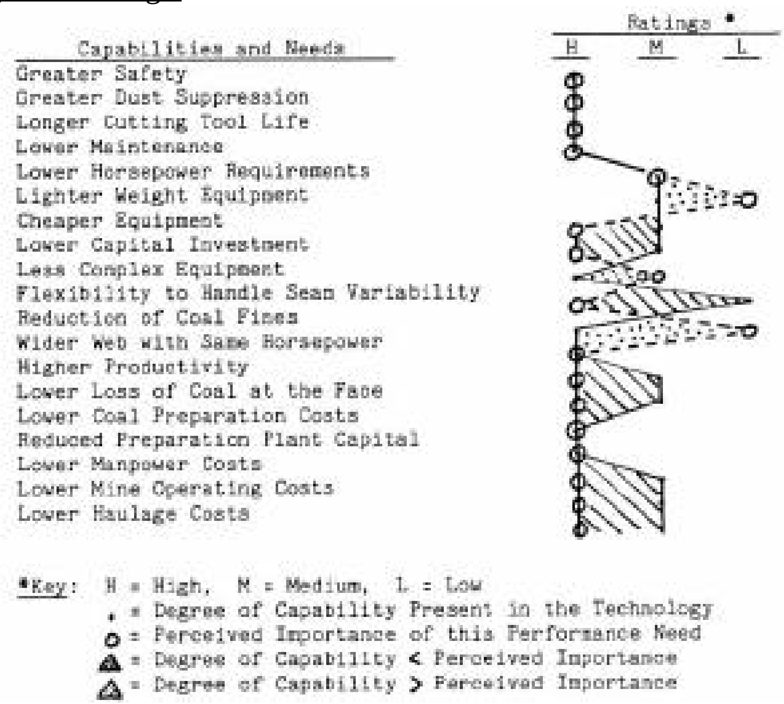
Table 3. Perceived disadvantages of water jet assists.

1. Potential operator safety hazards with jet pressures
2. Potential unreliabilities in the current technologies
3. Potentially higher start up costs for a new mine
4. Problems associated with hauling larger coal sizes

Table 4. Perceived barriers to the widespread adoption and use of water jet technologies.

1. The general uncertainty of the future demand for coal
2. The general uncertainties connected with any unproven technology
3. Uncertainties about reliabilities (even a known unreliable machine may be preferable to one of unknown reliabilities)
4. Apprehensions that jet assisted equipment will make the mine operator dependent on specific types of possibly hard to obtain equipment
5. Apprehensions about accessibility of the equipment for repairs

Table 5. Analysis of ratings.



Tale 6. Key Influences

1.Disembodied Know-How/Technological Potential
2.Need/Technology Pull; Timing
3.Need Response.
4.Knowledge Drive
5.Facilitating Technologies
6.R&D Funding
7.Development In Peripheral Fields
8.Capability Desire
9.Application Need
10.Bandwagon Effects

Table 7. Change Agent Actions.

1. Help user define needs.
2. Create need awareness.
3. Create bandwagon effects.
4. Encourage facilitating technologies.
5. Encourage desire for new knowledge.
6. Provide R & D Funding
7. Develop in a less demanding peripheral field.

THE NEW TECHNOLOGY OF HIGH PRESSURE JET CUTTING AND THE DEVELOPMFNI OF SUITABLE MACHINERY FOR ITS APPLICATION

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ABSTRACT

If the potential advantages and possibilities of new technologies are to be fully realized, they require "basic equipment" and 'basic techniques" for their mounting, for their support and to wring them to full efficiency. Experience has shown that in nearly all cases new basic equipment and basic techniques have to be developed for new technologies. This is time of the high pressure water jet cutting technique whose physical and technical fundamentals have been worked out 10 years ago with promising application possibilities in mining.

The reason for the current lack of widespread success in mines is, at least to a certain extent, that the necessary new basic equipment for the respective possible applications have not hitherto been systematically developed. The construction and development methodics open up the possibility of developing an entire series of alternative solutions to the requirements profile of conceivable applications of high pressure jet cutting technique in mining. These solutions could, in a subsequent assessment procedure, be compared or combined with one another and through a selection procedure, very probably permit the development of suitable basic equipment. Although this systematic approach leads to an increased accuracy in the search for new conceptions for the basic equipment, it does not obviate the need for detailed development and exhaustive testing and improvement up to final operational readiness.

The high pressure water jet cutting technique is suitable for high-grade cut-offs, bore-holes and for specific areas in the crushing and winning of minerals. From the many possibilities, we note as examples: the production of exact cut-offs at the roof and floor which follow the irregularities of the deposit formation precisely; the combination or integration of cutting tools and high pressure jet cutting to form chisels and picks resistant to abrasion; the production of bore-holes for roof bolting with curved boring equipment for "around the corner' boring and the production of contour slots for driving roadways, whereby the combination of driving equipment and the placing of the roadway support could be greatly improved.

INTRODUCTION

The hydro-mechanical winning of minerals has been successfully used in various coal fields of the world for a long time. In the mid 70’s, promising laboratory and test-stand investigations as well as tests under simulated operational conditions were undertaken at various locations and particularly extensively at this university, in order to develop thus technology up to high pressure water jet cutting. Hydromechanics, as it has already been, or is being applied with pressure of 1300 - 2600 psi above all in the coal fields of the Soviet Union, Canada, the People's Republic of China and Japan as well as in a pilot project in the Federal Republic of Germany, has numerous application possibilities in and outside mining. The list of lectures of this Second U.S. Water Jet Conference shows the multifarious and widespread use of high pressure water jet cutting with pressures of 8700 - 43500 psi. The possible breadth of application ranges from the elimination of mould remains on castings, from the removal of highly adhesive coatings e.g. the cleaning of coking oven doors, through metal cutting which cannot be effected by flame cutters, to the crushing and dissolving of rocks, minerals and concrete.

Under the breadth of application of the new technology "high pressure water jet cutting" which is constantly expanding, it is surprising that nearly 10 years after the working out of the technical and physical fundamentals in exceptionally promising laboratory and field investigations, the number of operating machines in mines using this technique is relatively low. In pursuing this question by applying the rules and findings of the construction and development methodics, it is far from my intention to diminish in the least the achievement of and praise due to the scientists and engineers who have hitherto devoted their efforts to the use of high pressure water jet technology. Rather, I wish to urge the enhancement of the procedures used so far through the addition of approaches in general development and construction methodics for the introduction of high pressure water jet technology in mining, in order perhaps thus to arrive at a more rapid and broader application of this technology.

The physical and technical processes which occur under the impact of high pressure water jets with starting pressures of for example 8700 - 43500 psi on rocks and minerals open up - as the already mentioned laboratory and field investigations have shown - completely new possibilities for the production of separation planes and bore-holes as well as the crushing and winning of minerals. High pressure water jets are after all only tools or components in the destruction and crushing of rocks and minerals. In order therefore, to fully exploit these possibilities, they must be brought into operation on suitable for the new technique optimally constructed machines, equipment and systems. In this context, I am thinking less of the obviously necessary operation orientated and durable construction of the machines and equipment as well as the individual components, which represents an unconditional prerequisite for mines. Far more, I wish to investigate the question whether, in their basic conception, continuous miners, coal ploughs, shearer loaders, part face heading machines, impact rippers or drill jumbos constitute the optimal realization of the possibilities offered by the high pressure water jet technique. Here, the results of investigations of basis equipment or basis techniques must be applied, which have been more or less consciously carried out in an aim-orientated fashion in the introduction of each new technology. I should like to illustrate this with the aid of two examples.

CONSIDERATIONS FOR THE DEVELOPMENT METHODIC

When the invention and development of an internal combustion engine independent of external cables offered the possibility (Figure 1) of a light self-driving rail-free vehicle - called the automobile -, the motor i.e. the new technology was immediately mounted onto the contemporary customary and widespread basis equipment namely the coach, as the upper part of Figure 1 shows . The motorized vehicle represented is the first motor car from Daimler in 1886, proceeding from the basic equipment of a "coach". The only concession to the new technology was the removal of the redundant draw bar and its substitution by another steering system . In 1903, just about two decades later it had been realized, that in order to retake full use of the new technology, the negligibly altered coach was obviously not the optimal basic equipment for the internal combustion engine, as the basic conception of a motor vehicle resembling today ' s automobile, in the lower half of the Figure (Figure 1(b)) shows.

In the first decade of our century the suggestion was made for tie first time and the attempt undertaken to install percussive compressed-air tools in mining (Figure 2). As the upper half of the picture demonstrates, the new technology "percussive compressed-air tools were at first adjusted to the basic equipment at that time current in coal winning, namely the pick. The equipment was thus also called "mechanized pick ax". In this form, the new technology could however not be rendered fully effective, because it was mounted on the basic equipment of a pick. It was also 20 years later with the discovery of a new basic equipment for the new technology, i.e. the pneumatic pickax, that compressed-air tools directly came to be distributed world- wide above all in coal mining. There are almost endless examples of this sort.

To apply this basic principle of development theory to the high pressure water jet technique (Figure 3), we must next look at the inherent advantages and disadvantages of the physical and technical processes of this technology. In the context of such a general coverage, these must be reduced to a very simplified level. Significant advantages to be cited are the extraordinarily high energy density in the area of impact in the mineral and rock and above all the high destruction and crushing effect, dependant on the formation and composition of the mineral and resulting from the complex processes in the area of impact and underlying mineral area. A further advantage is that at the point in the area of impact where the energy is introduced into the mineral, there is no wear or the tool. As disadvantages we must mention, above all in the basis of experience up to now, the relatively high specific energy input per volume unit of cut material as well as the technology specific input for equipment e.g. driving motor, pump, pressure controller, steering and control equipment, jet nozzle and movement mechanisms for jet nozzles. From this it can already be deduced that the high pressure water jet technique is probably only technically meaningful and economically feasible for high-grade crushing and winning tasks, which could be done by conventional tools and for operations, where the specific energy input per unit volume of cut mineral is not of primary importance.

In the next step, both the possible applications in mining and the point at which the machines and equipment hitherto employed have reached their limits or have been unable to realize or solve specific tasks within the existing methods and systems, must be examined. The requirements profile for such application possibilities will have to be precisely formulated and drawn up in detail with regard to technical and economic considerations as well as the optimization of the coal winning method or of the overall system.

Using the requirements profile and taking into account the advantages and disadvantages of the high pressure water jet technique, numerous or in any case several, alternative solutions for appropriate basis equipment or basis technologies must be looked for with which the new technique can be applied in the most effective manner possible. The construction and development methodics are the most useful procedures to find these alternative solutions. It is possible that in certain exceptional cases, the basic equipment or technologies presently in use can, under certain conditions, be combined with the high pressure water jet technique. As a rule, however, new basic equipment tailored to the specific need of the high pressure water jet technique will produce optimal effectiveness.

The final stage of the search for new conceptions for the optimal use of the high pressure water jet technique in mining is the technical and economic assessment of the individual solution possibilities for the basis equipment and the subsequent selection of combinations suited to their realization. Of course, this methodical procedure, which offers the chance of finding entirely new conceptions, can save no one the thorny journey of strenuous detail development, testing, improvement and constant requisite alterations up to operational readiness.

TECHNOLOGICALLY AND ECONOMICALLY FEASIBLE APPLICATIONS OF THE HIGH PRESSURE WATER JET TECHNIQUE IN MINING

An overview of the overall spectrum of the application possibilities for the high pressure water jet technique is in the present state of development certainly not possible. I can thus only try, with the aid of certain selected examples from my own experience and activities to print to what paths are opened by the development of new basic equipment adapted to the high pressure water jet technique. If, using the standard values in the literature, one compares the specific energy input per unit volume of cut mineral for mineral crushing, the winning process or for boring (Figure 4), it is immediately evident that the specific energy for coal winning by means of the coal plough, shearer loader or continuous miner is nearly two tenth powers lower than the specific energy input needed with the exclusive application the ultra high-pressure water jet technique.

With the jet-miner of the GHH in Oberhausen-Sterkrade which in an improved version is now being used in a second underground pilot application, the specific energy input values are of a similar order of magnitude with coal winning machines with mechanical cutting tools. The reason for this is that high pressure water jets and mechanical cutting equipment have in this machine been combined in a meaningful way. The values for the actual winning process with the coal winning machines with mechanical cutting tools, at present in operation are even lower, as the specific energy input also includes loading. The floor cut with the coal plough, which is to be effected precisely between the coal and the rock underneath by means of the bottom blade, is particularly difficult and numerous technical steering and design measures are needed to render this high-grade cut reasonably operationally satisfactory. Here, in longwall mining under certain conditions, the feasible application of the high-pressure water jet technique probably in combination with mechanical tools is a possibility. The same is true for the production of separating planes between seam and roof.

To attempt a general formulation of the requirements of the winning process in seam deposits can be said: at the roof and floor an exact cut must be effected which follows the irregularities of the deposit formation, while in the remaining area of the seam the winning can be carried out in any form which results in transportable fragmentation. Roof and floor cut in longwall mining would thus be conceivable areas of application for the high pressure water jet technique presumably in combination with mechanical tools.

The combination of, or better still, the integration of mechanical cutting tools and high pressure water jet technique appears to be an exceptionally fruitful development path (figure 5). In accordance with the above mentioned considerations, new basic equipment must be developed which is optimally tailored to such new tool forms. Mechanical tools for coal winning and still more for the cutting process in the stone must be produced with a relatively large blade angle for stability and durability. The burden then falls onto the angle of rake which leads to a reduction of the tensile stress which in turn is extraordinarily important for cracking and crushing. This also promotes the formation of the built up cutting edge which hinders the cutting process. Smaller blade angles which would suffer from these disadvantages to a lesser extent would, with the currently available strength of material, lead to an exceptionally high wear on the tools and an extremely short life span for the equipment. An integration of mechanical tools and high pressure water jets in the outlined form would on the one hand, enable an unusually small blade angle, and in the other exclude the formation of a built up cutting edge and lead finally to winning tools capable of continually renewing itself and thus not subject to wear and tear.

In European coal mining above all, where the longwall system with two parallel roadways is the most wide spread winning method. In this the surrounding rock is also usually extracted, roadway support is used in the form of rigid or flexible steel arches or frames. When the driving of the roadways is conducted with a part face heading machine, this is at a standstill during the installation of the standing support. On average, this stoppage time for all the machines used in Germany is 45% of the potential running time. Neither technical auxiliary devices nor organizational measures have up to now been able to alter this. Stoppages of more or less similar proportions result where, instead of steel arches, roof bolts are in use. Stoppage times for continuous miners during the installation of roof bolts - as far as I am informed - display similar trends. A basic change i.e. a considerable improvement in the running time of the part face heading machines in German coal mining and of the continuous miners in American and Australian coal mining could be attained if the anchor bore-holes were drilled from horizontal positions so to speak "round the corner" and if the bolts were to be inserted in a similar way.

Rock bolts would allow this way of setting. The high pressure water jet technique could lead to the appropriate boring method (Figure 6). The inner of two intermeshed steel spirals encloses the hose; the outer responds to feeder and steering devices. The suggested system is formed by high pressure water nozzles on the free end of the spiral ant hose combination, a guiding system for the start of boring and for deflecting the spiral hose system as well as a rotating conduit mounted on a gun carriage and connected to a stationary pump by means of hose piping. This system has not as yet been tested but is merely intended within the framework of this lecture, as an indication of new conceptions for the optimal application of the high pressure water jet technique in mining.

In boring, (Figure 4) the specific energy input per cubic meter of cut mineral is not of primary importance as the aim is the production of bore-holes in a given direction and of a given dimension as quickly as possible; not the economic cutting of mineral or rock. As the specific energy input in conventional boring with a percussion drill into rock is already extremely high, a further increase with the change over to the high pressure water jet technique could, viewed from the relatively small amount of power installed, be quite easily tolerated if the advantage of "round the corner" boring could be attained.

With part face heading machines, particularly in their application to rock, the specific energy input per cubic meter of cut material is substantially higher than at coal extraction or with full face heading machines. The entry cut into the rock when applying the part face heading machine requires, according to the standard values in the literature, as much as two tenth powers or almost two tenth powers greater energy input than coal extraction and approximately a tenth power higher energy input than driving with full face heading machines. This means that the additional application of high pressure water jets should be considered as a support for the mechanical tools of part face heading machines. It would, however, have to be exhaustively researched as to whether the tool carrier or also the machines would, as a basis technology, have to be substantially altered or conceived afresh.

HYDRAULIC MINING STUDIES OF STORM KING MINES

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INTRODUCTION

Storm King Mines began with the concept that the resource potential of thick and steeply pitching seams of the Grand Hogback region of Colorado could be mined profitably. Two large tracts of private land (fee coal) were acquired; approximately 23.8km^2 near Glenwood Springs, and 16.2km^2 near Meeker (Figure 1).

Although both properties have substantial coal reserves, one has several distinct advantages in developing a competitive mining operation. Table I illustrates these advantages. Among the most important considerations from an economical point of view are proximity to railroad, highway, labor force, quality of coal, seam thickness, and source of water. As can be seen, Storm King Mines' property is adjacent to the main line of the D & RGW Railroad, I-70, and the Colorado River. The cities of New Castle, Glenwood Springs, Silt, Rifle, Carbondale, and Parachute are within short commuting distances and have available skilled labor.

In evaluating the quality of coal, Storm King Mines carried out two years of exploration drilling. This drilling has established in-place reserves in excess of 500 million tonnes in seams greater than 1 meter thick, having an energy content in excess of 2.84 x 10^7 J.kg.k. Table II shows the reserves to l000 meters, and proximate coal analysis. The concept of using hydraulic mining technology has been considered by SKM and, in discussions with others in the coal industry, the responses were usually that the technique lacked credibility in the western coal environment. To test the validity of hydraulic mining, Storm King engineered and operated a test of full-scale hydraulic mining equipment on the Glenwood property.

GEOLOGY

Prior to discussinq the hydraulic mining test, a short description of the geology is helpful. In cross section (Figure 2), there are three zones where coal occurs in the Mesa Verde formation. The most prominent seam(s) is the Wheeler in the Lower group. Toward the western end of Storm King Mines' holdings, the Wheeler is over 13 meters thick. Further east, it splits into two parallel 5 and 7 meter seams. During the orogeny that created the Grand Hogback, slippage along bedding plane occurred. This slippage along with minute extension of the jointing system is the basic fracture pattern hydraulic mining exploits. Therefore, the deformation that created the steeply pitching seams is also responsible for extending the fracture patterns that make the coal more susceptible to hydrofracture.

Although there are bedding plane faults, it is important to note that Storm King Mines' drilling and mapping programs have not delineated any cross faulting of significance.

HYDRAULIC MINING TESTS

Site selection was made based upon several criteria. These criteria were set to evaluate the hydraulic miining method under what appears to be oour mostmost adverse conditions. Table III shows the criteria used to select the test site.

Storm King conducted 29 separate tests in order to evaluate various pressures, flow rates, nozzle sizes, energy requirements, and rates of production. Table IV depicts the ranges in operating parameters utilized throughout these tests.

Figures 3-5 show the trend of our findings. The test data show a transition series between the two extremes of drilling and stabbing. The series is defined by three categories, none of which represent a single phase, but a varying combination of each phase. These categories are shown in Table V.

The trend as seen by Figure 3 of flow rate to volume demonstrates that higher volumes of water are a key to successful hydromining.

Pressures throughout the range tested (6.9 - 17.2 x 106 Pa) were adequate to break the coal. Increasing the pressure without increasing the flow rate did not increase the volume of coal removed in quantities large enough to justify the energy required.

Both the graphs of flow rate and pump horsepower to volume of coal removed display the desired patterns. These are actually the graphs isolated the categories previously discussed. What they indicate is a transition series from purely drilling to purely stabbing to a free face. None of the tests represent either extreme, but, as stated by the categorical analysis earlier, various combinations. The line in blue on each graph represents Category I wherein the principal action was drilling and requires the greatest amount of energy per unit (specific energy: Es - J/m^3).

Cateogry II, the red line, represents primarily pressurization of fractures. The production rate was increased and the E lowered. The yellow lines represent Category III test results wherein the principal action was stabbing and the results were predictably highervolumes of coal removed.

Graphing the average for each nozzle size in each Category shows growth curves one would expect. An increased flow removes larger quantities of coal as shown in Figure 3. What is most significant about these curves is that they are not linear. Figure 5 shows the linear relationship expected as compared with the actual production rates. In each case, the actual production rate <u>exceeds</u> the linear relationship indicating again that the flow rate is critical to production as it lowers the E and therefore increases the quantity of coal produced per dollar spent on operating costs. The increase also reflects the overlap of the three categories.

The evidence points to the fact that increasing the size of the nozzles can benefit the production rate. The limits to the size increase are both natural and economically induced. The eoncomic factors deal primarily with material handling including rate of production, size of plarlt, water storage capacity, etc. These are engineering criteria that are easily handled. The natural limits, which have to be acquired by experimentation, are sometimes less obvious. They include the thickness of the coal and the relationship it may have to creating free faces, the dip and the role gravity plays, the weight the coal is carrying from the hanging wall, joints, fracture patters, hardness, and stand-off distance. Some of these may reach their practical limit before they reach their ultimate design limit. This is true for stand-off distance. We tested and found that at 27 meters, the fracking ability of the 2.54cm nozzle was not materially impaired, the 0.95cm nozzle at 5 meters seemed beyond its ability to produce coal. However, 27 meters may be the design limit in hydraulic stopes before failure of the hanging wall occurs. In our testing, several of the natural limits to producing coal were not tested in the extremes. Instead, drawing upon others' experiences, we tested most natural limits and found them not to impede production in this coal within ,

RATES OF PRODUCTION

In determining the rate of production, several factors are to be considered. When full-scale operations commence, we will be operating in two parallel 6.6 meter seams. During the tests, the excavation was not large enough that the effect of overburden pressures assisted in breaking the coal. During the test we operated principally in the pressurized fracturing category driving development headings. In full-scale operations, development headings will be driven by roadheaders and hydraulic mining will be utilized solely in production stopes whose dimensions (13.lm x 6.6m x 27m) will allow for wall rock pressures, gravity and stabbing to open faces to keep production rates high.

Our tests under less than favorable hydraulic mining conditions returned a production rate as high as 7 tonnes per minute. Experience in other hydraulic mines where wall rock pressure and gravity played a role were as high as 18 tonnes per minute sustained for the operating hours of a regular shift.

In putting together our mine feasibility study to determine the discounted cash flow, we utilized a rate of less than 4.05 tonnes during 4.65 hours of operating time per shift. Equipment was sized to handle twice this rate of production to allow for surges. The return on investment far exceeded our expectations even with the conservative rate of production.

Hydraulic mining has several advantages in the area of safety and rates of production. Its greatest advantage is to produce from thick coal seams. In many respects, the thicker the seam, the more advantageous. Table VI cites these advantages.

EQUIPMENT

The nozzle design was the standard set by the US Bureau of Mines. A straight section of 3 times the diameter and a tapered section at 0.23 rad. to a 10cm main line with straightening vanes. The entire line was rigid schedule 80 steel pipe with flexibility provided by swivel joints. Power came from two oil field hydrofracturing trucks, each having twin 532kw diesel engines driving their pumps. A blender assured proper feed rates and pressures to the high pressure pumps. Monitoring equipment with strip charts kept constant records of pressures, flow rates, and total volumes.

In order to provide movement to the monitor, it was mounted on the boom of a small roadheader. The cab on the miner was enclosed with plate steel and a small window was provided. Two-way radio communications were maintained to the pump operators and backup emergency signals devised.

SAFETY

All operations were carried out in compliance with MSHA standards. It is noteworthy, that hydraulic mining has a superior safety record. Table VII enumerates several of the areas where safer conditions exist.

CAPITAL

The capital requirements for hydraulic mining are about one-half that for a conventional mine of similar size. One million tonnes per year appears to be the optimum design for a single hydraulic panel. Other areas where operating costs and capital costs are reduced are summarized in Table VIII.

SUMMARY

In the hope of being repetitive, we have prepared a movie which summarized the development of the Glenwood Springs Coal Property. The effect of the hydraulic monitor on a section of outcrop opened for this purpose will be visually demonstrated.

TABLE 1. Competitive Mining Advantages

Dip	0.98 rad.
Thickness	1-14 m
Number of Seams	9

Distance To:

Water - Canyon Creek-Vulcan Ditch, a tributary of the Colorado River on theproperty.

Power - Public Service Company of Colorado transmission lines within 3 kilometers.

Rail -Main lines of the Denver and Rio Grande Western Railroad within 3 kilometers.

Highway - I-70 New Castle Exchange within 3 kilometers.

Labor -

Town	Kilometers
New Castle	5
Glenwood Springs	15
Silt	20
Rifle	30
Carbondale	35
Parachute	58

TABLE II. Quality and Reserves

Seams	Thicknesses	Reserves to 900 m
9	1.14m	380,000,000 tonnes

Recoverable Reserves	225,000,000 tonnes
Reserves in Thick Seams	160,000,000 tonnes

Coal Quality:

BTU	(2.838×10^7 J/kg.k)
Ash	6.5%
Moisture	6.2%
Volatile Matter	38.1%
ixed Carbon	49.2%
Sulphur	0.5%

TABLE III. Site Selection Criteria.

Hardgrove Grindability Index

Protodyakonov Test

Dip

Thickness

Wall Rock

Accessibility

TABLE IV. Operating parameters

	English	Metric
Nozzle Size	3/8-1 3/8 inches	.95-3.5 cm
Pressures	1000-2500 psi	6.9-17.2 x 10^6 Pa
Flow Rates	150-1850 gpm	9.5-116.7 L/s
Horsepower	600-3000 hp	450-2250 kw

TABLE V. Categories of Jet Mining

		Drilling	Slabbing
I	Penetration Drilling	90%	10%
II	Pressurized Fracking	60%	40%
III	Breaking to a Free Face	20%	80%

A category was assigned to each test.

TABLE VI. Advantages of thick seams.

1. Less dilution from wall rock
2. More coal produced by stabbing
3 Fewer equipment moves.
4. More developed coal for equivalent development
5. Presence of more zones of weakness
6. Full seam mining with higher coal recovery

TABLE VII. Safety

1. Less moving machinery
2. Less dust in the air
3. No electrical equipment in production stopes
4. Less manpower per ton of coal
5. No workers in the production stopes under unsupported roofs
6. No opportunity for dust or gas accumulation during transport

TABLE VIII. Operating and Capital costs.

1. Small capital requirement for face equipment
2. Less manpower - expected +23 tonnes per manshift
3. Less moving machinery, electrical motors, etc. and therefore, fewer maintenance costs
4. No rock dusting or supports in working stopes
5. More economic transport of coal

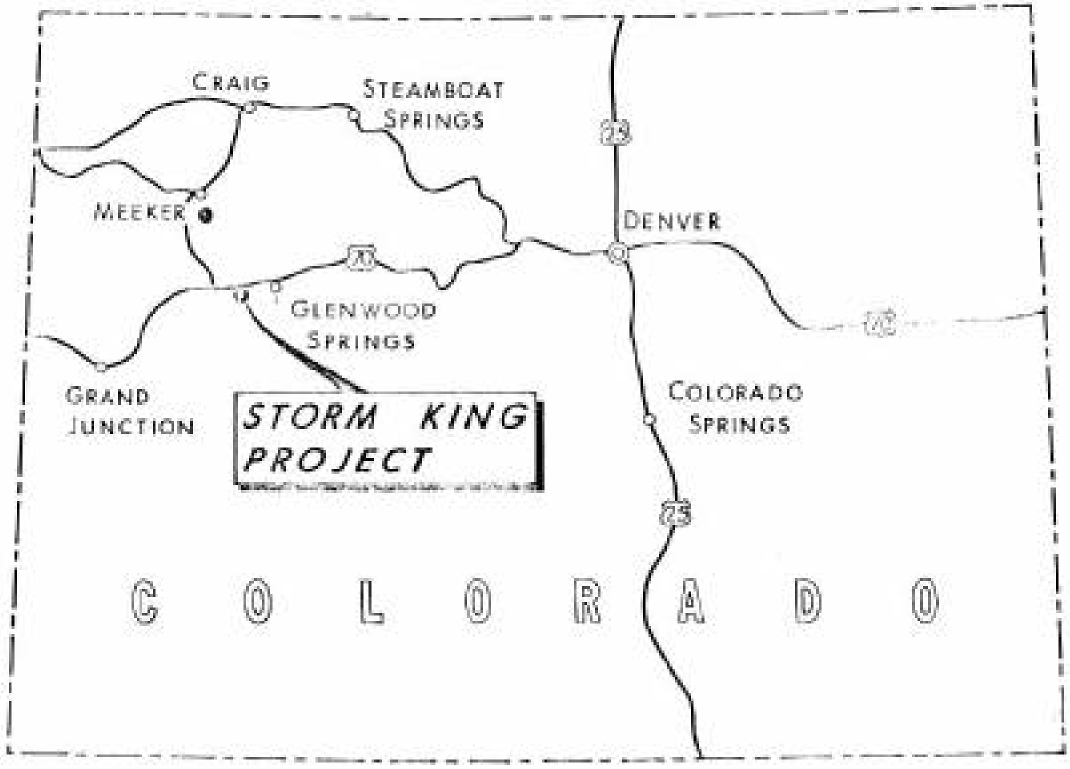

Figure 1. Location

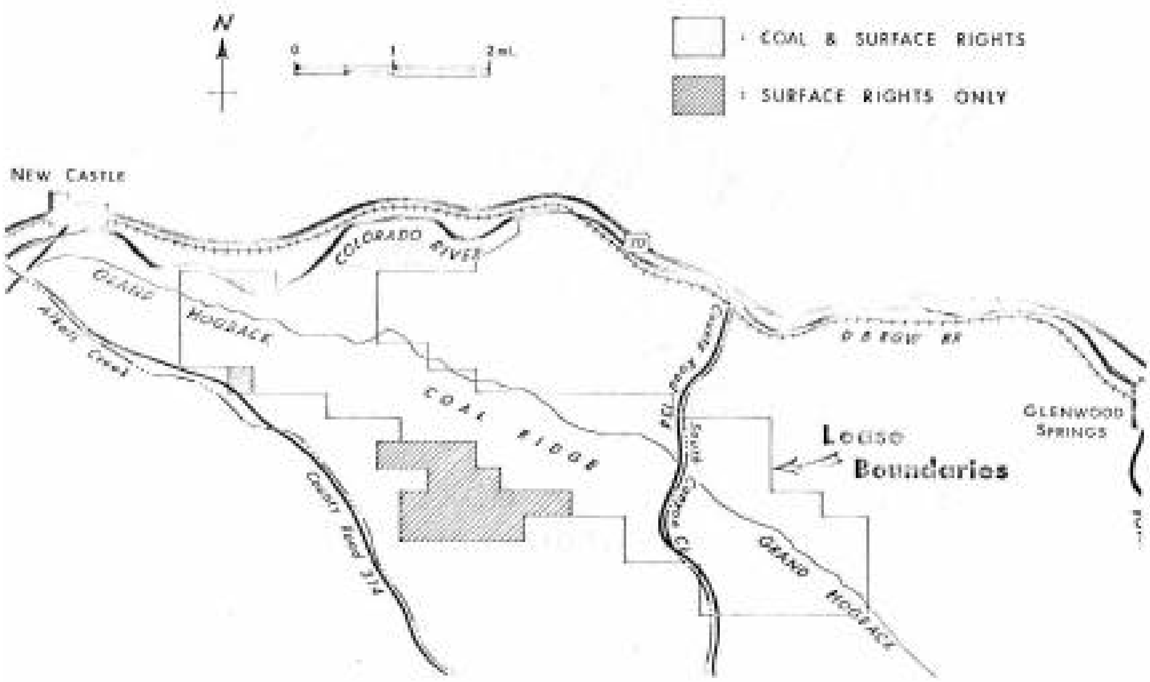

Figure 1A. Storm King Project Lease

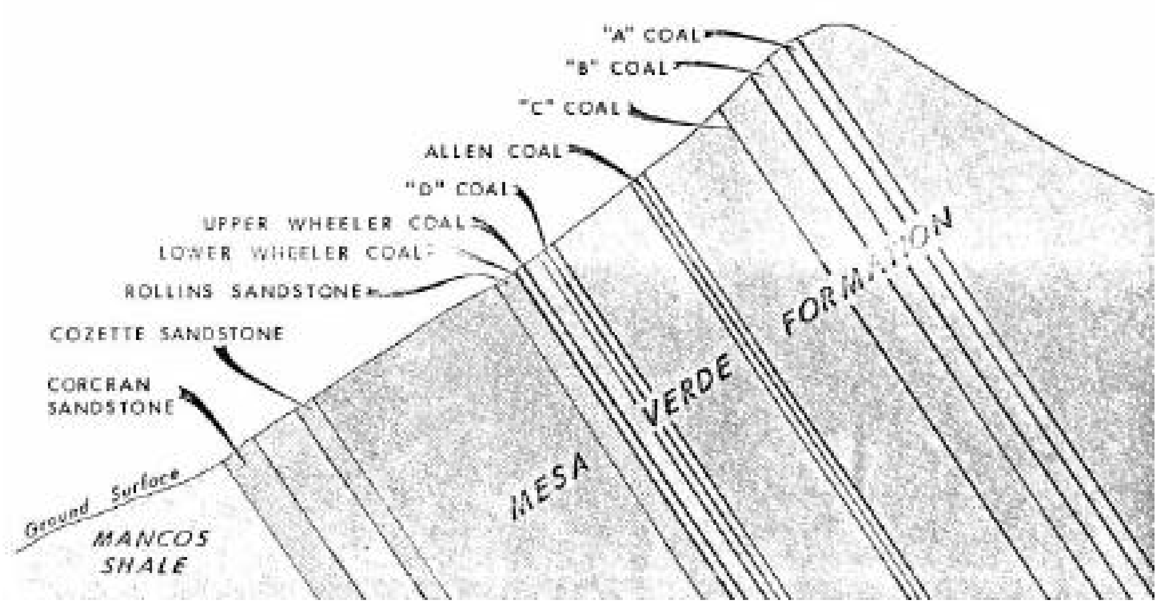

Figure 2. Generalized geologic profile.

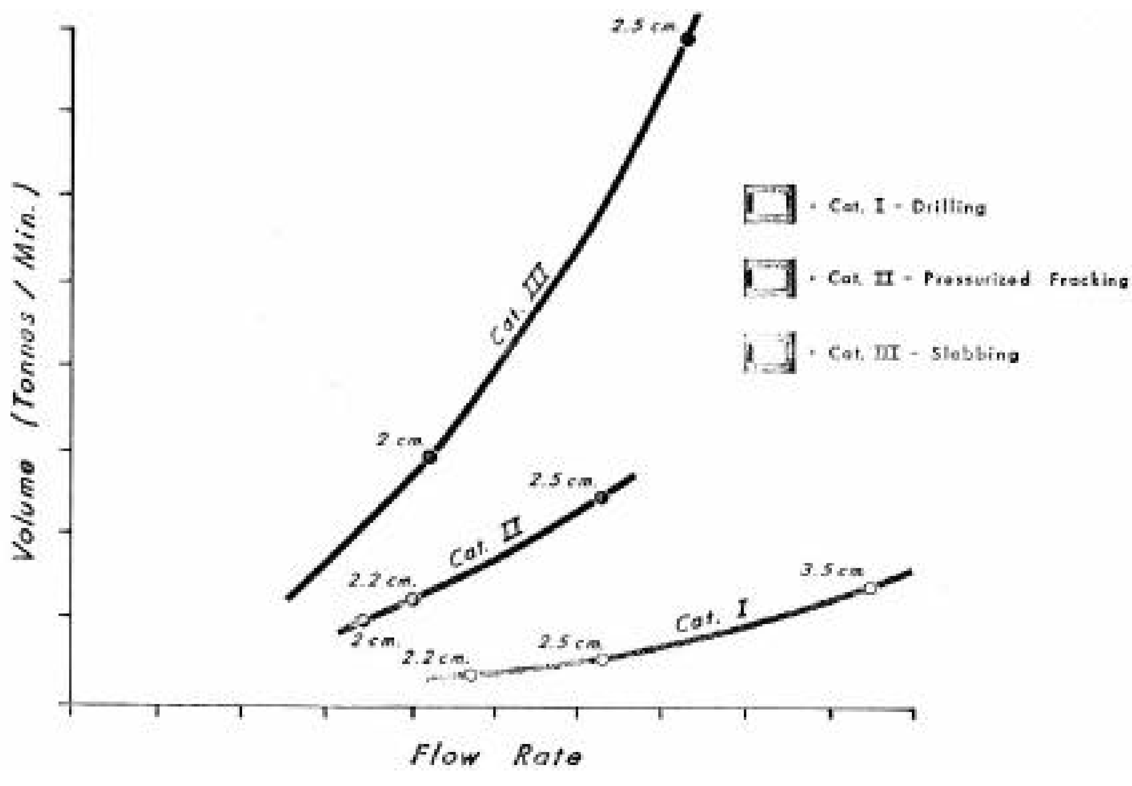

Figure 3. Actual volume of coal removed.

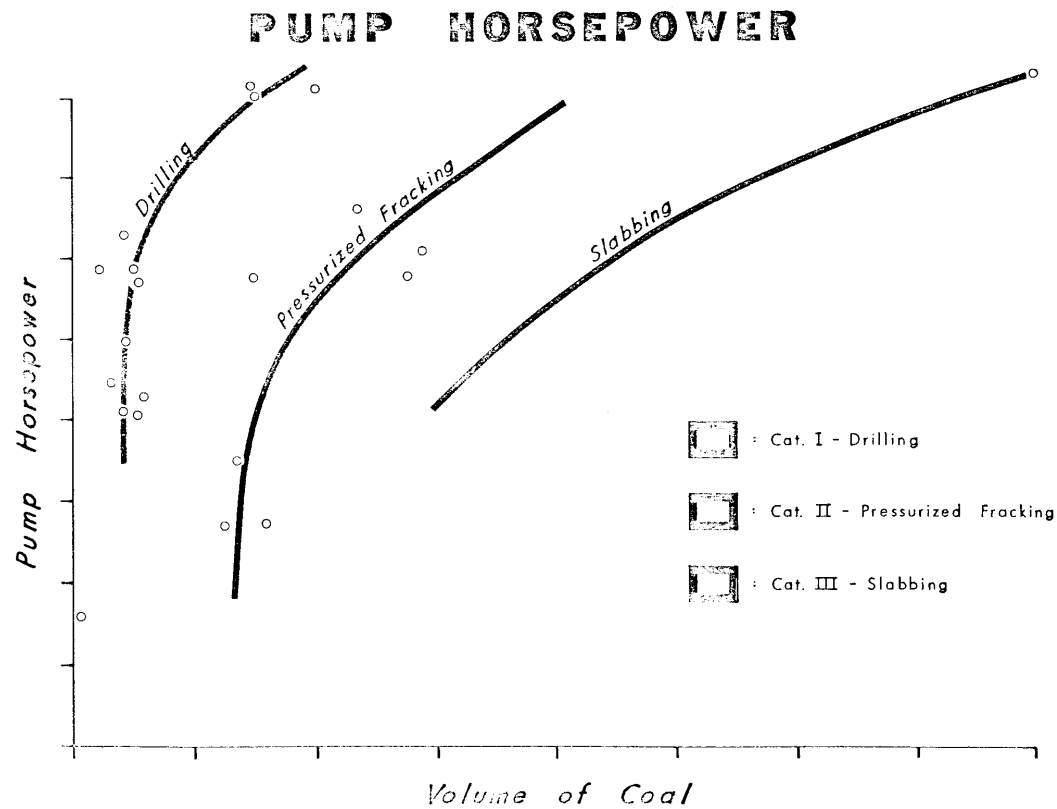

Figure 4. Pump Horsepower

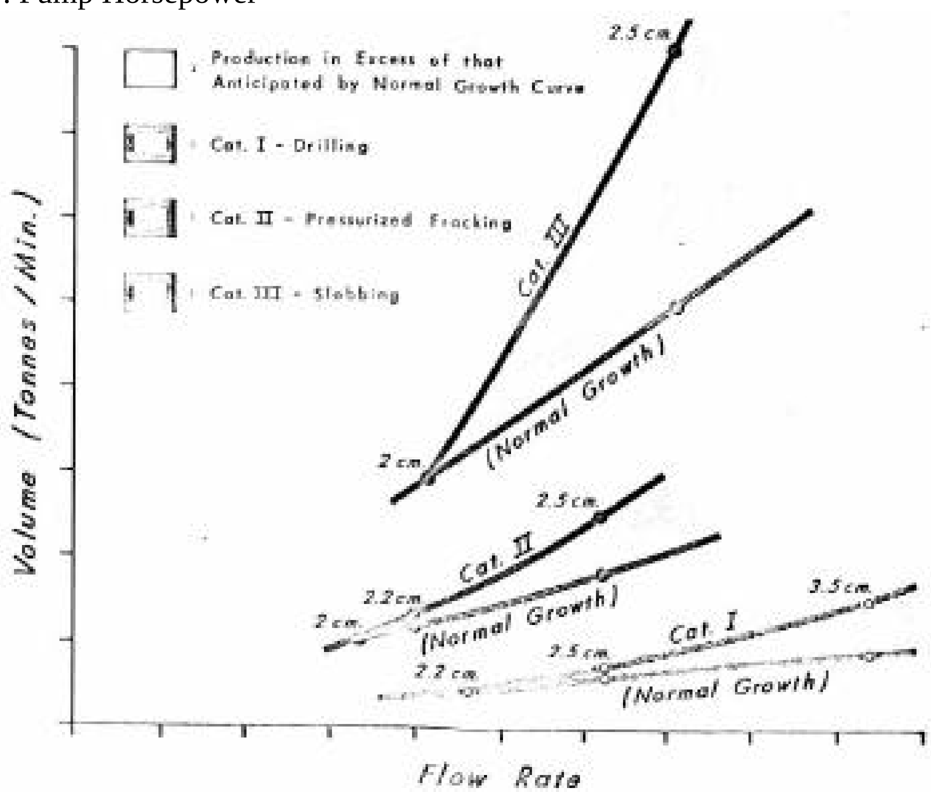

Figure 5. Actual versus anticipated volume of coal removed.

SECONDARY FRAGMENTATION WITH WATER JETS

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INTRODUCTION

In contrast to the extensive research employing high speed water jets to extract ores and cut and clean a variety of materials, the use of water jets to perform secondary fragmentation of rock-like material has not received much attention.

Besides applicability to conventional grinding processes, secondary fragmentation with water jets possesses a potential for use in the "borehole" mining process. In the borehole mining process, water jets at the end of a rotating mining "tool" extract the coal which is then pumped up the pipe holding the mining tool. In order to be pumped to the surface, the coal must first be reduced in size in order to pass through the inlet of the pipe.

In order to evaluate the use of water jet secondary fragmentation, a research program was designed and undertaken.

TEST PROCEDURE AND EQUIPMENT DESIGN

Evaluation procedures for conventional crushing and grinding devices are well established. However, these procedures cannot be used for water jet grinding. Consequently, a considerable amount of time and effort was devoted to devising a method to test and evaluate the use of water jet secondary fragmentation.

The original idea was to parallel the test procedure for batch grinding so that perhaps some comparisons could be made and some data determined for grinding kinetics. The original apparatus shown schematically in Figure 1 consists mainly of two concentric cylinders. Two water jet nozzles were mounted in the outer cylinder. The water jets passed through the inner cylinder by vertical slots cut in the wall. The inner cylinder was oscillated in an up and down fashion to expose the entire coal sample to the action of the jets. The top and bottom of the inner cylinders had small diameter holes drilled in order to allow the water to escape. Also the entire apparatus was submerged in a water tank so that the coal samples would be completely submerged in water. After several months of modification and experimentation, the approach was abandoned. The losses of coal from the inner cylinder could not be kept to an acceptable amount. In some runs the amount of the sample left in the container was less than 50 percent of the weight of the original coal sample. This could not be considered a "batch" grinding procedure.

A continuous process procedure was devised by removing the inner cylinder of the original apparatus, placing the jets closer to the lower end of the outer cylinder and angled slightly to create a swirling flow and placing a 2 inch opening grate over the lower end of the cylinder (Fig. 2). The cylinder was filled with the desired coal particle size and placed in a water tank so that the coal sample was always submerged.

The average exit velocity from the nozzle was determined by catching the flow from the nozzle during a period of time and utilizing an accurate measurement of the nozzle density. The average velocity was computed for the continuity equations.

RESULTS

If the production rate of ground product, in (lb/sec) is assumed to be a function of the density and strength of the coal, ρ_c and σ_c and of the initial coal size $d_{c\,i}$ and product size $d_{c\,f}$ as well as the fluid and nozzle variables, d_j, p_j, V_j, analysis yields the following non-dimensional ratios.

$$\pi_1 = \frac{\dot{m}\, d_{c,f}}{d_j^{\,2}\ \rho_j V_j^{\,3}} \qquad \pi_4 = \frac{d_j}{d_{c,f}}$$

$$\pi_2 = \frac{\sigma_c}{\rho_c\ g\ d_{c,f}} \qquad \pi_5 = \frac{\rho_c}{\rho_j}$$

$$\pi_3 = \frac{d_{c,i}}{d_{c,f}} \qquad \pi_6 = \frac{V_j}{g\ d_j}$$

In this study, the type of coal the final desired size and the jet fluid will not be varied so π, π_2 and π_5 can be dropped. π_1 contains the terms m/ρjVj^3 which is the amount of product produced per unit time per unit of fluid kinetic energy per unit time per unit nozzle area.

A logical way to present the data is to plot π_1 vs. π_3 holding the ratios π_4 and π_6 constant. For a different value of π_4 this plotting process can be repeated. If a curve exhibits a maximum, this is the optimum; that is, the d_j and V_j that produce the most product for the least energy input.

The ground product size distribution was determined by sieve analysis and was plotted on Rosin-Rammler plots. Representative plots are shown in Figures 3-5. Note the biomodal distribution shown in Figures 4 and 5. Our first thoughts are that this is due to the action of low pressure jets and high pressure jets to produce mainly fines, given two distributions in the product.

Also, another observed phenomena was that for a given nozzle and critical coal particle size the average size of the product decreased. Figures 6-8 are plots of π_1 vs. π_3. As mentioned earlier, what one hopes to find in these plots is a maximum point in the curve indicating the most product for the least energy for a given initial coal size and a given jet nozzle size (π_4 and π_6 held constant). This data is preliminary and more data points are needed for some of the plots. However, the data does indicate that the curves do possess a maximum point (Fig. 8). This is encouraging and work is proceeding to obtain additional data to find the maximum point for each of the curves.

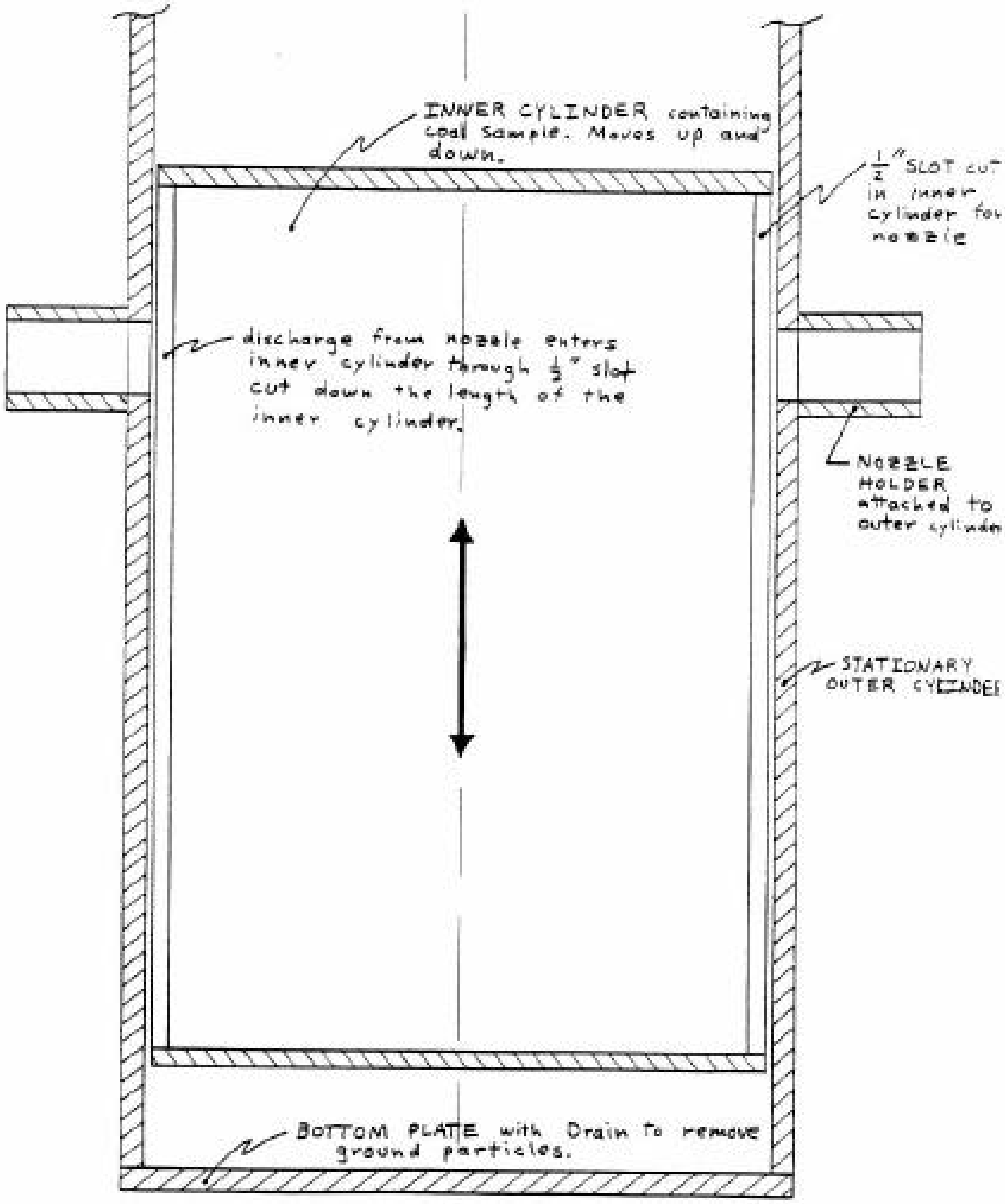

Figure 1. Initial Test Chamber design

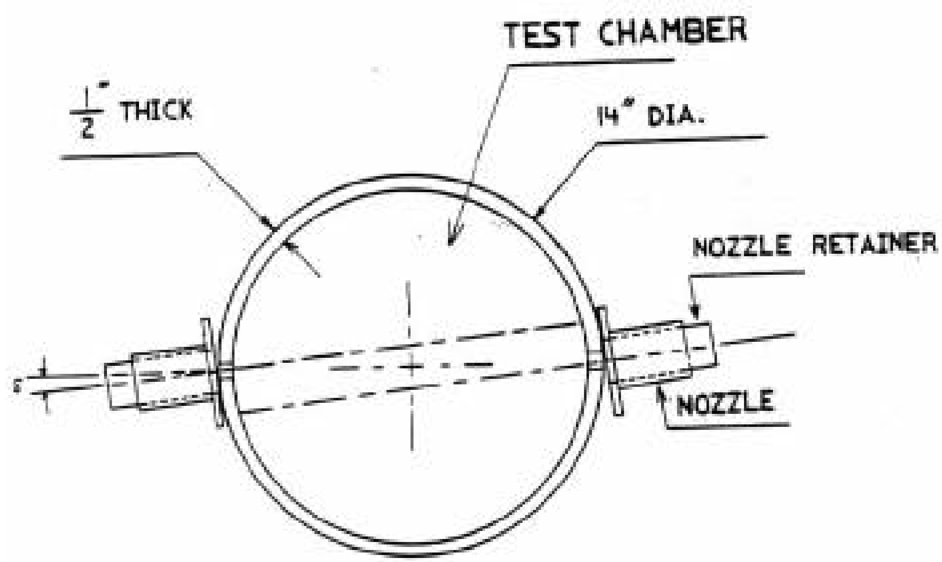

Figure 2. Final test chamber design.

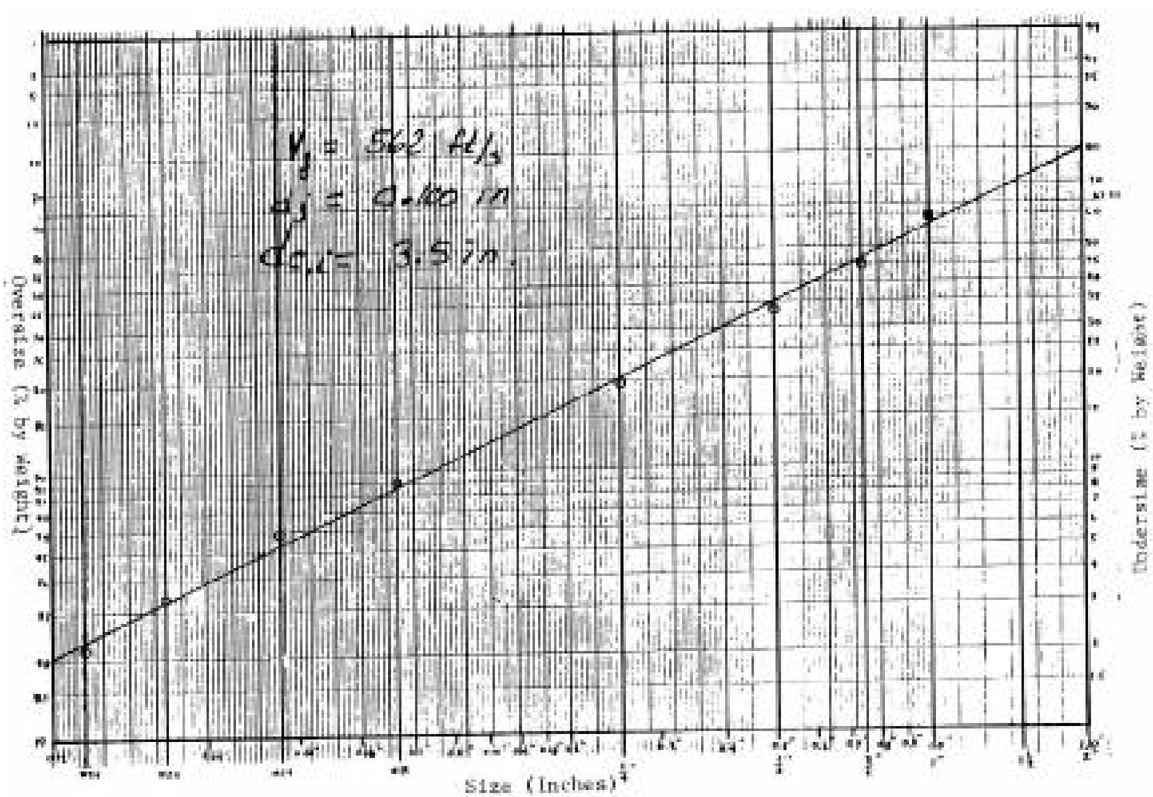

Figure 3. Rosin-Rammler plot.

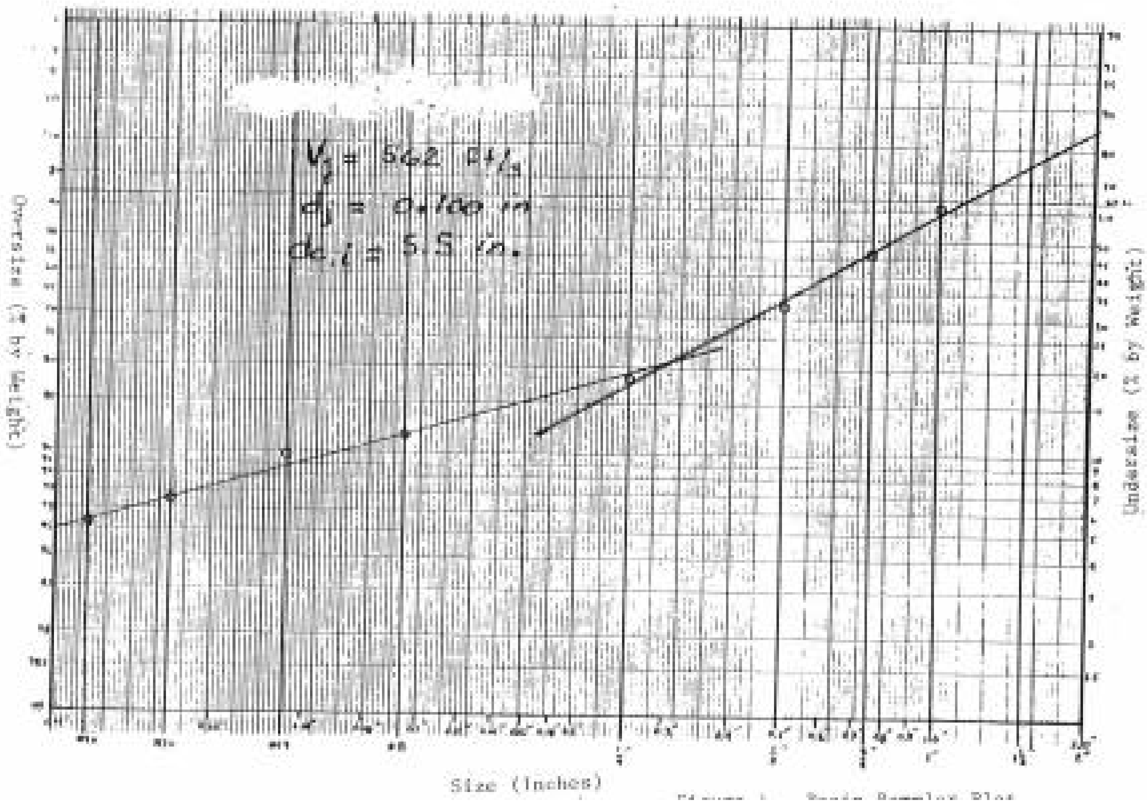

Figure 4. Rossin-Rammler plot.

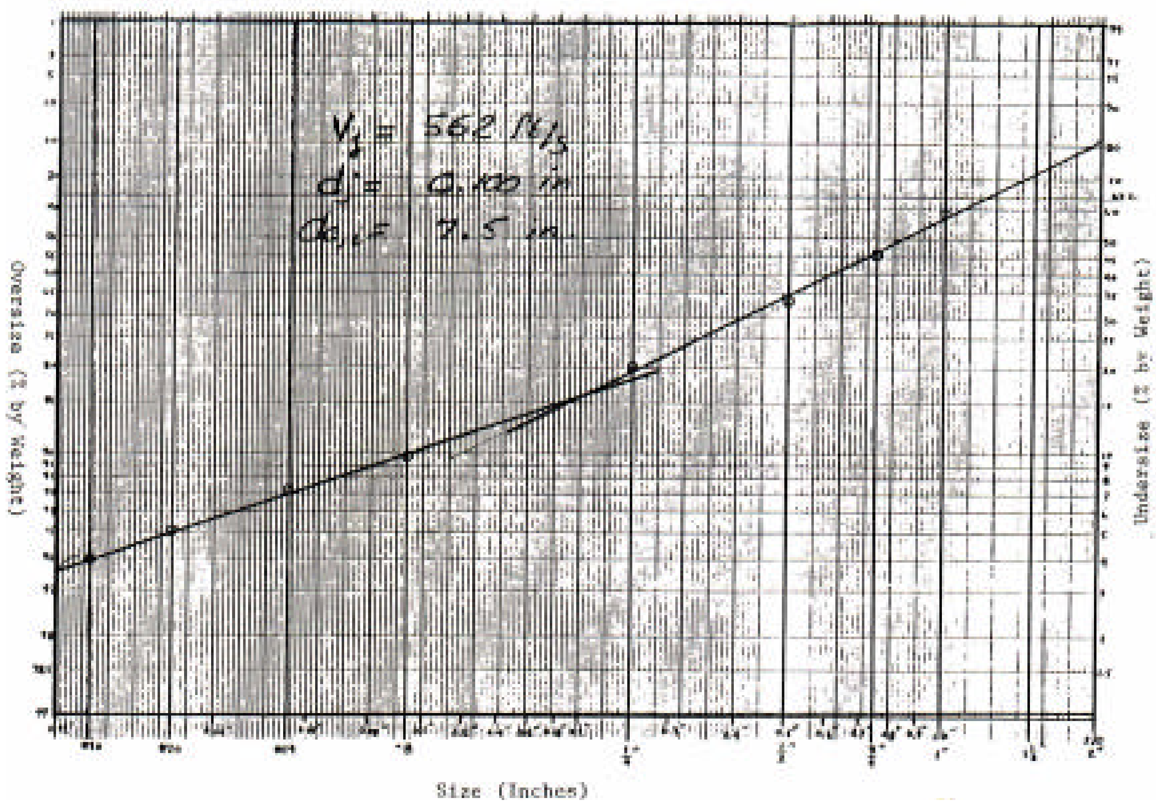

Figure 5. Rossin-Rammler plot.

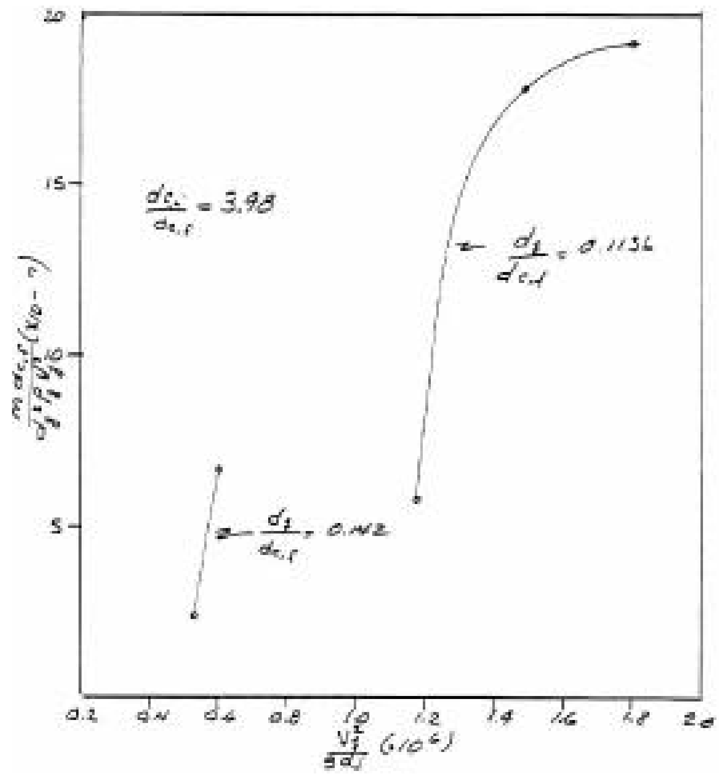

Figure 6. Non-dimensional plot for grinding study.

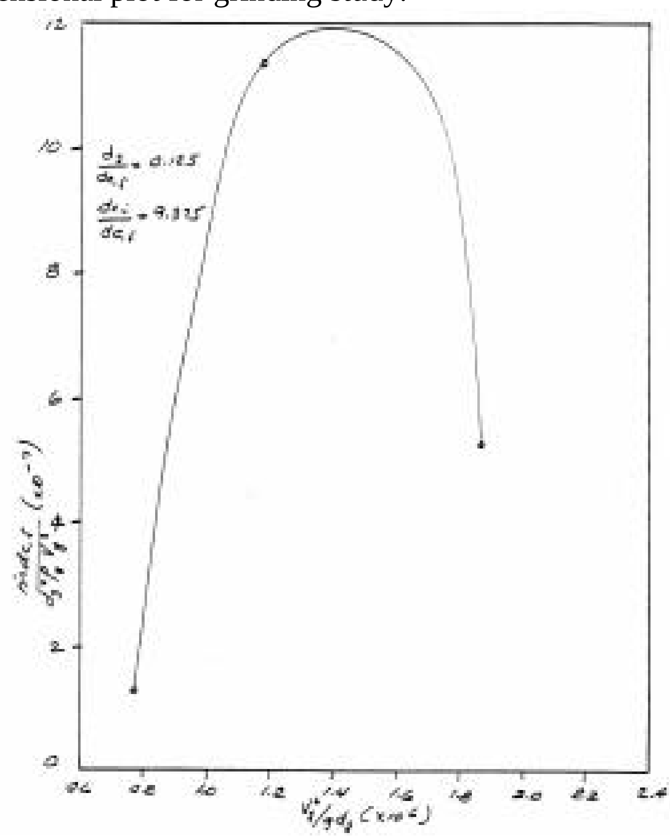

Figure 7. Non-dimensional plot for grinding study.

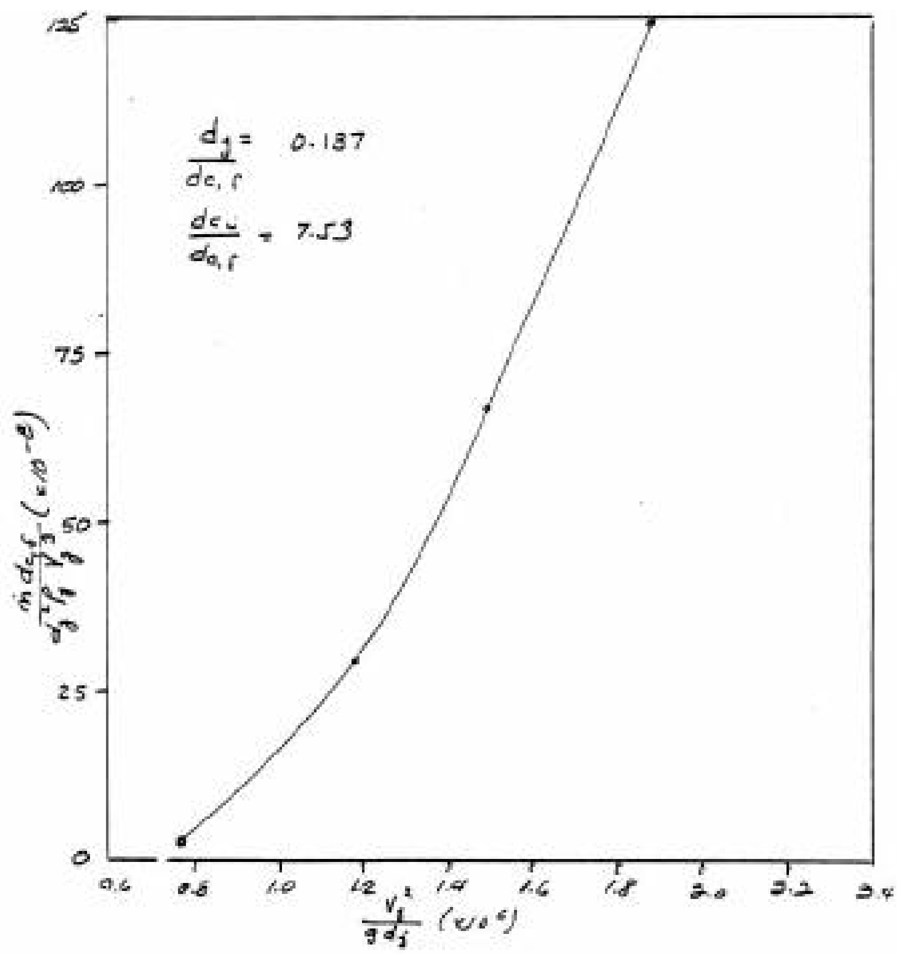

Figure 8. Non-dimensional plot for grinding study.

STATUS OF HYDRAULIC COAL MINING IN PEOPLE'S REPUBLIC OF CHINA

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ABSTRACT

A brief summary of hydraulic coal mining in People's Republic of China is presented in this report. The current production, seam characteristics, mine layout, sequence of extraction, coal transport and equipment used are discussed.

INTRODUCTION

People's Republic of China initiated its first hydraulic coal mining experiments in 1956. Following the successful demonstration, mine production began in 1958 and a peak production of seventeen million tons per year was reached prior to the 1977 major earthquake. The current production is about eight to ten million tons per year from over ten mines. There are three types of hydraulic coal mining used in China: (1) Complete hydraulic mining, (2) Hydraulic mining in combination with conventional mining, and (3) Conventional mining assited by hydro-transport.

SEAM CHARACTERISTICS

Hydraulic mining has been applied in single as well as multiple coal seams with 7^0 to 75^o dip and 0.7 to 14 meter thickness. Some of the seams are irregular in thickness caused by igneous intrusions, extremely gassy, soft and dusty and tend to have spontaneous combustion.

Five types of seams have been mined hydraulically. They are:

1. 4 to 8 m, gently pitching (10^0-30^0)
2. 1.0 to 2 m, gently pitching (10^0-30^0)
3. 10 to 14 m, moderately pitching (30^0)
4. 2 to 4 m, steeply pitching (30^0-75^0)
5. 0.6 to 1.3 m, steeply pitching (30^0-75^0)

Mine roof generally is stable shale, sandstone, or a mix of sandstone and shale. Floor is generally shale.

PRODUCTION

Field experiences (Ref. 1 & 2) indicate that under favorable geological conditions and a well equipped mine with competent management a complete set of hydraulic section may attain an annual production rate of:

600,000 tons/year in gently pitching, thick seam
450,000 tons/year in gently pitching, moderately thick seam
250,000 tons/year in steeply pitching, thin seam

The average productivity per manshift is 10 to 14 tons for an efficient mine and the overall average is 8 tons for hydraulic mining3 as compared to less than 2 tons for the conventional mining.

HYDRAULIC COAL MINING SYSTEM

Typical mine layouts for a multiple and a single seam are shown in Figures 1 and 2. The dimensions of the panel are: 500 to 600 m along the dip and 1,000 m along the strike. The schematic of a typical coal mining system is shown in Figure 3, and a hydraulic hoisting system with bin feeder in Figure 4.

EQUIPMENT

Hydromonitors used are manufactured in China. An operating pressure of 200 kg/cm and a flow rate of 300 to 750 m /hr. with a maximum distance of 600 meters are commonly used. The coal slurry pumps used are single, two and multiple stage pumps. The two stage pump is the most reliable and commonly used. It has a maximum of 100 m water head at a discharge rate of 450 m /hr. The life of the pump is about 1,000 hours. In addition to the application of hydromonitor for coal mining, China is actively experimenting with pulse, rotational and oscillating jets in the field for mining flat seams and for entry development in shale and sandstone.

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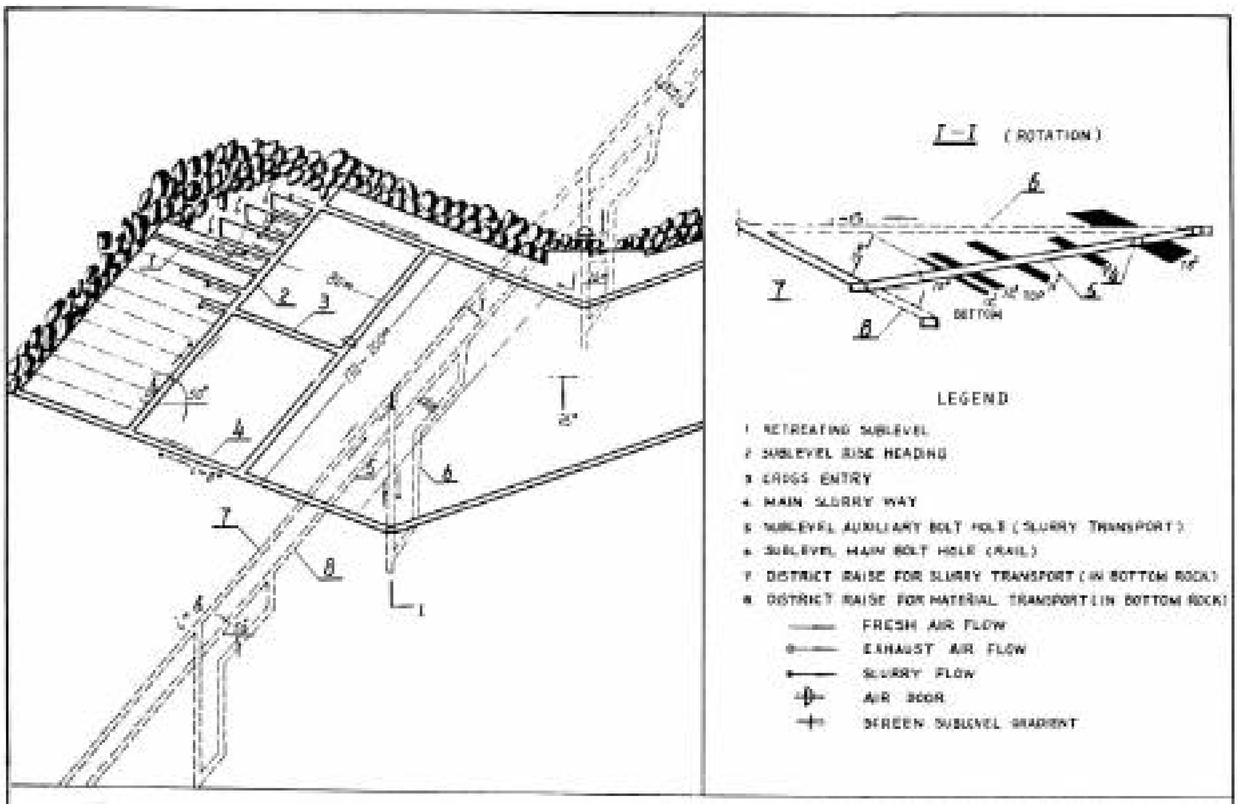

Figure 1. Layout of working area in multiple seams (seam 7 merges into seam 8) at Luchiato colliery.

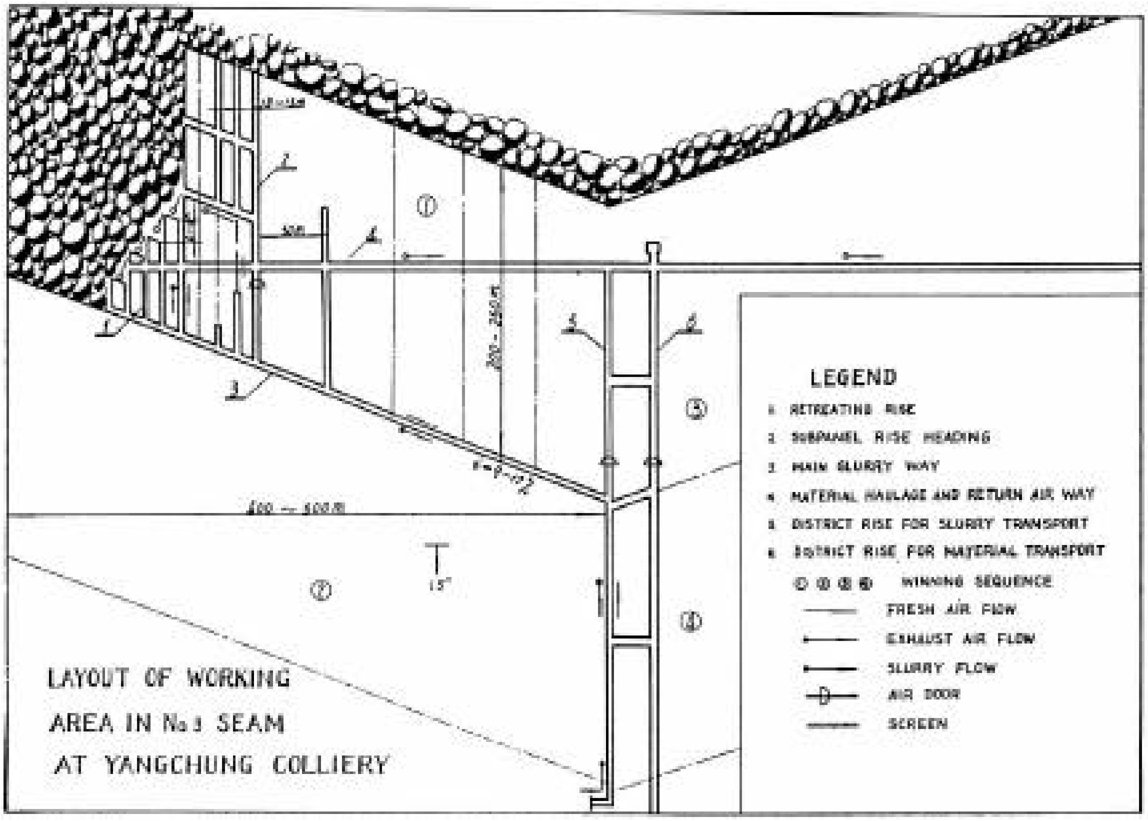

Figure 2. (from Ref. 1)

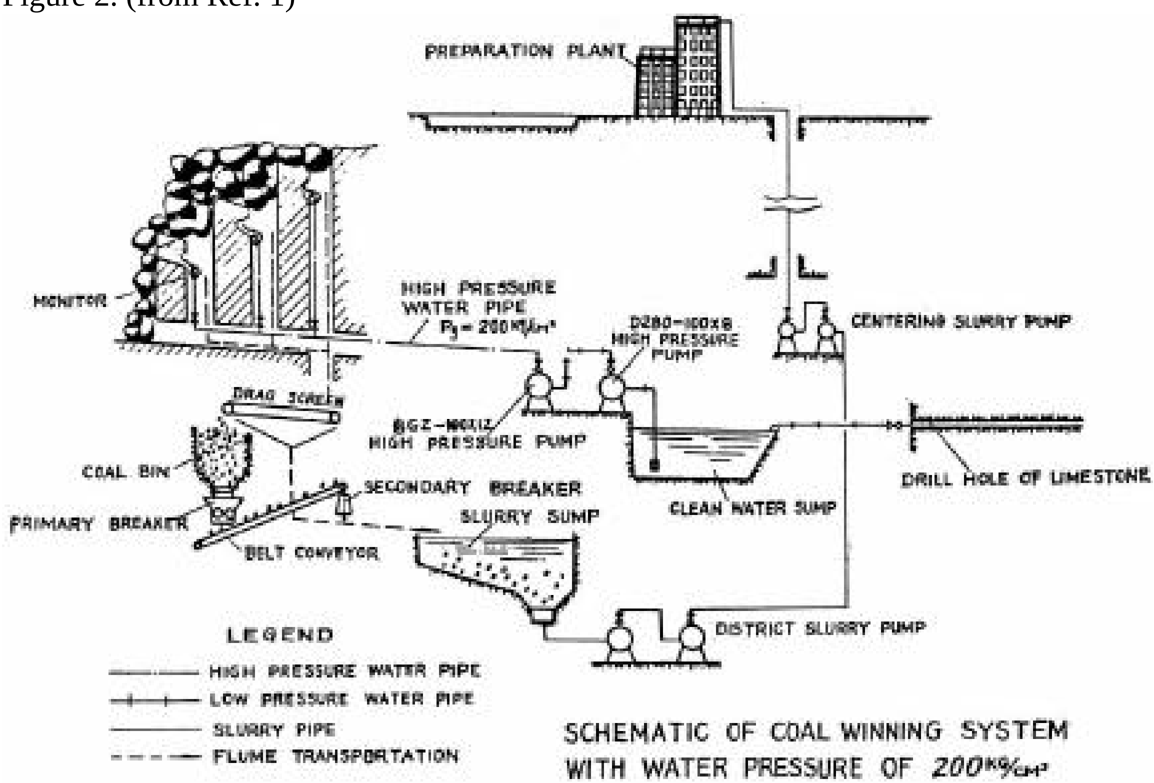

Figure 3. (From ref. 1)

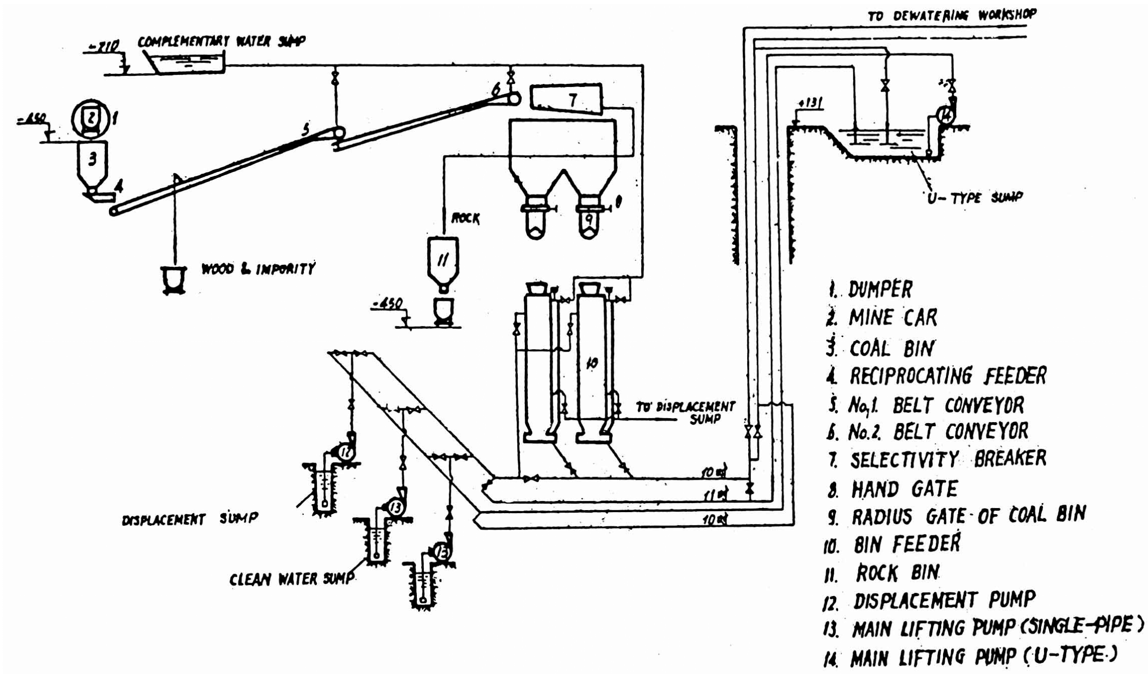

Figure 4. (From ref. 1)

DISCUSSION

NAME: James L. Evers
COMPANY: Southern Illinois University

QUESTION: "You said that pulsed jets were used to drive a 20 km tunnel. Do you have other information on the pulsed jets such as stand-off distance, pulsing technique, material removal rate, and material being cut?"

ANSWER: Yes. I have some information. The stand-off distance is about one meter, the pulsing technique is using nitrogen, driven piston and the material being cut is coal and coal measure rocks. There is no information on material removal rate.

PRELIMINARY PRACTICE IN THE USE OF A SWING-OSCILLATION WATER JET IN COAL MINE DRIVAGE

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ABSTRACT

Research and field trials into the development of coal mine drivage, by means of high pressure water jets are currently being carried out in the People's Republic of China, based on the successful practice of conventional hydraulic coal mining. A process of Swing-Oscillation Water Jet Cutting was selected in order to reduce the energy consumption during coal cutting and to enhance the coal heading advance rate.

As a result of research on the characteristics and principle patterns of the swing-oscillation water jet, optimum values and correlations were studied with the aid of a least-squares technique. On the above basis, pilot coal cutters, using a swing-oscillation jet, were built and employed in trials of underground drivages. A flow straightener device (i.e., a collimator) with a star-like section and a single fish shape in length was attached within the nozzle entrance pipe, to improve the structure of the swing-oscillation jet, and to extend the effective range of standoff distance.

In the process of coal drivage and coal cutting, the major means of enhancing the ability of coal cutting with a swing-oscillation jet were investigated during underground trials.

Preliminary practice of coal drivage shows that a higher rate of cutting, markedly less dust in the air of coal faces, and lower water content in raw coal could be obtained by means of a swing-oscillation jet. Advantages such as the improved quality of the roadways and the absolute elimination of explosions caused by sparks, could be achieved. Accordingly, the requirements for an increase in productive capacity, and safety, more satisfactory working conditions at coal faces, an extensive range of applications and a good quality of cut have basically been fulfilled. Therefore coal cutting with the swing-oscillation jet will become an appropriate technology in China.

By the end of 1982, a total coal drivage of more that 500 meters had been completed. At present, further trials and use of the equipment in continuous drivages, are being carried out more extensively.

INTRODUCTION

Developing an unique technology for coal cutting and rock fracturing to suppress dust; eliminate sparks; reduce the load acting on support systems, and to protect surrounding work from damage has been a most important problem for mining, all over the world.

For the last two or three decades, a wide variety of experimental studies into fracturing and cutting rock have been carried out in many countries. Of these, the cutting and fracturing of coal and rocks using high pressure water jets has been found to be one of the more effective and promising of the novel coal mining technologies examined to date. However, little is known of the best ways to raise the efficiency of coal cutting and rock fracturing, or to lower the energy consumption of coal cutting, in order to make a practical means available for productive application.

In 1970, an experimental study of high pressure water jet cutting technology was begun, based on the practice of hydraulic coal mining. In order to raise the technical level in coal mines more rapidly and to improve working conditions for miners, great attention has been paid to the new technology of coal cutting with high pressure water jets in China (3).

In an experimental study, aimed at the rapid application of high pressure water jets into production, three principles were followed:

1. The high pressure water system used, should be very reliable in underground operation to guarantee a long-term and continuous operation.
2. In order to make full use of the energy, an appropriate process of coal cutting with a water jet must be carefully designed.
3. Because drivage mechanization lags behind that of coal mining in general, at the present time, the technology of entry drivage using water jets

Starting in early 1979, the technology has been developed through theoretical design, evaluation of reliability, experimental studies and underground trials, based on the above ideas, and has been found to be basically effective and popular with mines.

BASIC THEORY OF COAL CUTTING WITH A SWING OSCILLATION WATER JET

To meet the requirements of coal drivage, it is necessary that any high pressure water jet should be capable of cutting and deep kerfing. In general, a jet should have three functions, which are as follows:

1. Maximizing the area cut per unit time.
2. Attaining the maximum depth of kerf.
3. Minimizing the energy consumption during coal cutting.

Through analysis and comparative studies of the energy consumption by various types of jets under different working conditions, the swing-oscillation jet was selected as the most favorable technology for use in coal drivage (1), (2).

From results obtained during the experimental study and field trials, the principal conclusions derived for the swing-oscillation jet use in cutting coal are that:

1. Combining a high frequency oscillating motion perpendicular to the traverse direction of the nozzle, with a high normal speed will produce a swing oscillation jet moving at a high relative velocity (i. e., traverse speed (Figure 1). Figure 1 shows the curves correlating resultant moving speed of nozzle exit (V), amplitude (a), oscillation frequency (F) and traverse speed (V). A review of Figure 1 reveals that by applying a swing oscillation motion, the traverse speed of the jet will be increased to a level it would be difficult to realize with conventional jets.
2. The area covered by the jet in unit time, using a swing-oscillation jet at high traverse speed is remarkably greater than that of a conventional jet (3,4)
3. By adjusting the amplitude of motion of the swing-oscillation jet it is possible to spall out the ridge left between slots when cutting. The water will also crack a zone around the cut, which will require a lower energy input to be removed later.
4. A wider broken zone will substantially improve the operational conditions for jet cutting. The depth of slot cut is linearly related to the number of traversing passes, over the range that the jet is efficiently cutting. Thus wider and deep kerfs can be created, i. e., a slot can be created of a certain width which can be cut to great depth.
5. In addition, any coal left between adjacent cutting paths can be broken down by vertical impact. Thus a much lower energy consumption would be necessary.
6. The excavation pattern for each site should be determined based on local geological conditions, and the mechanical and physical properties of the coal. Generally, deeply pre-kerfing the bottom of the coal face if done first will create favorable conditions for a large volume big amount of solid coal to fall out; as a result, the lowest energy consumption will be achieved and the maximum potential of the water jets for coal cutting would be achievable.

Figure 3 is a schematic diagram of a typical application, where the area surrounded by dotted lines indicates the amount of fallen coal which resulted from the extensive collapse of solid coal (4,5).

THE CUTTING ABILITY OF A SWING-OSCILLATION WATER JET AND THE TECHNICAL MEANS USED TO ENHANCE ITS ABILITY FOR COAL CUTTING

The schematic diagram given in Figure 3 shows that effective coal drivage with a swing-oscillation jet, requires that deep kerfs and surrounding fractured zones be created at high speed. As stated previously, it is necessary to study the jet coal cutting ability before studying the coal collapsing ability. With the aid of a Least Squares analysis of the experimental data, an equation predicting the ability of a swing-oscillation jet to cut material was derived and simplified. An appropriate equation is given as follows:

(EQUATION IS MISSING FROM THE MS)

The coefficients "b " in Eq. (1) can be obtained by experiment. In order to achieve a high rate of coal cutting, the optimum values of parameters, such as frequency of oscillation (F), amplitude (A), water pressure (P), traverse speed (V) and the relationship between them, could be approximately given by Eq. (1). On the basis of the above results, pilot swing-oscillation jet cutters were designed and put into operation in tests of underground coal drivage in 1981.

The curves given in Figure 4 show the correlation between the cutting ability and the effective range of jet cutting. As shown in Figure 4, the cutting ability where a swing-oscillation jet was used was much greater than that of a swing only jet for a given set of conditions, however, beyond a fixed standoff distance, the cutting ability of the jets would dramatically deteriorate. Oscillation of the nozzle assembly at a high frequency was clearly proven to aggravate the initial disturbance on the jet surface and worsen the jet structure.

In the research, a flow straightener with *like section and fish-shaped in length was used to improve the structure and behavior of the jet. Both experimental and field tests indicated that a correctly designed flow straightener correctly located relative to the nozzle, and with a flow channel designed according to the theory of hydrodynamics could enhance the jet quality, and considerably extend the effective range of stand-off distance. Typical results are seen as Figures 4, 5 and 6.

Results of the theoretical analysis confirmed that the influence of the flow straightener upon the structure could be estimated from the empirical constant related to the initial disturbance, with Equation 2 describing the distribution of axial dynamic pressure (6). This work was done by Miss Dang Ling under the direction of the author, Dr. Cheng Dazhong.

(EQUATION IS MISSING FROM THE MS)

Where:

C is an empirical constant describing the initial disturbance;
C = 0.64 - 0.69, without a flow straightener
C = 0.55 - 0.62, with a flow straightener

As a result of the above research, flow straighteners are being extensively employed. Operational performance has shown them to be effective.

The Ability of A Swing-Oscillation Jet Cutter to cut coal in a Coal Drivage Entry:

As shown in Figure 3, a typical procedure for coal cutting with a swing oscillation jet will generate a total volume of coal (W) for one of four ways:

$$W = W1 + W2 + W3 + W4$$

W1 is the volume of coal excavated due to direct cutting;

W2 is the volume of coal broken out from the ridge between adjacent slots when the jet is cutting;
W1 + W2 is the volume of the fractured zone cut by traversing the swing oscillation jet;
W3 is the volume of collapsed coal on both sides of the fractured zone caused by vertical impact and its own dead weight;
W4 is the volume of coal collapsing into the opening from beyond the area directly affected;

W1, the amount of coal excavated by direct cutting, should be minimized, while great effort should be made to increase W2, W3, and W4, and especially to maximize the volumes W3 and W4 (7).

Based on the above principles, an appropriate procedure should be designed as a function of the hardness of the coal, and the location and condition of joints and bedding planes in order to improve coal cutting ability. Although the water pressure used was not too high (e.g. 350-500 kg/cm^2); in a coal seam with a hardness coefficient of $\leq$ 1.2-1.5, a swing-oscillation jet cutter could mine in excess of 0.8-1.3 ton/min if an appropriate extraction pattern was adopted.

Figure 8 shows the nozzle assembly cutting from the left to right. Figure 9 shows the performance of a swing-oscillation jet while impacting on top coal. All the pictures were taken at the underground site. The working pressure was temporarily decreased for the photographs.

<u>Preliminary Trials of Coal Drivage Using the Swing-Oscillation Water Jet:</u>

The major conclusions which have been drawn from recent underground testing and production drivage of mine entries are as follows:

In the case of suitable coal seams coal cutting rates of between 4.0-7.5 m/ hr in coal roadways with a sectional area of 4.0-7.5 m, have been achieved.

1. During the drivage process, average dust measurements indicated less than 10 mg/m^3 of dust in the air around the coal face. This level is within the limits of "China Health Protection and Safety Standards of Coal Mines".
2. The measured water content of the fallen coal was less than 10% -11%, which results in better suppression of dust in the links of the production chair and means that the technique is applicable to coal mines with any means of coal transportation.
3. The quality of the road ways driven by the water jet was high. Routine maintenance operations were considerably simplified with significantly lower material costs because the rock surrounding the opening was undamaged.
4. Because of the complete absence of explosive sources, sparks, etc., a considerable improvement in safety conditions at the coal face was achieved.
5. Due to the very low reaction force required for jet clotting, much lighter and handier jet cutters, easier to transport underground, can be designed.

By the end of 1982, a total of more than 500 meters of coal drivage had been completed. Figure 10 shows an operating Swing-Oscillation Water Jet Cutter working in a narrow drivage.

Work is now continuing in driving coal roadways, and further tests of a developmental nature are being carried out.

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Figure 1. The performance of a swing-oscillation water jet. (This picture was taken from the cutting system with swing in the horizontal plane, oscillation in the vertical).

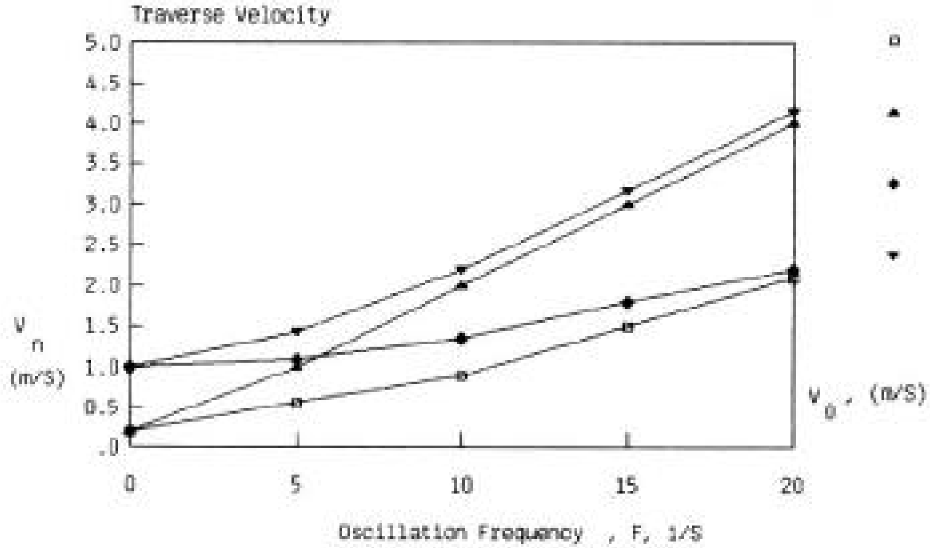

Figure 2. Correlation curves of the swing-oscillation jet between moving speed (V_n) and frequency of oscillation (F); amplitude of oscillation (A) amd traverse speed (V_0).

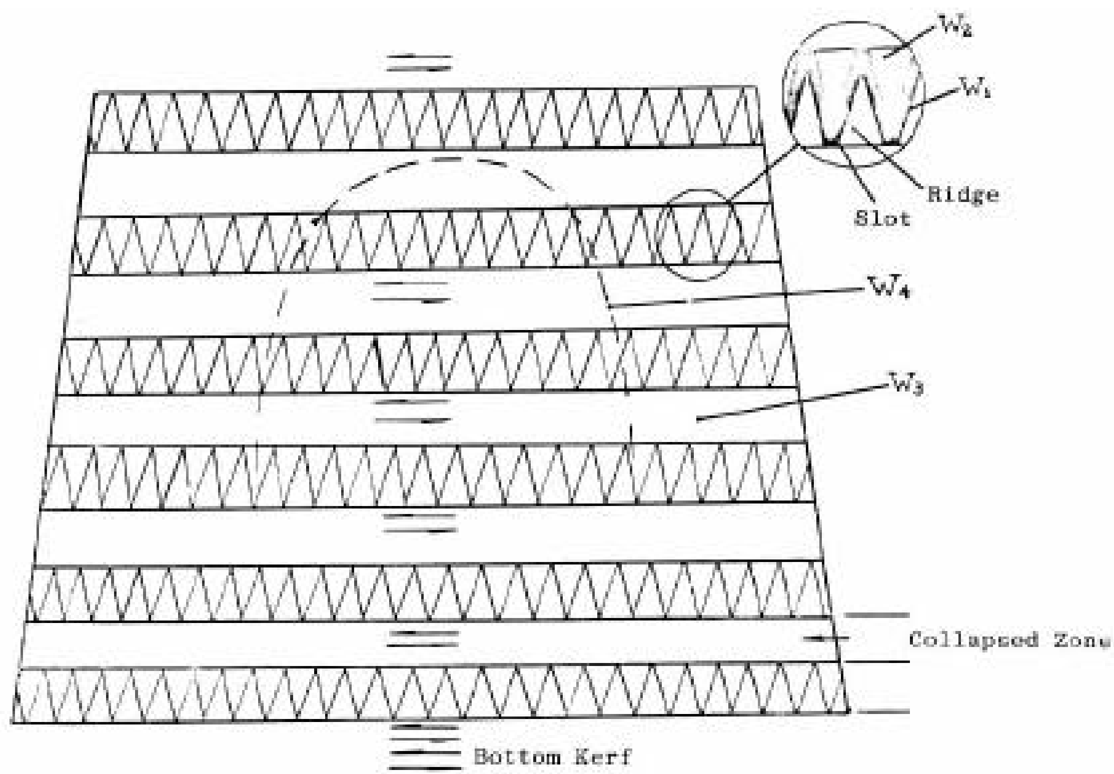

Figure 3. A schematic diagram of the typical process of coal drivage using a swing-oscillation water jet.

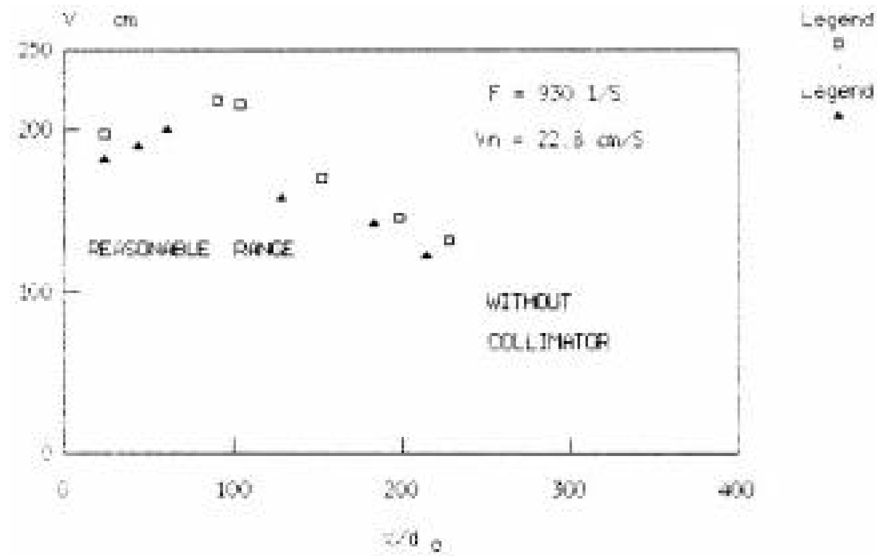

Figure 4. Cutting ability of swing-oscillation jet and its reasonable extent of working length.

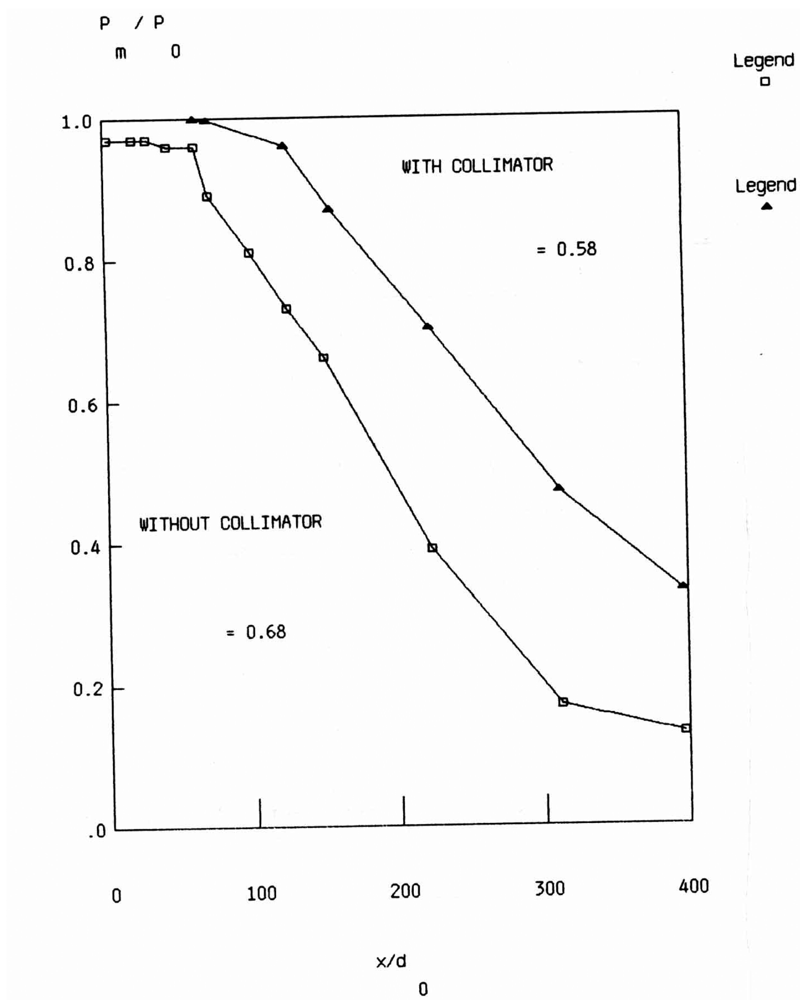

Figure 5. Comparison of the axial dynamic pressure distribution curves between the two cases, with a steady-flow device and without.

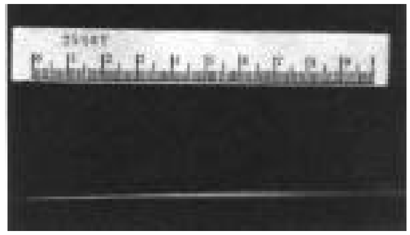

Figure 6. The photograph of a swing oscillation water jet without a steady flow device.

Figure 7. The photograph of a swing-oscillation water jet with a steady flow device.

Figure 8. The cutting performance (cutting from left to right).

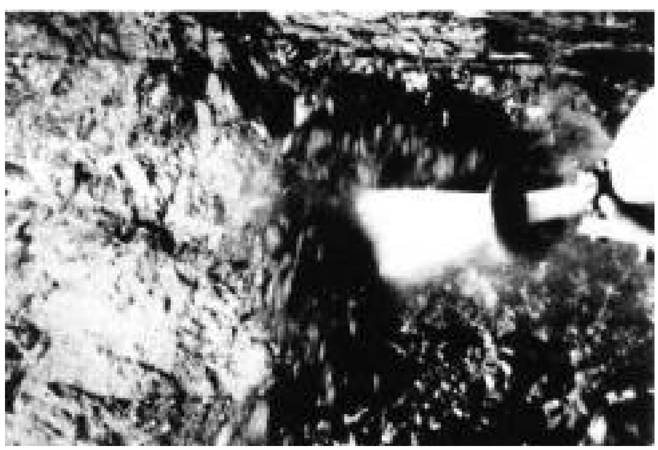

Figure 9. Impacting the top coal with a swing-oscillation water jet.

Figure 10. An operating performance in driving in a small cross-section roadway.

A PREVIEW OF METHODS FOR CUTTING CONCRETE

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INTRODUCTION

Background. In the event of bomb damage to a runway, military plans require repair and restoration of the pavement with priority over all but life-threatening activities. Minimum time has been allotted to the concrete cutting phase of the bomb crater repair to remove the distorted concrete.

The linear distance desired to cut around the crater is assumed to be 400 ft. A 1-hour cutting time is assigned. With these assumptions an acceptable cutting device should be capable of cutting 400 lin ft of 12-in.-thick concrete in 1 hour. The area of concrete thus exposed is 400 ft times 1 ft or 400 sq ft. Consequently all cutting methods reviewed are tabulated in terms of their exposure rate (the product of the linear foot of travel, in feet per hour, times the thickness of the actual cut reported, in feet) and compared to the 400 sq ft/hour target rate of cutting.

Scope. This study evaluates several means of cutting concrete including the water jet, water jet with abrasives, explosives, and mechanical means for cutting 12-in.-thick concrete. Both single and hybrid techniques were investigated.

CUTTING DEFINED

Concrete cutting will be arbitrarily defined as a method of parting a concrete monolith along some predetermined straight line (1/2-in. variance in 12 ft) in such a manner that a reasonably smooth surface is left along the cut face of the remaining concrete. Crushing or breaking the concrete is not accurate enough for the limitations imposed above.

METHODS EXPLAINED

Water Jet. The water jet is a method of cutting that employs a narrow stream of water at a high velocity to erode the concrete. The water may or may not cut through the aggregate used in the concrete and it has no effect on the reinforcing steel other than to thoroughly remove all concrete from its surface. The water jet device consists of a pump powered by electricity, gasoline, or compressed air to pressurize the water, an intensifier pump for pressures greater than approximately 2600 bars (38,000 psi), and a nozzle which controls the diameter of the water iet.

Water Jet with Abrasives. Abrasives can be included in the water jet for improved effectiveness. The abrasive is thoroughly agitated by the water jet and adds significantly to the cutting rate for any water jet as compared to a water jet without abrasives under otherwise similar conditions. Maximum effectiveness occurs when the abrasive is evenly distributed over the entire area of jet impingement. Obviously the type of material used for an abrasive, the abrasive particle size, and its rate of entry (usually measured in pounds per unit time) into the stream of the water jet will have a significant impact on the cutting rate and/or performance of the water jet. Since the water supplied (even potable water) is not sediment free, nozzle openings tend to enlarge with use. The average life of a water jet nozzle is estimated to be 800 hours. In general, water jets produce no dust, have a good maintenance record, and are relatively safe as long as contact with the jet itself is avoided. Operators can hand hold a water jet operating at up to 15,000 lbf/ sq in. without excessive fatigue (Barron and Nichols, 1973). Reducing water flow permits an increase in pressure and the upper limits of each vary with the operator.

Explosives. Explosives have the unique ability to concentrate a large quantity of energy in a very small area. Indeed, one of the chief peacetime uses of explosives is in the demolition of unwanted structures. Controlled explosions to produce a minimum amount of damage to adjacent structures have been developed. Shaped linear charges are being evaluated for cutting concrete and will be reported separately by another U. S. Army Engineer Waterways Experiment Station (WES) laboratory.

Mechanical Means. Rotary action diamond saws are the most common type of saw used to cut concrete (Kubo et al., 1981). Saws are capable of making large, precise cuts with a minimum of damage to the exposed concrete, produce no dust, and are relatively safe to operate. The cutting shape is limited as well as the cutting depth (less than one-half of the blade diameter). Blades tend to wear quickly and a high noise level is associated with saw operation. The saw is usually powered by a gasoline engine, although other means are available.

Thermal Means. Three thermal methods of cutting concrete by melting with intense heat are currently in use, the powder torch, the thermal lance, and the powder lance -- a combination of the first two. The powder torch employs intense heat generated by the reaction between oxygen and powdered iron and aluminum to melt a cut into concrete (Kubo et al., 1981). The thermal lance employs intense heat generated by the reaction between oxygen and mild steel rods to melt a cut into concrete. The powder lance combines some of the features of the powder torch and the thermal lance. Besides being much too slow, these three methods obviously damage the remaining concrete to a depth that has not been determined. The extent of damage depends on the degree of heat employed and the time of exposure. These devices are most effective in cutting reinforced concrete.

Method Combinations. The water jet with abrasives was explained previously. A water jet with mechanical pick is currently under study. This method uses a water jet located art finally in front of the mechanical pick hut now behind (or trailing) the pick. With the water jet trailing the mechanical pick, the vertical force could be reduced 40% to 50% and the cutting force approximately 33% (Styron, 1982a).

A device called a water wedge was demonstrated at the WES. In application the water wedge is loaded with a 12-gauge blank shotgun shell which provides a basic energy source. The device is then placed in a predrilled hole filled with water. When the shotgun shell is "shot," the energy is transmitted by the water to the concrete, causing it to crack. There is the possibility that several holes can be drilled in a line and "shot" simultaneously which would define a reasonably straight fracture plane. The design of this particular device is not suited for thin (12-in.-thick) concrete slabs. With some modification(s) it could prove to be a forerunner in the search for acceptable concrete cutting techniques. The speed of cut is mainly determined by the rate the holes can be drilled, assuming several holes/devices will be used in tandem (Styron, 1982b).

Another combination of methods employs a concrete saw to cut the slab 6 in. deep instead of 12 in. A mechanical impactor (jack hammer) device is used to break the concrete away from the remaining segment. The broken part of the remaining concrete may he used to provide a supporting surface for the replacement surface.

DATA TABULATION

The data search for specifics concerning various methods of cutting concrete was frustrating. At the start, a data sheet was prepared listing cutting method, cutting rate, fuel type employed, fuel consumption at the stated rate, water consumption (spillage/overflow that would probably find its way into the crater) , noise level, number of personnel required, and pertinent comments. The intent was to determine the time required to cut 400 sq ft and the gallons of fuel needed. Not one single report, brochure, or document listed all the desired information. Additionally, very few limitations were noted, although most mentioned that antifreeze would be necessary for water-lubricated/cooled cutting devices operating in freezing weather. Some indicated the most important variable in cutting un-reinforced concrete is the type of aggregate - those formed from silica and quartzite being extremely hard and difficult to cut. The lack of information is understandable when one considers that concrete hardness (compressive strength and type of aggregate employed) (Mellor, 1975) and concrete depth vary as does the depth of cut employed. These three variables have a significant impact on the rate of cut (exposure rate) realized.

The data presented below are separated by the type of technology used to cut/ break the concrete. The exposure rate and the technology level, if commercially available or a research and development item, are tabulated (Tables 1 - 4), together with major technical issues and/or remarks noted.

Concrete Saw. Concrete cutting saws are mechanical devices that currently enjoy 80% of the concrete cutting market (Kubo et al., 1981). Two basic types are in use -- the carbide-tipped saw and the diamond-blade saw. The carbide-tipped saw technology is based on the abrasive removal of concrete using a hard carbide-tipped saw with sufficient power. Blade speed is important since blade wear and maintenance increase with blade speed. Blade life is the main issue. Equipment mobility under the expected use conditions are suspect. Blade lubricant/coolant is required by some but not by others. The diamond-blade saw is generally the faster of the two (Table 1) and reportedly quieter (no details/ data stated). Most saws in operation today use diamond blades. Care must he taken and a certain amount of expertise is desirable because at high blade speeds the impact between the diamond particles used to cut the concrete and the concrete can break the particles, while at slower speeds the diamond particles take deeper cuts in the concrete which tends to dislodge the diamond particle from the blade. The exact nature of the material bonding the diamond particle to the blade varies between "hard" and "soft" concrete and is proprietary among companies. Diamond blades can cut reinforcing steel but blade wear increases approximately one and three-quarter times (Kubo et al., 1981). Diamond blades must be water cooled. Most commercial sawing today is only 2 to 4 in. deep and the (commercial) equipment available is relatively small. Current thinking concerning developing a concrete saw to satisfy military requirements seems divided over whether to make several passes with currently available but modified equipment (or one pass with multiple blades in tandem), or to try to increase the equipment size, power, etc., to meet the 400 sq ft/hour goal. This is seen to be a sizable undertaking, hut is at least theoretically feasible.

Water Jet. In order to cut concrete with a jet of water, the water pressure generated must be greater than the compressive strength of the concrete (Barren and Nichols, 1973). This fact is also advantageous when cleaning concrete surfaces whereby keeping the water pressure below the compressive strength of the concrete, a deposit can be selectively cleaned from the surface without fear of abrading the concrete. The biggest use of water jets around concrete is for cleaning and scarifying damaged concrete both on concrete surfaces and in concrete cracks. Besides water pressure and nozzle diameter, other factors affecting cutting performance are the nozzle angle with respect to the surface to he cut, the rate of cut, the nozzle height above the surface, the nozzle size, and whether or not a free surface is available for the swarf (the abraded concrete and water mixture). Commercial intensifiers are available to boost pump pressures up to 60,00 psi. There are two nozzle designs: the standoff nozzle that does not enter the cut region (kerf) and the penetrating nozzle that does enter the cut region. Penetrating nozzles are usually designed to rotate and provide clearance in the kerf for the nozzle. Closely associated with the high operating pressure is a corresponding requirement for pure water. Critical equipment parts, i.e., seals and nozzles, and their maintenance history closely reflect the purity of the water used. Efforts are currently directed to improving the effectiveness of the pumping systems, the nozzle swivels, and the pressure rating of the flexible hose. Specific exposure rates noted in the literature and the major technical issues/remarks are listed in Table 2.

Combinations of Cutting Methods. Current concrete cutting methods indicate "hybrid" technologies may achieve concrete cutting rates significantly faster than those produced by the current state of the art.

One hybrid technology incorporates a high-pressure water jet with a mechanical tool. The research and development efforts on this device have currently been directed toward use in rock excavation. Tests have been conducted on rocks with compressive strengths from 10,000 to 40,000 psi. The preliminary results indicate the technique could be used for cutting concrete with high productivity.

In the technique used for rock cutting, a mechanical tool such as a carbide tip mounted on a wheel rotating at high speed produces a crack zone and a water jet is directed at this zone which removes and further fractures the rock. The apparatus can make numerous passes over the area to be excavated and on each subsequent pass, the cutting tool wil1 remove the loose rock produced by the water jet from the previous pass in addition to providing the initial cracks that the water jet can exploit. This combined operation substantially reduces the requirements on both the water jet and cutting tool . The water jet can be operated at hydraulic fracturing pressures in the range of 5,000 to 10,000 psi instead of the much higher pressures which are needed for cutting the rock with .iets alone while the thrust and cutting forces for the use of the cutting tool are reduced by as much as 50 %.

The water-jet-assisted mechanical cutter has been used on 20,000-psi sandstone achieving a removal rate of 5 - 1() sq. ft/min (equivalent to a cut 12 in. deep and 5 to 10 ft long per fin). The device used a pointed pick that was forced into the concrete in conjunction with a 10,000-psi jet (Wang and Wolgamott, 1978). Later developments indicate the jet is more beneficial trailing (behind) the bit forming an angle of 45° with the bit (Styron, 1982a). This information is being analyzed and the report should be published this year (Table 3).

A water wedge, a device designed and developed by Mr. Don Hadak, was demonstrated at the WES (Styron, 1982b). The water wedge is a cylindrical device approximately 3 ft long composed of two basic parts: a perforated tube and a powder chamber. The perforated tube is inserted in predrilled holes in the concrete to be cut. The holes are 1-1/2 in. in diameter and filled with water. Energy for the device is supplied by a common 12-gauge shotgun shell. then the shell is set off, pressure is forced through the holes in the perforated tube, causing the concrete to crack. Unfortunately, the line of cracks did not follow the line of predrilled holes. This method has the definite advantage of being easily carried through debris strewn areas caused by bomb damage and operated safely even when several are operated in nearby areas at the same time. This method does not require nearly as much water as the more promising methods previously mentioned. Some additional research and development is required to ensure that cracks produced in the concrete follow the line of predrilled holes. A hydraulic wedge is reported to cause concrete to crack within the diameter of the predrilled holes (1-5/8-in. diam). This device is still in the research and development stage also (Kubo et al ., 1981) .

A water jet with abrasives is currently under study. Multiple jets (2 to 6) are arranged symmetrically around the periphery of a nozzle with an abrasive introduced through a hole in the center of the nozzle (Styron, 1982a). water pressure ranges from 10 to 30 ksi; abrasives are added at the rate of 2 lb/min. Difficulty was experienced when the dry abrasive that is force-fed by air pressure was dampened by back-spray from the jetted water. One possible solution to this problem might be to suspend the abrasive in a liquid prior to its point of introduction into the water jet.

In Germany engineer construction units are being trained to repair bomb damaged runways with commercially available concrete saws that are used to cut around the periphery of bomb craters to shallow depths (15 cm) followed by an impact device used to break away the remaining segment.[1] Specific rates noted in the literature and the major technical issues/remarks are listed in Table 3. Reported rates of cut/repair using this procedure vary considerably. The extent of future interest in this method of repair will hinge directly on the success or failure of the repairs/tests under study when subjected to actual expected loads.

Other Methods. Lasers cut materials by melting or immediately vaporizing the material. Currently, a slow cutting rate, shallow cutting capability, large immobile size, inaccuracy of alignment, and a lack of remote cutting application limit the laser's potential (Kubo et al., 1981). Other electrical methods fare no better.

The use of chemicals for cutting concrete at the desired rate for military planning is beyond the current state of technology and would require an extensive research and development program.

Thermal Devices. The powder torch, the thermal lance, and the powder lance all consume large amounts of fuel and produce some damage to the face of the remaining concrete to an unspecified depth, and are best used to burn numerous holes in the concrete with the section finally broken away (Kubo et al., 1981). In addition, the rate of cut is slow and vast quantities of smoke are produced; similarly, the fuel oil and compressed air method (Fay, 1971) uses a lot of fuel, causes the rock to. spall, and produces an undetermined amount of damage to the remaining concrete (Stanley, 1980). Specific rates noted in the literature and the major technical issues/remarks are listed in Table 4.

RESULTS, CONCLUSIONS, AND RECOMMENDATIONS

Results. As a result of this review, exposure rates are observed to vary from 1.25 sq ft/hour (laser cut) to 375 sq ft/hour (water jet with ahrasives). Most of the reported rates are the best values obtained from laboratory studies. In some cases these rates involve shallow cuts of 1 in.; in other cases the concrete specimen was passed under the cutting device. This is not meant to cast doubt on the validity of laboratory cases, but simply to point out the large technological gap between the reported laboratory cutting rates and the commercial cutting rates of today (and the desired exposure rate of 400 sq ft/hour). Further, practically all tabulated values lack verification. After reviewing the referenced literature, it appears the most reliable method of making long cuts of 12-in.thick concrete is with the concrete saw. The saw equipment is commercially available and the average exposure rate varies from 50 to 70 sq ft/ hour.

Conclusions. The concrete saw with diamond blade is the best method currently available with a demonstrated capability of cutting large amounts of concrete. Further, commercial units are available today .

[1] Brabston W.N. 1982, personal communication with Mr Van Orman Tyndall Air Force Base, Florida.

Commercial equipment capable of cutting 400 lin ft of 12-in.-thick concrete in 1 hour is not currently available, nor does there appear to be an imminent breakthrough available in the foreseeable future.

Impact devices, thermal methods, and chemicals appear to hold little promise as potential concrete cutters for future programs.

Recommendations. Research and development must and should continue, preferably at an accelerated rate, to exploit the most promising methods listed below:

1. Diamond-blade saw with more horsepower for a single pass and with multiple blades arranged in a train formation of some type
2. Multiple water jet with abrasives
3. Water jet with mechanical pick
4. Water wedge
5. Carbide-tipped saw with impactor

Large expanses of unjointed concrete pavements should be scored initially in some predetermined pattern to improve the ability of cutting crews to cut square patterns.

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Technology	Cut/ Break	Exposure Rate sq ft/hr	Technology Level	Major Technical Issues/Remarks
Carbide-tipped saw	Cut	60-300	Commercial*	Lubricant/coolant not required by all; blade wear
Carbide-tipped saw	Cut	109-165	Commercial	Includes 4.5 in. asphalt and 22 in. frozen ground
Carbide-tipped saw	Cut	--	Commercial	Cutting bits variable
Carbide-tipped saw	Cut	250	Commercial	10 in. concrete; gas or diesel fuel
Diamond-blade saw	Cut	20	Commercial	Free surface required; blade wear
Diamond-blade saw	Cut	60	Commercial	Type of aggregate important
Diamond-blade saw	Cut	20	Commercial	
Diamond-blade saw	Cut	295-354	Commercial	19,000 gal water/hr
Diamond-blade saw	Cut	58.3	Commercial	
Diamond-blade saw	Cut	11.7	Commercial	Reciprocating action saber saw

* Commercial means as opposed to R&D (research and development).

Table 1. Concrete saw technology.

Technology	Cut/ Break	Exposure Rate sq ft/hr	Technology Level	Major Technical Issues/Remarks
Water jet	Cut	240	Commercial	High water consumption
Water jet	Cut & break	60	Commercial	
Water jet	Cut	8	Commercial	12 gpm; open hand hold to 15,000 lbf/sq in.
Water jet	Cut	240	R&D	Dual rotating nozzle 12 sq ft/hr
Water jet	Cut	48.4	Commercial	Slotting concrete
Water jet	Cut	58.3	Commercial	Ultrahigh pressure, 40 ksi
Water jet	Cut	3.9-13.6	Commercial	
Water jet	Cut	4.2	Commercial	6.5-ksi concrete with 1-1/4-in.-diam bars

Table 2. High-pressure water jet technology

Technology	Cut/ Break	Exposure Rate sq ft/hr	Technology Level	Major Technical Issues/Remarks
Combinations				
Water jet with pick	Cut	up to 200	R&D	Water jet trails pick
Water wedge	Break	unknown	R&D	WES demo indicated high potential
Water jet with abrasives	Cut	375	R&D	Fluidyne demo; abrasives 2 lb/min
Hydraulic wedge	Break	unknown	R&D	1-5/8- to 1-3/4-in.-diam holes; 12 in. on center
Concrete saw/ impact device	Cut/ break	52.5	Commercial	Telephone conv. from Mr. Van Orman in Germany to Dr. Brabston; 33F, no rain; 16-man team
Concrete saw/ impact device	Cut/ break	325	Commercial	Telephone conv. from LTC Bergholz, Ramstein AFB, FRG to Dr. Brabston

Table 3. Combinations of concrete cutting methods.

Technology	Melt/ Cut	Exposure Rate sq ft/hr	Technology Level	Major Technical Issues/Remarks
Powder torch	Melt/ cut	8.3	R&D	Damages exposed face of concrete
Thermal lance	Melt/ cut	8.3	R&D	Damages exposed face of concrete
Powder lance	Melt/ cut	8.3	R&D	Damages exposed face of concrete
Fuel oil & compressed air	Melt/ cut	115.7	R&D	8-1/2-in.-diam well @ 52 ft/hr
Thermojet air fuel	Melt/ cut	12-18	R&D	Kerosene or diesel (9-15 gal/hr); burnt concrete forms a slag that is not easily removed
Laser	Cut	390	R&D	Undocumented
Laser	Cut	1.25	R&D	Theoretical analysis

Table 4. Thermal methods of cutting concrete

DISCUSSION

NAME: Tom Brunsing

COMPANY: Foster-Miller

QUESTION: "Do cuts in concrete need to go around curves or can it be all straight cuts?"

ANSWER: Straight cuts with square corners are preferred in order for the crater to be "capped off" with some synthetic material, probably square or rectangular shaped.

NAME: Andrew Conn

COMPANY: Tracor Hydronautics

QUESTION: "Other than your specification of 400 ft /hr for cutting the concrete, do you have any others, such as a limit on total water used, power, weight or size of the system?"

ANSWER: Presently, we are only interested in some means of cutting concrete at the rate of 400 sq ft per hour. If a solution presents an additional problem(s) it (they) will have to be judged separately. Should two solutions be found, then obviously the faster, lighter, cheaper, etc., system would be selected.

NAME: Larry Pater

COMPANY: Dravo

QUESTION: "What are the physical characteristics of the runway concrete (compressive/ tensile ultimate strength, aggregate size and type, reinforcement, etc.)?"

ANSWER: The compressive strength of the runway concrete will vary over a wide range with an average of about 6000 pounds per square inch. The maximum aggregate size is 2 inches, and the actual size encountered will vary from a maximum of 2 inches down to sand-size particles. (Reference Department of the Army Military Construction Guide Specification MCGS 02611, December 1975, for details.) Reinforcement, where encountered, may be cut by some other means; for example, with an acetylene torch or bolt cutters--we're strictly interested in cutting concrete (only) at the rate of 400 sg ft per hour.

NAME: Dr. M. Vijay

COMPANY: National Research Council of Canada

QUESTION: "Since you are cutting large areas of concrete, have you considered the use of pulsed jets for initial rough fracture?"

ANSWER: Pulsed jets were not noted in the literature reviewed; only water jets with mechanical picks, water jets with abrasives, and water jets used alone. I would suspect any two-phased operation would find it very difficult to meet our time constraints. There is the additional concern for water in the crater. Most of this water will probably find its way into the crater making the sub-base compaction repair phase more difficult.

FEASIBILITY STUDY OF CUTTING SOME MATERIALS OF INDUSTRIAL INTEREST WITH HIGH PRESSURE WATER JETS

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ABSTRACT

Recently a number of experiments have been conducted in the laboratory to studio the feasibility of using high pressure water jets for the following applications:

1. Cutting thin layers of thermoplastic rubber material for use as shoe heel covers,
2. Cutting furs,
3. Processing sea food products, and
4. Removing baked enamel from thin sheets of aluminum, used as sidings for buildings.

The objectives in each of these investigations were
(i) to establish if water jets could indeed perform the required operations and
(ii) if found promising, to collect data to fully assess their capabilities.

Experimental results pertaining to these investigations are presented in this paper.

INTRODUCTION

The experiments describes here were carried out in the laboratory at the request of industries that had experienced some difficulties in processing materials of their interest with conventional cutters and therefore were in search of alternative methods. The purpose of the investigations was to acquire data, on the basis of which appropriate recommendations could he made to the company or organization concerned. Of the several enquiries dealt with recently, the following four were particularly interesting.

(1) Cutting thermoplastic rubber material,
(2) cutting furs,
(3) processing sea food products and
(4) removing baked enamel from thin aluminium siding strips.

These experiments are described in the following sections.

1. THERMOPLASTIC RUBBER MATERIAL (REF. 1)

Thin slices of this material are used as shoe heel covers, Fig. 1. The material is fabricated with layers of different colors as shown in Figs. 2 and 3. The blocks can also have various shapes, mostly dictated by the shape of the heel, Fig. 3.

Cutting thin slices of this material with conventional methods posed certain problems. First, as shown in Fig. 3, the material peeled off frequently causing the seizing of the blades or the teeth of the saw. Second, due to the heat generated by friction, color was transferred from one layer to another (Fig . 3) rendering the material unusable. These problems resulted in loss of productivity. The firm concerned was interested to learn if the use of high-pressure water jets would overcome these problems. From the standpoint of rate of production, the firm stated that water jets should cut thin (~ 1.0 mm) slices fom pre-shaped blocks of 200 mm (thickness) x 400 mm (length), at a traverse speed of 0.8 m/min. The results sought were: the quality of cuts, the magnitudes of pressure and power.

Tests were conducted in the laboratory using an intensifier type of pump (the sane pump was used for all the experiments reported in this paper) capable of delivering 2.4 liters/min of water at the rated pressure of 310 MPa. Since thin slices were required to be cut, nozzles of diameter <0. 254 mm were employed, the standoff distance being 3.2 mm. The traverse speed was maintained at the required value of 0.8 m/min for all the tests.

Fig. 4 shows that high pressure water jets could be employed to cut this material. The quality of the cut was judged to he better than that obtained with the conventional cutting systems (Fiqs. 2 & 3), except near the bottom of the sample from where the jet emerges. The rough or irregular cuts in this region were due to lack of sufficient power in the jet. This could be corrected by (i) increasing the pressure (ii) adding long chain polymers, such as polyox to increase the coherent length of the jet or (iii) cutting the material with two jets, placed on opposite sides of the block. Fig. 5 shows a minor problem encountered in the process of cutting this material with the jets. The particles (debris) of material removed were found to agglomerate and adhere to the face cut at some locations. This was not serious as long as the slices were separated from the parent block immediately after cutting and were washed in a stream of water. Otherwise, the debris adhered permanently to the slices, marring their appearance.

<u>Conclusions:</u> The test results showed that a 0.254 mm jet at 207 MPa could cut through a 7.6 cm thick block of the material, traversed under the jet at a speed of 0.8 m/min. Linear extrapolation of the data (Ref. 1) indicated that to cut through a 20 cm thick block of the material with a single jet of the same diameter would require pressures in the neighborhood of 550 MPa. This pressure is rather high and could be reduced, for example, by employing two jets positioned on either side of the block to be cut. With such an arrangement, tin 0.254 mm jets at 276 MPa would be able to cut through the 20 cm block of the material, requiring a total power of 22.4 W.

This information together with an approximate estimate of costs for adapting the water jet system for this application was conveyed to the firm concerned .

2. FURS WITH AND WITHOOUT STITCHES (REF. 2)

Fig. 6 shows a sample of fur in which several cuts have been made with a gang of mechanical knives. Althouqh the cuts made by the knives were clean and sharp they posed a problem in that the spacing between adjacent cuts was difficult to control. The company concerned was interested to find out if the application of high pressure water jets would simplify this problem. Tests were conducted in the laboratory to obtain data on hydraulic power required to cut through samples of plain furs and also those containing some stitches (see Fiq. 8). It was also important to examine the quality of the cuts.

Before the tests could be run it was necessary to hold the fur in tension while traversing under the jet. The required tension was applied by two rubber bands incorporated into a clamping device shown in Figure. 7. Experimental data for plain furs are summarized in table 1 and for furs containing stitches are depicted in Fig. 9. The cuts made by the water jets (Figure. 8) were as good as those made by the mechanical knives (Figure. 6) . As is clear from table 1 and Fig. 9, the magnitudes of pressure and power required to cut through the material were dependent on the presence or absence of stitches. Since the traverse speed was stated to be of minor importance, the requirements would be much lower than those reported here. Fig. 9 shows that the pressure decreases substantially as the nozzle diameter is increased from 0.076 to 0.203 mm. However, minimum power levels are achieved for nozzle diameters in the range of 0.10 to 0.12 mm. Also, from the standpoint of the amount of water, quality and the spacing of the cuts, nozzle diameters in this range would be well suited for this application.

<u>Conclusions:</u> The recommendation to the firm was that it is possible to use high pressure water jets for fur cutting applications. For a typical fur sample containing stitches, traversed under the jet at a speed of 30.5 cm/sec, the power required to cut through the sample was approximately 0.6 kW (nozzle diameters 0.1 to 0.12 mm; pressure = 100 to 130 MPa).

3 . SEA FOOD PRODUCTS (REF . 3)

There appears to be a considerable interest in processing food products in general with high pressure water jets (Ref. . 4) . Work reported here had the objective of determining if water jets had any distinct advantages over the band saws for processing fresh cod fish, frozen ocean perch (redfish), lobster and frozen cod fish fillets, as limited work conducted earlier in the laboratory (Ref. . 5) had indicated that it might be worthwhile to pursue this investigation. Processing sea food products with conventional methods is highly labor intensive. For example consider lobster . At present meat is re novel from lobsters by a four-step process (Ref. 6). Lobsters are first boiled whole. This serves to cook the meat and drive out some of the moisture contained in the meat, thereby shrinking it. The lobster is then frozen so that the meat solidifies. This is done to make the meat resistant to mechanical shock. The lobster is then exploded to remove the meat. The exploding is accomplished by placing the lobster in a vacuum chamber where it would explode due to the pressure differential between its interior and exterior. The meat is then separated from the shell. Also, these manual and mechanical operations apparently involve unavoidable wastage of edible meat. Therefore, the fishing industry is on the lookout for new methods that would improve the production rates and minimize the losses. Study of computerized prototype machines incorporating high-pressure water jets for processing fish is already in progress in some countries (Ref. 7).

Tee requirements (type of cuts, etc) and the outcome of the tests are displayed in Figs. 10 to 22. A limited number of experimental data pertaining to cutting of fresh cod fish, frozen ocean perch (red fish), lobster and frozen blocks of cod fish fillets are summarized in table 2.

Fresh Cod Fish: Tests were conducted to assess the possibilities of gutting, throat cutting, bob-tailing (Fig. 10) and filleting (Fig. 11). The traverse speed in all of these tests was maintained at 7.6 cm/sec. As has already been reported (Ref . 5), high speed water jets were quite effective for gutting operations. Belly could be cut (to remove the viscera, etc) at a relatively low pressure of 69 MPa, consuming only 0.83 kW of hydraulic power. Cutting the throat, bob-tailing and filleting required rather high pressures (=276 MPa). Moreover, as shown in Fig. 11, the texture of the flesh deteriorated as there was not enough power in the jet to cut through the bottom layers of the fish, yielding pulpy or mushy type of meat. It was felt that the quality of the meat could be improved by increasing the jet (diameter <0.25 mm) pressure to about 400 MPa.

Frozen Ocean Perch (Redfish) The object of these tests was also to determine if water jets could be used to remove the head and viscera of a whole frozen perch by making cuts in various directions as shown in Fig . 12. The first direction involved severing the head by means of one straight cut through the gills (Fig . 13A) . In the second, a cut was required from the back of the head to the anus (Fig. 13B) . In the third direction, two cuts were made, one from the back of the head to the gills and the other from the gills to the anus, resulting in a fillet as shown in Fig. 14. Excellent cuts were achieved (in two passes) with a 0.203 mm nozzle operating at a pressure of 276 MPa (hydraulic power = 6.6 kW) . The traverse speed in these tests was 2.5 cm/ sec .

Fresh & Cooked Lobster: Of the 27% of the lobster that is edible, most of the meat is concentrated in the tail (49.5%) and the claws (25.6%, Ref. 8). The tests on lobster involved the extraction of meat from the tail and claws (Figs . 15 to 18) . The tail meat of a fresh lobster was removed by traversing the lobster lengthwise under the jet (Fig. 15). In such cuts it was desirable for the jet to penetrate the shell but not the meat underneath. However, the pressure required to fracture the shell was such that the jet passed right through the meat rendering it mushy or guppy (Fig. 16). It is known that when a lobster is cooked, its meat shrinks by about 5.28% inside the shell (Ref. 8). To take advantage of this shrinkage, it was decided to use cooked lobsters for extracting the meat. Cuts were made along the outer edges of the tail or the claw (Fig. 17 & 18) where the jet passed through easily without damaging the meat. These clean cuts were obtained at a pressure of 242 MPa (hydraulic power t 5.4 kW) . The traverse speed was 2.5 cm/sec.

Frozen Block of Cod Fish Fillets: Fig. 19 shows a typical deep frozen block (46x29x6 cm) consisting of layers of cod fish fillets. At present conventional band saws are used to cut from these blocks saleable portions of fish fillet such as fish sticks . The purpose of the tests conducted in the laboratory was to determine if water jets could perform the same task with minimal wastage and clean cutting. Initial tests were conducted at a traverse speed of 2 cm/sec and at pressures as high as 310 MPa using a nozzle of 0.203 mm in diameter. The results were unsatisfactory because the cuts were not clean (Fig. 20) resulting in considerable wastage of meat. Subsequent tests were carried out at a traverse speed of 0.66 cm/sec using nozzles of diameter in the range of 0.076 to 0.203 mm.. The pressure was varied from 103 to 310 MPa. Figs. 21 and 22 show typical cuts obtained in these tests . The wastage was minimal (=0 .6%) .

Conclusions: From the limited number of tests conducted in the laboratory it was concluded that (a) processing of fresh Cod fish with high pressure water jets requires further investigation and (b) high pressure water jets have good potential for cutting frozen whole fish, lobsters and frozen blocks containing layers of fish fillets.

The organizations for whom this work was performed are at present investigating the economics of introducing the jet cutting systems for processing various sea food products.

4. REMOVING BAKED ENAMEL

Aluminum siding strips coated with baked acrylic paint are used extensively to cover the outside surfaces of wooden buildings. Since the sidings are manufactured in standard lengths, a large quantity of short pieces of strips remain after these standard lengths are cut to the lengths required to cover the buildings. Therefore, there is an economic incentive to reclaim the metal for other uses and this requires removing the paint from the strips. As heat or acid assisted scraping was found to be uneconomical, the company concerned was in search of an alternative method to perform this task. The purpose of the laboratory investigation was therefore to find out if water jets could be used for this application.

Experimental results were obtained at traverse speeds of 5.1 and 28 cm/sec. Pressure varied from 69 to 138 MPa, the nozzle diameter from 0.203 to 0.356 mm., and the standoff distance frown 2.5 to 15.7 mm.. At both traverse speeds, water jets removed the paint completely without damaging the metal. For a given nozzle at a fixed pressure, maximum width of the paint was removed when the standoff distance was about 9 mm (see Fig. 23).

Conclusions: The experimental data have indicated that high pressure water jets could be used for removing baked acrylic paint from aluminum siding strips. In practice, however, two or more oscillating jets would be required to remove paints from 8 to 10 cm wide strips. The company is presently investigating the economics of a water jet system for this application .

ACKNOWLEDGEMENTS

Thanks are due to Mr. W. J. Goulding for assisting in the laboratory on jet cutting of fish, etc. The authors appreciate the interest shown in this project by Mr. R. A. Tyler, Head, Gas Dynamics Laboratory, N.R.C. Thanks are also due to Mrs. J. Robinson for typing the manuscript of the paper.

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Nozzle Pressure (MPa)	Traverse Speed (cm/sec)	Flow Rate (liter/min)	Hydraulic Power (kW)	Remarks
103.5	101.6	0.221	0.380	The samples were
138.0	101.6	0.255	0.590	cut through in all
172.5	101.6	0.286	0.820	of these tests.
138.0	154.9	0.255	0.590	
172.5	154.9	0.286	0.820	The cut edges
207.0	154.9	0.313	1.080	were sharp and
				clean.
138.0	190.5	0.255	0.590	
172.5	190.5	0.286	0.820	
207.0	190.5	0.313	1.080	

Summary of Experimental results (Fig. 8)
Thickness of the skin of the fur sample = 0.635 +/- 0.065 mm.
Nozzle diameter = 0.102 mm
Standoff distance 3.2 mm

Table 1. Cutting plain fur with high pressure waterjets

Test No.	Nozzle Diameter (mm)	Nozzle Pressure (MPa)	Remarks
1	0.203	69.0	Belly cut but not collar bones.
2	0.203	138.0	Belly cut & collar bone severed.
3	0.203	207.0	Throat cut not satisfactory
10	0.254	276.0	Tail cut through.
13	0.203	276.0	Tail cut through. Good cut.
17	0.203	207.0	Filleting cuts. One cut on
18	0.203	276.0	each side of fins. Good cuts,
19	0.203	276.0	but flesh made "mushy" (Fig. 11).

2.1 Heading and gutting fresh cod fish (Figs. 10 & 11)
Traverse speed = 7.62 cm/sec

Test No.	Nozzle Diameter (mm)	Nozzle Pressure (MPa)	Remarks
1	0.203	276.0	Neck cuts. Fish completely
2	0.203	276.0	cut through in two passes (Fig. 13A)
5	0.203	276.0	Neck and belly separation. Cut through in two passes (Fig. 14)
6	0.203	276.0	Fillet cut. Cut through (Fig. 13B)

2.2 Heading and Gutting frozen ocean Perch (redfish)
Traverse speed 2.5 cm/sec

Test No.	Nozzle Diameter (mm)	Nozzle Pressure (MPa)	Remarks
1	0.203	207.0	Cutting shell of fresh lobster. Traverse speed = 7.6 cm/sec (Fig. 16)
2	0.203	172.5	Cutting claw of cooked lobster (Fig. 17)
3	0.203	207.0	Claw cut through (Fig. 17)
6	0.203	241.5	Cutting claws along the edges.
7	0.203	241.5	Good cuts (Fig. 18)

2.3 Cutting fresh and cooked lobster (Figs. 15 – 18)
Traverse speed 2.5 cm/sec (except where noted

Test No.	Nozzle Diameter (mm)	Nozzle Pressure (MPa)	Remarks
2	0.203	310.5	Traverse = 2 cm/sec. Cut through, but quality no good (Fig. 20).
5	0.203	310.5	Cut through. Quality good (Fig. 21).
7	0.127	310.5	Cut through. Quality good.
9	0.076	310.5	Cutting small slices (Fig. 22)
10	0.076	241.5	Thickness of slice = 2.9 cm.
11	0.076	172.5	Excellent cuts, except No. 13
12	0.076	138.0	
13	0.076	103.5	Not cut.
14	0.076	138.0	In these tests loss of meat due
15	0.076	207.0	to width of cuts was measured. Wastage
16	0.076	207.0	was minimal (Average loss = 0.6%).

2.4 Cutting frozen blocks of cod fish fillers (Figs. 19 – 22)
Traverse speed = 0.66 cm/sec exccept where noted.

Table 2. Processing Sea Food Products with high pressure water jets.
In the above tables the experimental results are shown in figures 10 – 22.
The standoff distance in all tests was 6.35 mm.

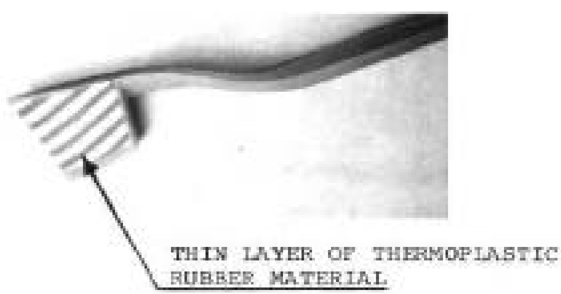


Figure 1. Thermoplastic rubber material used as a shoe heel cover.

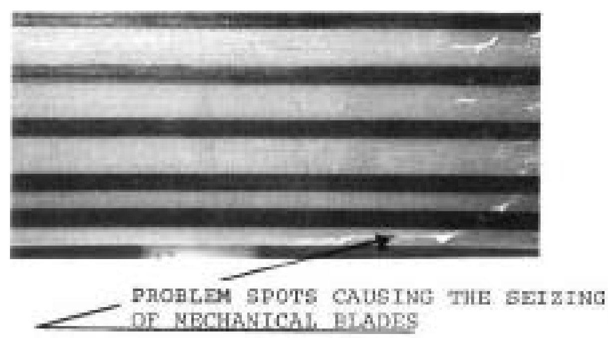


Figure 2. A block of thermoplastic rubber cut with a mechanical cutter.

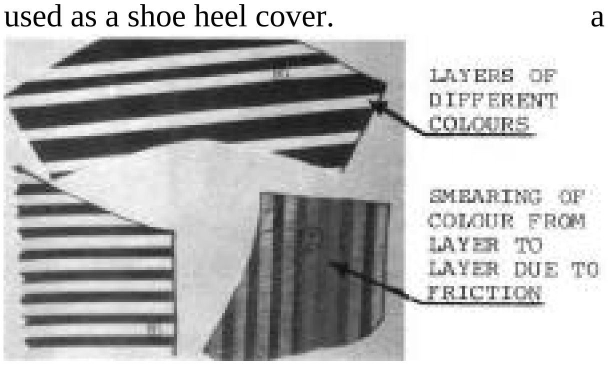


Figure 3. Thin slices of thermoplastic rubber cut with a mechanical cutter.

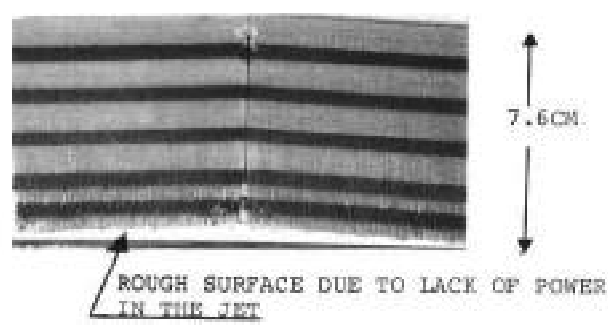


Figure 4. Sample of thermoplastic rubber cut through with water jets.

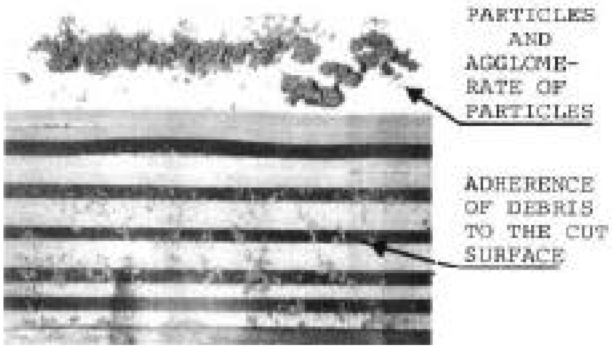
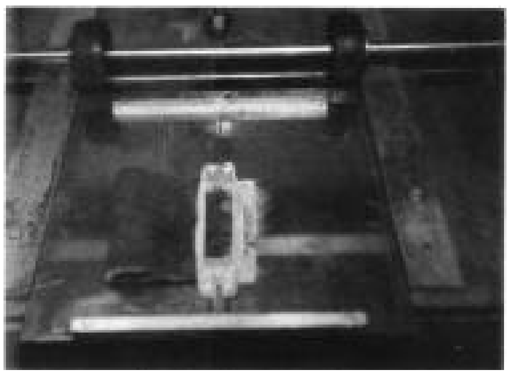


Figure 5. The cuts surface of a block of thermo-plastic rubber showing the adherence of debris.

Figure 6. Sample of fur showing cuts made with a gang of mechanical knives.



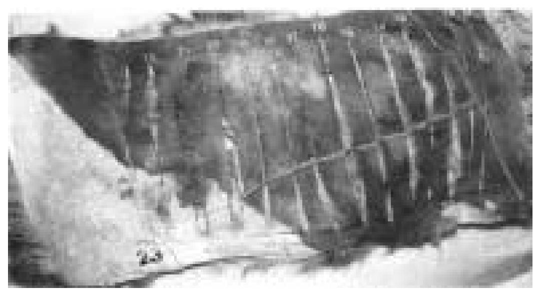


Figure 7. Sample of fur held in tension for Traversing under the jet on a pinch roller.

Figure 8. A sample of fur showing cuts made with high pressure water jets.

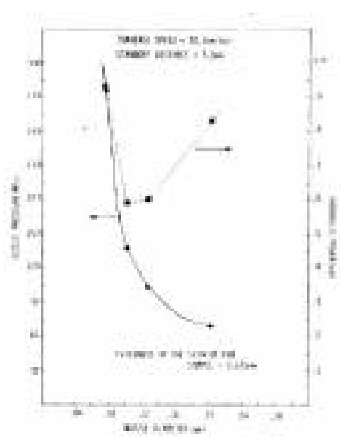


Figure 9. A plot of pressuer and power required to cut through samples of fur with stitches.

Figure 10. Tail cuts of fresh cod fish.

Figure 11. Fillet cuts of fresh cod fish.

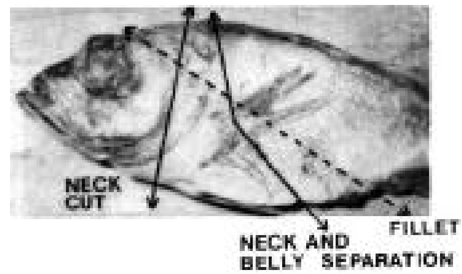


Figure 12. Frozen ocean Perch showing the desired directions of cut.

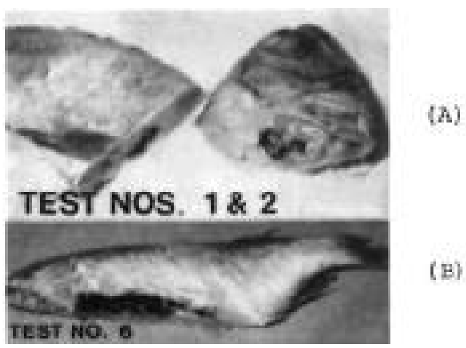


Figure 13. Frozen fish cut through in two passes.

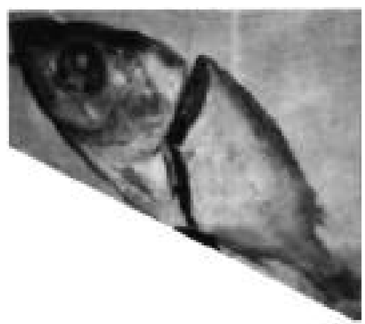


Figure 14. Fillet cuts of the frozen fish made in two passes.

Figure 15. Cooked lobster showing the desired locations of cut.

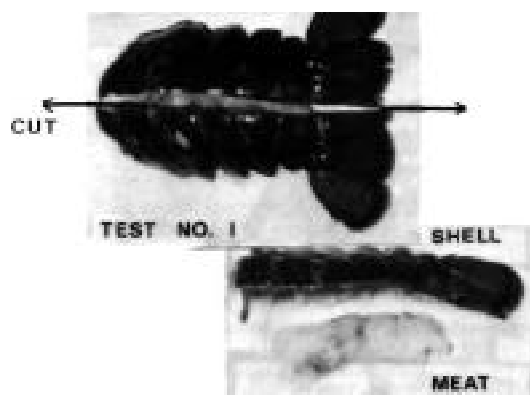


Figure 16. Cut made through the top shell of a fresh lobster.

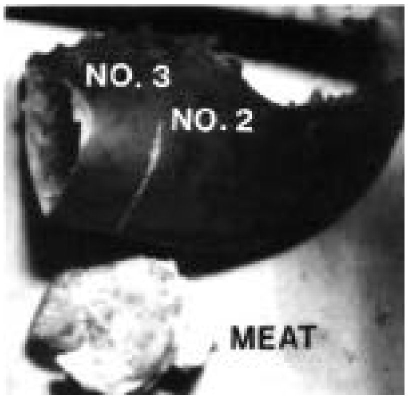


Figure 17. Cuts made across the claw of a cooked lobster.

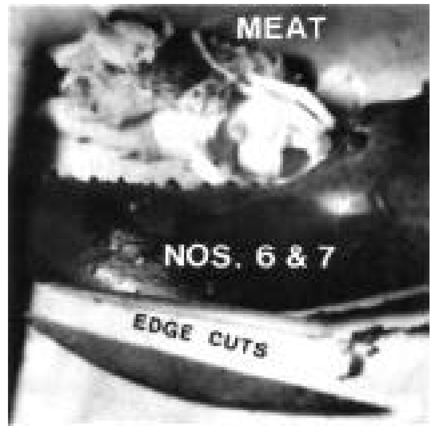


Figure 18. Excellent edge cuts of the claw Meat not damaged.

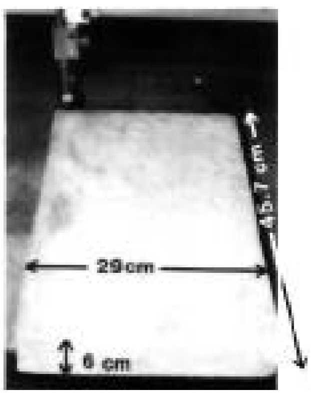


Figure 19. A frozen block consisting of layers of cod fish fillets.

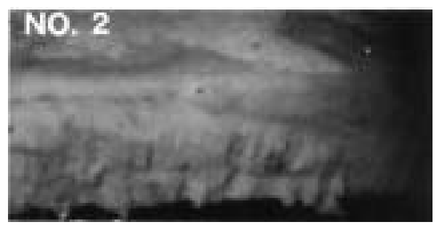


Figure 20. Frozen block of cod fish fillets Cut through. Quality of cut not satisfactory.

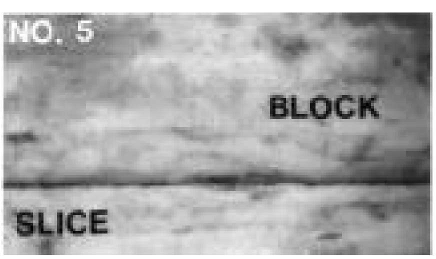


Figure 21. Frozen block of cod fish fillets cut through. Quality of cut Excellent.

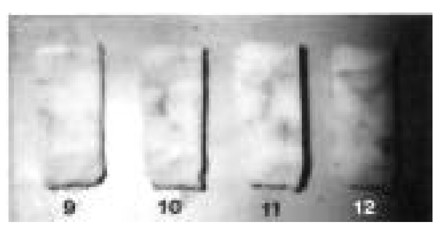


Figure 22. Typical 2.9 cm thick small slices (Fish fingers) cut with a 0.076 mm nozzle.

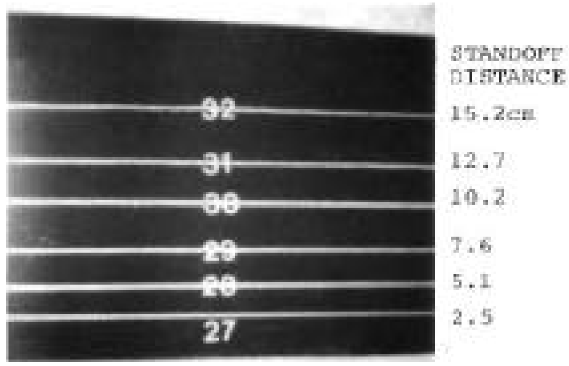


Figure 23. Removing baked enamel from a sheet of aluminum.
(Nozzle diameter = 0.254 mm, pressure 138 MPa, Traverse speed = 28 cm/sec)

DISCUSSION

NAME: Andrew F. Conn
COMPANY: Tracor Hydronautics

QUESTION: "In your tests for removal of baked enamel from aluminum siding, your optimum X/d (standoff: X = 10.2 cm, nozzle dia.: d = 0.254 mm) was 402. At this distance, would you agree that the erosive mechanism was droplet impingement, and not a steady erosive jet (l/2 pv2) mechanism?"

ANSWER: The authors do not agree The erosive mechanism is not due to droplet impingement. The width of the paint removed increases because the cross sectional area of the jet increases due to aerodynamic drag. The coherent length of a good jet can persist up to a distance of 2000 nozzle diameters. In any case, if the erosion was due to droplet impingement, the width of the paint removed would be irregular (random patches), not clean as shown in the photograph of the paper

NAME: Steve Sugino
COMPANY: Sugino Machine Limited

QUESTION: "When you cut the frozen fish or fish fillet, didn't you have any problem of dehydration on its surface? What was the cutting condition?"

ANSWER: No, we did not find any dehydration problems. Frozen samples of fish were stored in a deep freezer prior to testing They were cut immediately after removing them from the freezer. At the feed rates involved, dehydration would be a problem. If anything, softening of the top layers of the tissue due to defrosting would be a problem, but only a minor one.

JET NOTCHING USED IN THE CONSTRUCTION OF A CHEMICAL WASTE ISOLATION BARRIER

Thomas Brunsing
Foster-Miller, Inc. Waltham, Ma

ABSTRACT

Jet notching can be used in conjunction with hydraulic fracturing and vertical displacement for constructing a complete isolation barrier around contaminated ground. This "Block Displacement" process has been demonstrated adjacent to a chemical waste site near Jacksonville, Florida. The process can be used for isolating chemical solid or hazardous waste as well as for other construction barrier needs. Geologic considerations dictate applicability and cost of the process and influence the sequence of construction of bottom and perimeter barriers.

INTRODUCTION

Block Displacement is a method for vertically lifting a large mass of earth. The technique produces a fixed underground physical barrier placed around and beneath the earth mass. The barrier is formed by pumping slurry, usually a soil bentonite and water mixture, into a series of notched injection holes. The resulting barriers completely isolates the earth mass from groundwater migration.

A bottom barrier is formed when lenticular separations extending from horizontal notches at the base of injection holes coalesce into a larger separation beneath the mass block of earth (Figure 1). Continued pumping of slurry under pressure produces a large uplift force against the bottom of the block and results in vertical displacement proportional to the volume of slurry pumped.

A perimeter barrier is constructed in conjunction with the bottom barrier either prior to or following bottom barrier construction. The perimeter barrier can be constructed by various means including slurry wall, vibrating beam or jet grouting techniques. The perimeter wall constructed prior to bottom separation can be used to ensure a favorable horizontal stress field for proper orientation of the propagating bottom separation. In geologic formations not requiring control of horizontal stress, the perimeter may be constructed following initial bottom separation or following the completion of block lift.

The Block Displacement Method can be used to increase the width of an initially thin perimeter barrier such as might be constructed by vibrating beam or jet grouting techniques. To increase perimeter width by means of the block lift the thin perimeter must be constructed on a slight angle off of the vertical prior to a substantial portion of the lift (Cleary 1979).

POTENTIAL COST BENEFITS

The Block Displacement process can produce complete in-situ isolation to any desired depth. Complete in-situ isolation using a slurry wall or other conventional vertical barrier alone requires keying into a naturally occurring impermeable stratum such as clay or unweathered bedrock. Thus, Block Displacement is economically attractive where the depth to such a natural stratum is greater than the depth of contaminated soil. Figure 2 shows the tradeoff between Block Displacement and slurry wall for different combinations of contaminant depth and depth to impermeable stratum.

Block Displacement realizes additional cost savings by isolating the minimum volume of soil, thus minimizing the fluid handling requirements for subsequent in-situ and flushing treatment techniques.

Block Displacement can be more economically attractive than all alternative techniques under suitable site conditions. For the example shown in Table 1, it is 1/3 the cost of a slurry wall barrier and 1/2 the cost of a well point system. Relative cost of incineration has been included for reference.

These relative cost figures are based on standard construction costs and have not considered costs peculiar to a specific hazardous site operation (Walsh, 1982).

<u>Block Displacement at the Whitehouse Site</u>

A demonstration of Block Displacement was recently conducted adjacent to a Superfund site in Whitehouse, FL. The demonstration "block" was 60 ft. in diameter by 23 ft. deep, and composed of unconsolidated marine sediments (Brunsing and Grube, 1982).

The test program successfully demonstrated the fundamental aspects of the Block Displacement process. Figure 3 shows a perpendicular cross section of the displacement of the earth mass. Core samples were taken to verify the continuity of the bottom barrier in-situ, and barrier thicknesses ranging from 5 to 12 in. were obtained.

<u>Bottom Notching and Fracture</u>

The original demonstration plans called for injection hole bottom notching using an elaborately plumbed system of high pressure - high volume air, sand and slurry to be pumped to the base of each of seven casings. A notching nozzle would be inserted to the bottom of a casing and high pressure and volume air would be pumped in thus displacing the mud and groundwater and keeping the hole open. Next, sand would be injected into the line and used to erode a notch radially out from the hole bottom. The volume of eroded material captured at the surface would give an approximation of the notch diameter. When the desired notch size had been reached, the air would slowly be replaced with bentonite slurry until the entire notch and casing were filled. The process would then proceed with displacing the ground upward by injecting slurry under low pressure into the notches. The major drawback to this system is that air back pressure control, required to stabilize the notch opening, is difficult to maintain both during the erosional process and during slurry backfilling. For this reason, a slurry jet was used for notching. Initially a 2% bentonite slurry was jetted in a slurry medium in all seven injection holes yielding 2 feet diameter notches in each.

Bottom notch diameter was determined both indirectly by measuring the volume of cuttings removed during notching and directly by means of a down hole caliper. The volume of cuttings were measured by recording the change in height of settled material in a 55 gallon drum used as a settling tank. The down hole caliper measured the insertion distance of a spring steel tongue, Figure 5, which was forced into a horizontal orientation over a series of rollers when engaged. The caliper was manually operated from the surface by pushing a sliding rod downward relative to a fixed position caliper body which could be lowered to the desired depth in the injection hole.

Initial analysis indicated horizontal separation should initiate from 2 foot diameter bottom notches at an injection hole pressure of approximately 20 psi. Initial attempts at injection required pressures in excess of 40 psi to induce slurry flow indicating that renotching of each injection hole followed by immediate slurry injection was required. The resistance to separation propagation at the tip of the notch was attributed to excessive filter cake build-up resulting from slurry leak off occurring over a period of a few weeks. By inducing propagation immediately after notching this build up would be avoided.

Higher injection pressures also lead to leakage up along the casings. Leakage into the overburden in general often reached the surface in the vicinity of injection holes. In the course of modifying the notching system to achieve the desired performance over 20 leaks to the surface were encountered requiring the development of a routine leak seal off procedure. Initially portland cement grout was pumped under low pressure into the leaking injection hole, allowed to set and through drilled in preparation for another notching attempt at slightly greater depth. This operation proved time consuming and made subsequent notching operation more difficult.

A second method of sealing off leaks was therefore adopted and refined. A thick slurry containing swelling particles of varying diameter was pumped into the leaking injection hole and injection pressure was gradually increased allowing these particles to bridge in the leak channel and seal off with filter cake build-up. Upon attaining a satisfactory pressure (35 to 40 psi) the hole was renotched followed by immediate separation propagation.

Deformation analysis of an infinite half space indicated injection pressure should be inversely proportional to notch diameter. The notching mechanism was therefore modified to increase notch diameter to 4 feet. The jet system was now designed to erode with a jet of 2 percent bentonite slurry in an air medium (figure 6). Slurry was pumped down a 2 inch diameter line to a 1/2 inch diameter horizontal jet (figure 7). Slurry and cuttings and excess air then returned up a 1 1/2 inch diameter return line from a sump beneath the jet to a settling tank on surface. The air medium was maintained by a slow inflow of compressed air at the casing collar. Back pressure was maintained by a flow control valve throttling the return flow at the settling tank. Excess air bubbled into the return line in the sump and was released at the settling tank. This system was used to notch and propagate separations from intermediate injection holes. The initial 4 foot notches were each expanded until they intersected the perimeter of the block. Each separation was therefore propagated until it obtained a diameter of approximately 26 ft.

The % bentonite slurry in air notching operation was applied to the central hole but injection pressure again was too high. A high pressure (150 psi) jet pump, was brought in to increase the jet efficiency yielding a 6 foot diameter notch on the central hole. Injection then proceeded smoothly at 25 psi with the central hole connecting to each of the intermediate injection holes after 500 gallons of slurry had been injected.

At this point, visible upward displacement was measured at inner points around the block by means of a surveyor's level located 50 feet beyond the block perimeter. Immediately following observed pressure transfer to intermediate injection holes, flow was observed at the perimeter of the block. A total of 2,000 cu. ft. of soil bentonite slurry was then pumped into the coalesced bottom separation displacing upward a comparable volume of ground (figure 8).

GEOLOGIC CONSIDERATIONS

Three basic geologic considerations must be addressed when evaluating block displacement applicability. Fluid flow characteristics and discontinuities within the vicinity of bottom barrier construction must be compatible with the process. A site which meets these first two criteria must be evaluated with regard to insitu material properties and stresses within the strata for design purposes.

Soil stratification can act as a separation guide. Typically less consolidated and more permeable material act as the separation conduit with less permeable more consolidated material above and beneath acting as the guide boundaries. Sand interbedded between silt layers is an example.

On a micro scale, anisotropy in tensile strength can control separation orientation. The separation will propagate in a plane perpendicular to the direction of minimum tensile strength of the material. Horizontally oriented consolidated sediments generally have less tensile strength in the vertical direction normal to bedding, which is desirable.

In-situ stresses also act to orient separation propagation. Separations propagate in the plane normal to the direction of minimum principal stress. The in-situ stress relationship is the result of geomorphological events and is difficult to assess in general. The vertical stress is equal to overburden weight. The horizontal stresses however should be measured in-situ. Alternately, horizontal stresses can be controlled by constructing and surcharging a perimeter barrier prior to bottom separation propagation.

A relationship between in-situ stress and material strength can be derived yielding a safety factor for assuring maintenance of horizontal orientation of the bottom separation.

$$S.F. = \frac{T_H + \sigma_H}{T_V + \sigma_V}$$

where S.F. = safety factor

T_H = tensile strength in the horizontal direction

T_V = tensile strength in the vertical direction

σ_H = in-situ stress in the horizontal direction
σ_V = in-situ stress in the vertical direction

For cohesionless material $\sigma_H/\sigma_V = 1$. Furthermore for homogeneous material in a uniform stress field (S.F. = 1) a separation propagating parallel to a nearby free surface will yield a disk shaped surface ultimately intersecting the free surface. Proper injection hole spacing can compensate for this gradual upward migration of the propagating separation.

In addition to evaluating separation propagation, separation initiation must be examined. The initiation pressure, Pi_H, required to propagate the bottom separation must be kept below the pressure required to induce vertical fracture, Pi_V, near the injection well casing. The desired initiation pressure is

$$Pi_H = \sigma_V + T_V + K_1 D\mu$$

while the undesired initiation pressure is

$$Pi_V=\sigma_H+T_H+K_2 L\mu$$

where D = bottom notch diameter
L = effective unlined length of injection well
μ = viscosity of injection fluid
K1 and K2 are constants associated with notch and well geometry

Lastly, the area and shape of the site being displaced will impact the displacement construction sequence. Excessively large areas may be lifted by section. Elongated areas may be lifted progressively from one end. Every site will require geologic and geometric consideration as part of the design process.

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2. Cleary, J. M., 1979, A Method for Displacing Large Blocks of Earth: U.S. Patent No. 4,230,368.
3. Walsh, J. J. et. al, 1982, Costs of Remedial Actions at Uncontrolled Hazardous Waste Sites: EPA Eighth Annual Symposium on Solid and Hazardous Waste Management, Cincinnati, Ohio.

METHOD:		COST
1.	Slurry wall to impermeable stratum	1.0
2.	Well point system	.63
3.	Block displacement (using a slurry wall perimeter)	.32
4.	Removal and incineration	6.38

Table 1. Life cycle cost comparison. (Note the cost comparison is for a 1 hectare square site contaminated to a depth of 5 m in 30 m of permeable soil, Groundwater migration rate is 100 m/year.

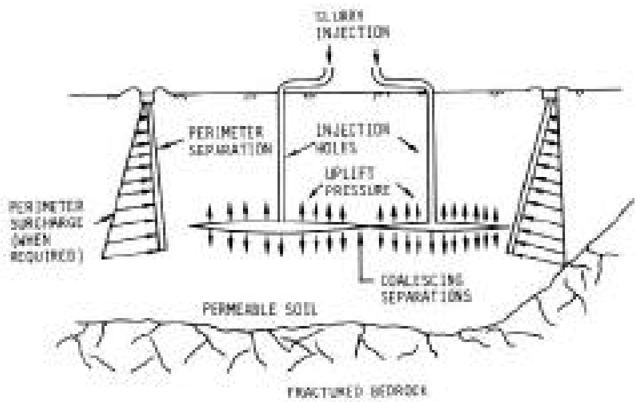

Figure 1.Creating separation to induce displacement.

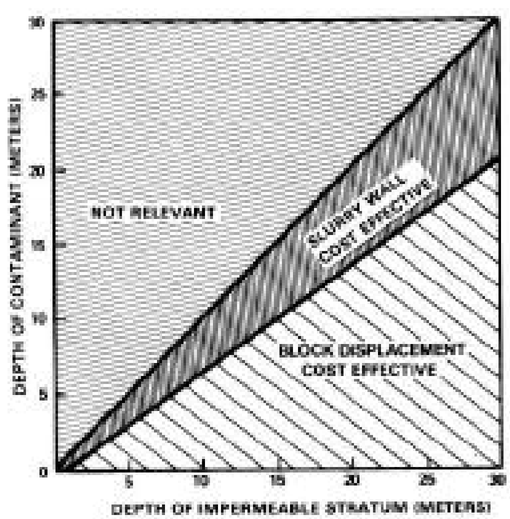

Figure 2. Cost comparison of block displacement versus slurry wall for a one hectare site (10,000 m^2)

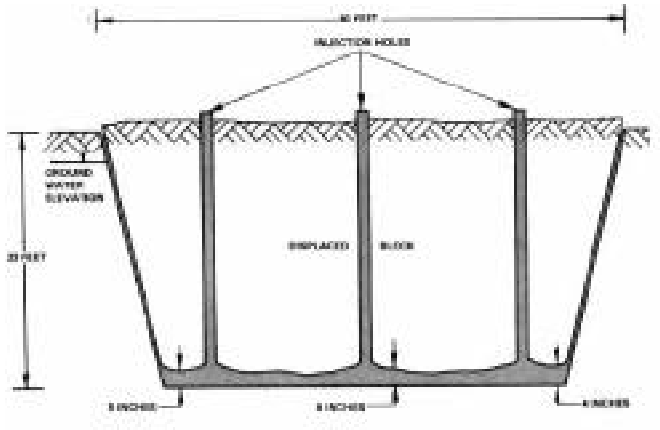

Figure 3. Final block displacement.

Figure 4. Upward displacement at surface due to block lift.

Figure 5. Down hole horizontal notch radius caliper.

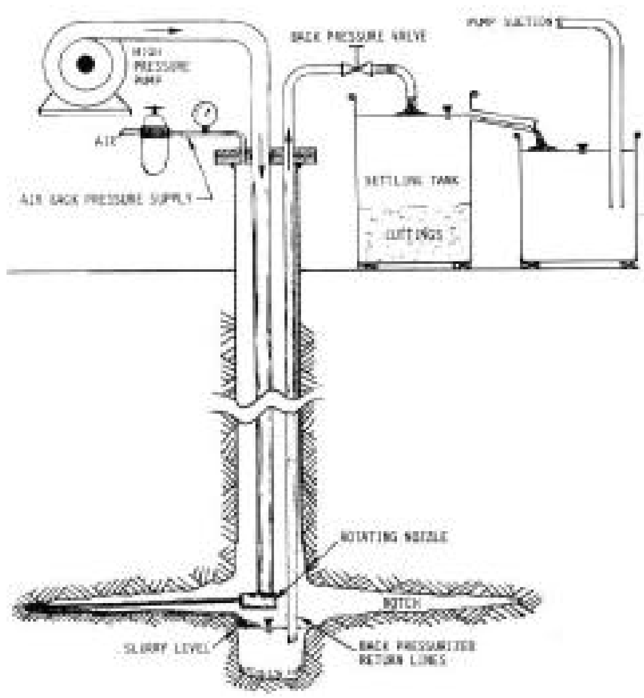

Figure 6. Slurry jet in air notching operation.

Figure 7. Slurry notching jet.

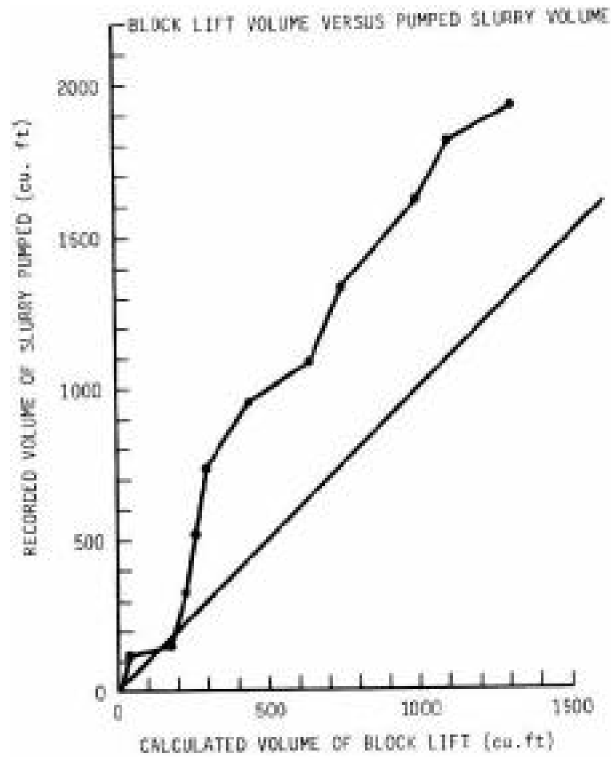

Figure 8. Comparison of Slurry Pumped to Volume Displaced During Block Lift

DISCUSSION

NAME: George Savanick
COMPANY: Twin Cities Research Center, Minn., MN

QUESTION: "It is difficult to see how unconsolidated material can be fractured. Would you please enlarge upon your use of the term 'fracture'?"

ANSWER: It is certainly true that the term "fracture" implies there is a fracture toughness associated with the material. In the Florida demonstration, the soil was cohesionless material thus having negligible toughness. In the text, I have tried to use the term separation. The resistance to separation opening is principally the in situ compressive stress field which is altered at the tip of the separation by the hydraulic pressure applied along the separation surface. The distribution of this separation opening pressure is a function of the separation shape, slurry properties and pumping rate. There are, therefore, similarities between the separation process which occurred in the unconsolidated sediments and hydraulic fracture in deep rock formations from an analytical standpoint.

Although I do not pretend to thoroughly understand the difference in behavior between these two processes, I feel that the major differences are in stability of the material at the separation tip. I hypothesis that the cohensionless soil below the groundwater table will experience a reduction in pore pressure as the separation extends thus leading to temporary liquefaction at the tip. This localized soil density reduction will redistribute with time spreading away from the unsupported separation tip. The length of unsupported tip is dependent on the deformation of the soil as there is a critical minimum width of separation or gap beyond which the ground supporting slurry will not penetrate within the separation. The localized density reduction yields a localized change in soil modulus which alters the size and orientation of the stress field at the separation tip. This alteration in stress field orientation will thus alter the direction of separation or "fracture" propagation as predicted by rock fracture theories. This direction of propagation is the characteristics of greatest significance to the Block Displacement process.

THE FURTHER DEVELOPMENT OF AN UNDERGROUND CABLE FOLLOWING TOOL

James M. Reichman, Douglas P. Kelley, Mark Marvin
Flow Industries, Inc.
Kent, Washington 98032

ABSTRACT

Flow Industries has been conducting a program aimed at developing a tool for replacing failed underground cable. The tool uses fluid jets to remove the soil surrounding the cable, thus facilitating the cable replacement without excavation from the surface. Field tests conducted during the summer of 1982 demonstrated that the tool was capable of working to distances in excess of 300 ft under various test conditions. The advance rate averaged 2 ft/min for sand and sandy loam. The cost of replacing cable in these types of soils compared favorably to current methods. However, tool advance was stopped in hard clay and cohesive soils. The tool performed satisfactorily in all soil conditions after changes were made in the cutting head. The reasons for the changes and the specific adjustments are discussed.

INTRODUCTION

Flow Industries, Inc. has been conducting an ongoing program for the Electric Power Research Institute (EPRI) for the development of tools used to replace directly buried electric cable. The tool currently under development is a cable follower. The tool is designed to advance along a buried cable and cut the soil around it, allowing the failed cable to be removed and a new cable put in its place. The motivation behind the development of this tool is the reduction of replacement costs associated with surface excavation for cable replacement. The need to cut road surfaces and landscaping can increase the cable replacement cost to as high a $95/ft. The savings that can be realized through the development of a tool that minimizes the required excavation is tremendous. In addition the use of the existing cable assures that the tool will exit at the desired point, thus eliminating the need for a guidance system. The tool also assures that damage to adjacent utilities does not occur.

In the course of developing the cable follower various changes were required to enable the tool to operate in a wide range of soils. Field tests and the modifications to the system are discussed in this paper, and an economic analysis of cable replacement using the system is presented. The results of the testing conducted to date and the economic evaluation are very positive.

CABLE FOLLOWER DESIGN

The cable follower has previously been described (Reichman and Kelley, 1982). For completeness the basic tool description is presented here. The tool operates by:

1. Using an array of fluid jets to remove the soil around the cable.
2. Incorporates an internal flushing system to minimize wear.
3. Pulling itself along the cable by means of an internal gripping system.

A brief description of the soil removal system and advancing system is presented below.

Fluid-Jet Soil Removal

Directly buried cable is generally put in the ground by two methods. In the cut and fill method, the cable is bedded in a select backfill material which can consist of a sand backfill or a screened native fill. In either case the soil in the trench is generally softer then the native soil even after many years. The second method of installation involves the use of cable plows. With this method the soil surrounding the cable will be in the same condition as the native soil. The cable follower must be capable of operating in various soil conditions, ranging from unconsolidated sand to hard cohesive soil, to gravely soil. The variation in soil type causes some problems in creating a stable hole. Therefor, most tests were conducted in the two extremes of soil types-sand and hard cohesive clay.

Linear cutting tests were conducted in a wide range of soils. The general finding of the study was that the soil types generally encountered in the vicinity of a directly buried utility could be cut at pressures below 5,000 psi. It was also found that at this pressure the jets would not damage adjacent cables. The tool was therefore set up to run at 5,000 psi and a maximun flow rate of 4 gpm. This represents approximately 12 hydraulic horsepower being supplied to the cutting head. The cutting of the soil is accomplished by covering the face of the tool with a series of jets which can cut the soil surrounding the cable. The exact position of the jets would be determined through tests.

Advancing

The tool advances along the cable by means of a double-acting internal gripping system. The tool's operating cycle is as follows:

1. The internal grippers are activated,
2. the advancing side of the cylinder is pressurized causing the nozzles to press forward into the soil,
3. the internal grippers are deactivated while the rear gripper activates to hold the tool in place, and
4. the retraction side of the cylinder is pressurized to pull the body forward.

This cycle is illustrated in Figure 1. The tool stroke is 6 in. in length. The tool is capable of a maximum advance rate of 4 ft/minute. An additional feature of the tool's current configuration is that the grippers can be placed in a neutral position and the tool pulled backward. This feature assures that the tool can be recovered should an impenetrable object be encountered.

Field System

The testing of the cable follower tool requires that the system be tested outside the facility. Consequently, it is necessary to develop a self-contained system even for initial testing. The system must be capable of supplying all the necessary controls to the tool as well as the cutting fluid. A water supply is necessary, since it is not always possible to use taps or hydrants. To achieve the design goal of operation to distances of over 300 ft. the tool requires water for 2.5 hours - even when advancing at 2 ft/min. Using water at 4 gpm, the tool requires a supply of 600 gallons.

The system, as built, consists of a trailer containing the hydraulic drive for the tool, a hydraulic intensifier pump, a hose reel, and shifting controls for the tool advance. A picture of the trailer-mounted system is shown in Figure 2. The water for the system is supplied by a water trailer if necessary or from taps when available. A brief description of the various components is presented below.

Trailer. The trailer consists of a power unit, hose reel, and system controls. The power unit includes an intensifier pump capable of delivering up to 5 gpm at 12,000 psi. The intensifier has been modified to operate on a variety of fluids including various drilling muds. Components that have been changed include the seals and the check valves. The component life with these fluids has been found to be acceptable. The intensifier is powered by a pressure compensated pump and operates through an open loop hydraulic system.

The unit also has a second hydraulic system which supplies the hydraulic power for the cable follower. This system is completely independent of tile intensifier system so that the operating pressures can be adjusted independently. This system has a maximum working pressure of 2,000 psi.

The third component of the power unit is an air compressor. This is driven off the follower hydraulic system. The power unit also has a feed pump which is used to supply fluid to the intensifier pump. This is necessary since it is not always possible to have water supplied at the proper pressure particularly when water is supplied from a water tank.

The prime mover for the power unit is a gas engine. The various components are housed in a frame. The placement of the components within the frame has been made with ease of service in mind. The power unit is shown in Figure 3.

System Controls. The cable follower operates in a cyclic manner. In order to effectively operate the tool it is necessary to use a control system. The system controls are mounted on the power unit. Their function is to operate the advancing cylinder in the tool. The system control is achieved with two timers. One timer controls the time of the advance stroke while the second controls the retraction stroke. The time duration can be set to optimize the advance rate in various soils. The system can be operated in a manual mode should it be desired. The system can be controlled either from the trailer or by means of a pendant control at the pit.

Hose Reel. The cable follower requires a high pressure line, and two hydraulic lines. In addition a retraction cable and air line are also required. In order to facilitate the deployment of the system all utilities have bee combined into an integral hose. The length of hose currently being used is 330 ft. The power unit output is supplied to the hose through a multi-passage swivel mounted on the hose reel. This enables the hose to be played out as the tool advances and the power reel can be used to pull the tool out of the hole.

System Deployment

The cable follower tests were conducted with the system described above. The tests have been conducted both at Flow and at utility sites. The procedures used for the operation of the tool consist of the digging of a starting pit and sump and the excavation of an exit pit. The tool must be placed on the cable in the pit. The actual operation of the tool is controlled from the surface. A photograph of a typical field test is shown in Figure 4. The operation of the system can easily be handled with a crew of two. In general the only support equipment required is a backhoe.

TEST RESULTS

The cable follower was first tested at Flow in a loamy soil. The tests were conducted to distances of up to 150 ft. The results of these tests were very encouraging. The tool advanced along the cable at speeds up to 2 ft/min. The fluid used was water and the nozzle configuration is shown in Figure 5. The hole created by the tool was generally stable and the cable could be pulled out of the ground by hand. Over 20 tests were conducted under similar soil conditions all with equally successful results.

The next series of tests were conducted on actual utility cable that was embedded in sand. The cable in this case was buried for 15 years and would permit us to see if old cable presented a mechanical problem for the tool. The test run was 230 ft. The results of the test showed that the tool could create a hole in sand allowing the tool to advance; however the hole was not stable. After advancing 70 ft the supply lines were stuck in the hole. The tests showed that the tool in its present configuration could not maintain an open hole in unconsolidated soil. It was therefore, necessary to understand the mechanisms that were stopping the tool and what could be done to solve the problem.

In order to facilitate testing in various soils it was decided that an above ground test facility would be advantageous. The simulation of various soil conditions could be easily done and the recovery of the tool could be achieved with minimal hand excavation. To this end a 20 ft long test trench was fabricated. The trench had a viewing window through which visual observations could be made. The test facility is shown in Figure 6.

The tests were conducted on sand buried cable. The test facility was capable of duplicating the problems encountered in the field. The problems that were stopping the tool was the unstable hole formation. This caused the hole to collapse around the tool, which in turn applied a high holding force on the tool.

The first approach to solving this problem was to inject air through the head of the tool. This created a highly turbulent flow and prevented collapse in the area immediately around the tool. The injection of air allowed the tool to move through the soil but it didn't create a stable hole. In actual operation the tool would be stopped by the drag force on the supply line. It was determined that a different solution must be found in order to make the tool operational in sand.

The next step was to examine the use of drilling mud as the cutting fluid. Tests were conducted with various fluids to determine their ability to suspend sand. The second characteristic of the fluid was the ability to "case" the hole. Of the fluids tested bentonite was found to meet the requirements. Bentonite when used in the cable follower allowed the tool to create a stable hole in sand. The hole formed remains open indefinitely allowing for easy replacement of directly buried utilities. At this point a decision was made to use bentonite as the fluid for the system.

The use of bentonite required modifications to the system. The principle change was in the fluid supply system. The bentonite must be mixed several hours prior to use in order to completely hydrate the bentonite. During operation bentonite must be completely agitated and filtered so as not to plug the nozzles. In order to accomplish this a mixing pump and filtration system was added to the system. A photograph of this is shown in Figure 7.

The testing of the bentonite cable follower system was conducted during the summer of 1982. The tests were conducted on buried cable of lengths up to 300 ft. In addition tests were conducted on cables with adjacent utilities in the trench. In all the above tests the cable was buried in unconsolidated soils.

The tests demonstrated the ability of the tool to penetrate the soil to the maximum test distance of 300 ft. The advance speed in this type of soil ranged from 2 fpm to a maximum of 4 fpm. The hole formed by the tool was in all cases stable. It was possible to extract the buried cable with a pulling force of less than 200 pounds. The tests conducted with random lay cable demonstrated the the jets do not damage adjacent cables. When equipped with a streamlined head, the tool can adequately handle crossing cables. At the conclusion of this series of tests it was determined that unconsolidated soils required no further testing.

The next series of tests with the tool were conducted in compacted clay. This represents a different problem for the tool. When the tool operates in unconsolidated soil it relies on stationary jets washing the soil from in front of the tool. In consolidated soils the washing action does not occur to the same extent. As a result when the head developed for unconsolidated soils was tested in compacted clay it could not advance. In order to remedy this situation two approaches were tried. The first examined the jet density required to clear the soil in front of the tool. The second examined the cutting of the soil with a rotating head.

The tests conducted to examine the ability of stationary jets to advance through cohesive soil involved first a determination of the maximum spacing between jets that results in complete washing and then the best shaped head for soil penetration In order to have a standard sample for testing, clay was obtained from one source and then compacted in a tube by means of a vibratory compactor. The resultant hardness of the clay was above that normally measured by standard shear testers. Consequently, a indenter was used to measure hardness. This allowed for repeatable data to be obtained. The tests found that the jet nozzles had to be spaced at a distance of less then 1/8 in. for complete removal. In addition it was found that since the jets must diverge in order to cover the entire face any material not washed by the jets formed a very stable "pyramid" which stopped the tool advance. Various head designs were tested in order to examine the feasibility of advancing through the soil. A test sled capable of advancing th head was used for the tests. The sled could apply constant force to the head at level equal to the force the tool can exert on the cable. A photogrpah of the various head designs and the pusher are shown in Figure 8.

The results of these tests were not encouraging. It was found that, in order to successfully negotiate the soil, the total power required for the head was three times that currently available on the machine. This implies that three times the flow rate would be necessary for advancement. This would pose a problem in both mixing of mud and disposal of slurry. In addition the advance of the head through the soil is dependent on the operation of every nozzle in the cutting head. Since the jet diameters must by on the order of 0.012 in. to keep the flow rate as low as possible the chance of nozzle plugging during the course of a run is high. The plugging of one nozzle can potentially stop the advance of the tool. This reduces the reliability of the tool. Consequently, the use of a cutting head with stationary jets does not appear to be feasible at the present time.

An alternative is to use a rotating head, which assures totak coverage of the soil in front of the tool. In order to test this approach, a Flow Jetminer™ was used with a low pressure mud supply. The tool was mounted on the pushing sled and advanced into the clay samples. The variables used in the tests included rotation speed, jet pressure, and nozzle pattern. The measured quantity was the hole diameter and the thrust. The test set up is shown in Figure 9.

The use of a rotating jet allows a unifourm diameter hole to be cut in all cohesive soils. Some drill samples are shown in Figure 10. Results of the parametric study demonstrated that pressure abouve 4,000 psi had no measurable effect on the maximum hole diameter. The maximum diameter that can be obtained from a given nozzle is controlled by the standoff distance from the jet. This indicates that the pressure decay is rapid in the mud filled hole. As a result, the maximum radius that can be reached is 1 in. beyond the jet. Based on the tests it was found that a 5-in. diameter hole could be created in the hardest clays with a rotating head utilizing 4 gpm of fluid.

The use of a rotating head as a general soil cutting tool is only possible if a stable hole can be created in non-compacted soils. Tests with the rotating head were conducted in sand. The results indicated that a stable hole is formed due to the bridging action of the mud. The control of the hole diameter is more difficult in that it requires the head to be in close proximity to the soil or the hole diameter will be too large. This tendency can be overcome by changing the power density on the head. Concentration of the cutting force away from the outer reaming jet reduces the tendency to overcut the hole.

The rotating head requires major modifications to be made to the tool. Included in this is the development of the cutting head, development of a compact down hole mud swivel, a down hole rotary motor, and rotational power and controls. The development of this hardware is currently underway. It is anticipated that this development will enable the tool to penetrate any soil type quickly and economically.

OPERATING ECONOMICS

In order to justify any development it is necessary to determine if the system presents a viable economical alternative to currently used practices. In the case of cable replacement the cost can vary greatly. In undeveloped areas the costs can be as low as several dollars per foot. In developed areas where extensive restoration is required the costs of replacement can run as high as $85/ft. The general range is however, from $25 to $65/ft. In order for a new system to be competitive it must be below this cost. An economic evaluation of the replacement costs for a cable follower was made. This is a simple model and an attempt was made to make the cost estimate conservative.

In order to develop the economics for the system it is necessary to make several assumptions on the operation of the equipment and the cost of money. Based on these assumptions a sensitivity study can be made. For the purpose of this analysis we assume:

The total system cost is $100,000. The cost of capital is 17%. Maintenance, parts and fuel total $20/hr. The cost of two operators is $40/hr. With these assumptions and asssuming a 3-year depreciation, it is possible to determine the hourly operating cost based on the hours of operation per year. This is shown graphically in Figure 11. The hourly cost is based on the capital cost per hour which is the dashed curve. This is a function of the machine usage per year. The maintenance and labor cost are a fixed rate. The resultant curve of the cost components shows the cost to be a strong function of usage up to the 1000-hr per year level. Above this level of usage the fixed operating costs dominate. At a 1000-hr utilization which is approximately 4 hr/ day utilization the cost of the equipment per hour is approximately $100/hr. Once the operating cost has been determined, it is possible to determine the cost per foot based on an average advance rate. This is shown graphically in Figure 12.

It can be seen that at an average advance rate of 2 ft/min, the cost per foot to remove a failed cable is on the order of one dollar ($1.00). This is only a portion of the total cost of replacement. The replacement cost involves the cost of excavating the starting and exit pit, the cost of restoring the surface, the cost of removing the spoils and the cost of replacement cable. If we assume the cost of excavation of the pits, the cost to dig the tool up once per 200 ft and the cost to restore the hole is $1000 ($5/ft for 200-ft run) and the cost for replacement cable is $3/ft and add a 10% contingency the total cost for the replacement is $10/ft. This represents a significant improvement over conventional methods.

This analysis demonstrates the commercial viability of the cable follower concept. It will be necessary to conduct tests with the system to validate the economics of the system and to determine what problems still have to be solved to make this a viable tool.

ACKNOWLEDGEMENTS

The authors wish to gratefully acknowledge the support of EPRI and especially the helpful ideas and insights provided by Mr. Tom Kendrew, EPRI Project Manager, and Mr. Bill Shula, Project Manager, Electric Systems Division, during the course of this work.

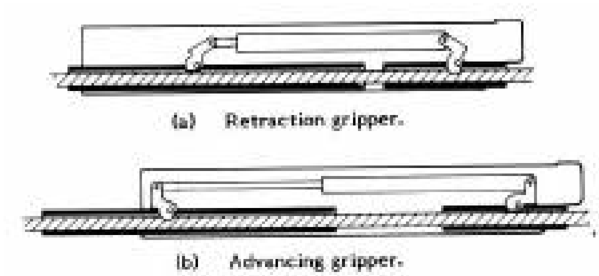

Figure 1. Cable follower advancing cycle.

Figure 2. Trailer mounted system.

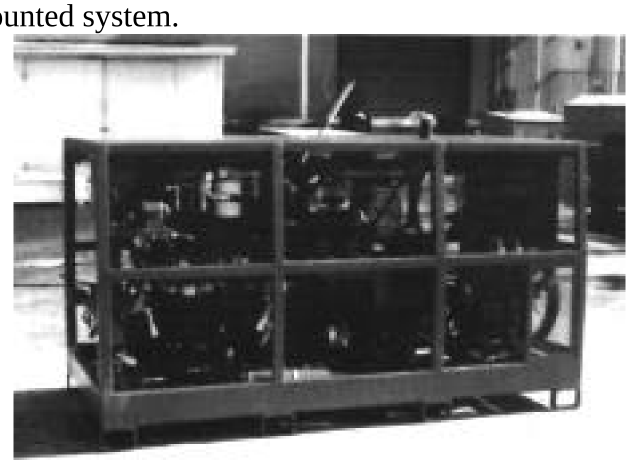

Figure 3. Power unit for high pressure fluid and tool advance.

Figure 4. System field test set-up

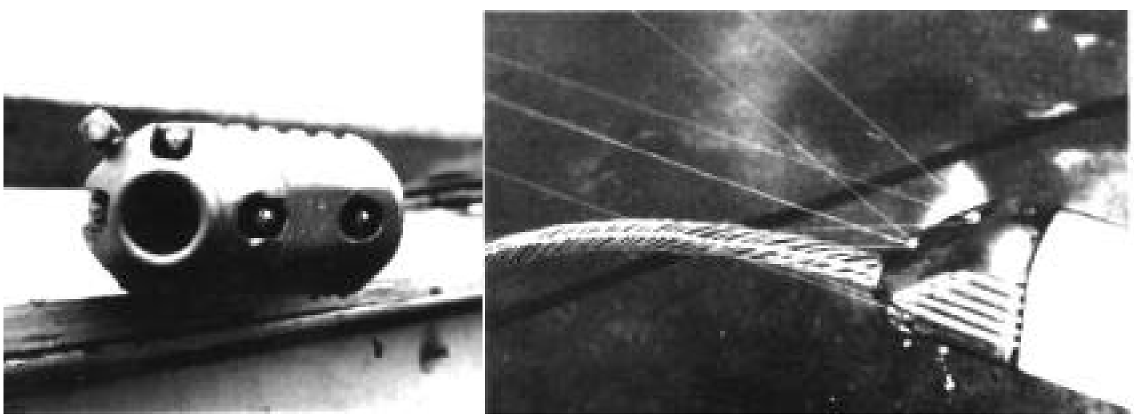

Figure 5. Original nozzle configuration for sandy soil.

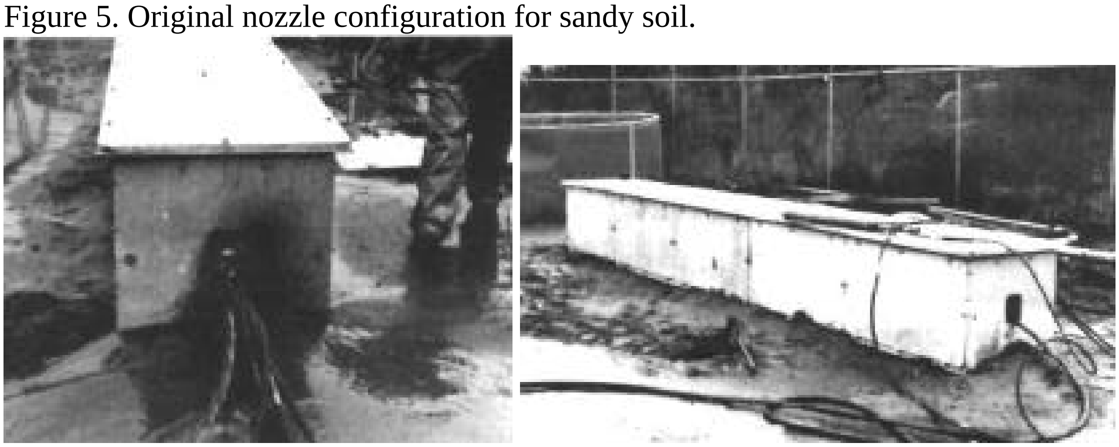

Figure 6. Laboratory test facility.

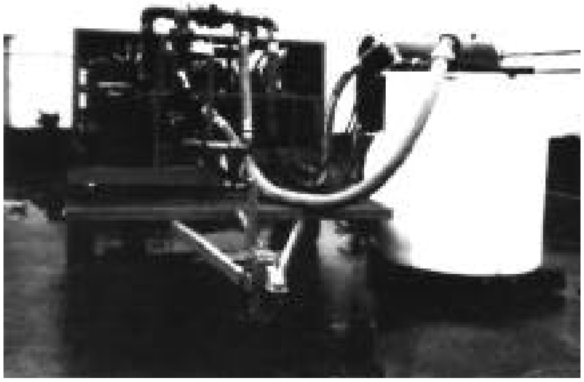

Figure 7. Mud mixing and filtering system.

Figure 8. High concentration of nozzles in test heads.

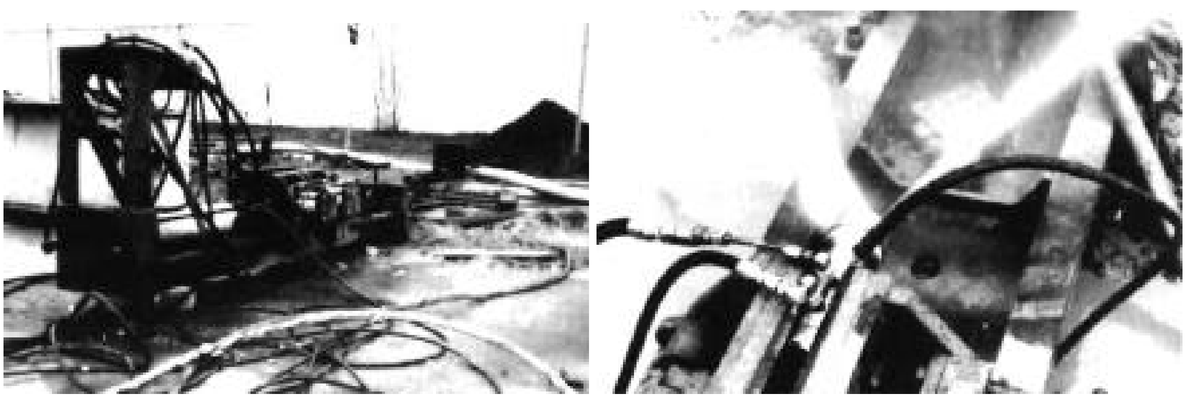

Figure 9. Rotating jet test set-up

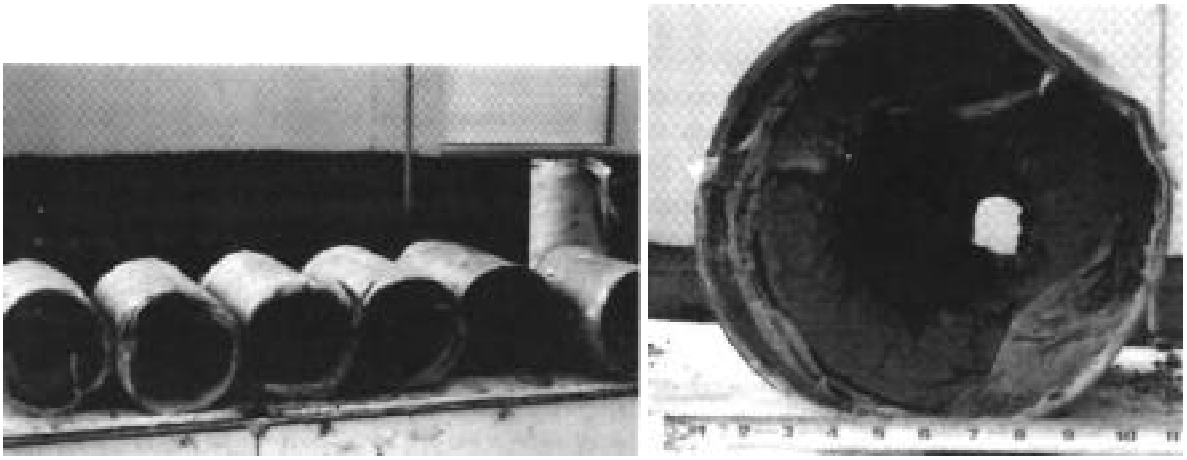

Figure 10. Bentonite drilled clay samples

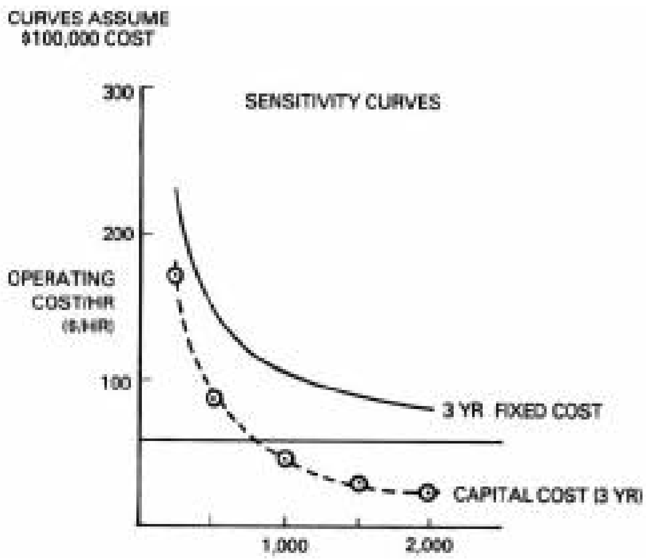

Figure 11. System hourly operating cost as a function of usage.

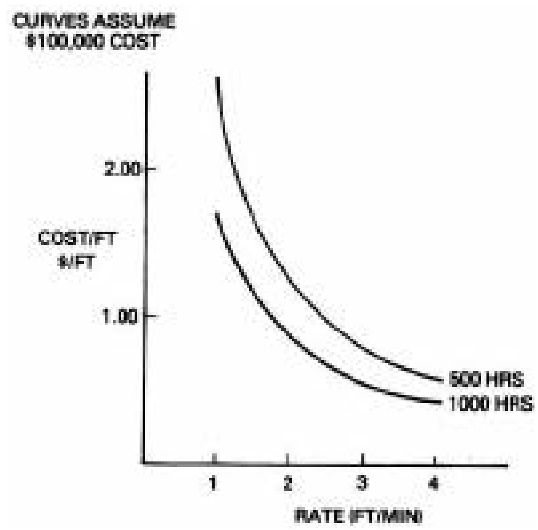

Figure12. System operating cost on a per foot basis.

WATER JET ASSISTED MINING TOOLS: WHAT TYPE ASSISTANCE AND WHAT TYPE MINING MACHINE?

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ABSTRACT

Water jet assisted mining tools have been discussed in the literature for many years, but virtually no discussion exists of the specific modes of assistance and the benefits to be derived from each. This paper will identify two modes of assistance and show the effects that each has on the specific energy related to rock cutting. The cutting mechanism of a mining tool without water jet assistance will be examined briefly and its effect on tool spacing on a mining machine developed. Next the effects of kerfing (cutting thin slots) on specific energy will be examined and a technique for the optimization of kerf spacing and kerf depth as a function of depth-of-tool-cut will be presented. A discussion of the stable crack extension mode will be presented and its effectiveness shown. Finally the requirements for a mining machine which could take maximum advantage of these assistance modes will be presented.

INTRODUCTION

Water jets, because of the great range of operational parameters, are used for many tasks. One use of water jets is to cut rock or to assist mechanical tools used for cutting rock. Two mechanisms by which water jets create a failure in the rock are: (I) erosion of the rock by removing or weakening the material cementing the grains together and removing the grains with hydrodynamic drag forces (Rehbinder, 1980; Crow, 1973) and (2) hydraulic crack extension (Hood, 1977).

The efficiency of these water jet mechanisms depends on several factors:
(I) water pressure at the nozzle,
(II) nozzle diameter,
(III) traverse rate, and
(IV) rock properties.

When used in conjunction with mechanical cutting tools, water jets also improve the cutting efficiency by lubricating the cutting surface and by removing rock chips.

This paper is concerned with the overall efficiency of the mining process. The overall efficiency is affected by the individual efficiencies of the cutting devices. Therefore, the individual processes and their efficiencies will be examined. The results will then be applied to describe the characteristics of a mining machine which can best utilize the most efficient mechanisms and characteristics. This work is supported by the U. S. Bureau of Mines, Pittsburgh Research Center.

MECHANICAL TOOLS

Cutting Tool Energetics Without Kerfs

The most efficient placement of tools on mining machines requires that the specific energy be minimized. The specific energy is the energy required to remove a unit volume of material. Several factors influence the specific energy, E_v, required for cutting.

N	= Number of cutters
F_c	= Cutting or drag force (tool and rock dependent)
F_n	= Sumping or shearing (normal) force (tool and rock dependent)
L_c	= Distance along tool path
L_n	= Sumping or shearing distance
d_c	= Depth-of-cut
W	= Width of tool
θ	= Break-out angle (rock dependent)
S	= Tool-to-tool spacing
K_t	= Tool constant

The specific energy for a single tool, ignoring supping or shearing, is given by

$$E_v = \frac{F_c}{\left(W \ d_c + d_c^2 \ \tan\theta\right)}$$

F_c is linearly proportional to the depth-of-cut, d_c. The break-out angle, θ, is independent of d_c. A tool constant may be defined as follows:

$$K_t=F/(d_c{*}W)$$

where

$$F = \sqrt{F_c^2 + F_n^2}$$

K_t has units of force/area. When N is more than one and S is less than 2*dc*Tan θ, the cutting tools interact and the volume of material removed for N cutters at spacing S is

$$Vol = L \left\{ N \ W \ d_c + d_c^2 \ \tan\theta + \left(N-1\right) \left[d_c^2 \left(S - \frac{S^2}{4\tan\theta} \right) \right] \right\}$$

Where

$$L = \sqrt{L_c^2 + L_n^2}$$

For tool configurations other than abreast (e.g., an echelon) optimum spacing (neglecting stress concentrations) occurs when

$$S = \sqrt{\frac{A^2}{B^2} + \frac{(A\ D - B\ C)}{B\ E}} - \frac{A}{B}$$

where

A=K_t*W*d_c+(N-1)*K_t*W*d_c/2

B= (N-1)*K_t*W/(4*Tanθ)

C=N*W*d_c+(d_c^2)*Tanθ

D=(N-1)*d_c

E=(N-1)/(4*Tanθ)

These results are for "first pass" cuts on competent rock (no damage from previous cuts). (See Ford, Longfellow & Friedman, 1983, for other configurations and "subsequent pass" equations.)

<u>Cutting Tool Energetics With Kerfs</u>

Tool forces and specific energy are reduced if kerfs (thin slots) are cut between tools. Ford et al (1983) derived the following expression for the optimum efficiency center-to-center spacing and depth of the kerf

S=2*R*Sinψ+W+w

D_k=R*Cosψ+d_c

where

$$R = r\sqrt{k}$$

r = d_c/Cos θ, the break-out radius without kerfs

k = stress concentration factor at bottom of kerf (approximately 2 to 10)

w=kerf width

$$\Psi = \cos^{-1}\left\{ \frac{\sqrt{\left[-\frac{d_c}{R}\right] + \left[\frac{d_c}{R}\right]^2 + 2)}}{2} \right\}$$ for maximum volume

or

$$\Psi = \cos^{-1}\left\{ \frac{\sqrt{\left[-\frac{d_c}{R}\right] - \left[\frac{d_c}{R}\right]^2 + 2}}{2} \right\} \text{ (for minimum volume)}$$

The minimum specific energy for the "first pass," Ev, is:

Ev={p*Kt*W*dc/2}/{[W*dc+(R^2*Sin*ψ)/2+R*dc*Sinψ}+C/w

where

p = ratio of cutting force with kerfs to cutting force without kerfs, typically about 0.4 to 0.6 (Ford et al, 1983)

C = Water jet energy per depth-of-cut

Because of the stress concentration at the bottom of the kerf the rock will fracture from the bottom corner of the tool to the bottom of the kerf. This results in an "inverted breakout angle" (Figure 1). This could represent up to 75% reduction in specific energy, depending on the efficiency of the kerf cutting operation. These equations hold for "first pass" cuts only; work is progressing on the "subsequent pass" case(s) but was not ready for publication at this time.

<u>Discussion of Tool Energetics</u>

Results of this study indicate that tool spacing, specific energy, and therefore cutting efficiency, are very sensitive to depth-of-cut. Deep tool cuts produce lower specific energy. These results are believed to be conservative since they do not include the effects of stress concentration or stress field interactions produced with multiple tools. If water jets are used to cut the kerfs cooling and lubrication should further improve these results. The water also lowers the respirable dust concentrations produced and reduces the probability of methane ignition when methane is present. When cutters are placed on rotating drum machinery such as continuous miners, alpine miners, and longwall and shortwall machines, the efficiency suffers because the cutters are usually making shallow entry or exit cuts. This continuous variation in cutting depth on rotating drum machines limits the gains in efficiency which can be attained when cutter spacing is optimized.

Much larger gains in efficiency are possible if a low energy kerf cutting system is employed in addition to the mechanical cutting tools. Because of the problems with rotating drum machines the greatest gains in efficiency will be achieved with kerfs between tools making deep linear cuts.

WATER JET KERFING

Water jets used in rock kerfing are not required to remove large amounts of material; the goal is to cut as narrow a slot as possible. This allows small nozzle diameters (within the constraints of other factors, such as clogging), low flow rates (which are highly desirable in most mining situations), and low power pumps (within the constraints of the pressure requirements). The effect of the traverse velocity and the required depth-of-cut then allow a trade-off to be made which will minimize the specific energy. The specific energy for a kerf cut by a water jet is given by:

$$E_v = \frac{\pi \ \rho \ a^2 \ V^3}{\left(8V_t \ w \ D_k \ g_c\right)}$$

where

ρ	= density of fluid
a	= nozzle diameter
V	= jet velocity
P	= pressure
V_t	= traverse velocity
D_k	= depth-of-kerf-cut
g_c	= gravitational acceleration
w	= width of slot (approximately 2 or 3 times a)

The volumetric flow rate, Q. is:

$$Q = C_d*V*A$$

where

A	= nozzle area
C_d	= discharge coefficient (approximately O.7)

The pump power is

$$H=Q*P.$$

As the kerf depth, D_k, increases, the required specific energy also increases. Apparently, viscous and frictional losses reduce the cutting efficiency as the depth-of-cut increases. It is extremely difficult to track a narrow slot on a vibrating, rotating drum to achieve a deeper kerf by using multiple passes. The solution appears to be to use multiple nozzles which are fixed together at a specific spacing and which are traversed simultaneously. This gives a result somewhat like the multiple pass (depending on the spacing) and yet allows relatively high traverse rates and tool speeds such as those associated with continuous miners.

WATER JET CRACK EXTENSION

Water jets may also be used in conjunction with mechanical tools by employing them slightly ahead of the tool (and aimed in the approximate direction of cut) to assist in forming chips. Hydraulic pressure entering the fracture can, if the pressure is high enough, cause the fracture to propagate. Factors which affect this mechanism are: (1) width of the existing fracture opening, (2) length of the existing fracture, and (3) roughness of the fracture surfaces. These determine, for a fixed jet placement, how much water enters the fracture and its dynamic pressure distribution within the fracture. This mode also contributes to chip flushing, i.e., removing the chips from the tool path, thus reducing the cutting force by eliminating a large portion of the material which would otherwise be reground. The water ahead of the tool also lubricates the tool and cools it, thus reducing wear and is also useful for suppressing dust and reducing methane ignitions. This mode of operation has been shown (Hood, 1977) to reduce cutting forces by approximately 30% and the normal force by about 50%. In general, the water pressures required for this mode of assistance are lower than those required for kerfing.

MINING MACHINE REQUIREMENTS

Reduction of the specific energy involved in the coal mining process is the key to more efficient mining, longer lasting tools, reduced dust production and a lower probability of methane ignition. Cutting kerfs of proper spacing and depth between tools has been shown (Ford, Longfellow and Friedman, 1983) to achieve a dramatic reduction in specific energy. These kerfs may be cut efficiently using relatively low (for water jets) pressure (8,000 to 10,000 psi). The mechanical tool specific energy has been shown to be sensitive to depth-of-cut; deeper cuts require less energy. Water jet spacing for optimum kerfs is also a function of the tool depth-of-cut. Rotating drum equipment causes shallow entry and/ or exit cuts, thus reducing the efficiency of such machines over a large percentage of the tool path. Also, water jets on rotating drum equipment require a high pressure rotating seal; these are usually prone to failure. Mechanisms are needed on rotating drum equipment to "turn on" and "turn off" the jets when they are not in a cutting position. These problem areas are eliminated with a linear cut machine.

Water jets cut deeper at low traverse velocities; however, they require more specific energy for deeper cuts because of frictional and viscous energy losses. A compromise traverse speed must be chosen to achieve the optimum depth of cut and minimize the specific energy. By using small diameter nozzles the amount of material removed by the water jets may be minimized; this reduces the ratio of water jet to mechanical energy. Multiple passes with very small nozzles probably won't cut in the same track. The design should be for a single pass kerf cut, possibly with multiple nozzles. Wedge shaped longwall cutters (shearers) have been used in tests (Ford, Longfellow and Friedman, 1983) to produce the "inverted break-out angle" and would function well on a linear cut machine without producing large amounts of rebroken coal. Water jets of lower pressure (3,000 to 5,000 psi) could be used just ahead of these tools to enhance fracture, chip formation, lubrication, dust suppression, and chip flushing.

From an energetics standpoint, hybrid systems look very promising. The coupling of conventional mechanical tools and water jets offers substantial gains in efficiency with improvements in production and increases in health and safety. Thus, from these considerations, the most promising mining machine may be a deep, linear cutting machine operating at moderate speed using optimally spaced low pressure water jets to cut kerfs on each side of longwall shearers which have lower pressure jets just ahead of each tool.

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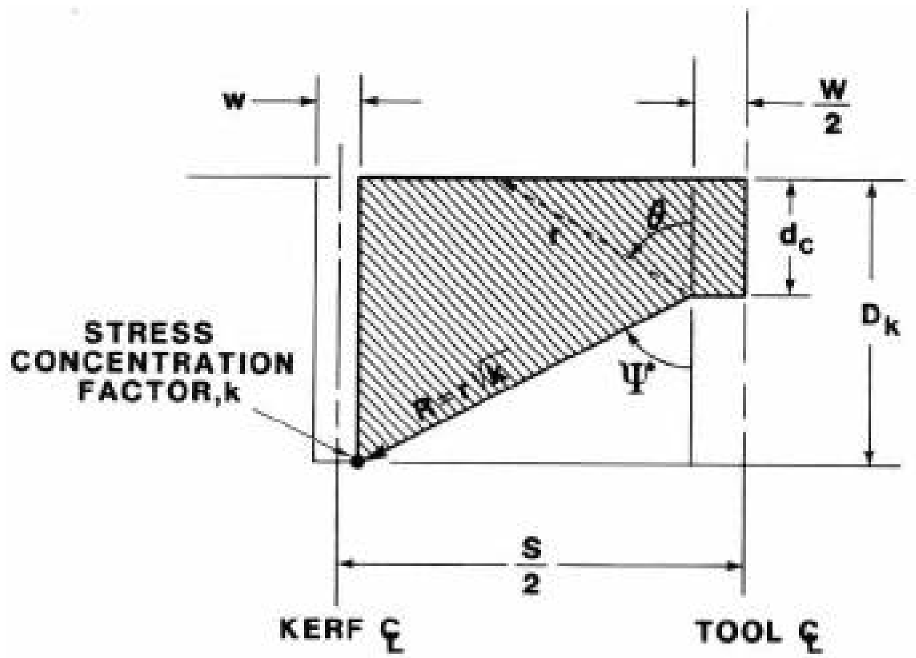

Figure 1. Tool – kerf – spacing geometry for producing the inverted break-out angle Ψ.

DISCUSSION

NAME: Simon Johnson
COMPANY: Newcastle University

QUESTION: "What was the material used in the test, and what are the relevant material properties - namely - compressive strength, tensile strength, shale hardness, abrasivity?"

ANSWER: The material used in our tests was Tennessee Sandstone, sometimes known as Crab Orchard Sandstone. The compressive strength ranged from 21,0()0 psi to 23,000 psi (145 to 159 MPa); the tensile strength ranged from 800 to 1150 psi (5.5 to 7.9 MPa); the porosity was from 5 to 8 percent; the permeability was from 0.06 to 0.10 md. No additional mechanical properties data were measured.

NAME: Tom Brunsing
COMPANY: Foster Miller

QUESTION: "What percent of energy in the process was in the jetted notch?"

ANSWER: Typically, for a small diameter nozzle diameter, and relatively low pressure jet, the percentage of specific energy used to cut the kerfs (or the notch) is between 2 and 10 percent.

NAME: George Savanick
COMPANY: Bureau of Mines, Twin Cities Research Center

QUESTION: "Do rotary couplers exist appropriate to feeding water to the water jet augmentation nozzles on mining machines?"

ANSWER: I'm not an expert on rotary couplers, but I'm sure that they exist, at least on some tunnel boring machines. They are, from what I am able to find in the literature, a definite problem area, at least from a reliability standpoint. This is one of the reasons why I feel a linear cutting system is to be preferred

NAME: Michael Hood
COMPANY: University of California, Berkeley

QUESTION "Did the specific energy figures quoted include the energy consumed by the water jet?"

ANSWER : The specific energy value quoted did include the water jet energy. By using very small nozzles and low to medium pressure, the energy and flow rate are minimized. The goal is to remove as little material as possible, thus creating a higher stress concentration at the base of the kerf with as little energy and water consumption as possible.

HIGH-PRESSURE WATER JET-ASSISTED TUNNELING TECHNIQUES

Dipl.-Ing. Wilhelm Knickmeyer, Essen, Bergbau-Forschung GmbH, Federal Republic of Germany
Dipl.-Ing. Lothar Baumann, Langenberg, Bergbau-Forschung GmbH, Federal Republic of Germany

INTRODUCTION

Some 500 km of underground roadways are driven in the Federal Republic's hard coal industry on an annual average. Of these 500 km, 27% are driven mechanically, but only about 2% are realized by means of full-face tunneling machines. Figure 1 shows full-face tunneling machines with cutter diameters of approximately 6.0 m. Such heading systems have a length of about 200 m. They require high capital costs and considerable expense for underground transport and assembling before and after operation. A minimum drivage distance of 3,000 - 4,000 m is necessary to allow for an economical drivage of hard headings with such full-face tunneling machines. This is the reason for their so far strongly limited use.

As the majority of hard headings to be drifted in the German hard coal industry is characterized by only short lengths up to 2 km, current development efforts center on the expansion of the application range of mechanical heading systems. The construction of light-weight mobile heading systems is to allow for the economical drivage of short, hard headings too. To this purpose, Bergbau-Forschung GmbH, in cooperation with the mines and the mining suppliers, is conducting in-depth tests with high-pressure water jet-assisted conventional cutting tools.

These research efforts are supported by the Minister of Economy, Small-Scale Industries and Transport of the Land North Rhine-Westphalia.

HEADING TECHNIQUES

Mechanical rock cutting by means of picks and roller tools is approaching its technological limit due to tool wear, limited energy transfer, high forces of thrust and torques, and consequently due to the heavy machines of relatively limited heading performance and mobility. Both picks and roller tools incorporate extremely sophisticated mechanized technologies which have been used for years. Regarding the long period of development and the high technical standard of conventional methods any related substantial improvements can no longer be expected.

As a result of this situation, new techniques for rock breaking without any explosives were developed and tested early in the 170's, aiming at an essential reduction of the high thrust forces of statically acting cutting tools and thus the construction of light-weight and mobile machines (Fig. 2).

One of the most promising methods is a combination of high-pressure water jets with roller tools or carbide picks, respectively, which have good prospects for medium-term use in tunneling.

ROLLER TOOLS COMBINED WITH HIGH-PRESSURE WATER JETS

Previous studies on the application of high-pressure water jets on full-face tunneling machines ranged from extensive basic studies up to technical scale testing in a quarry.

These tests were carried out with a TB-I-260 tunneling machine of Messrs. Wirth Maschinen- und Bohrgerate-Fabrik GmbH, having a cutter diameter of 2.6 m, in a upper carboniferous sandstone quarry near Dortmund-Hohensyburg (Fig. 3). They were conducted in cooperation with Messrs. Wirth in the context of a cooperation agreement with the U.S. Bureau of Mines (today's Department of Energy) which supplied parts of the high-pressure water system on a loan basis.

The principle of a combined water jet/roller tool system for rock destruction consists in cutting kerfs and thus exposed areas into the rock on both sides of the cutting tool by means of water jets, with the roller tool subsequently shearing off the rock rib left in-between the kerfs (Fig. 4). According to the comprehensive strength/tensile strength ratio of the rock, the static forces required for the operation of the roller tools can be reduced as a result of the modified stress acting on the rock when destroyed by shearing action as compared with a purely mechanical pushing/pressing rock destruction. A possible reduction of the machine thrust required and thus of the machine weight in the order of 50% against a purely mechanical operation was taken as a basis for planning the tests.

The tests conducted fully confirmed the initial assumptions. The reduction of the necessary thrust forces by additional high-pressure water jets compared with purely mechanical tools in the order of 50% (Fig. 5) is in line with the objective and can be expected to allow a corresponding reduction of the weight of such roadheaders with all the resulting advantages regarding the mobility of such a heading system.

This positive influence of high-pressure water jets can also be utilized for an increase in the rate of machine advance. The results achieved in this respect were even more favorable as the rate of advance, with unchanged thrust, could immediately be doubled using high-pressure water jets (Fig. 6).

As a next step in this field, high-pressure water jets are to be used underground on a Mannesmann-Demag AG full-face tunneling machine in 1983. The main objective of this test is the limitation of the energy requirement by selective application of high-pressure water jets within the center of the cutter head and in the caliber section. It must be noted that peak loads and extreme wear were observed within these sections. The high-pressure nozzles are arranged in four groups on the cutter head in such a way that various combinations of roller tools and high-pressure nozzles are possible (Fig. 7).

Apart from confirming the results achieved underground so far, the proposed heading tests are to furnish new findings, particularly regarding the effects of the high-pressure water jet-assisted cutting technique on tool wear and mine climate.

CUTTING OF THE ROADWAY PROFILE

The roadway profile cutting system developed by Bergbau-Forschung GmbH together with the Bochumer Eisenhutte Heintzmann GmbH & Company KG and Ruhrkohle AG has already reached the stage of underground testing (Fig. 8). Apart from its advantages regarding supports requirement and rock stability, this system ensures high profile accuracy. The tool consists of carbide cutting tips and high-pressure nozzles, i.e., 2 leading and 3 trailing nozzles (Fig. 9).

The tests were successfully concluded both in solid sandstone and in softer stratified rock as well as in the coal seam/rock interface area. They illustrated that the forces acting upon the carbide picks can be reduced on average by 50% by means of "forced" high-Pressure water jets. Furthermore, the tests showed that, by use of high-pressure water jets, the pick life can be substantially extended, thus yielding a new application potential for carbide cutting tools in breaking solid rock (Fig. 10). Work is currently under way aimed at the development of a tool continually cutting a roadway profile us to 1 m deep before extracting the remaining roadway core (Fig. 11).

In parallel, a profile cutting system, working with high-pressure nozzles alone, is being developed in cooperation with the U.S. Department of Energy. The cutting tool comprises two rotating nozzle heads provided with two nozzles each (Fig. 12). It was possible to cut an accurately delimited slot, roughly 70 mm wide, into a hard and highly abrasive Ruhr sandstone without any thrust. As the test installation served only as a tool guide, it could be designed as a very light unit (Fig. 13).

For reasons of priority, work in this field was discontinued for the time being since the cutting performance achieved in the tests conducted so far does not yet satisfy the requirements.

WATER BLAST UNIT

In the context of core recovery during roadway profile cutting, a novel device is being developed which uses a high-pressurized water projectile instead of explosives for rock destruction.

First tests were carried out with water volumes of 50 ml projected at 6,000 bar max. with velocities of some 1,000 m/sec through a nozzle onto the rock surface by ignition of a propellent charge (Fig. 14).

The destruction effect can be improved substantially by directing the water jet into a borehole.

Compared to conventional shotfiring, waiting time is eliminated since there is no risk of blasted-off mineral (Fig. 15).

Small dimensions, light-weight design and no need for supply equipment ensure high mobility and a broad range of application.

Various designs and applications are feasible, e.g., for cleaning work and for the destruction of connecting elements during salvage work in the face/ gate road intersection (Fig. 16).

SUPPLY BOREHOLES

A further task results from the need for coal gasification underground. In order to ensure the recovery of deposits in great depths which poses numerous problems to conventional mining, a concept of borehole sinking for the supply of gasification agents was developed. The high-pressure water is supplied via a steel tube fed from a tube reel via a guide unit to advance the nozzle ahead (Fig. 17). The borehole diameter is 80-100 mm, the desired borehole length some hundred meters.

MINI-FULLFACER WITH HIGH-PRESSURE WATER JETS

The so-called rock back-cutting principle was again adopted for mechanical roadway drivage. The tests carried out in the hard coal industry more than 20 years ago with the Wolmeyer roadheader have gained renewed importance as a result of the new application potential of high-pressure water jets (Fig. 18).

In the context of a cooperation agreement with Messrs. Atlas-Copco Sweden, who made available, free of charge, a roadheader of the Mini-Fullfacer type, first preliminary tests have been started in the Herdecke quarry.

As with profile cutting, the standard cutting pick is equipped with high-pressure nozzles to extend the application range of this roadheader to hard and abrasive rock (Fig. 19).

In the meantime, a roadway length of 20 m has been drifted, initially without high-pressure water. In sandstone, the back-cutting depth is 100 mm and more with a cutter width of only 35 mm (Fig. 20).

The wedge-shaped cutter pick breaks the back-cut rock rib. This results in a very favorable breaking performance/cutting work ratio.

LICENSING PROCEDURE AND SAFETY AT WORK

Both commercial use and testing of roadheaders as well as testing installations require the official approval in the Federal Republic of Germany. All underground cavities with a cross-section of more than 8 m^2 extracted without direct access from the surface are subject to Para. 130 of the Mining Law. Besides, the mutual indemnity association of the mining industry and the trade supervisory office are involved in the licensing procedure as official authorities.

The risks associated with the use of high-pressure water jets essentially relate to the high pressure and the noise level. Water pressures are up to 4,000 bar, noise levels up to 97 dB(A) when cutting rock and 122 dB(A) for open jets. The essentially higher noise level for open jets can largely be prevented during practical application of high-pressure water.

To reduce any related effects, the following personal protective measures are made compulsory, apart from technical requirements for machinery (Fig. 21):

- reinforced rubber boots
- protective suit against splash water
- protective gloves
- helmet with eye protection
- noise protection
- respiratory protection (for use of additives).

EVALUATION

Summarizing the results and experience made so far with the novel heading systems, the potential advantages of a technical-economical and reasonable application in the hard coal industry may be evaluated as follows:

The applicability of the techniques studied could basically be confirmed. As to conventional operation of cutting tools in particular, the high cutting force components required could substantially be reduced, or the rate of advance could alternatively be increased using the same thrust. Besides, there were both positive effects on tool wear and important economic advantages.

The tests have also shown, however, that the energy requirement is relatively high. A reduction of the energy requirement could, for example, be achieved by selective application of the high-pressure water jets in particular areas, such as at the periphery and in the center of the roadway.

High-pressure water jets combined with carbide picks have proved to be highly effective. In contrast, rock destruction by water jets alone has an extremely high specific energy requirement since the water jet, as an excellent means of cutting, is not suited to extract larger rock volumes.

Further tests were conducted to improve the effect of high-pressure water jets by additives. The results have confirmed that, by using additives in the high-pressure water, the slot depth can be increased by up to 100% (Fig. 22), or that the pressure can be reduced by 30% for the same slot depth.

Most of the above-mentioned development work of Bergbau-Forschung GmbH is still in the laboratory stage. Based on the results achieved, however, field tests on large technical scale could already be carried out which will, in one or another case, certainly lead to a further improvement of roadheading techniques.

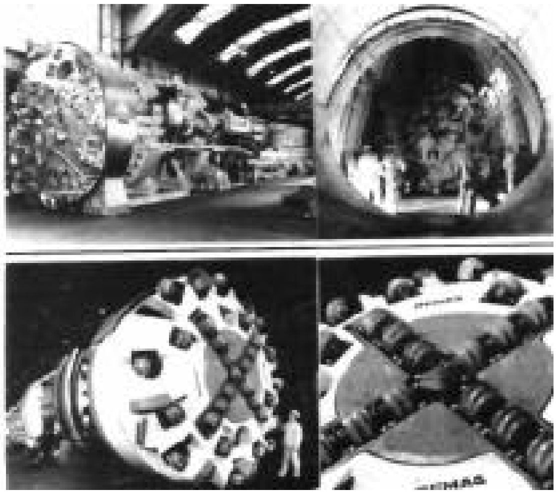

Figure 1. Full-face tunneling machines.

Figure 2. Objectives

Figure 3. Tunneling machine for roadheading tests with high-pressure water jets.

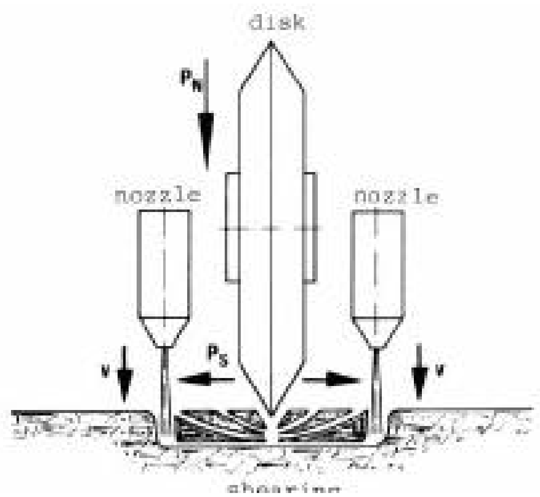

Figure 4. Combined cutting system.

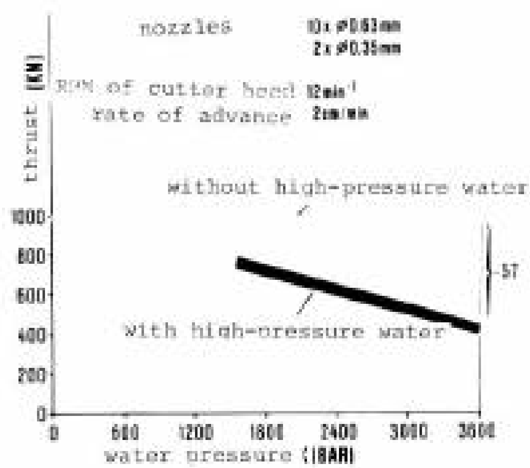

Figure 5. Test results

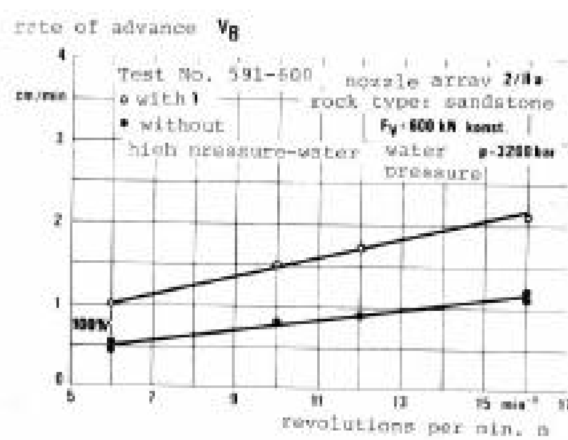

Figure 6. Influence of high-pressure water on rate of advance.

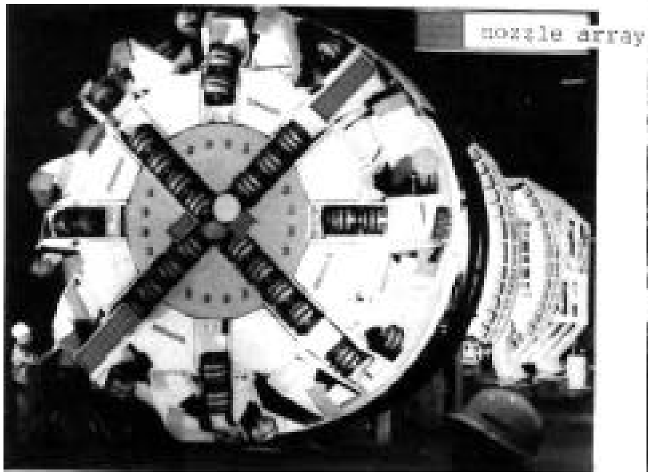

Figure 7 High-pressure jet assisted roadheader.

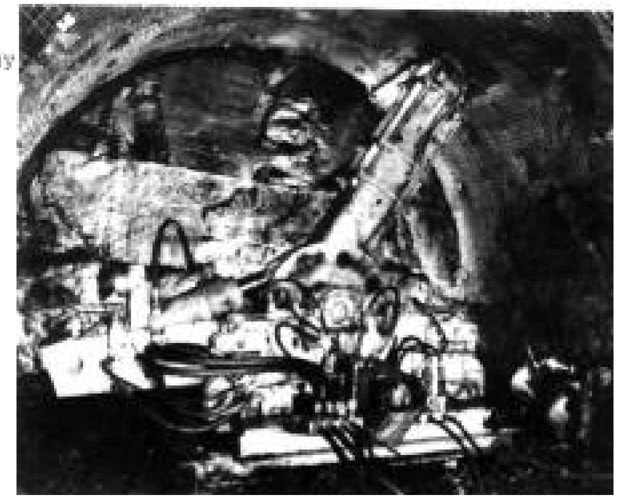

Figure 8. Roadway cutting profile at Rossenray colliery.

Figure 9. Cutting arm equipped with hard-metal tools and high-pressure nozzles.

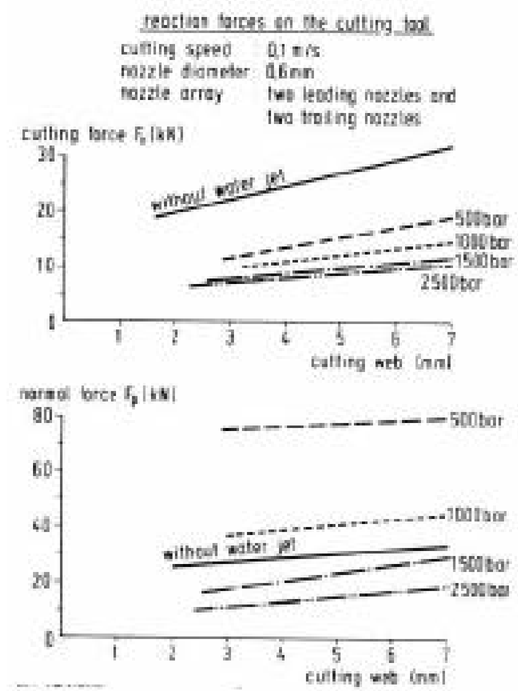

Figure 10. Test results

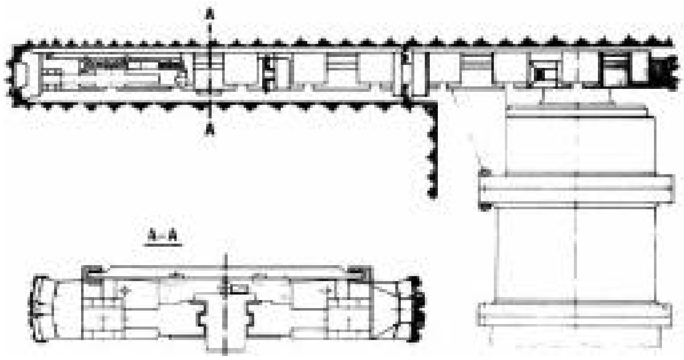

Figure 11. roadway profile cutting assisted with High-pressure water jets.

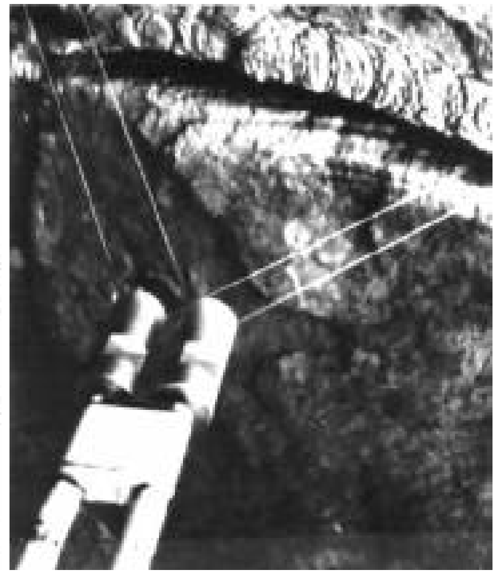

Figure 12. Rotating cutting nozzles.

Figure 13. Road profile cutting by high-pressure water jet only.

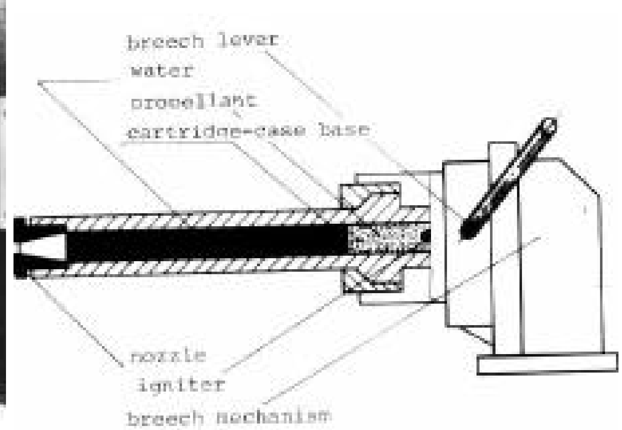

Figure 14. Water blast unit.

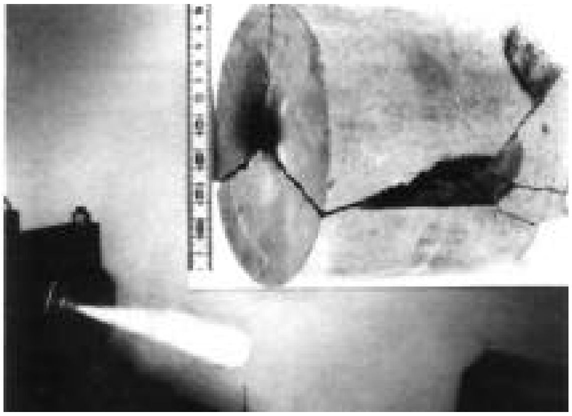

Figure 15. Rock destruction by pulse water jet.

Figure 16. Prototype of water blast unit.

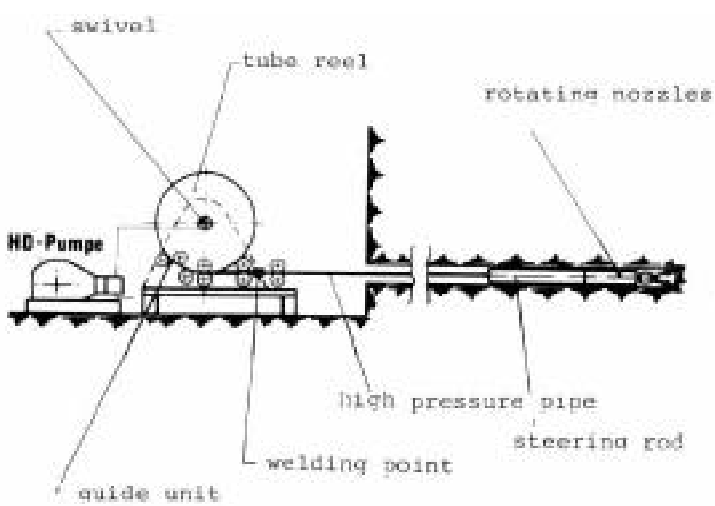

Figure 17. Horizontal boring concept

Figure 18. Mini-Full-facer, roadway profile.

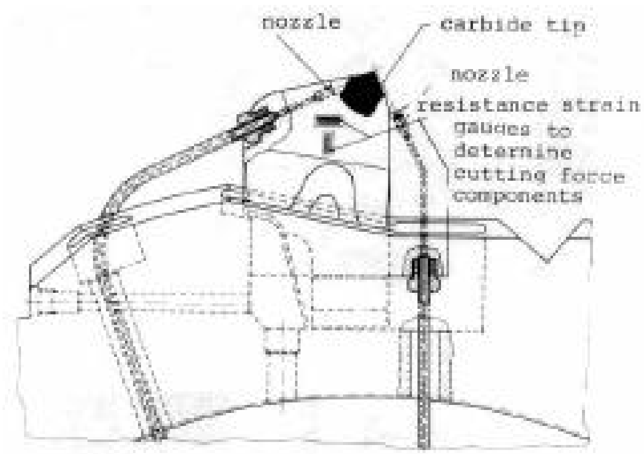

Figure 19. High-pressure water jet – assisted cutting tool.

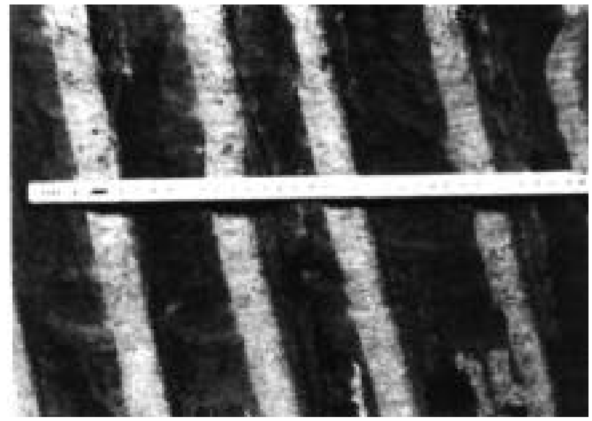

Figure 20. Mini-fullfacer roadway wall -. Sandstone.

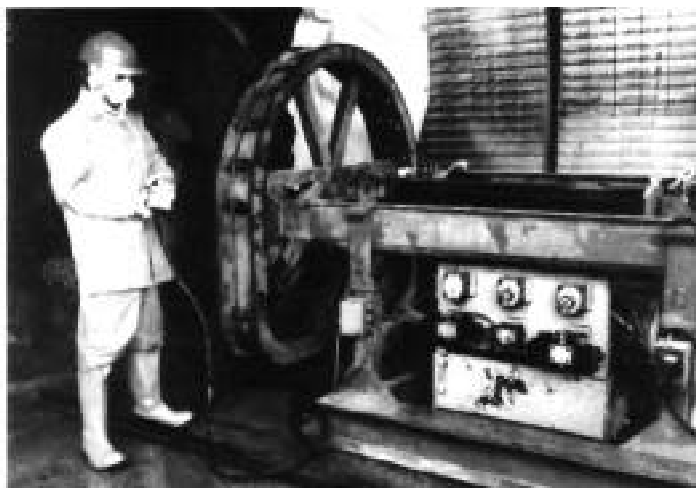

Figure 21. Protective clothing for use with high-pressure water jet application.

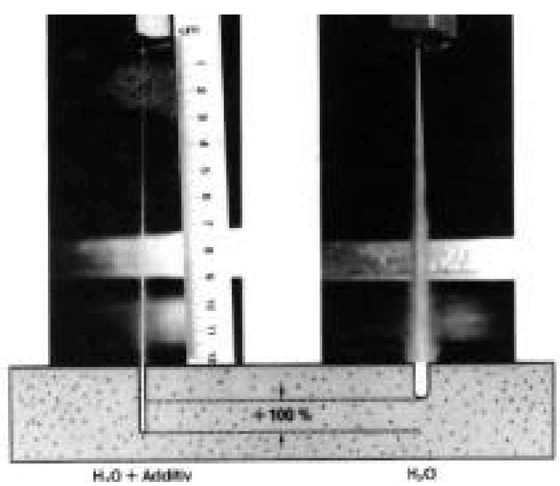

Figure 22. Improved water jet effect by use of additives.

DISCUSSION

NAME: R. Pootmans
COMPANY: Indescor

QUESTION: "What was the gallonage of the jets at 16,000 psi?"

ANSWER: Subject to the number of nozzles, nozzle diameter and water pressure we applied up to max. 60 1 water/min during the trials.

NAME: Michael Hood
COMPANY: University of California, Berkeley

QUESTION: "This question relates to the profile cutting machine. Could you describe how the machine is advanced into the face when the bit is embedded in a slot?"

ANSWER: The advance of the cutting tool in forward direction is carried out by a hydraulic feed cylinder.

NAME: Dr. M. Vijay
COMPANY: National Research Council

QUESTION: "Why are you using two rotating nozzles at the same time for making slots? The first rotating nozzle does all the cutting. What does the second one do?"

ANSWER: Our trials were carried out in a dense and abrasive sandstone. The application of two nozzles in coincidence served - in this case - for an increase of the cutting rate, and for a better evacuation of the crests remaining between the slots cut.

DESIGN AND OPERATION OF TWO LARGE-SCALF LABORATORY ROCK DRILLING AND BORING EQUIPMENT FOR WATER-JET ROCK CUTTING RESEARCH

Levent Ozdemir, Director
Russell Miller, Associate Research Professor
William Sharp, Associate Research Professor
Earth Mechanics Institute
Colorado School of Mines, Golden, Colorado 80401

ABSTRACT

This paper describes the design, operation and capabilities of two large-scale laboratory rock drilling and boring equipment for conducting research with mechanical and water-jet assisted cutting tools. The first machine is a laboratory drill rig capable of drilling holes up to 3-ft. in diameter. The machine is all hydraulic drive, allowing a wide range of thrust and speed control. This equipment is currently fitted with a 28-in. diameter Dosco roadheader cutting head which will be used to perform water-jet assisted drag bit cutting tests in oil shale and coal measure rocks. The second equipment is a rotary cutting machine capable of accommodating cutter heads up to 8 ft. in diameter. This machine is pivoted on a support frame to permit tilting into any direction to simulate tunnel, raise or shaft boring conditions. It is operated through total electronic servo-control using a PDP-1103 computer. The computer and associated data acquisition system can monitor up to 32 channels of data including 16 channels of cutter force data transmitted via wireless telemetry from triaxial load cells installed under cutter saddles. This machine also has the capability of being equipped with low or high pressure water-jets to evaluate the performance of hybrid water-jet mechanical cutting systems.

INTRODUCTION

Significant advancements have been realized in underground excavation technology in recent years. Major design improvements coupled with a better understanding of rock fragmentation processes have resulted in more effective application of mechanized rock boring techniques. Today, mechanical tunnel, raise and shaft boring machines are being utilized in ever increasing numbers in nearly every aspect of underground construction field. Under favorable conditions, these machines have been proven to offer spectacular advance rates at lower costs compared to conventional excavation methods.

One major obstacle to manufacturers achieving even further design and machine performance improvements is the general lack of accurate data related to cutter forces and the performance of critical machine components and ultimately their overall effects on the rock cutting efficiency. If such data were available, further improvements could be realized, resulting in a wider and more cost effective application of mechanized boring techniques to underground excavation field.

Realizing this shortcoming, the Colorado School of Mines (CSM) was awarded two separate research contracts by the Department of Energy to design and fabricate a

3-ft. and a 6-ft. diameter laboratory rock drilling and boring equipment. This paper presents the basic capabilities and operation of these machines along with their potential utilization for carrying out water-jet assisted rock cutting research.

LABORATORY ROTARY CUTTING MACHINE

Machine Description

The condensed specifications of the rotary cutting machine are listed in Table 1. The major components of the machine include a main drive unit, a rock box and a support frame on which the machine is mounted. Figure 1 shows two overall views of the machine.

The rotary cutting machine is currently fitted with a 6 ft. diameter Cutter head. It utilizes a total of 15 cutter assemblies incorporating 12 inch diameter replacable disc cutter rings. This particular cutter head also allows a 14 cutter configuration which is achieved by a simple rearrangement of the cutters. Each cutter ring describes an individual radial pattern as the cutter head rotates through a complete revolution (Figure 2).

The cutter head is attached to the base frame through a 2 inch thick adapter plate. The purpose of using an adapter plate is to permit installation of different cutter head assemblies on the machine. The maximum cutter head diameter that can be tested with present configuration is 8 ft. This constraint is brought about by the dimensions of the rock box.

The cutter head thrust is provided by four hydraulic cylinders working in the retracting mode. The total thrust force capacity in the forward propel direction is 813,900 lbs which is generated at a hydraulic pressure of 3500 psi. The maximum thrust stroke is 41.9 inches.

For cutter head rotation, the rotary cutting machine incorporates two hydraulic motors capable of generating 100,000 ft-lbs of torque up to a rotational speed of 12 rpm. Above 12 rpm, torque decreases at constant horsepower input. The cutter head rotational speed is infinitely variable within a range of O to 30 rpm.

To provide cutter head steering, the entire drive assembly is pivoted about front ball bearings attached to main guide columns. A steering actuator mounted at the back end of the main beam assembly provides the hydraulic force for steering the cutter head +/- 5 degrees in one plane. This actuator can generate a steering force of 278,000 lbs.

The hydraulic power unit for rotation includes two 150 hp AC electric motors driving 2 variable volume hydraulic pumps delivering 93 gpm total volume of oil at 4,500 psi maximum pressure. These pumps supply the necessary oil flow to the drive motors which are attached to gear reducers of planetary type. The output pinions of the gear reducers drive into the common main ring gear to produce a maximum cutter head speed of 30 rpm.

The hydraulic system for the thrust and steering actuators includes a 35 hp AC electric motor driving 1 each 30 gpm charge pump and 1 each variable volume pump that can deliver 8.7 gpm at 3,500 psi maximum pressure.

Apart from the main hydraulic system, a lubrication unit is also available to provide the means for lubricating the main bearing, the bull gear and the two gear reducers. The system also permits the extraction and routing back to the reservoir the oil that accumulates in the bearing and gear reducers to achieve a constant recirculation of lubricating oil. Included in the lube unit are a 7.5 hp AC electric motor driving a three-stage, 15 gpm pump and a single stage, 12 gpm return pump.

To allow rotation into any direction, the entire machine body is pivoted on a heavy structural support frame. Tilt capability is provided by two double acting hydraulic cylinders reacting against the base of the support frame. Once the machine is tilted to a desired position, lock-beams are installed to securely hold the machine in that position.

The rock box consists of a heavy structural frame which is flange-mounted to the main machine assembly. The rock box, weighing over 60,000 lbs with rock sample is installed onto the machine when it is in the vertical down position. After the mounting bolts are securely tightened, the entire assembly is tilted to the desired direction using the erection actuators.

The rotary cutting machine also includes removable face and liner shields for use in cutting tests through loose or highly fractured rock to simulate bad ground tunneling conditions. With the face and non-rotating liner shields, the machine can closely simulate the operational characteristics and features of soft ground tunneling equipment.

Machine Instrumentation and Controls

The rotary cutting machine incorporates a very elaborate, custom designed, computer operated data acquisition and control system (Figure 3). The computer and ancillary equipment is designed for 3-channel servo-control and 32 channels of high speed data acquisition. All machine functions including thrust, rpm and steering are servo-controlled to maintain precise control of machine operation during rock cutting. Through custom developed software, the computer operates all servo controllers, checks for limit conditions and records 32 channels of data during machine operation. Thus, the entire machine operation and data recording is accomplished under total computer control. However, a manual override switch is also provided to operate the machine with manual control in case of computer breakdown.

Various types of transducers are used for measuring machine operative parameters including thrust, torque, penetration, steering angle and cutter head rotational speed. Among these transducers, the thrust, steering and ram transducers serve dual purpose as they also provide the feedback signal to the computer which in turn supplies the command signals to the servo-controllers.

The machine thrust load is measured by two custom-built load links installed on the clevis mounts of two opposite thrust actuators. The internally gaged load links only respond to the horizontal load components on the actuators and therefore output is not affected by the change in geometry resulting from forward movement of the cutterhead.

The cutterhead torque measurement is accomplished with a diaphram-type pressure transducer which monitors the hydraulic oil pressure on the two drive motors. The pressure reading is then converted to cutterhead torque using manufacturer supplied performance charts for drive motors.

The steering angle is measured by a linear potentiometer which actually measures the linear displacement on the steering actuator. Through a simple lever arm ratio, this displacement is then converted to a corresponding steering angle of the cutterhead. As with torque output, this conversion is again accomplished by the computer.

The cutterhead rpm is monitored by a magnetic pickup transducer installed on the main bearing housing. By knowing the time interval between successive pulses, the computer then calculates the rotational speed of the cutterhead.

The cutterhead penetration into the rock is measured using an ultrasonic displacement transducer, which provides information about the location of the cutterhead with respect to a fixed starting point. The computer again converts this information to a corresponding penetration rate value.

A unique feature of the rotary cutting machine is the capability to monitor and measure various cutter forces. Presently, the machine is fitted with 8 load cell assemblies installed on selected cutters on the cutter head. These are triaxial load cells capable of measuring three orthogonal force components, vertical, side and rolling on individual cutter assemblies. The load cells were designed and developed at CSM and have proven extremely reliable in past laboratory and field instrumentation programs. They are totally enclosed with steel cover plates and are sealed to provide protection against rock chips or moisture. Temperature compensation is provided through the use of full-bridge circuitry for each load channel. The load cells were designed to incorporate high loading capacity while maintaining excellent measurement sensitivity to any direction of load.

The transfer or cutter force data from load cells to the data acquisition system is accomplished by means of a multi-channel wireless telemetry device. The system basically consists of a multiplexer/signal conditioner, a transmitter and a receiver/demodulator unit. In operation, data from load cells is sampled at a rate of 6400 readings per second, digitized and transmitted serially in a modified NRZ Pulse-Code format consisting of synchronization information and 16 data words. At the receiver end, the data is then decoded and reconverted into analog voltages which are presented in parallel and continuous fashion updated with each sampled value. These data are then fed into the computer for storage on hard disc.

Results of Trial Tests

As part of machine verification and trial tests, a complete series of tests were run with the rotary cutting machine. The rock sample used was a fabricated sample consisting of various size boulders of soft and hard rock cast in a concrete matrix. The purpose of using such rock sample was to simulate mixed face tunneling conditions in order to verify the proper operation of all machine controls and data monitoring systems under most adverse cutting conditions.

Various cutter forces were measured and recorded during trial tests. Tests were run at various machine thrust loads and cutter head rotational speeds. The analysis of test results showed a wide degree of cutter force fluctuations over the cutter head. In particular, very high peak loads were encountered on gage cutters, confirming the hypothesis that gage cutters are subject to more force fluctuations and higher shock loads than the center and inner cutters. By combining the load histories of instrumented cutters, a three-dimensional plot of cutter force distributions over the cutter head for one specific period of cutting was generated (Figure 4). Such a "fingerprint" of the tunnel face is useful in determining the hard inclusions in the face and the resultant force distributions over the cutter head.

LABORATORY DRILLING FIXTURE

Description and Capabilities

An overall view of the laboratory drill rig is shown in Figure 5. The basic machine frame consists of a heavy I-beam weldment which serves to maintain all thrust and torque reactions within the system. The frame is bolted to the concrete foundation to provide stability during drilling. The condensed specifications of this equipment are listed in Table 2.

The machine drive system consists of a Dresser Model-800 raise bore drive unit which features two pinions driving into a common gear. The drive unit includes four guide shoes which slide on structural guide columns. The columns are bolted to the base frame and further secured with a series of L-shaped steel plates, as can be seen in Figure 5.

The cutterhead rotation is provided by a variable volume hydraulic motor. A 150 hp AC motor driving a variable volume hydrostatic pump delivers the necessary oil to the drive motor. Due to the hydraulic drive feature, the rotational speed of the drill bit is infinitely variable from O to 50 rpm. The torque generating capability is 25,000 ft-lbs at 20 rpm. Beyond this speed, torque decreases with constant horsepower input.

A double-acting hydraulic actuator is used to provide forward thrust to the bit. The maximum thrust capability currently available is 100,000 lbs. The machine is designed to withstand higher thrust loads which can be generated by adding a second actuator to the hydraulic thrusting mechanism. Power to the thrust actuator is supplied by a 20 hp AC motor driving a 10.9 gpm variable volume pump. The maximum forward stroke of the drill rig is 54 inches.

The drilling fixture is designed to permit testing of bits of up to 3 ft. in diameter. The drill pipe is fitted with an adapter plate onto which various types of drill bits can be mounted. Presently, the machine is fitted with a 28-inch diameter roadheader cutting head dressed with conical bits (Figure 6).

Controls and Data Monitoring System

The laboratory drilling fixture is equipped with various types of transducers to monitor and measure machine operative parameters. Presently, instrumentation is available for measuring thrust, torque, rpm, penetration rate, water-jet pressure and flow. Provisions have been made for expanding the number of data channels that can be measured as need arises.

Machine thrust is measured with a pressure transducer installed on the thrust actuator hydraulic line. A similar type pressure transducer is also used for measuring the torque pressure supplied to the drive motor. Penetration rate measurement is accomplished with a 10-turn potentiometer mounted on one of the guide shoes. This potentiometer is calibrated to produce a certain voltage output for a given linear movement of the drive assembly with respect to the fixed base frame. A DC-generator type transducer is used to measure the rotational speed of the cutterhead.

The drilling fixture is currently operated with manual hydraulic controls. A control panel located behind the machine in an elevated position contains all necessary controls and meters for the manual operation of the equipment. Data from all transducers is routed to a multi-channel light-beam recorder which features high frequency response and a wide selection of chart speeds.

Future plans for the drill rig call for its totally automated operation using the commuter system that is currently available with the rotary cutting machine. This will permit electronic servo-control of drill rig operative parameters and enable high speed acquisition and analysis of data generated from testing. Thus, the main computer control system described earlier will incorporate the capability of running either one of the two machines.

Design work is also underway for installing a 6-channel wireless telemetry unit on the drill rig. The system will be mounted on the center rotating tube which passes through the drive head. Selected bits on the cutterhead will be instrumented and data telemetered to recording devices in order to measure bit forces during rock cutting.

Test Plans

As noted previously, the drill rig is currently fitted with a 28 inch diameter roadheader cutting head equipped with conical bits. The system is being used as part of a research program sponsored by the Department of Energy to evaluate various cutting techniques for the mechanical mining of oil shale. This effort will primarily concentrate on conducting a series of tests to assess the applicability of roadheader type machines to cutting of oil shale. Included in the investigation will be the evaluation of obtainable cutting rates, degree of bit wear, effect of cutting direction with respect to bedding and the overall economics of this cutting technique. In terms of bedding effects, previous tests performed on CSM linear cutting machine have concluded that direction of cutting in relation to the bedding has a significant influence on bit forces and wear, as well as the size distribution of the cuttings produced. Best cutting performance was obtained when bit was oriented to penetrate oil shale in a direction perpendicular to bedding planes (i.e. bedding planes are parallel to cutting surface). This effect will be investigated in more detail during tests with the drill rig.

USE OF LABORATORY DRILLING AND BORING MACHINES FOR WATER-JET CUTTING RESEARCH

Both the laboratory drill rig and the 6 ft. diameter rotary cutting machines were designed to permit the use of water jets to assist mechanical cutting performance.

For incorporating water jets, the rotary cutter includes a 6-inch diameter center rotating tube which runs the entire length of the main machine body. This tube provides access to the cutterhead through four, Ii2 inch holes drilled at the center portion of the cutterhead. In order to add water jets, a swivel can be installed on the back end of the center tube and high pressure water routed to the cutterhead using either stainless steel pipe or flexible high pressure hose located inside the center pipe. Once the high pressure water is brought to the cutterhead, final plumbing can be devised to route the water to individual nozzles mounted on selected cutters. Thus, the rotary cutting machine incorporates all the design features and means to permit easy installation of water jets on cutter bits.

The drilling fixture is also equipped with a 6-inch center rotating tube which can be used to route the high pressure water to the cuttinghead.

Various types of low and high pressure water pumps are available at CSM that can be used to generate the desired water pressures and volumes for use with either one of the two machines. Currently available are four low pressure triplex pumps and three high pressure duplex intensifier pumps, as well as a large selection of jet nozzles, holders and high pressure tubing. Through various combinations of these pumps, it is feasible to generate a wide range of water jet pressures and flow volumes.

Test plans are currently being devised to install a 3 ft. diameter cutterhead on the drill rig for conducting water jet assisted drag bit cutting tests. These tests which will be performed within the next few months are designed to provide data to evaluate the effect of traverse speed on the degree of assistance which the water jet produces on bit forces and wear.

CONCLUSIONS

Both the laboratory drilling fixture and the rotary cutting machine provide unique capabilities for conducting research programs to evaluate and optimize the factors involved in water jet assisted mechanical cutting of rock. The extensive data acquisition and control capabilities of both machines will assure that tests are conducted under strictly controlled conditions and that a large data base is generated corresponding to all pertinent test variables and bit forces.

Both pieces of equipment are available to industry or to any other organizations interested in carrying out research and development programs in mechanical or water jet assisted cutting of rock.

TABLE 1 Condensed Specifications of Rotary Cutting Machine

Diameter of Head	6 feet
Cutterhead Toorque	2 each 124 hp hydraulic motors provide 100,000 foot pounds at 12 rpm (decreasing torque at constant hp above 12 rpm)
Cutterhead Speed	ariable 0-30 rpm
Thrust Force	813,900 pounds
Thrust Stroke	41.9 inches
Steering Force	278,000 pounds (hydraulic force maximum)
Steering Offset Angle	+/- 5° in one plane
Tilt Capability	+/- 90° from horizontal up or down
Hydraulic Power Unit Rotation	2 each 150 hp AC electric motors driving 2 each variable volume pumps delivering 93 gpm total volume at 4,sno pss maximum
Charge Thurst & Steering	1 each 35 hp AC electric motor, driving 1 each 30 gpm charge pump and 1 each variable volume pump delivering 8.7 gpm at 3,500 psi maximum thruststeering hydraulic supply
Lube Power Unit	1 each 7.5 hp AC electric motor driving a three stage 15 gpm total volume and 1 each sinnle stage 12 gpm return pump
Total Connected Power	342.5 hD 3 phase 460 vac
Overall Dimensions, Including Anchor Frame	23 ft long x 21.5 ft wide x15.8 ft high in the Maximum height, rock down - 19.9 ft

Overall approximate Weight:

Basic Machine without Rock Sample or Rock	
Container Box	71,500 lbs
Anchor Frame	24,000 lbs
Rock Container	18,500 lbs
Rock Sample	52,800 lbs

TABLE 2 Condensed Specifications of Laboratory Drilling

Fixture

Cutterhead Diameter	0 to 3 feet
Cutterhead Torque	25,000 ft-lbs at 20 rpm
Cutterhead Speed	variable 0-50 rnm
Thrust Force1	100,000 pounds
Thrust Stroke	54 inches
Hydraulic Power Unit	one 150 hp AC electric motor driving one variable volume hydrostatic pump
Hydraulic power Unit Thrust	one 20 hp AC electric motor driving one variable volume 10.9 gpm pump
Drilling Parameters Measured	thrust, torque, rpm penetration rate, water jet pressure and flow

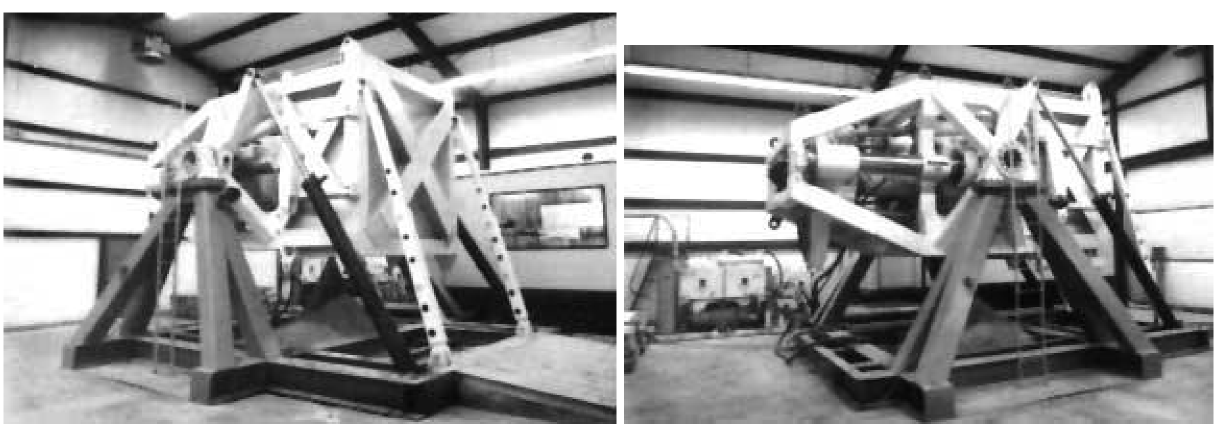

Figure 1. Views of Laboratory rotary cutting machine

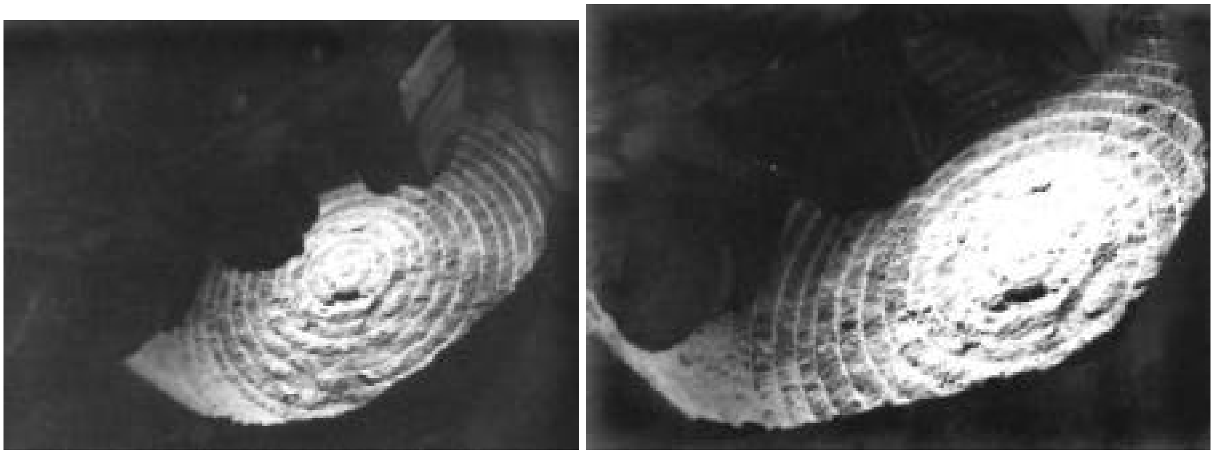

Figure 2., Rock face created by the rotary cutting machine.

Figure 3. The computer control and data acquisition system for the rotary cutting machine.

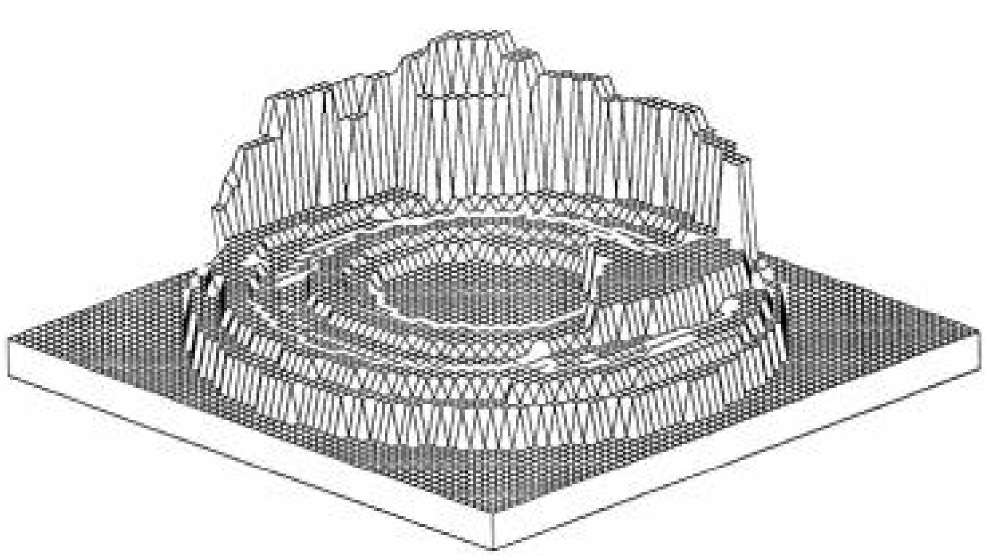

Figure 4. 3-D plot of cutter force distribution over the cutterhead.

Figure 5. 3 ft. diameter laboratory drilling fixture.

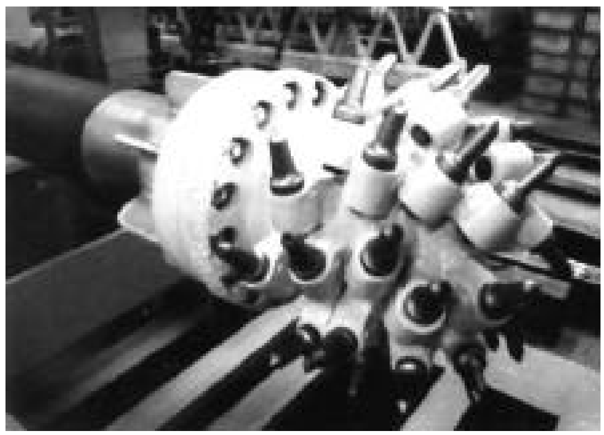

Figure 6. Roadheader cuttinghead installed on the laboratory drill rig.

DISCUSSION

NAME: Dr. Henkel
COMPANY: Bergbau-Forshung

QUESTION: "Have you got already results from the optimum position of the nozzle in order to reduce dust, and/or to increase the cutting performance?"

ANSWER: We believe the optimum nozzle positioning for a water jet assisted disc cutting system is behind the cutter with jet oriented to impinge the cutter path. The jet helps remove the crushed zone remaining in the cutter path to permit higher penetration for the successive cutter passes. An added benefit is the reduced cutter wear. Our results to date indicate that this beneficial effect of water jet assist can be obtained at pressures below 5,000 psi.

HYBRID ROCK CUTTING : FUNDAMENTAL INVESTIGATIONS AND PRACTICAL APPLICATIONS

Orhan Tecen, Mining Division, Etibank, Turkey
Robert J. Fowell, Lecturer, University of Newcastle upon Tyne, U.K.

ABSTRACT

Early experiments with water jet assisted drag tool cutting have shown that this mode of cutting has considerable promise in extending the strength and abrasivity range of rock materials that can be economically excavated by drag tools.

The paper will describe some recent tests undertaken at the Department of Geotechnical Engineering in the University of Newcastle upon Tyne, England, where an extensive programme of fundamental studies on this aspect of mechanical excavation is currently being undertaken.

Despite the promise of the early experiments the fundamentals of rock failure under the combined attack of both jet and drag tool have yet to be fully understood.

This paper concentrates on the influence of cutting speed, jet pressure, and nozzle diameter on cutting efficiency and tool forces. The practical implications of the results obtained indicate that it is not always the highest jet pressures that provide the best results.

INTRODUCTION

Assisting the cutting action of drag tools by preceding their use with high pressure water jets has shown great promise in increasing the range of rock materials than can be excavated economically by roadheader tunnelling machines (Plumpton and Tomlin,1982).

To investigate further the mechanisms of rock failure under this hybrid excavation system, initial studies were undertaken in the University of Newcastle upon Tyne in two sandstones with differing rock properties.

The advantages of water jet assisted rock cutting appear to be:

- Reduction in cutting forces acting on the tool and reduced torque fluctuations on the head.
- Cooling for the tungsten carbide tipped cutting tool extending its economical life.
- Cool the rock in the track behind the tool eliminating hot spots that could cause ignitions.
- Suppress dust at source
- Provide lubrication at the tool/rock interface.
- Assist with debris removal.

This paper reports the results for a limited range of tests which formed the first investigation undertaken for a much wider study of the issues involved. Jet pressure, nozzle diameter and cutting speed were considered to be of fundamental importance to practical application of this technique.

ROCK CUTTING RIG

The tests were conducted on a modified shaping machine having a stroke of 800mm and capable of providing 50 kN in-line thrust (Fig. l).Rock specimens up to 500mm x 500mm x 300mm dimension may be accommodated on the machine table, which could be raised or lowered and laterally traversed with respect to the cutting tool. The cutting tool was mounted in a triaxial dynamometer on the machine crosshead. A Uraca three-piston, positive displacement pump was used in jet assisted cutting tests. This was powered by a continuously rated 24 kW electric motor and delivered 13.0 litre/min at a pressure of 62 MPa. Nikonov type nozzles, which have a 13^0 contraction angle, followed by a straight section of 3 times the nozzle exit diameter, were used. These were made of silver steel and oil hardened to prevent brittleness.

EXPERIMENTAL PROGRAMME

A programme of laboratory tests was carried out to determine the rocks' properties considered relevant to water jet cutting (Table 1). Experiments were conducted at a range of levels to determine the corresponding tool forces, quantities of debris and mechanical specific energies for the tool and hybrid cutting systems (Fig.2). Hydraulic variables are given in Tables 2, 3 and 4.

For the variable pressure experiments, it was considered that the influence of jet pressure on cutting parameters might vary with the depth of the drag tool. Therefore, the experimental programme incorporated an investigation of the effects and interactions of pressure and tool depth of cut.

For cutting speed experiments the water jet pressure for each rock type was chosen such tat at the slowest speed, the jet penetrated the rock surface to a depth slightly more than the mechanical cutter and, for nozzle diameter experiments, the water jet pressure was changed in addition to the variation of nozzle diameter to determine the optimum nozzle performance for each nozzle used.

The positioning of the nozzle in the nozzle holder were noted and the same positions were maintained throughout the experimental programme repeated four times and carried out in random order to minimise any changes that may have occurred in the rock samples or operating conditions during the cutting investigations. Computer curve--fitting analysis was performed on the experimental output and best-fit functions were chosen.

DISCUSSION OF RESULTS

1. Influence of Pressure Investigations

The experimental conditions are given in Table 2 and Figures 4 to 12 show the results obtained.

The results indicate that the penetration depth of the water jet and hence its pressure has a strong influence on the efficiency of a hybrid cutting system. Penetration depth varied directly with jet pressure, showing a linear relationship within the experimental range. Threshold pressures required to initiate penetration were found by extrapolation to be 9.12 MPa for Darney Sandstone and 6.3 MPa for Springwell sandstone. Cooley (1970) has reported that "threshold pressure is typically 20 to 50 percent of the rock's compressive strength". Results obtained from these cutting tests contradicted this, as threshold pressure for Darney Sandstone was 14.13% and for Springwell Sandstone, 14.5%. Experiments were conducted at a constant cutting speed of 165 mm/sec. This speed is slow in comparison with speeds attained by rotating cutting heads of excavation machines, though within the range attained by coal ploughs If it is accepted that the value of threshhold pressure varies with cutting speed (Harris, 1974) this may explain the difference.

Increase in water jet pressure results in increased energy input available for rock cutting which leads to deeper jet penetration. For the most efficient hybrid cutting system it was found that the water jet pressure should be such that it will not penetrate the rock more than the depth taken by the mechanical tool. For example, taking a tool depth of 11 mm, it can be seen (Figs. 5 and 6) that when the pressure was increased above 44.8 MPa, no significant reduction in the cutting and normal forces had taken place. This is believed to be due to the jet penetrating the rock to the same depth as the tool at this pressure. When the pressure was further increased, this resulted in deeper jet penetration. Since the mechanical tool depth was less than this, no useful gain was made by this increased pressure. In fact, it had a harmful effect in terms of overall cutting efficiency as power requirements for achieving higher pressure increase directly with the increase in pressure.

Increasing both the mechanical tool depth of cut and the water jet pressure have caused reductions in mechanical specific energy. But when both hydraulic and mechanical specific energies were taken into account, the total specific energy for water jet assisted cutting was much higher than when cutting with a mechanical tool alone at the same depth of cut.

The increase in mechanical tool depth of cut led to production of more yield at a constant water jet pressure (Fig. 11). Curves showed a power relationship between the variables, demonstrating the advantages of taking deeper depths of cut. However, at a constant depth of cut, increase in water jet pressure resulted in differing yield values (Fig.6). After a certain depth of cut the yield was found to decrease with increase in jet pressure. For Darney Sandstone, up to and equal to 5mm depth of cut,yield increased linearly with pressure. But at higher depth values, yield decreased. An increase in yield with increase in pressure at shallow depth of cuts may be explained by the fact that, at these levels, the water jet has produced more yield than the mechanical tool. The cutting action of the mechanical tool was changed at deeper depths by the assistance of the deep penetrating water jet. Instead of tool tip initiating the fracture process it is the sides of the mechanical tool which do the work. As a result, less yield is produced. Therefore, for a hybrid cutting system the water jet should penetrate the rock less than the mechanical tool depth of cut.

2. Cutting Speed

The influence of cutting speeds will be discussed separately on the mechanical and hydraulic components of a hybrid cutting system.

(l) Mechanical Component: The effect of speed becomes more apparent with increases in the distance cut by the tool in the field situation and the condition of the tool determines the degree of influence of cutting speed. At the beginning, when the distance cut is comparatively small, the cutting tools are in sharp condition and tool forces and mechanical specific energy are not affected by the variations in cutting speed. As the distance cut increases, due to the rise in temperature at the tool/ rock interface, wear of the cutting tools progresses. Once a critical speed is reached, depending on the critical temperature of the tungsten carbide, wear increases dramatically. Altinoluk (1981) has found that the factors such as the geometry of the tool, properties of rock, and cutting conditions, also modify the temperature attained by a worn tool. The hardness of tungsten carbide decreases rapidly with increase in temperature of the tool/rock interface and, in some instances, falls below the hardness of the quartz grains it is cutting, which are at ambient temperature. This leads to rapid degradation of the tool. The rapid increase in tool wear causes corresponding increases in tool forces and mechanical specific energy. But, in the absence of wear it has been shown that cutting speed has negligible influence on the mechanical part of hybrid cutting systems. The action of the water jet is considered beneficial in removing heat from the tip and lubricating the interface.

(2) Hydraulic Component: The effect of time on water jet penetration depth is made apparent by the change in cutting speed. There is a limit to the amount of penetration depth that can be attained by a slow traversing jet once the water jet pressure exceeds the threshold pressure for the rock. At the slowest speed - when the exposure time over a point on the rock is in excess of 60 seconds - no increase in penetration depth takes place after 30 seconds (Summers, 1977). When a certain crater depth is reached, any further increase in depth is prevented by the water cushion formed in the crater. Most of the penetration takes place in very short periods of time ˘(0.0ls).

Between the minimum and maximum ends of the speed spectrum, there is an optimum jet traverse speed, at which hydraulic specific energy reaches its most efficient value which, according to Harris (1974) changes with water jet pressure.

The results for Springwell and Darney sandstones have shown that the penetration depth varies inversely with the cutting speed, exhibiting a power relationship. The power value for Springwell Sandstone was found to be -0.5 and -0.45 for Darney Sandstone. Results had shown more sensitivity to change at slow traverse speeds, especially between 50-100 mm/sec. Cutting and normal forces had increased as a result of decrease in jet penetration depth. Curves tend to reach a constant value and run parallel to the speed axis with increases in speed. Mechanical specific energy increased at a decreasing rate with increase in speed, exhibiting a hyperbolic relationship (Figs.13 and 16).

Cutting speed has different effects on the components of a hybrid system. Increasing cutting speed led to a decrease in hydraulic specific energy and to an increase in mechanical specific energy. Therefore, a compromise must be found for each cutting condition. For the most efficient hybrid cutting system, the water jet pressure should be high enough to cause a penetration less than, or equal to, that of mechanical tools at a selected speed.

3. Nozzle Diameter

At a constant stand-off distance generally an increase in nozzle diameter leads to a corresponding increase in jet penetration depth due to increased power into the rock. The rate of increase being greater at higher water jet pressure. Due to the increase in jet penetration depth all the tool force components were decreased. This decrease was more pronounced between 0.6mm and 0.85mm diameter (Figs.17 and 20). Three nozzles have performed equally well on Springwell Sandstone at a constant stand-off to nozzle diameter ratio at 13.8 MPa. Mechanical specific energy decraesed with increasing nozzle diameter; more so at high pressures.

The optimum nozzle exit diameter is dependent on the type of mechanical tool with which it is to be used. The location of the nozzle with respect to the mechanical tool, as well as on the power available and desired pressure range, must also be taken into consideration. The width of the cut is dependent on the nozzle diameter and was found to be approximately equal to three times the nozzle diameter. When the larger diameter l.l mm nozzle was used with a point attack tool it produced a slot approximately 3mm wide which was greater than the point attack tool tip diameter. Hence only the upper edges of the carbide tip came into contact with the walls of the kerf and the cutting became less efficient.

CONCLUSIONS

Operating efficiency of an hybrid cutting system is dependent on the penetration depth of the high pressure water jet. Penetration can be increased by increasing the jet pressure or jet diameter, or decreasing cutting speed. Changing the properties of the fluid used and by using several jets in tandem can also influence the penetration depth.

At a constant mechanical tool depth, increasing pressure leads to a decrease in the forces acting on the tool tip. The magnitude of reduction is dependent on the penetration depth of the water jet. The optimum pressure for water jet assisted cutting is not necessarily the highest pressure that can be attained. It is dependent on the rock type and on other hydraulic variables such as nozzle diameter, cutting speed and nozzle positioning. The pressure of the water jet should be such that the jet will not penetrate the rock surface more than the mechanical tool depth of cut to keep power requirements to a minimum.

Tool forces increase with the increase in cutting speed due to decreasing depths of jet penetration. Curves show a tendency to reach a constant value and run parallel to the speed axis at high speeds, indicating that at high speeds tool forces are independent of speed. Although an increase in cutting speed causes a decrease in hydraulic specific energy it effects an increase in mechanical specific energy. Hence, a compromise must be found for every cutting condition. For the most efficient cutting at a selected speed the jet should not penetrate the rock surface deeper than the mechanical tools.

At a constant waterjet pressure and constant stand-off distance, the jet penetration depth increases directly with increase in nozzle diameter; more so at higher pressures. Because increasing nozzle diameter causes an increase in jet penetration depth, the tool forces are reduced as a result. Three nozzles have performed equally well at a constant stand-off/nozzle diameter ratio. The optimum nozzle diameter is dependent both on the type of mechanical tool with which it is to be used and on the location of the nozzle with respect to the tool tip. Also, on desired pressure range and available power. Power requirements for water jet cutting increase with the first power of pressure and second power of nozzle diameter. Therefore, the nozzle diameter should be selected such that it will cost less to penetrate to a required depth. This necessitates the use of smaller diameter nozzles whenever practicable.

A large rock cutting rig is presently being modified to increase the cutting speed to lm/s to permit a more detailed investigation of this promising rock excavation system.

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Rock Property	Springwell Sandstone	Darney Sandstone
Uniaxial Compressive Strength (MPa)	43.20	64.53
Indirect Tensile Strength (MPa)	3.00	4.34
Static Elastic Modulus (GPa)	15.40	22.50
Dynamic Elastic Modulus (GPa)	17.90	9.97
Poisson's Ratio	0.26	0.28
Bulk Density (g/c)	2.21	2.27
Apparent Porosity (%)	16.40	8.50
Shore Rebound Hardness Number	36.70	35.3
Plasticity (%)	42.20	30.60
Schmidt Hammer Rebound Hard. No.	52.03	43.40
NCB Core Indenter Hardness	1.98	2.53

Table l Various Rock Properties.

Experimental Variables	Levels
Nozzle diameter (mm)	0.85
Cutting speed (mm/sec)	165
Stand-off distance (mm)	45
Lead-on distance (mm)	5
Water jet pressure (MPa)	13.8, 24.2, 34.5, 44.8, 55.2
Mechanical tool depth of cut (mm)	3, 5, 7, 9, 11

Table 2 Pressure Experiments on Darney Sandstone.

Experimental Variables	Levels: Springwell s/s	Darney s/s
Depth of cut (mm)	8	7
Water jet pressure (MPa)	44.83	27.60
Nozzle diameter (mm)	0.85	0.85
Lead-on distance (mm)	2	5
Cutting speed (mm/s)	61, l00, 120, 167, 216	56, 121, 145, 168, 189, 205
Stand-off distance (mm)	45	45

Table 3 Cutting Speed Experiments

Experimental Variables	Springwell s/s	Darney s/s
Depth of cut (mm)	8	7
Cutting speed (mm/s)	165	165
Stand-off distance (mm)	45	45
Lead-on distance (mm)	5	5
Water jet pressure (MPa)	13.8, 34.5	13.8, 27.6
Nozzle diameter(mm)	0.6, 0.85, 1.10	0.6, 0.8, 1.12

Table 4 Nozzle Diameter Experiments.

Tip angle (degrees)	87
Off-set angle (degrees)	6.5
Angle of attack (degrees)	45

Table 5 Mechanical Tool Variables.

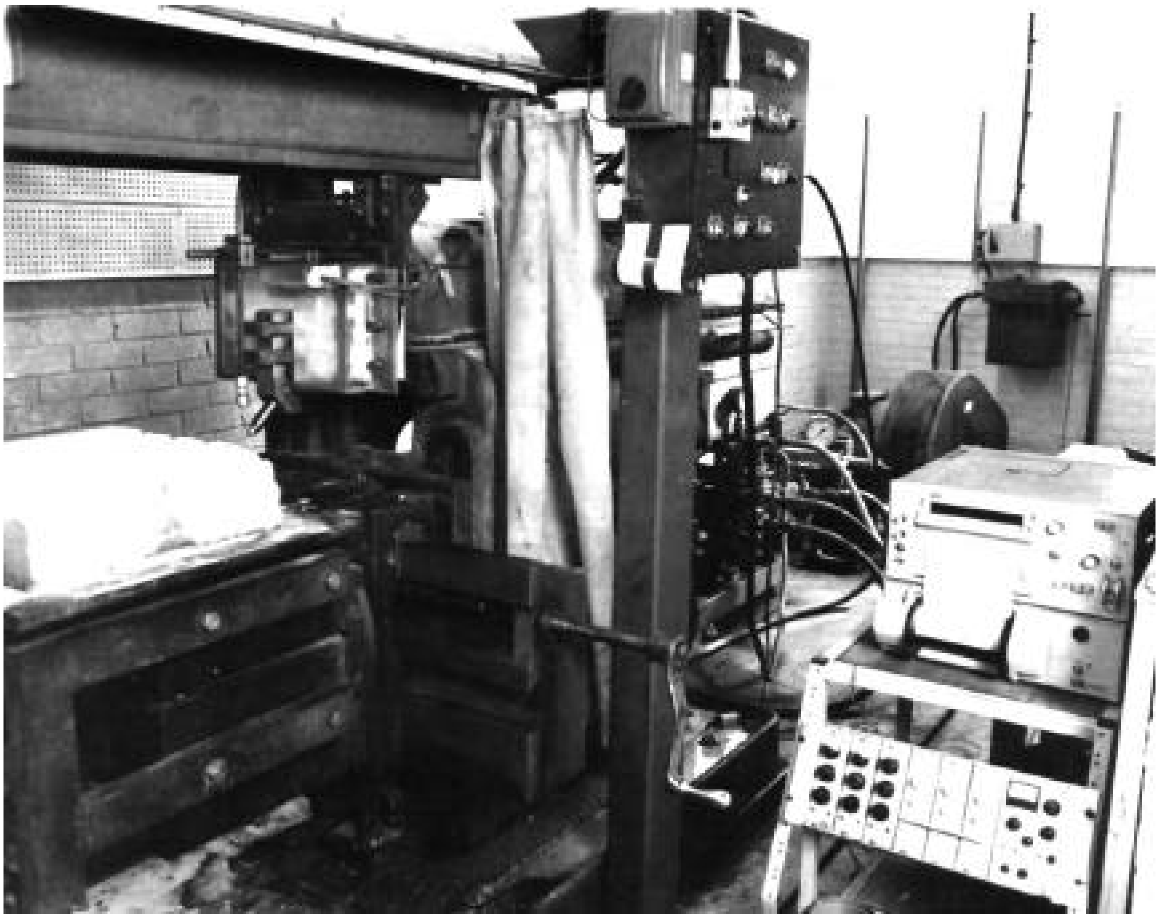


Figure 1. General view of hybrid cutting room

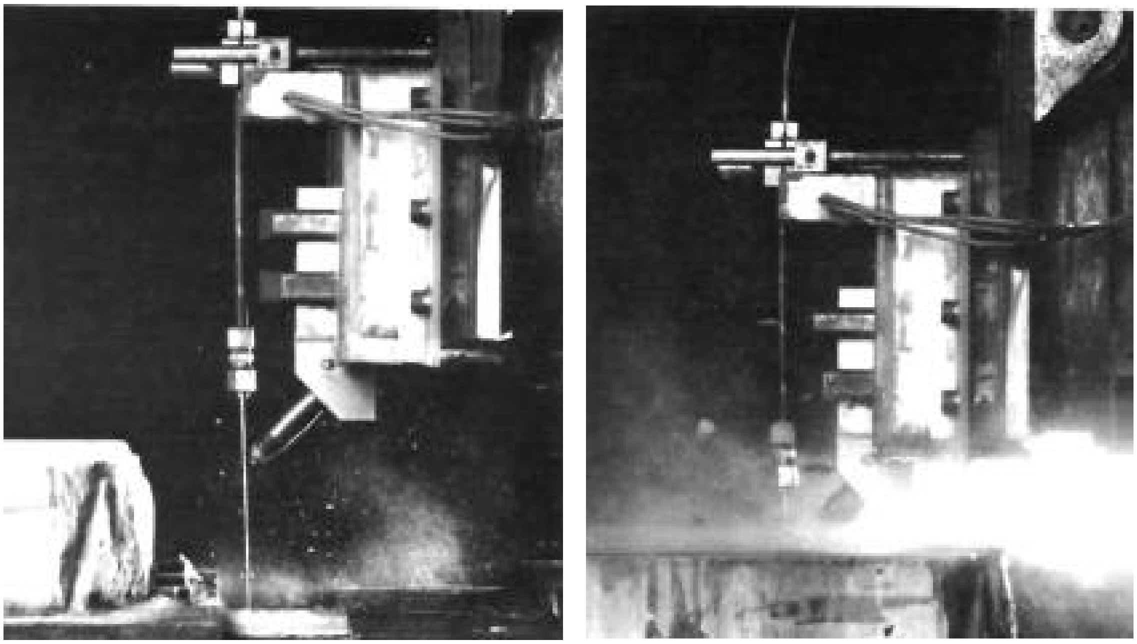
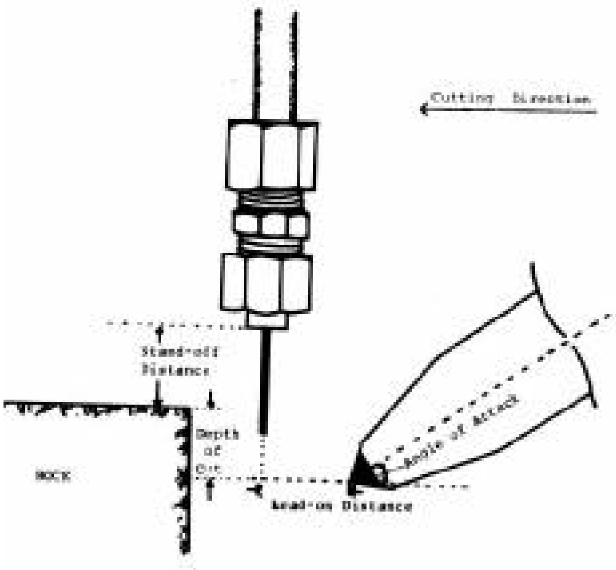


Figure 2. Hybrid cutting - before and during cutting.



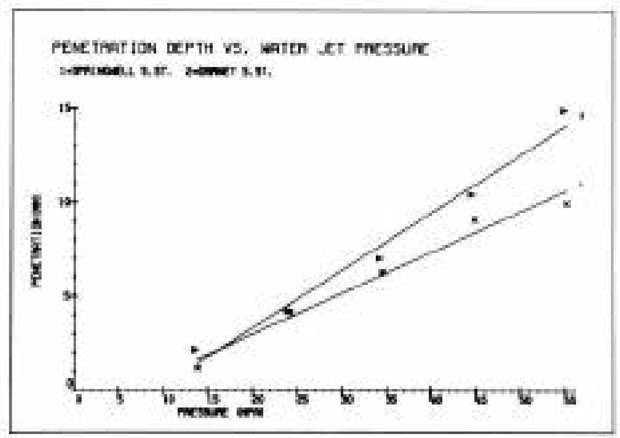


Figure 3. Variables

Figure 4.

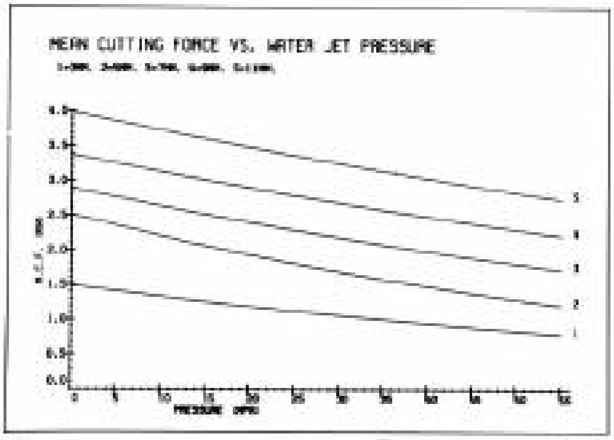
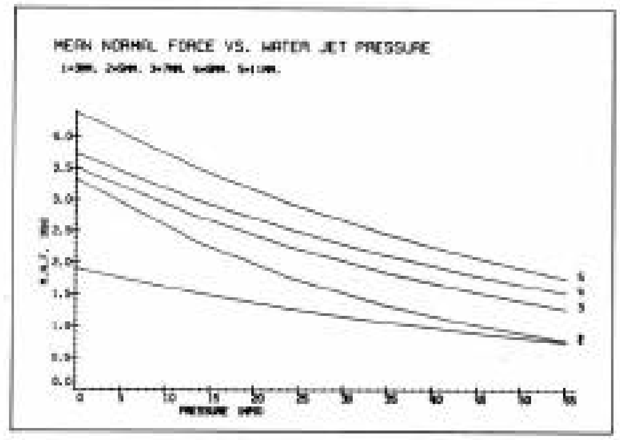


Figure 5.



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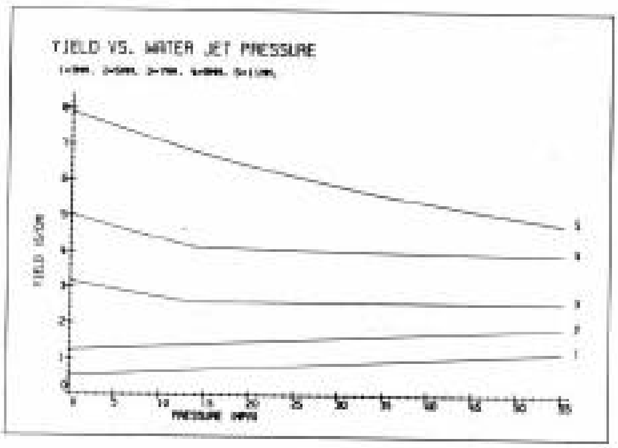
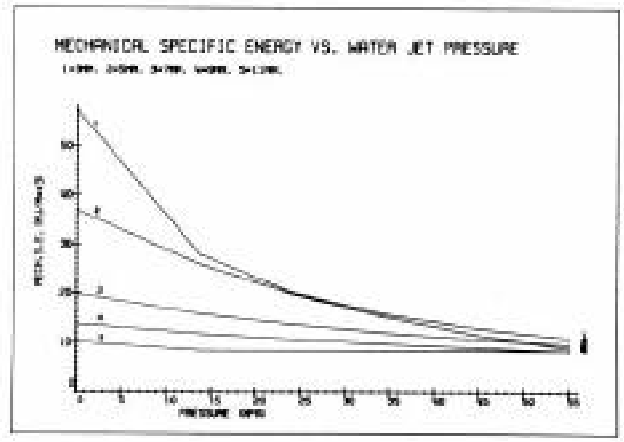


Figure 7



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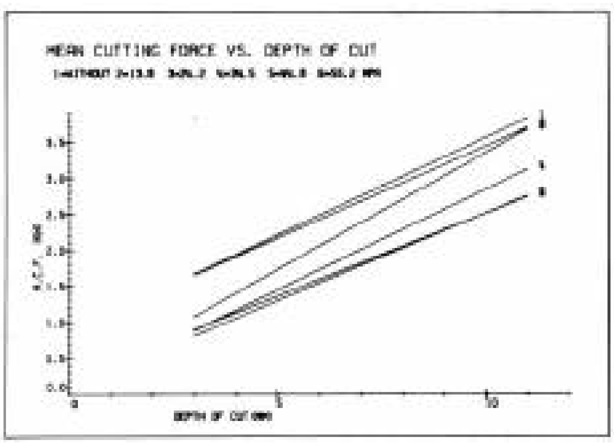

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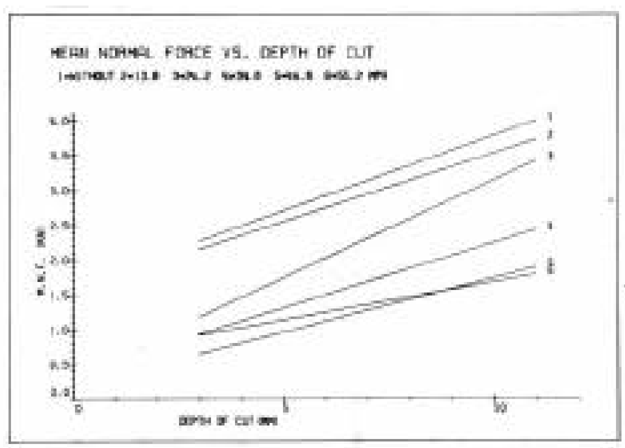


Figure 10.

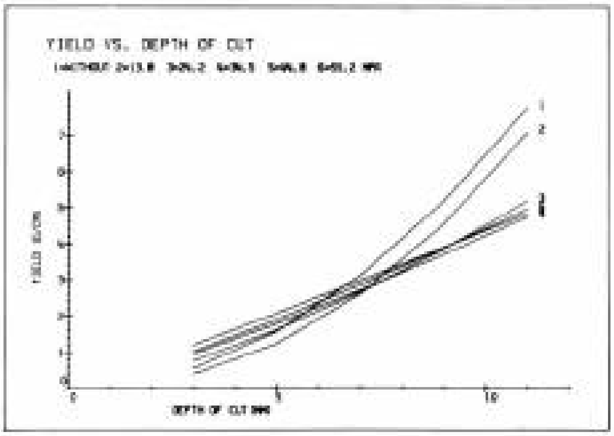
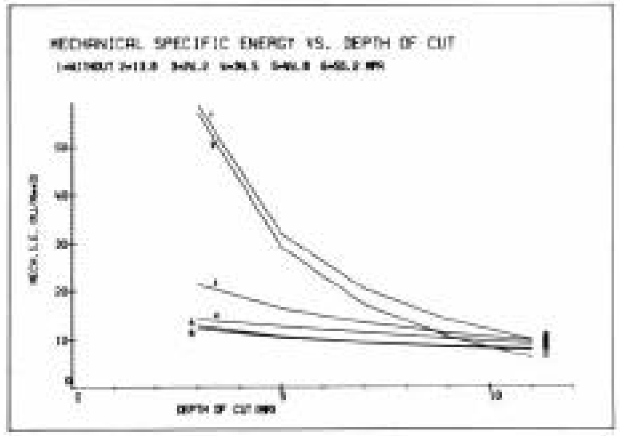


Figure 11.



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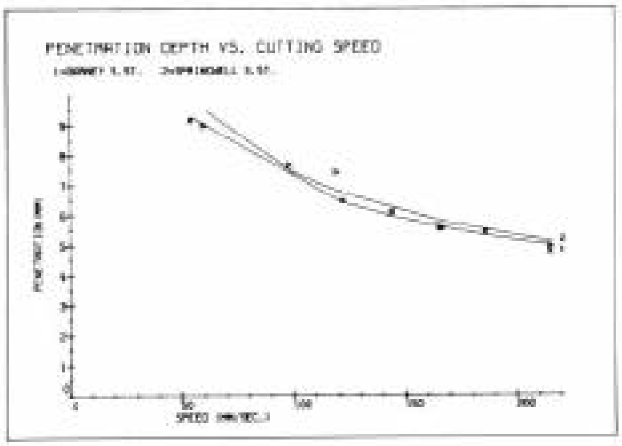

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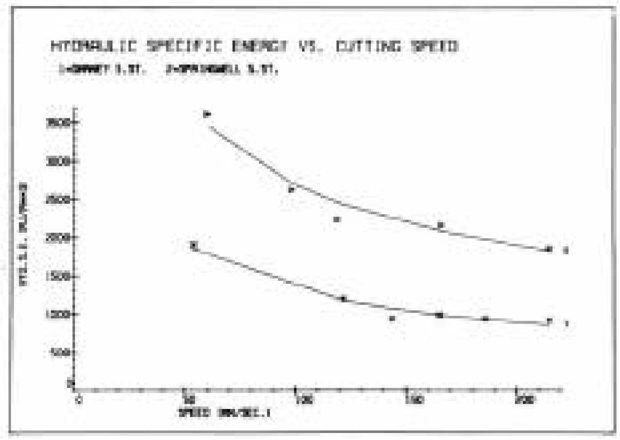

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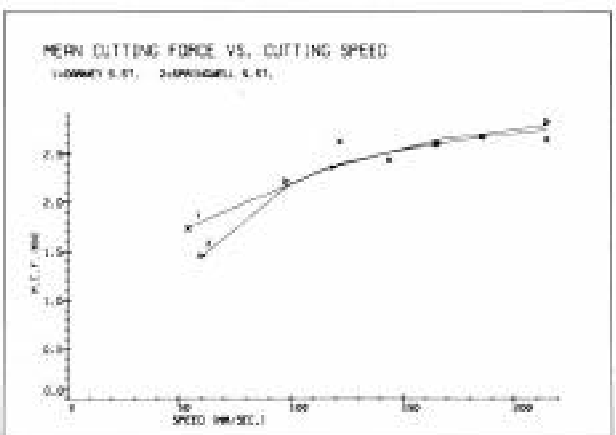

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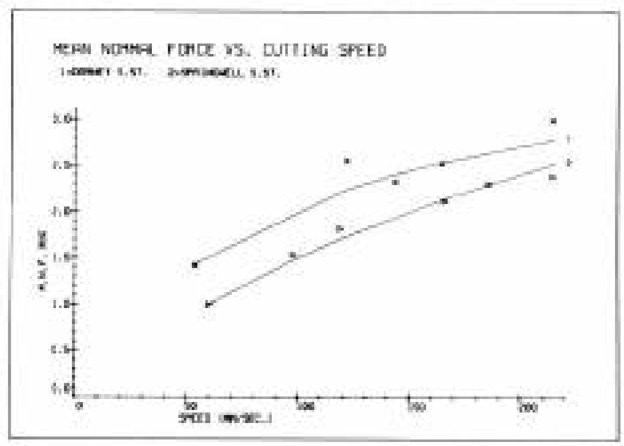

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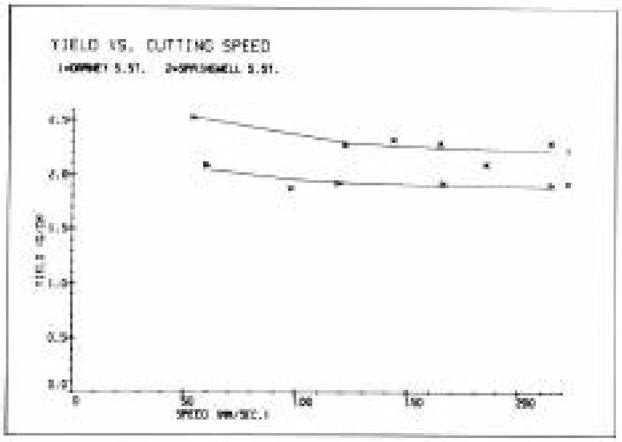

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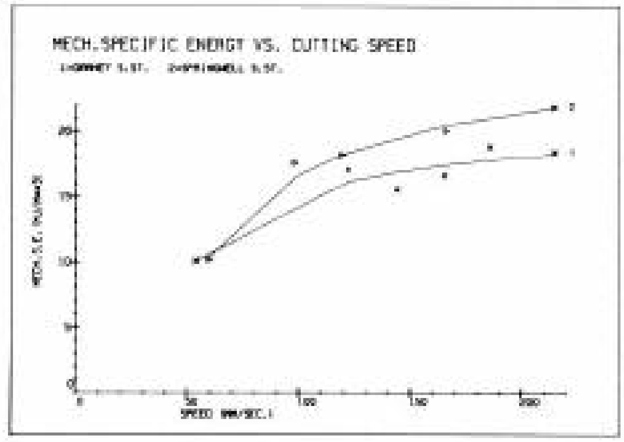

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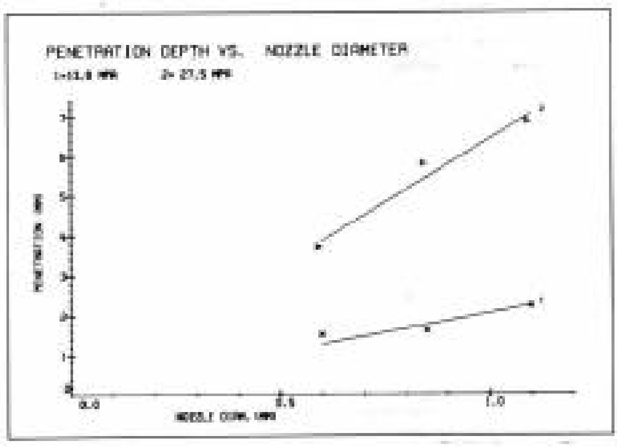

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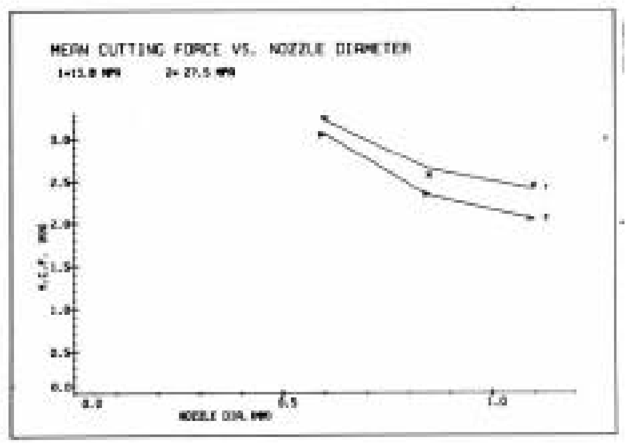

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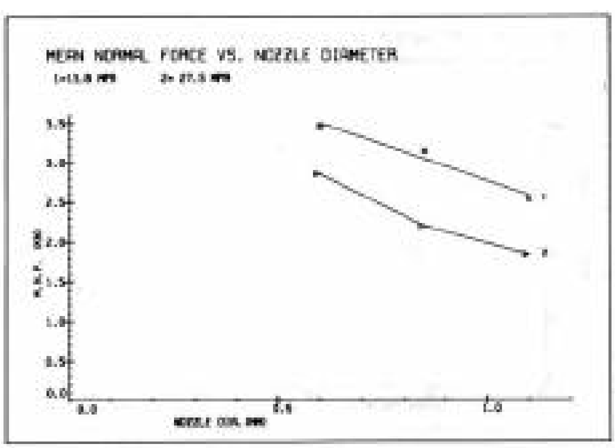

Figure 21.

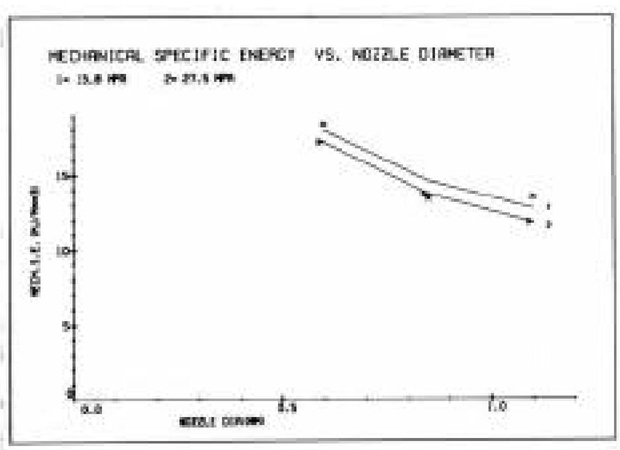

Figure 22.

DISCUSSION

NAME: John E. Wolgamott
COMPANY: Stone Age, Inc.

QUESTION:"Why did you choose to place the jet in front of the pick? I found in my earlier work at the Colorado School of Mines that this location suffered greatly because of disruption to the jet by the generated chips from the pick action. Aiming the jet from behind the pick avoids this problem, allowed lower jet pressures to be effective, and had a dramatic effect on lowering the normal force required to keep the pick cutting in the rock.

ANSWER: There were a number of reasons for placing the jet in front and not behind the tool, namely:

- at the time of the study this was the obvious position, hypothesised by many to be desirable;
- this position in conjunction with a tool mounted on a modified metal planer, presented both least technical difficulty and allowed for both easy and flexible variation of stand-off distance and lead-on distance.
- you will appreciate the large number of variables involved in linear hydrid cutting, and the jet in front offered the most progressive experimental program within the philosophy of the pilot study.
- The study looked at effects of speed, stand-off distance, various lead-on and offset geometries for the nozzle, nozzle diameter, water pressure, depth of cut and rock type, and their effects on specific energy, cutting and normal forces, jet penetration, hydraulic specific energy for a point attack tool, for both single and multiple passes

However, this is not to say that any other jet position is to be ignored.

Indeed, we plan in our present work on a much larger linear rig cutting at 1 m/s with commercial tools, to change jet position, and we believe that the jet impinging from behind the tool may well prove effective in reducing forces, especially the critical normal force at these speeds. We also feel that any jet impinging in this way may well display an advantage in coal cutting by cooling the path of the tool, and so preventing the incidence of frictional ignition in potential methane environments.

We hope, in due course, to have the opportunity of presenting the results of our present research.

SCHEMES OF COAL MASSIF BREAKAGE BY DISC CUTTER AND HIGH-VELOCIEY WATER JET

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V.G, Merzlyakov, Cand. Sc. (Tech)
Skochinsky Institute of Mining, 140004, U,S,S,R,

ABSTRACT

Paper deals with some research results obtained by schemes of coal massif breakage by a disc cutter and a high-velocity water jet. Dependences between force parameters, specific energy consumption and granulometric composition of broken coal and regime parameters, shape of disc cutter and coal massif strength are established.

A scheme of coal massif breakage by a disc cutter and a high-velocity water jet is given. The scheme provides for reduced specific energy consumption, better coal grade and for lower dust content in the air as compared with the mechanical method.

INTRODUCTION

The hydromechanical method which combines a high-velocity water jet and a cutting or shearing instrument (Ref. 1-2) is a method of advantages.

Great sliding frictions along the lateral and rear sides of the instrument and considerable space of its contact with the coal massif accompany the process of coal massif breakage by a cutting instrument both under the hydromechanical and mechanical methods. This cuts off the instrument life and efficiency, leads to a higher specific energy consumption to break coal, and provides for a higher yield of fines. The hydromechanical method with a shearing instrument (disc cutter) instead of cutting instruments allows for reduction of rolling force and specific energy consumption, and for improvement of coal grade (Ref. 3-5).

It is necessary to investigate some schemes of joint action of disc cutters and high-velocity water jets onto a coal massif so as to rationally choose the parameters of hydromechanical instruments with disc cutters. For this purpose a certain amount of research has been carried out in Skochinsky Institute of Mining.

RESEARCH METHODS

The combined method of breakage suggests that the water jets should cut slots to form inter-slot coal pillars to be sheared by the disc cutters. The disc cutters can act onto the pillar in two ways:

i) either by directly introducing the wedge rim top of the disc cutter into the pillar, or
ii) by affecting the lateral surface of the pillar with the wedge rim side of the disc cutter while rolling along the slot.

Development of various schemes for coal massif combined breakage by a disc cutter with double-sided (Fig. l,a-d) and with single-sided wedge rim (Fig. l,e-f) and a high-velocity water jet is based on these ways the following schemes of breakage by a water jet and a double-sided-wedge rim disc cutter are possible:

(a) a front scheme when the inter-slot pillar is broken by a disc cutter rolling directly along the pillar in its symmetry plane (Fig. l,a);

(b) an one-sided scheme (Fig. l,b) when the pillar is broken towards its exposed surface by the side of the wedge rim of the disc cutter thrust against the side surface of the pillar and rolling along the slot between the pillar and the main part of the coal massif;

(c) a two-sided scheme (see Fig. l,c) when the disc cutter breaks simultaneously both pillars while rolling along the slot cut by a water jet between these two pillars; and

(d) an one-sided scheme of breakage with a floating attachment of the disc cutter (Fig l,d). In contrast to the rigid attachment (Fig l,b) which allows but one degree of freedom for the disc cutter (its rotation), the floating attachment provides for two degrees of freedom (besides rotation the disc cutter can reciprocate).

For breaking a coal massif by a water jet and a disc cutter with single-sided wedge rim, two one-sided schemes are possible: either the depth of the disc cutter thrust is greater (Fig. l,e), or it is smaller (Fig. l,f) than the depth of the slot cut in the massif by the water jet.

The main feature of the last scheme (Fig l,f) as compared with all the others is that the disc cutter while rolling along the slot acts upon the coal massif not only by its rim sides but also by its rim top.

Scheme of breakage, breakage parameters (depth of disc cutter thrust h, slot cutting pitch t, depth of slot h_s, distance between the disc cutter axis and the water jet contact point with the coal massif a), geometric parameters of disc cutter (wedge rim angle δ and wedge outside diameter (D) and physical and mechanical properties of the coal massif constitute the main factors responsible for the combined breakage.

The schemes of the combined breakage were studied with constant velocity of water jet and disc cutter movement, while the depth of slot (produced by different water pressure in nozzle outlet) was assumed as a unifying hydraulic factor.

The mechanical breakage specific energy consumption H_w was assumed as the main criterion to evaluate the combined breakage scheme efficiency. Besides, this efficiency was evaluated according to: total specific energy consumption E_o including the mechanical breakage energy consumption H_w and hydraulical breakage energy consumption E_o (with V_p = 2.75 m/s, d_o = 2 mm); cutting force P_z, forward force P_y and side thrust P_x; percentage of coal grade 0-6 mm (R_{-6}) and +25 mm (R_{+25}) and specific coal dust formation q.

Coal masif breakage was studied both for the levelled surface and in the established regime. The studies of breakage from the levelled surface have made it possible to better find out the special features of the combined breakage and to establish its most efficient schemes. The studies in the established regime were carried out to find out thoroughly an efficient scheme for the combined breakage and to establish its rational parameters and dependences:

$$P_z, P_y, P_x, H_w, R_{-}6, R_{+25}, = f\,(h, t, h_s, a, d, \delta, k) \qquad (1)$$

Advantages of the hydromechanical method for coal breakage were defined by a comparative research of the mechanical and hydromechanical methods.

The research was carried out by breaking a coal-cement block on a planer. A tensodynamometer (with an amplifier 8ANCh-7M and an oscillograph H-115) was fixed on the support of the planer to register the force parameters of the process. A special hydromechanical device was attached to the dynamometer Water was supplied to the device by two pumps (water pressure up to 55 MPa, water discharge 200 l/min.). The stand was provided with a special device to collect the dust formed by breaking the coal block.

CHARACTERISTICS OF COAL MASSIF COMBINED BREAKAGE AND DETERMINATION OF ITS RATIONAL SCHEME

The research has proved that for the breakage while $h_s > h$ the depth of slot cut by a water jet does not tell upon P_z, P_y and P_x and upon the energy consumption H_w because the pillars (being heterogenious) are broken at a smaller height than the depth of slot. When $h_s > h$, the deeper the slot, the more reduced the force parameters of the process and the specific energy consumption for mechanical breakage, because a bigger h_s provides for a smaller area of contact of the cutter with the coal massif and a lower pillar resistance to breakage.

For all the breakage schemes any increase of the parameters h and t leads to a reduction of the energy consumption H_w according to the curve dependences, and it reaches the minimum at its certain rational values which correspond to the maximum volume of breakage. Any further increase of the mentioned parameters leads to a higher specific energy consumption because the volume of breakage remains practically constant, while the rolling force of the disc cutter increases.

It is determined that when the disc cutter is introduced directly in the inter-slot pillar (Fig. l,a), its breakage happens along the direction perpendicular to the plane of rolling of the disc cutter and provides for a considerable output of coal fines. Whereas the disc cutter introduced in the slot formed by the water jet provides for breaking the coal massif due to the tension and bending moment both perpendicular to the plane of rolling of the disc cutter and along its movement in the slot. That is why the cutting force P_z is by 1.3-1.5 times and the energy consumption H_w is by 3.0-3.5 times greater for the front scheme than for the one-sided and two-sided schemes of breakage. So the front scheme is responsible for output of about 30 per cent of coal grade 0-6 mm (which is by 4-5 times more than that of the one- and two-sided schemes) while its output of coal grade +25 mm is but almost 2 times less.

After this research of the schemes of breakage by a water jet together with a disc cutter with double-sided wedge rim it is established that the best results are obtained by the one-sided scheme of breakage with a floating disc cutter (Fig. l,d). The floating attachment has made it possible to completely exclude the side thrust P_x, to reduce P_z by 1,3-1,8 times and P_y by 2.4-4.8 times (Fig. 2), to cut the energy consumption H_w by 20-60 per cent (Fig. 3) and to improve the coal grade Ref. 6) what cannot be provided for by the rigid attachment of the disc cutter.

For rigid attachment the inter-slot pillar breakage is accompanied by a considerable pressure exercised over the side surface of the coal massif part left, because all the time the disc cutter affects not only the side surface of the pillar which it breaks, but also the side surface of the pillar which it leaves unbroken. As the one-sided scheme allows to shear the pillar by parts both perpendicular to the plane of the disc cutter rolling and ahead of it along the slot, then, in case of the rigid attachment the sides of the disc cutter wedge rim affects the side surfaces of the pillar as well as the part of the massif left. This takes place only during wedging out the pillar by the disc cutter. It lasts till the tensile stress and tension grew sufficient enough to break this part of the pillar. After this moment the disc cutter moves along its axis and away from the side surface of the pillar left until it comes across the next virgin part of the pillar. Due to a wedge-like shape of its rim, the disc cutter enters the slot by moving along its axis towards the side surface of the massif which is left, in order to get wedged out between it and the pillar to shear the pillar. So, the floating attachment of the disc cutter helps to considerably reduce the area of its contact with the side surface of the part of the massif left unbroken, thus practically completely eliminating the side thrust effect on its axis.

However, the one-sided scheme of breakage by a double-side wedge rim disc cutter has but a certain demerit. While rolling the disc cutter touches not only the side surface of the pillar to be broken but also the side surface of the coal massif left unbroken. It tells negatively upon the process parameters.

The research results of the breakage schemes (Fig l,e,f) have demonstrated that when the disc cutter enters the slot at a depth $h > h_s$, the inter-slot pillar is sheared by large lumps practically along its base which is subjected to the maximum tensile stress. When rolling along the slot the disc cutter affects not only the side surface of the pillar by its side rim, but also shears the base of the pillar by pressing its top in the massif. This results in reducing the energy consumption for hydraulic breakage considerably because the depth of slots required is smaller. So,the energy consumption H_w was reduced by 25-40 per cent while the energy consumption E_o was cut by 3-4 times by application of the breakage scheme with a depth $h > h_s$, as compared with the thrust depth $h < h_s$ (Fig. 4). By introducing the disc cutter in the base of the pillar it has become possible to cut the output of 0-6 mm coal grade almost by 3 times and to increase the output of +25mm coal grade by 35-45 per cent.

The scheme of breakage by a water jet and a single-sided wedge rim disc cutter with $h > h_s$ (Fig l,f) is found out to be most efficient and to provide for the minimum specific energy consumption for the mechanical and hydromechanical breakage.

Research analysis has proved that the combined application of a water jet and a single-sided wedge rim disc cutter with $h > h_s$ allows for:

(1) a cut in energy consumption H_w by 20-30 per cent and total energy consumption by almost 3 times (see Fig. 5) as compared with the one-sided scheme of breakage by a water a et and a floating disc cutter. Besides, the output of 0-6 mm coal grade is reduced, while that of +25 mm coal grade is increased by 10-15 per cent;

(2) a reduction of the forces P_z by 1.5-2.0 times, P_y by 1.2-1.4 times and P_x by 1.7 - 4.4 times (see Fig. 6), as well as that of energy consumption H_w by 1 9-2.2 times (see Fig. 7). It is responsible for 15-20 per cent more of +25 mm coal grade output and 10-15 per cent less on 0-6 mm coal grade output as well as for a lower dust content. (The efficiency of dust suppression for the hydromechanical method of breakage is 99.7-99.8 per cent as compared with the mechanical method of breakage by a disc cutter); and

(3)) a cut in energy consumption H_w by 1 5-2.0 times as compared with the hydromechanical method of breakage by a water jet and a cutting instrument like IT-2C or I-901B type.

INFLUENCE OF PHYSICAL AND MECHANICAL PROPERTIES OF COAL MASSIF UPON POWER AND ENERGY INDICES OF COMBINED BREAKAGE

Coal characteristics widely used as criteria for the mechanical and hydraulic methods of breakage were taken as the physical and mechanical property basic indices in order to choose a criterion to evaluate coal massif resistance to the combined breakage. They are given below.

Table I

Indices	Research range
Strength index F according to Prof M.M..Protodyakonov scale	1.2 - 2.3
Temporary coal uniaxial compression strength σ_{com}, MPa	6.2 -14.5
Temporary coal uniaxial ten sile strength, σt_{en} MPa	0.81- 1.23
Coal resistance to cutting, A, N/cm	1180 - 2080
Conventional breaking strength $R_y = \sqrt{\frac{\sigma_{com} \ \sigma_{ten}}{3}}$,MPa	1.29- 2.43

The criterion for an estimate of coal massif resistance to the combined breakage has been determined with the help of twin correlation analysis method which means finding an interrelation of the power indices of the process with every property separately on the basis of the experimental data results. We applied the results obtained during combined breakage of coal massif in the standard regime according to the scheme (see Fig. 1,e) as the experimental data. In all the experiments we resorted to a disc cutter with a single-sided wedge rim and constant geometric parameters (D = 200 mm, g = 35^o), while the breakage parameters h, t, h_s and a were of different values.

As it was demonstrated by the research results, any increase in the strength properties of a coal massif led to a linear increase of the forces P_z and P_y and practically did not tell upon the formation of the force P_x.

It is possible to choose the index A of coal resistance to cutting among the indices given in Table I, as an integral index of the coal resistance to the combined breakage with the help of which one can evaluate the strained state of the massif in situ and for which at present there are sufficiently reliable methods to determine it in the section of the coal massif close to the face. The research has proved that as compared to the calculations the minimum variation in the experimental data takes place when the forces P_z and P_y are compared with the conventional breaking strength R_y which simultaneously takes into account both the tensile and compression resistance of the coal massif. Both these processes belong to the combined breakage: compression when the cutter rim top enters the coal massif, and tensile forces to break the pillar along its base due to the tensions. That is why we recommend the index R_y which corresponds to the physical essence of the process as a criterion to evaluate the coal resistance to the combined breakage in the stand conditions, while for the calculation of parameters for the hydromechanical tools of the coal-winning machines we recommend the index A which takes into account the state of the coal massif in situ.

Provided both cutting force and side thrust of the disc cutter are assumed equal to 1.0 during breakage of a massif with the conventional breaking strength R_y =2.11 MPa, then, the coefficients ${}^{K}R_{yz}$ and ${}^{K}R_{yy}$ which take into account respectively the influence of the index R_y on the force P_z (energy consumption H_w) and on the force P_y, can be determined from

$$ {}^{K}R_{yz} = 0.37 + 0.28\ R_y \qquad (2) $$
$$ {}^{K}R_{yy} = 0.44 + 0.26\ R_y \qquad (3) $$

In the same way we obtained the ratios to determine the coefficients K_{Az} and K_{AY} which take into account the index A and its influence on the forces P_z and P_y respectively and characterize their relative change as compared with the standard regime of breakage (A = 1600 N/cm):

$$ K_{Az} = 0.37 + 0.9003\ A \qquad (4) $$

$$ K_{AY} = 0.58 + 0.0002A \qquad (5) $$

DEPENDENCE OF POWER INDICES OF COMBINED BREAKAGE 0N GEOMETRIC PARAMETERS OF DISC CUTTER

Research to determine the influence of the geometric parameters of the disc cutter on the power indices has been carried out by breaking the coal-cement blocks with the resistance to cutting A = 1600 N/cm according to the scheme (see Fig l,e). Cohen determining the influence of the disc cutter sharpening angle on the power indices. We used the disc cutters with an outer diameter D = 200 mm and a sharpening angle δ = 20-40° and when determining the influence of the diameter we used the disc cutters with a diameter D = 140-260 mm and a constant sharpening angle δ= 35°. the experiments have been carried out with different values of the parameters h, t, h_s and a.

It is established that the forces P_z and P_y linearly increase together with the greater angle δ as the contact area of the disc cutter with the pillar which is being broken increases too. The intensity of the forces P_z and P_y increase is approximately the same: the angle δ growing from 20 to 40,° the force P_z increases by 2.2 times while the force P_y increases by 2,3 times. the side thrust P_x for a greater angle δ becomes smaller, because the pillar resistance increases too. This resistance is directed along the axis of the disc cutter and compensates the force which presses the cutter away from the side surface of the part of the coal massif left.

Upon processing the experimental data we have obtained the dependences to deterrmine the coefficients $K_{\delta z}$ $K_{\delta y}$ and $K_{\delta X}$ which allow for the influence of the angle δ upon the forces P_z, P_y and P_x and characterize their relative change as compared with the standard regime ($\delta = 35^0$).

$$K_{\delta z} = K_{\delta y} = 0,029\delta \qquad (6)$$

$$K_{\delta X} = 0.42 + 0.086\delta - 0.002\,\delta^2 \qquad (7)$$

'When the disc cutter rolls along the slot its main part does not contact with the nassif which is being broken. That is why any increase of the disc cutter diameter does not practically exercise any influence upon formation of the forces P_z and P_y and only slightly tells upon the side thrust $P_{x.}$

For example, when D increases from 140 min to 260 mm, i.e. by 1.85 times, the force P_y increases by 20% . The degree of influence of the disc cutter diameter on the force P_y can be allowed for with the help of the coefficient K_{DY}, which characterizes the relative change of this force as compared with the standard regime (D = 200 nmm,):

$$K_{Dy} = 0.62 + 0.002\,D \qquad (8)$$

RATIONALPARAMETERS OF COMBINED BREAKAGE

In order to determine the influence of the parameters h, t, a and h_s on the forces P_Z, P_y and P_X, the energy consumption H_w and the output of coal grades R_{-6} and R_{+25} research has been carried out for the combined breakage of the coal-cement blocks with a resistance to cutting T = 1600 N/cm, according to the scheme (see Fig. 1,f) in the standard regime (Ref. 7). For that purpose we used the disc cutter with diameter of 200 mm and the sharpening angle of 35^0. The research has been carried out according to the planning matrix with a number of factors k = 4 varying within the range: h = 2-6 cm h_s /h = 0.4+0.8, t/h = 1.0+2.6 and a = 220 + 380 mm.

The research results indicate that a deeper slot provides for a smaller contact area of the disc cutter and the massif, and thus, for weakening the massif. That is why, as h_s increases, both these forces on the disc cutter as well as the energy consumption for the mechanical breakage (Fig. 8 and 9) are reduced to reach the minimum value at h_s /h = 0.7+0.8.

For example, for h = 6 cm, t/h = 1.8 and a = 220 mm as the slot becomes deeper from 2.4 to 4.8 cm the force P_Z is reduced by 1.4 times, the force P_y by 1.2 times and P_x by more than 2 times; at the same time the energy consumption H_W is reduced from 1.8 to 0.7 MJ/m^3 or by 2.6 times.

As the slot becomes deeper, besides the formation of coal fines (grade 0-6 mm, within the slot), the content of grade +25 mm increases due to providing favourable conditions for the disc cutter to break the inter-slot pillar into larger fractions. And the research has proved that the dependences of the output of grades 0-6 mm and + 25 mm upon the depth of the slot are of a curved type. For any parameters of breakage as h_s is increasing, the output of grade 0-6 mm, is reduced to reach the minimum for h_s /h = 0.6, while the content of grade +25mm is increasing to reach the maximum for a certain rational value of the ratio h_s /h which depends on the breakage span and is determined according to the expression

$$h_s/h = 0.9 - 0.2\ t/h \qquad (9)$$

It is established that for every depth of the disc cutter thrust for the hydromechanical method of breakage there is a certain rational dependence t/h when breakage requires a minimum energy consumption (see Fig. 10) and produces a sufficient number of large lumps (Fig. 11) and a minimum output of coal fines (Fig. 12). In order to determine this rational dependence t/h we have obtained the expression:

$$t/h = 3.8 - 0.4\ h \qquad (10)$$

When a becomes longer, the forces P_z and P_y, the energy consumption H_W; and the output of coal grade 0-6 mm increase, while that of coal grade +25 mm decreases, and the side thrust P_x does not practically change. While cutting a slot in a coal massif the water jet produces a strain condition due to its pressure in the spot of contact with the inter-slot pillar. As the pillar is broken for a certain depth of shear along the slot, then, it is obvious that the effect of the water jet pressure upon the strain condition of the pillar will be considerable in its shear zone (producing cross-cut fissures) provided the distance a is rather small. That is why the most efficient combined breakage is produced by the minimum distance a which can be permitted by the design of the cutting instrument of the shearer.

The experiments have shown how the main indices of the combined breakage depend upon the depth of the disc cutter thrust (Fig. 13 and 14), and the rational values of this depth to produce but the minimum coal fines 0-6 mm have been determined:

$$h = 6.8 - t/h \qquad (11)$$

as well as that to produce the maximum of +25 mm coal lumps:

$$h = 9.6 - 1.9\ t/h. \qquad (12)$$

CONCLUSIONS

I. There are some schemes of the combined breakage of a coal massif by a high velocity water jet and a disc cutter developed and studied. Their characteristics have been determined, the main being as follows:

(a) When introducing the disc cutter in the slot for a thrust depth which is smaller than the slot depth, a deeper slot leads to no change in the power parameters of the process and in the energy consumption for the mechanical breakage because the inter-slot pillar is broken but not for its entire height;

(b) When introducing the disc cutter in the slot for a thrust depth which is more than the slot depth, the inter-slot pillar is broken practically along its base. And a deeper slot provides for a reduced resistance of the pillar to breakage and a smaller contact area of the disc cutter with the coal massif, thus reducing the disc cutter forces and the specific energy consumption for the mechanical breakage, and all of them reach their minimum at $h_s = (0.7+0.8)\ h$; and

(c) Application of a floating attachment of the disc cutter of a front type makes it possible to avoid the side thrust acting on its axis, and to reduce the cutting and forward forces, and to considerably lower the energy consumption for the breakage.

2. The most efficient scheme of the combined breakage of a coal massif providing for a minimum specific energy consumption for the mechanical and hydraulic breakage and the best coal grade is the scheme when the water jet cuts a slot to form an inter-slot pillar and to shear it towards the recently-exposed side surface with the one-sided wedge rim disc cutter while the thrust depth is more than the depth of the slot cut.

3. The efficiency of a water jet applied together with a disc cutter to break a coal massif depends upon their mutual disposition. Rational is the minimum distance between the axis of the disc cutter and the contact point of the water jet with the massif as far as the power parameters and specific energy consumption are concerned.

4. Besides the depth of the slot and the distance between the water jet and the disc cutter, the parameters of the combined breakage depend also upon the cutting pitch and thrust depth of the disc cutter, its geometric shapes and the coal strength properties.

5. It is established that for the hydromechanical method of breakage the use of the disc cutter with one-sided wedge rim instead of cutting instruments provides for a reduction in specific energy consumption for the mechanical breakage by 1.5-2 times.

6. When the disc cutter is used together with a high velocity water jet to break a coal massif, then, as compared to the mechanical method of breakage with a disc cutter, the forces acting onto the disc cutter are reduced: the rolling force by 1 5-2 times, the forward force by 1.2-1.4 times and the side thrust by 2.7-4.4 times and the specific energy consumption by 1.9-2.2 times. Besides, the coal grade is improved and dust content in the air is reduced to the permissible concentrations. The efficiency of dust suppression is 99.7-99.8 per cent for the hydromechanical method of breakage.

NOMENCLATURE

h = depth of thrust of disc cutter (web width), cm
t = slot cutting pitch, cm
h_s = depth of slot cut by water jet in coal massif, cm
a = distance between disc cutter axis and water jet contact point with coal massif, mm
D = outside diameter of disc cutter wedge rim, mm
δ = sharpening angle of disc cutter wedge rim, degrees
H_W = mechanical breakage specific energy consumption, XJ/m
E_h = hydraulic breakage specific energy consumption, I';J/m)
E_o = total specific energy consumption for hydromechanical breakage, RJ/m3
U_c = speed of cutting (speed of water jet and disc cutter movement), m/s nozzle diameter, mm
P_z = cutting (rolling) force acting onto disc cutter axis during coal massif breakage, kN
P_y = forward force acting onto disc cutter axis during coal massif breakage, kN
P_x = side thrust acting onto disc cutter axis during coal massif breakage,kN
R_{-6} = percentage of coal grade 0-6 mm
R_{+25} = percentage of coal grade +25 mm
Q = specific dust formation, g/t
K = criterion for coal massif resistance to combined breakage
f = coal strength coefficient according to Prof.Protodyakonov scale
σ_{com} = coal resistance to uniaxial compression, MPa
σ_{ten} = coal resistance to uniaxial tension, MPa
$\bar{A}$ = coal resistance to cutting, N/cm
R_y =conventional compression strength of coal, MPa
$^{K}R_{yz}$ = coefficient correcting for force P (energy consumption Hw)depending
$^{K}R_{yy}$ = coefficient correcting for force Pv depending on Ry
$^{K}A_z$ = coefficient correcting for force P (energy consumption Hw) depending A

$^{K}A_{v}$ = coefficient correcting for force P depending on A

$K_{\delta z}$ = coefficient correcting for force P_z (energy consumption H_W) depending on disc cutter angle of sharpening

$K_{\delta y}$ = coefficient correcting for force P_y depending on disc cutter angle of sharpening

$K_{\delta x}$ = coefficient correcting for force P_x depending on disc cutter angle of sharpening

$^{K}D_{y}$ = coefficient correcting for force P_y depending on disc cutter diameter Y

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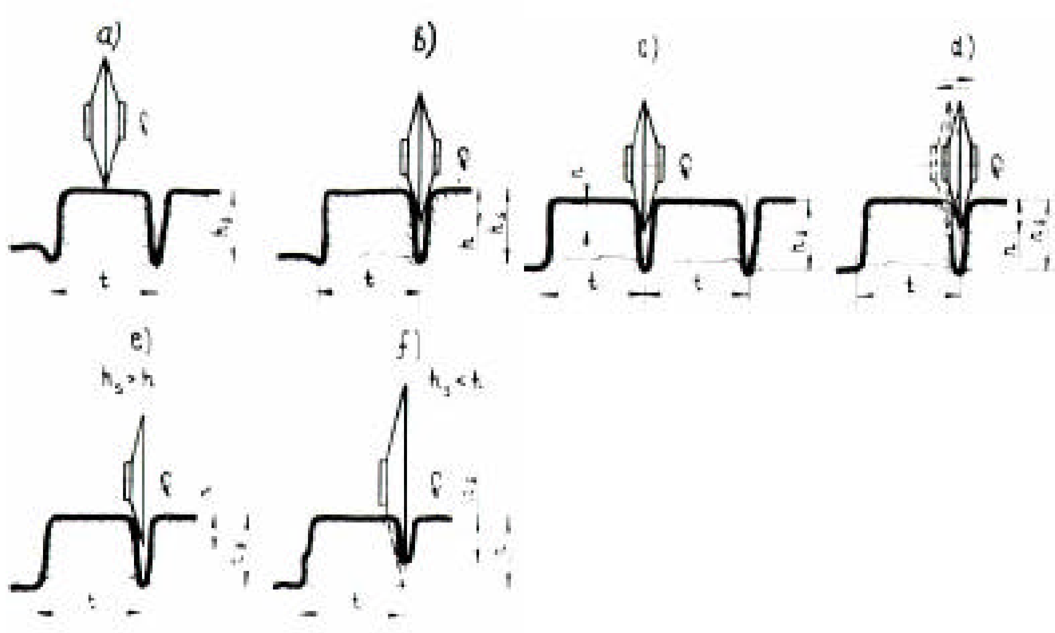


Figure 1. Schemes of combined breakage of a coal massif by a water jet and a disc cutter with a two sided wedge rim (a- front; b – one-sided; c – double-sided; d – with floating disc cutter) and with one sided wedge rim (e, f).

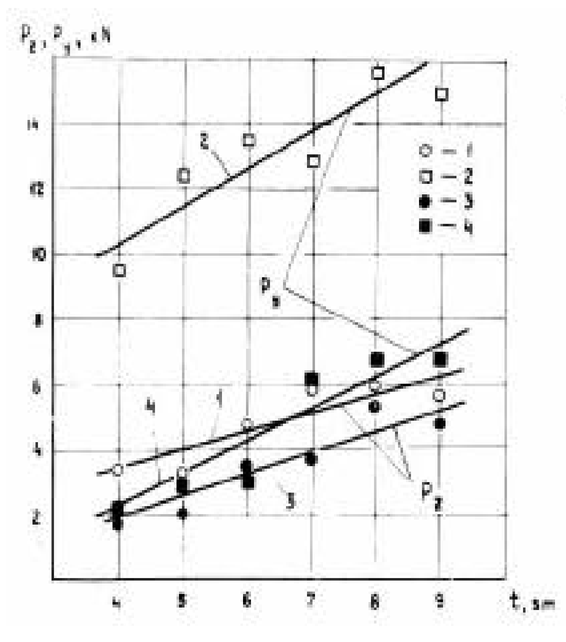


Figure 2. Dependance of rolling force P_z and side thrust P_x on pitch t when breaking a coal massif by a water jet and a disc cutter of rigid (1,2) and floating (3,4) attachment with: h = 4 cm; $h_s/h = 1.1$; a = 300 mm; A = 1600 N/cm; D = 200 mm. And $\delta = 30^o$.

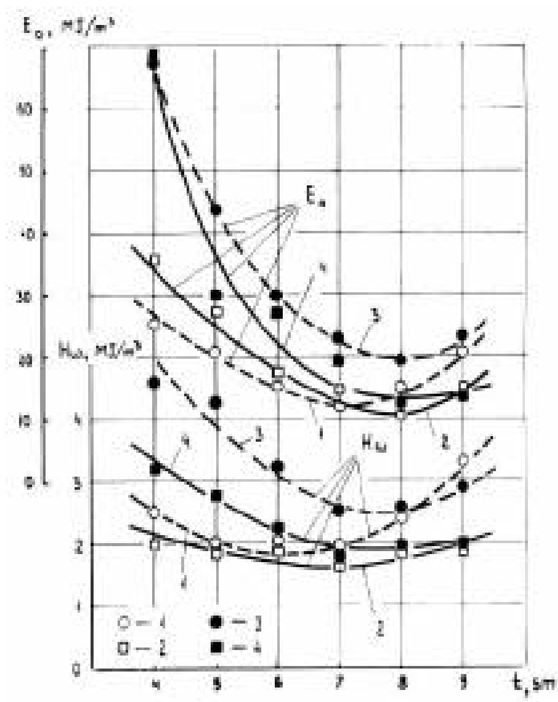


Figure 3. Dependence of specific energy consumption H_w and E_0 on pitch t when breaking a massif by a water jet and a disc cutter of rigid (1 & 3) and a floating (2 & 4) attachment with:
h = 4 cm;(1 & 2) and h = 5 cm (3 & 4); hs/h = 1.1; a = 300 mm, A = 1600 N/cm; D = 200 mm; and $\delta = 30^o$.

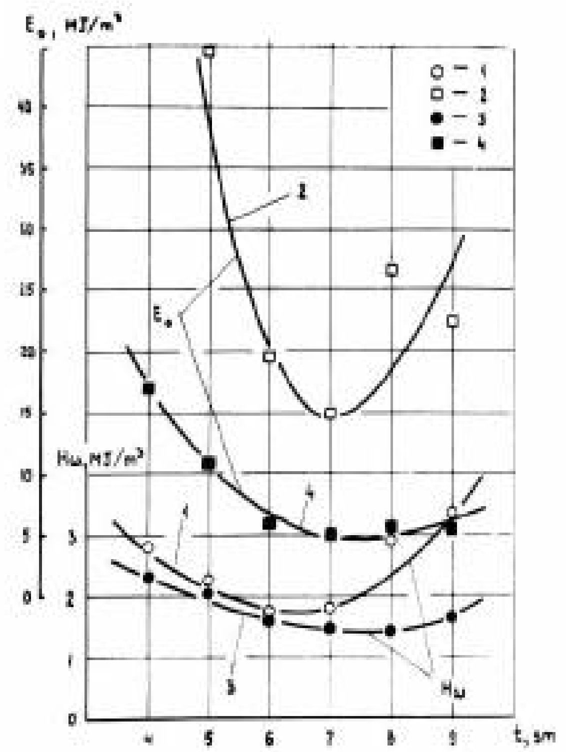


Figure 4. Dependence of specific energy consumption H_w and E_0 on pitch t when breaking a massif by a water jet and a disc cutter with one sided wedge rim with: h = 4 cm; a = 300 mm , A = 1600 N/cm; D = 200 mm; and $\delta = 30^o$.For (1&2) e at $h_s = 1.1$ h; (3 & 4) f at $h_s = 0.6$h.

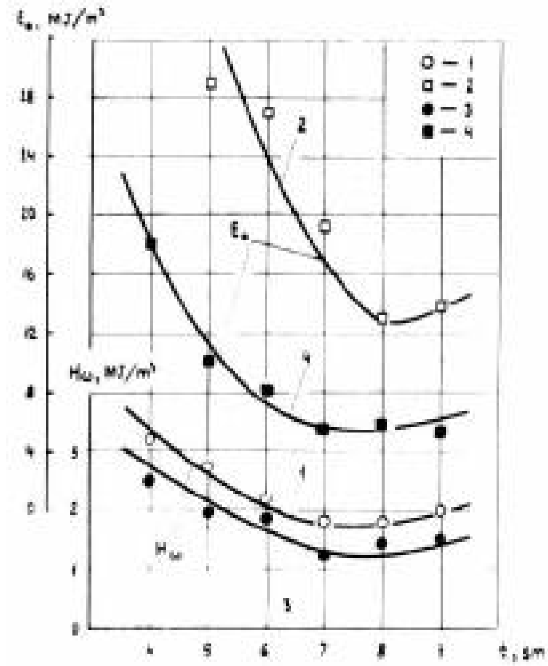
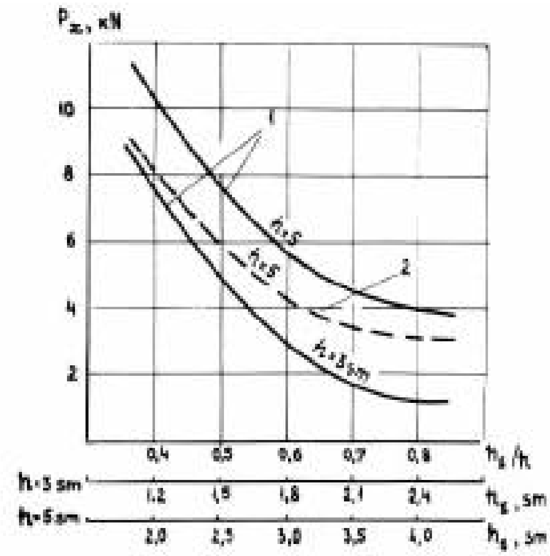


Figure 5. Dependence of specific energy consumption H_w and E_0 on pitch t when breaking a massif by a water jet and a disc cutter with: h = 5 cm; a = 300 mm; , A = 1600 N/cm; D = 200 mm; $\delta = 30^1$ & 2 for the scheme d with floating disc cutter at $h_s = 1.1$ h;
3 & 4 for the scheme f with disc cutter with one-sided wedge rim at $h_s = 0.6$ h.



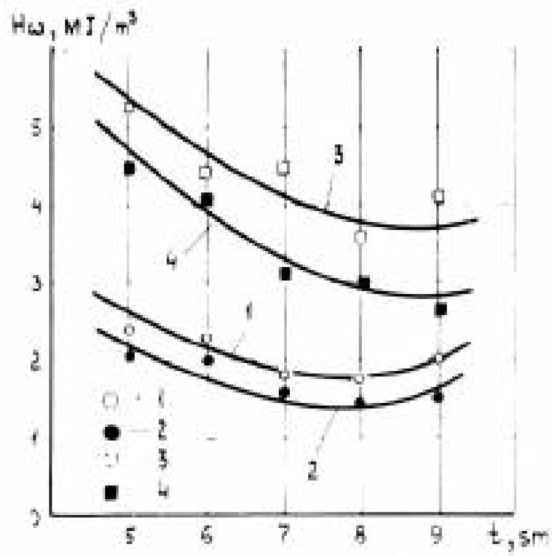


Figure 6. Forces Pz, Py, Px as functions of Pirch t for hydromechanical (1,2 & 3) And mechanicaal (4, 5, & 6) methods of Breakage at : h = 4 cm; a = 300 mm; A = 1600 N/cm; D = 200 mm; $\delta = 35^1$ h_s/h = o.6.

Figure 7. Specific Eenrgy Hw consumption as a funciton of pitch t for hydromechanical (1 & 2) and mechanical (3 & 4) breakage at hs/h = 0.6; a = 300 mm; A = 1600 K/cm; 1 & 3 at h = 4 cm
2 & 4 at h = 6 cm.

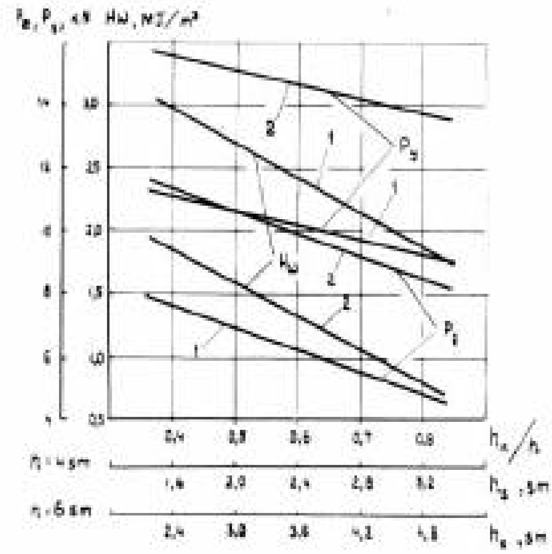


Figure 8 Forces Pz and Py and energy Consumption Hw as a function of slot depth Hs with: t/h = 1.8; a =220 mm; h = 4 cm (I) H = 6 cm (2)

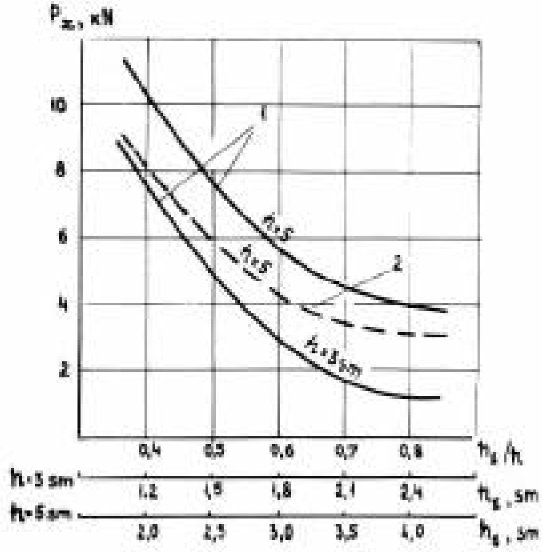


Figure 9. Force Px as a function of slot depth hs (a = 220 mm) at: t/h = 1.4 (I) and t/h = 1.8 (2).

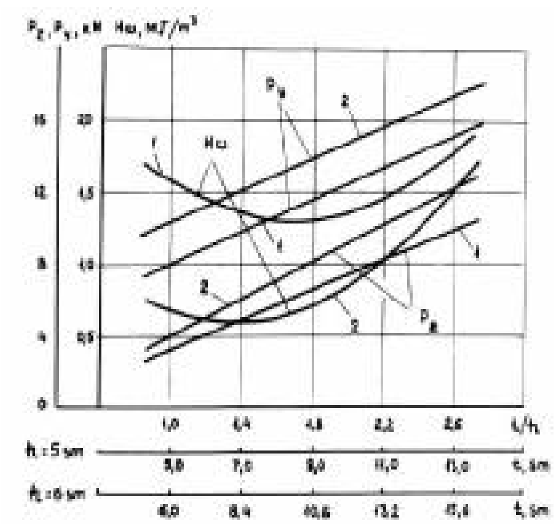


Figure 10. Forces Pz and Py and energy Consumption Hw as functions of pitch t (a = 220 mm; hs/h = 0.8) at (1) h = 5 cm (2) h = 6 cm.

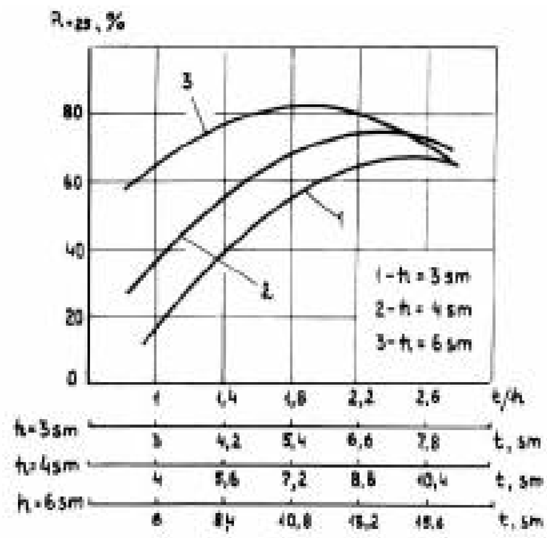


Figure 11. Percentage of grade R $_{+25}$ in the broken off coal as a function of pitch t at: hs/h = 0.6 and a = 220 mm.

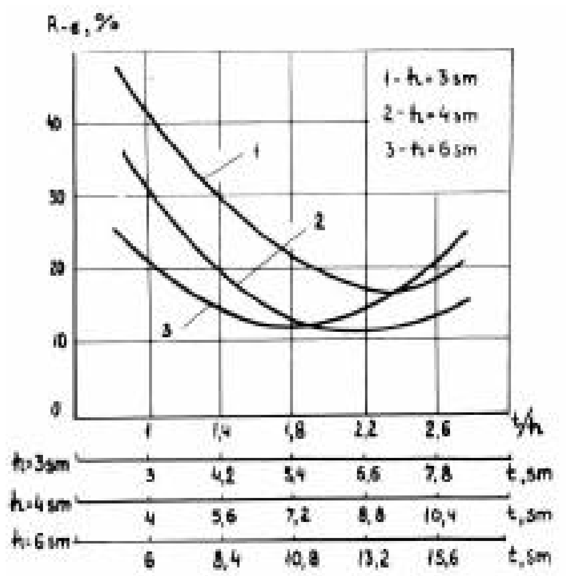


Figure 12. Percentage of grade R –6 in the Broken off coal as a function of pitch t at: hs/h = 0.6 and a = 220 mm.

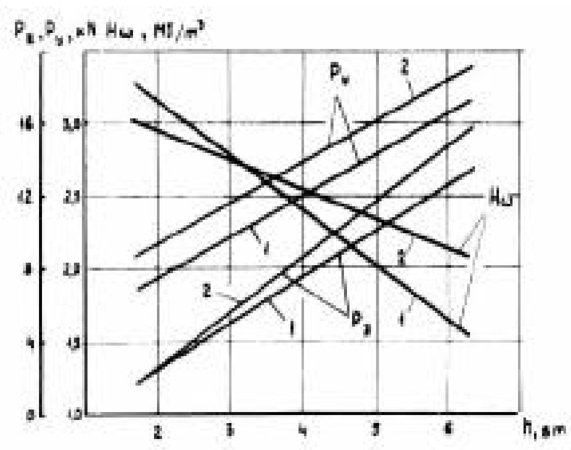


Figure 13. The forces Pz, Py and the energy consumption Hw as functions of the disc Cutter depth of thrust h at: hs/h = 0.6; a = 220 mm 1) t/h = 2.2 (2) t/h = 2.6

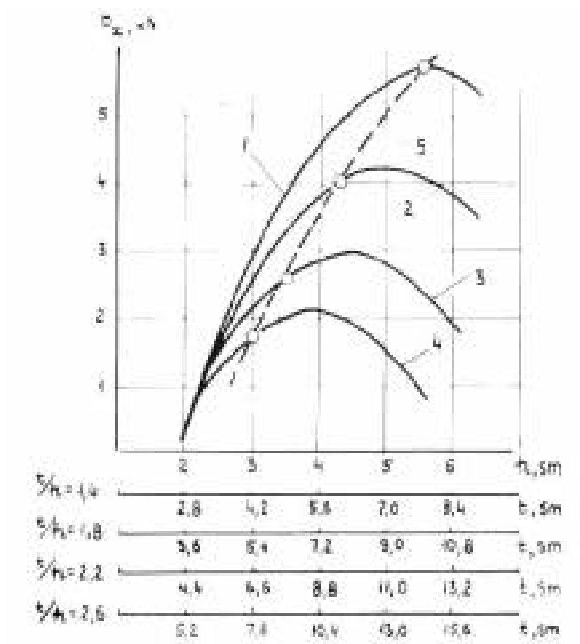


Figure 14. The side thrust Px as a function of the disc cutter depth of thrust h at: hs/h = 0.6; and a = 220 mm; 1) t/h = 1.4; (2) t/h = 1.8; (3) t/h = 2.2; (4) t/h = 2.6 and (5) t/h = 7.8 cm.

DISCUSSION

NAME: George Savanick
COMPANY: Bureau of Mines

QUESTION TO: MIKE HOOD

QUESTION: "There appears to be an inconsistancy between your cumulative curve and your 'sawtooth' force distribution curves. The cumulative curves indicate that cutting with the water jet augmented bit is no more efficient than cutting with bits along but your 'sawtooth' curves of water jet augmented bits are less accentuated than those of unaugmented bits. Would you comment on this apparent contradiction?"

ANSWER: "The cumulative curve that you refer to is the particle size analysis plot of cumulative weight percent finer than a given size fraction vs. particle size (Figure 1 below). You are correct in your observation in that the slope of these curves is a measure of the efficiency of the rock breakage process and, as can be seen from Figure 1, the efficiency of this process is the same for both dry and water jet assisted cuts.

The force:time or force:displacement measurements (Figure 2a and b) on the other hand, as you correctly point out, indicate that water jet assisted cutting is more efficient. The area under these force:displacement curves is a measure of the energy expended during the cutting process. Thus, the smaller this area, the more efficient the cutting operation.

The reason for this apparent contridiction lies in the energy expended by the jets. This component of energy is accounted for in Figure 1 but is not included in Figures 2a and 2bo Unfortunately, because most laboratory tests are conducted at slow cutting speeds, where the fraction of the energy exerted by the jets is large in comparison to the mechanical energy exerted by the bit, it is not possible to qualify meaningful specific energy figures for the over-all cutting process. However, field trials that have been conducted using water jet assistance have shown that the specific energy is decreased when water jets are used (Hood, 1978; Tomlin, 1982). That is, the efficiency of the overall cutting process is increased by the action of the jets. The reason for this is that the rate of rock excavation is increased by more than the increase in the energy required for the jets.

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TOMLIN, M. (1982) "Field Trials with a 10,000 psi Prototype System"; Proceedings of Seminar for Water Jet Assisted Roadheaders for Rock Excavation, sponsored by U. S. DOR/U.K.NCB, Pittsburgh, PA, May 26-27, pp. C1-C11.

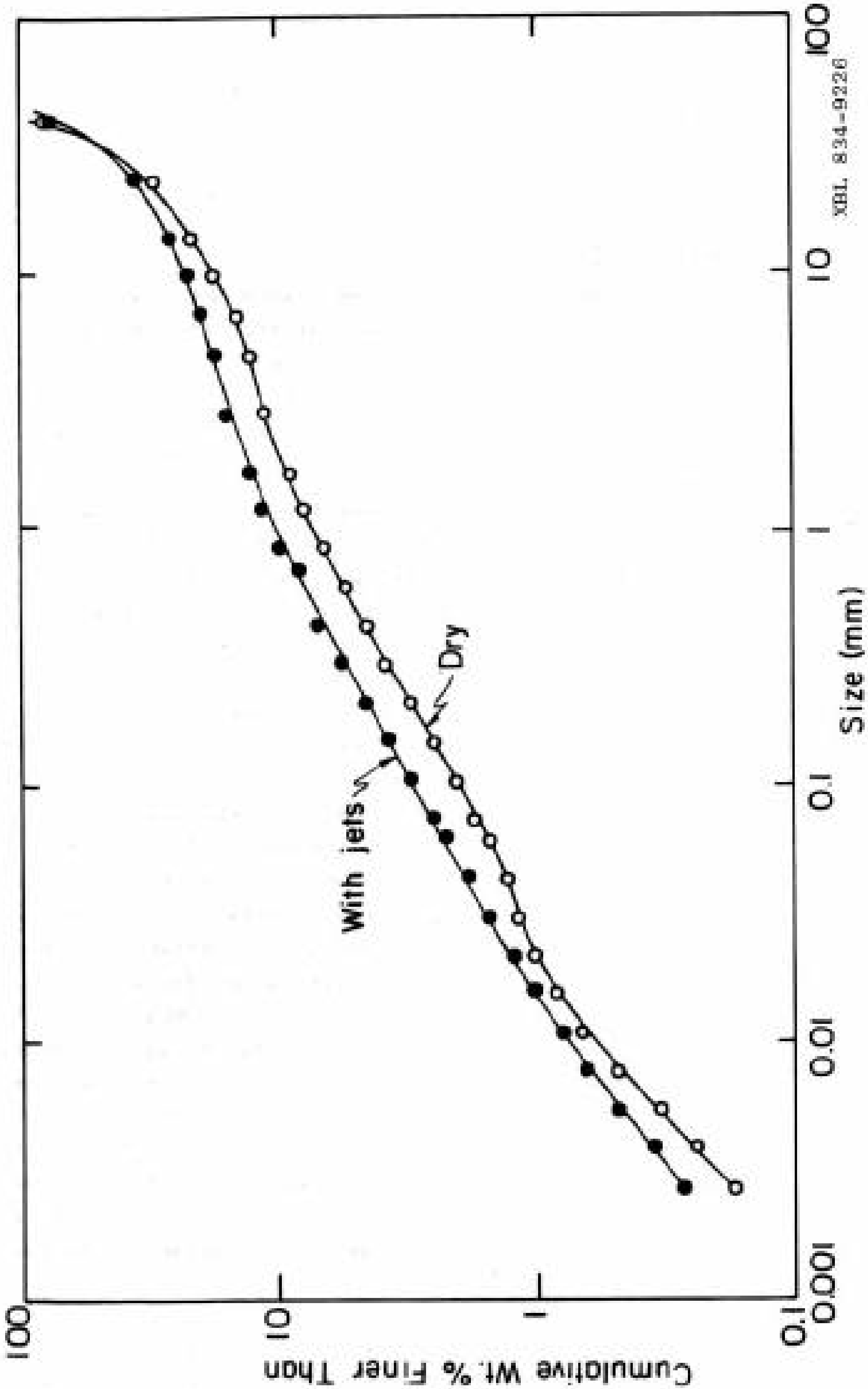

Figure 1. Particle size analysis. The slope of these curves is a measure of the efficiency of the rock breaking process: the steeper the curve is the more efficient the process. Note that the slope of both curves is approximately the same.

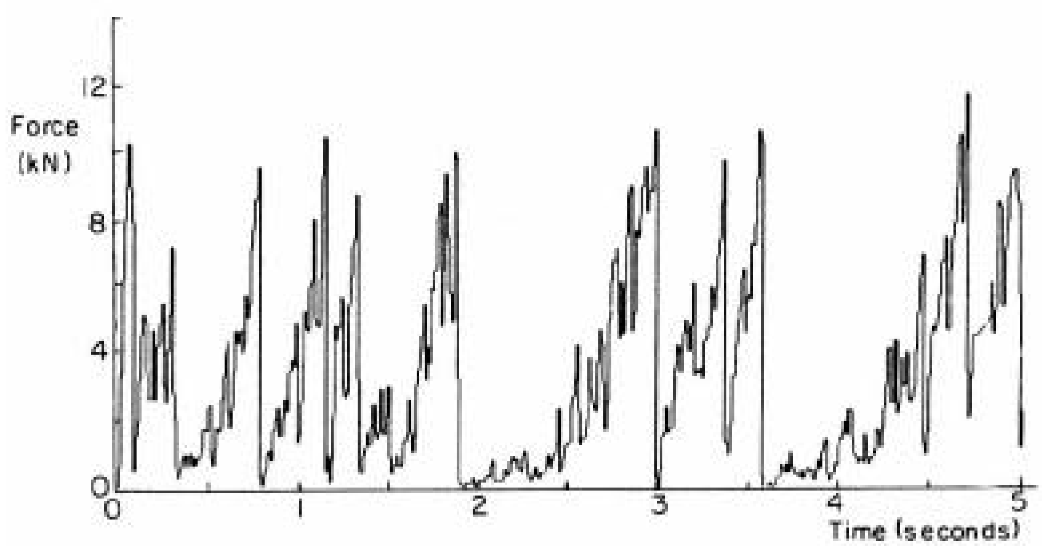

Figure 2 (a)

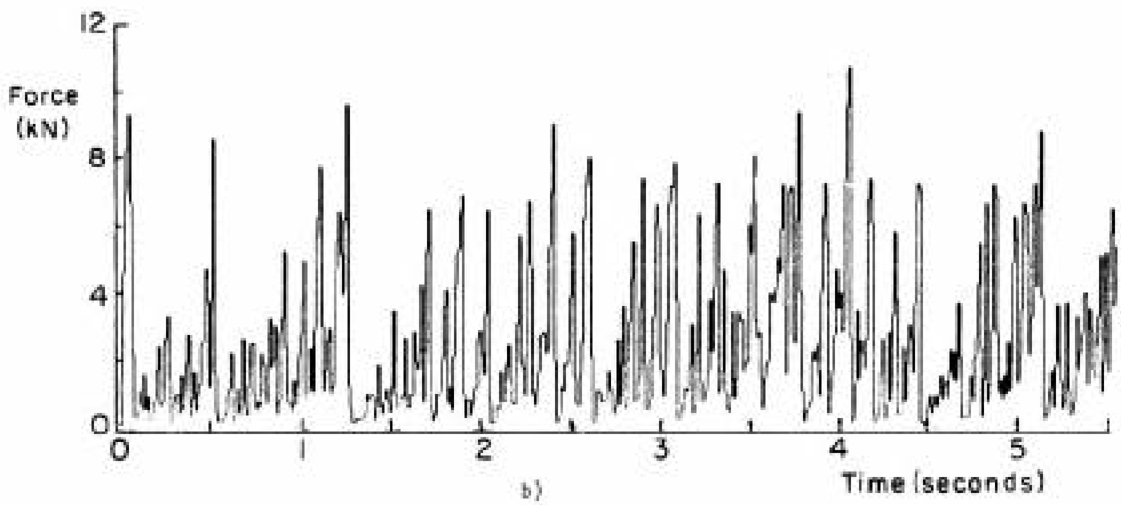

Figure 2. Strip chart recordings of cutting force measurements in Indiana limestone. Depth of cut 15 mm.

a) without water jets, mean force = 4.13 kN; mean peak force = 8.50 kN; Note the sawtooth characteristic of this curve.
b) With a 35 MPa water jet directed immediately ahead of the leading edge of the bit. Mean force = 2.60 kN; mean peak force = 7.11 kN. Note the lack of sawtooth response indicating that the jets result in a smoother cutting action.

NAME: Dr. Henkel
COMPANY: Bergbau-Forshung

QUESTION TO: MIKE HOOD

QUESTION: "Did you get any effect on dust reduction; was it measured?"

ANSWER: Dust particles are those fine particles, generated often during the cutting process, that become entrained in the ventilation air stream. It is known that only a small fraction of these fine particles become so entrained. This particle entrainment process is poorly understood, but it is known that it has much to do with pick velocity and the air flow around the cutting drum which is rotating at high speed. Since all of our laboratory tests were conducted at low speed, and we had no way of determining which particles would have become entrained in the air stream, we made no attempt to measure dust. However, careful measurements of all particles down to a size range of 2µm were taken.

EXPERIMENTAL STUDIES OF CUTTING WITH ABRASIVE WATERJETS

Mohamed Hashish
Flow Industries, Inc.,
Kent, Washington

ABSTRACT

Experiments were conducted to study the cutting of solids with abrasive waterjets. The abrasive-waterjet parameters that were varied include pressure, water flow rate, abrasive flow rate, abrasive particle size, traverse rate, number of passes, relative hardness and standoff distance. The cutting process, observed by taking high-speed movies of Plexiglas being cut with an abrasive waterjet, was found to be cyclic in nature and to consist of two dominant cutting mechanisms along the surface. At the top of the kerf, erosion at shallow impact angles controls the abrasive wear mode, while deformation wear by impacts at large angles takes place further down the kerf. The experiments revealed some of the global cutting process kinematics and helped us to develop a better understanding of site erosion phenomena, while providing direction for further necessary research in these areas.

INTRODUCTION

The mixing of abrasive particles with a stream of high-velocity water to form a slurry cutting jet is a new and promising technology. Materials that can be cut with this jet include metals, hard rock, steel reinforced concrete, high-strength advanced composites, glass and ceramics. The depth and the quality of cut for a given material depend on parameters that include water pressure and flow rate, abrasive particle size and flow rate, mixing chamber dimensions, traverse rate, number of passes and standoff distance. Figure 1 is a schematic of abrasive waterjet nozzles and parameters. Because of the large number of parameters, the task of optimizing the jet's cutting performance is quite difficult. An analytical model describing the relationships among the dependent and independent parameters would be useful to predict and to optimize this cutting performance. The development of such a model requires extensive experimental investigations to:

(1) elucidate the microcutting mechanism,
(2) identify the role of the carrying liquid on the erosion process as it superposes hydrodynamic loading on particle impact,
(3) determine the effect of particle fragmentation upon impact, and
(4) define material properties relevant to the cutting.

These areas of research will lead to an understanding of "site" phenomena of material removal mechanisms. Another area to study and describe is that of "global" cutting process kinematics.

Although a great deal has been learned about erosion by solid particle impact, as summarized by Ruff and Wiederhorn (1979), this knowledge cannot be directly related to abrasive-waterjet cutting. However, similar methods of analysis and experimentation can be used to study abrasive waterjets, especially to understand site erosion mechanisms.

This paper, the result of a relatively limited study, presents observations gained from actual cutting tests rather than simulated tests that have been used in classical erosion investigations. Trends of abrasive-waterjet cutting data are discussed and observations of the cut kerf are presented. This study, of course, is not thorough enough for a complete understanding of the processes involved, but it has already provided direction for further research efforts.

TRENDS OF ABRASIVE-WATERJET CUTTING DATA

In this section, we discuss trends of the experimental data and, in some cases, we extrapolate the trends beyond their experimentally confirmed ranges. We only address qualitative behaviors that cover a wider range of parameters than those already obtained experimentally (Barker and Mazurkiewicz, 1982; Barton and Saunders, 1982; Hashish, 1982a, 1982b, 1983; Saunders, 1982). The trends described are of data from actual cutting tests in different materials along with some previously published data.

Effect of Waterjet Pressure

Experimental observations (Hashish, 1982a, 1982b, 1983; Saunders, 1982) of randomly selected conditions show that the effect of waterjet pressure on the depth of cut is approximately linear, as shown in Figure 2. Also, a critical pressure, P_c, has been observed to vary in a narrow range (~ 1,000 psi) for a given material. By analyzing the mixing process, this critical pressure can be related to a critical particle velocity, which is a material erosion characteristic. If the material could be cut with a plain waterjet, the critical pressure would be independent of abrasive and mixing parameters. Both the critical particle velocity concept and the threshold pressure for cutting with plain waterjets have been confirmed experimentally in many studies on erosion mechanics (Ruff and Wiederhorn, 1979) and on waterjet cutting (Crow, 1973). However, there has been no agreement on the correlation of this critical pressure to material properties.

When a critical pressure has been identified for a given material, an optimum working pressure can be derived for the maximum efficiency of power consumption. If the depth of cut, h, is expressed as $K_l(P-P_c)$ and the power, E, as $K_2P^{1.5}$, where K_1 and K_2 are constants for a given h-versus-P line and depend on other parameters, the critical pressure that satisifies $\delta h/\delta E = O$ is $3P_c$. This value, however, may be low for deep penetration, unless high water and abrasive flow rates are used, as can be predicted from Figure 2. A typical critical pressure value for mild steel is between 3000 and 4000 psi (Hashish, 1983), which has also been confirmed by Saunders (1982).

Effect of Waterjet Nozzle Diameter

Increasing the water flow rate increases the depth of cut. When this is accomplished by enlarging the waterjet nozzle diameter, d_n, a trend results, as shown in Figure 3. Initially, the depth of cut, h, varies linearly with d_n, but further increases in d_n will be associated with only slight increases in h. For two different abrasive flow rates, m_l and m_2, as shown in Figure 3, there exists a nozzle diameter, d_e, at which depth-of-cut results are equal. That happens because, for the lower abrasive mass flow rate, m_l, the abrasive particle velocities are higher and the number of impacts less, respectively, than they are for the higher abrasive flow rate, m_2. For the nozzle diameter, d_e the effect of reducing the velocity is counterbalanced with an increase in the number of impacts.

Just as there is a critical water pressure, a critical nozzle diameter exists at which very little, if any, cutting occurs. Our experimental data (Hashish, 1983) were limited to the linear range and did not extend to critical nozzle diameters. Extrapolation of this data, however, indicates that there are critical diameters at which the water momentum transferred to the abrasives will not be enough to deliver them with velocities higher than the critical particle velocity required for cutting.

On the other side of the curve (Figure 3) where nozzle diameters and water flow rates are large, the rate of increase in the depth of cut will be low and eventually no increase will be observed, i.e., $\delta h/\delta n = 0$. This will define another critical nozzle diameter, which can also be predicted theoretically. The maximum possible abrasive particle velocity in this range can be obtained from

$$V = \frac{V_j}{(1+R)} \qquad \text{where } R = \frac{m}{m_w}$$

For a high water flow rate, m_w (i.e., when large diameters are used) compared with the abrasive flow rate, m, the loading ratio, R. approaches zero and the particle velocity V approaches a maximum possible value nearly equal to the waterjet velocity, V_j, assuming that, for simplification, we ignore mixing losses. If the lower portions of the curves in Figure 3 are assumed to continue linearly to h = 0, the depth of cut can be written as $h \propto (d_n - d_c)$. The optimum nozzle diameter satisfying $\delta h/\delta E = O$. where $E \propto d_n^2$, will be $2d_C$.

Effect of Traverse Rate

Increasing the traverse rate will generally reduce the depth of cut (Figure 4). However, the rate of kerf area generation, h_u, will show a peak (Hashish, 1982a, 1982b, 1983) at a critical traverse rate, u_c. Experimental observations suggest that the relationship between h and u can be approximated as

$$h = h_{\max}\left[1 - e^{\frac{-\beta}{(\mu - \mu_0)}}\right] \qquad u \geq u_0$$

where u_o is the traverse rate below which no increase in depth of cut occurs, and h_{max} is the maximum possible depth of cut achieved when u is equal to u_O. The parameter β is a function of other abrasive-waterjet variables.

An equation similar to the one above has been developed (Hashish and Duplessis, 1978) for plain waterjet cutting. The critical traverse rate, u_s, at which the h_u curve peaks can then be determined using the above equation. However, when we observed the cutting process in transparent materials, we found this to be an oversimplification of a complex behavior (Hashish, 1983). There is more than one cutting mechanism along the kerf that affects the depth of cut to a degree that varies with the traverse rate. This is likely to produce a complex function for h-versus-u relationships other than the one mentioned above. The effect of the traverse rate on shifting the mechanism of cutting is a concept that has also been discussed (Hashish, 1981) for plain waterjets. For high traverse rates, the abrasive flow rate, as a variable in the parameter β, does not significantly affect cutting results. For these high traverse rates, the abrasive waterjet performance approaches plain waterjet performances, especially for nonmetallic materials. The steadiness of the traverse has been observed to affect cutting results significantly because of the nature of the cutting process, which continues to develop a kerf in a cyclic manner. An unsteady traverse will interrupt the cyclic sequence of kerf development, resulting in rough cuts.

Effect of Abrasive Particle Size

For a given material, or group of materials, there exists a range of optimum abrasive particle sizes that produce deeper cuts. This range is wider for brittle materials, such as glass, than for ductile materials, such as metals (Figure 5). Results, however, have been confusing in terms of identifying a general trend for the shape of depth-of-cut-versus-particle-size curves. A decreasing h-versus-d_p curve, with d_p as the average abrasive particle diameter, may be convex or concave for the same material being cut, depending on other parameters. The role of particle size in the area of erosion mechanics needs to be identified (Ruff and Wiederhorn, 1979). No studies exist that provide an explanation of the optimum particle size. However, in abrasive-waterjet cutting, this optimum particle size could be related to the efficiency of the mixing process, if the material removal rate is directly proportional to the abrasive flow rate, as classical erosion theories and experiments indicate. In other words, for a given mixing chamber configuration, there exists an optimum particle size for the maximum transfer of the momentum of the water to abrasives. Consequently, changing the particle size would require a different mixing chamber, so that comparisons of the effects of particle size could be more accurate and relate only to the material being cut.

<u>Effect of Abrasive Flow Rate</u>

The approximately linear, initial relationship between the depth of cut and the abrasive flow rate has been observed to be as shown in Figure 6. However, additional data (Hashish, 1983) showed a trend for decreasing depths of cut at higher abrasive flow rates beyond a critical one. This trend is expected because, with an increase in the abrasive flow rate for a fixed water flow rate, particle velocities will decrease faster than the number of impacts will increase. This critical abrasive flow rate, at which a maximum depth of cut occurs, is generally less than the water flow rate, which is less than 1 for a loading ratio, R. However, for ideal conditions, it can be shown mathematically that this critical abrasive flow rate occurs when the loading ratio equals 1. Based on simple erosion theories (Fannie, 1958), the depth of cut can be expressed as proportional to the particles' kinetic power mV^2, where V is the particle's exit velocity and can be related to the waterjet velocity, V_j, as V ~ Vj/(l + R), leading to

$$h = m\left[\frac{V_j}{(1+R)}\right]^2$$

The condition δh/δR =0 occurs when R equals 1, as can be determined from the above equation. The discrepancy between experimental and theoretical values for the critical value of R is attributed to mixing losses, which result in particle velocities less than what the simple momentum equation yields. Also, the use of the simplified erosion equation ($h \propto V^2$) may be inaccurate.

Notice that the lines in Figure 6 pass through the origin, as no depth will be obtained for zero abrasive flow rate. However, this is true only for hard materials that cannot be cut with a plain waterjet using a nozzle as shown in Figure 1. Research is needed to determine whether the rate of impacts per unit time or the total number of impacts is more directly related to the cutting process. If the former proves to be the case, then a material fatigue property must be among the material parameters that affect cutting. Erosion theories (Crow, 1973) show a relationship between material removal and the total number of impacts, but the effect of the rate of impacts was not discussed. Also, research is needed to determine whether the abrasive particles are reentrained in the jet after initial impacts at the top of the kerf to cause further cutting at lower portions of the kerf.

Effect of Number of Passes

Depending on the traverse rate, trends for the effect of the number of passes on the depth of cut will differ, as shown in Figure 7. For each of the dashed lines in Figure 7, when u/N = constant, the total jetting time is equal. The top dashed line shows that the depth of cut can have a maximum value at a certain combination of traverse rate and number of passes. The bottom dashed line indicates that single-pass cutting will always be the most efficient for a certain required depth of cut. The middle dashed line shows that the number of passes does not increase or decrease the cutting depth for the same amounts of hydraulic energy and abrasives consumed. It is important to select traverse rates carefully so that such a plot as Figure 7 can be generated and used for optimization.

The experimental data in Hashish (1982a, 1982b, 1983) and Barton and Saunders (1982) show trends similar to those discussed above. For high traverse rates, equal depths result with every pass, at least for the passes at the start of cutting. For low traverse rates, the depth of cut obtained in the second or third pass is much shallower than that resulting in the first pass, although $\delta h/\delta N$ may seem higher than for high traverse rates. Mathematically, $(l/h_l)(\delta h/\delta N)$ will always be higher for high traverse rates, where h_i is the depth of cut obtained in the first pass. Most observations of plain waterjet multipass cutting showed that cutting results always improve at a high number of passes, which is not true with abrasive waterjets. The existence of an optimum combination of the number of passes with the traverse rate depends on the abrasive flow rate to a great extent. For high abrasive flow rates, a single pass is more efficient, and multipasses are better for low abrasive flow rates at u/N = constant.

The lower curve in Figure 7 shows an initial positive $(\delta h/\delta N)$ behavior in contrast to the negative $(\delta h/\delta N)$ that has commonly been observed for both abrasive-waterjet and plain-waterjet cutting. This peculiar behavior occurs when the mixing process is inefficient, producing incoherent jets. The kerf obtained in the first pass with these jets acts to focus the jet for the second pass, and again for the third and any subsequent passes. This in situ focusing process raises the rate of increase of the depth of cut with the number of passes until it is counteracted by the actual standoff distance. Data supporting this trend are found in Hashish (1982) and Saunders (1982). This phenomenon can provide a means of judging the mixing effectiveness of nozzles and can help to optimize the parameters of nozzle configurations.

Effect of Standoff Distance

The data reported in Hashish (1982b) and Saunders (1982) indicate a linear trend for a decreasing depth of cut as the standoff distance decreases. However, we have recently observed that within a change of up to 1 in. in the standoff distance, no decrease in the depth of cut results (Figure 8). This happens only when the mixing process is efficient. With multiple waterjet nozzles, results are most sensitive to standoff distances, especially when no mixing chambers are used. With these nozzles at a zero standoff distance, the jets merge together. Negative standoff distances are generally associated with dramatic reductions in depths of cut when compared with results obtained when the target is lower than the merging point, as shown by the dashed line in Figure 8. For material volume removal results, the effect of the standoff distance is dramatic, especially for brittle materials. A one-hundred-fold increase in volume removal can be obtained in glass if the standoff distance is changed from 0.25 in. to 5 in. For ductile metals, such a change in standoff distance will change the abrasive waterjet from a cutting tool to a cleaning tool.

Effect of Abrasive Hardness, Shape and Fragmentation

Figure 9 shows the trend for the relationship between the depth of cut and the relative hardness number, Hr. which is defined as the ratio of abrasive particle to material hardness, H_a and H_m, respectively. The trend indicates that no further improvement in depth of cut can be obtained when the hardness ratio exceeds a critical value. We have observed that, at the same parameters, aluminum can be cut with softer abrasives, such as silica sand, as well as with harder abrasives, such as garnet. Because the cost of abrasives increases with their hardness, it is important to determine the critical hardness number, H_r. at which $\delta h/\delta H_r = 0$. However, changing abrasive materials also changes other characteristics, such as the specific weight and shape of particles. The effect of particle hardness alone on cutting results requires additional research.

Round particles have proved to be less effective than sharp-edged particles for both brittle and ductile materials. Cutting with glass beads (Figure l0a) produces up to 70% shallower depths in both ductile and brittle materials than does cutting with sharp silica sand particles (Figure l0b). Both have similar hardness values. For ductile materials, sharp particles help to remove material by plowing and raising lips (Finnie, 1958), while, for brittle materials, sharp particles induce more stress concentrations than round particles.

We have observed that particles fragment during the cutting process, depending on the relative hardness between the abrasive and target materials. however, abrasives harder than the target material also fragment. When cutting steel or aluminum, silica sand with a mesh number of 70 breaks down to a mesh number of 100. No quantitative data are available yet to characterize the percentage of abrasive particles that break or to what degree these particles fragment. Silicon carbide abrasives showed minimum fragmentation when cutting steel and were used again after drying with similar results. The design of an abrasive recycling system will depend on particle fragmentation data, as recycling may not be feasible if excessive fragmentation occurs. Also, to understand the cutting mechanism, it is important to identify the role of particle fragmentation in producing erosion in excess of that obtained by the initial impact.

OBSERVATIONS OF THE CUT KERF

Visualization of the cutting process was accomplished by taking high-speed movies of the cutting of transparent samples of Plexiglas, as shown in Figure 11. Figure 12 shows cutting surface contours at different intervals with the cutting process divided into two regions. In the first, the entry region, the jet travels a certain distance before reaching the maximum depth, which is equal to $h_i + h_2 + h_3$, where h_i, h_2 and h_3 represent depths in which the amount of material removed varies as described below. A steady depth, h_i, occurs primarily by the jet's leading "edge." Particles strike that surface at shallow angles of impact causing cutting wear to control the mechanism of material removal (Ruff and Wiederhorn, 1979, and Finnie, 1958). Over the cutting surface corresponding to h_i, the material removal rate equals the material displacement rate. Because the material removal rates at deeper distances from the jet are less than the material removal rates at upper locations, the curvature of the cutting surface increases with the depth. At the end of the depth h_i, particle deflection causes the cutting wear mechanism to terminate.

A small traversal distance creates a step at the end of that depth, which is immediately subjected to particle impacts at large angles of attack, bringing about the deformation cutting mechanism (Bitter, 1963) which then controls the cutting action in depth h_2. This step is removed while another, wider one is formed. This process continues until the jet becomes ineffective in removing material. The jet at this location, h_3, deflects in an increasing upward direction, which results in further penetration because of the increase in the rate of change of water momentum which imparts higher hydrodynamic loading on the particles that, consequently, remove additional material.

Further jet traversal will engage the jet in the second region cutting pattern, as shown in Figure 12, for one cycle of penetration. This approximately equals the jet diameter. During this process, a rough bottom will result that has also been observed for other materials (Saunders, 1982). When the jet reaches the exit edge and is approximately half way off the material, it will deflect to the other side of the material causing a contour (a) in Figure 12. The uncut portion, shown in the same figure, results from curvature of the cutting surface and jet exit behavior.

The photographs shown in Figure 13 were taken of Plexiglas cut by an abrasive waterjet and clearly show the step formation and progression. Observations of the jet exit flow during the cutting of relatively thin materials showed the steadiness of the angle of deflection at the sample bottom surface, γ_2, in Figure 14a. This steadiness supports the previous description of the cutting process when the sample thickness is equal to or less than the depth of the cutting wear mode, hi. The complete penetration of relatively thick samples showed that the jet exit flow oscillates backward and forward, as shown in Figure 14b. At intervals the jet cannot be seen exiting the bottom surface, but it gradually reappears with a decreasing angle, γ_3 (Figure 14b). This again supports the previous description of the cutting process during the penetration from h_i to h_2. Material thickness, t, in this case can be expressed as $h_2 > t > h_l$. If the sample depth is less than $h_i + h_2 + h_3$ but larger than $h_i + h_2$, as the level A indicated in Figure 12, the bottom of the sample will show holes, rather than a continuous cut as shown in the lower sketch of Figure 12.

The picture in Figure 15 is of the bottom surface of an aluminum sample showing that phenomenon. The angle γ_1 at the top of the cutting surface (Figure 14) is more dependent on target material properties than abrasive waterjet and cutting parameters. Relating this angle to material properties is a key requirement for the development of a cutting model (Hashish, 1983). Careful experimentation needs to be conducted to determine the sensitivity of this angle to abrasive waterjet parameters.

Figure 16 shows that the smoothness of the cut varies with its depth, changing from a smooth cut at the top of the kerf to a straight one for the rest of the depth. This variation can be related to the different mechanisms of cutting in h_i and h_2 described earlier. However, an additional factor related to the effective jet diameter may be more relevant to this observation. As the jet penetrates a material, its effective diameter becomes smaller, due to wall friction and a reduction in abrasive impacts caused by rebounding and jet deflection. The width of the cut will consequently be smaller as the depth increases, as shown in Figure 17. Because the jet is round, similar narrowing occurs in the third dimension perpendicular to the directions of both the traverse and the depth.

In some situations, especially in cutting nonmetallic or relatively soft materials, an opposite phenomenon occurs in which the kerf widens with an increase in depth, at least to a certain point, because the jet spreads but remains effective over its increased diameter. Pictures in Barton and Saunders (1982) of concrete cutting support similar observations, as shown in Figure 18.

CONCLUSIONS

The trends of the effect of different abrasive waterjet parameters on depth-of-cut results can be summarized as follows:

Parameter	Trends of depth-of-cut data
Pressure	- Approximately linear - A minimum critical pressure exists
Waterjet nozzle	-Initially linear diameter -A minimum critical diameter exists
Traverse rate	- Complex function - An optimum traverse rate exists for maximum rate of kerf area generation
Abrasive particle	- Requires additional research size - There is an optimum size range for different target materials
Abrasive flow rate	- Initially linear - High flow rates may reduce cutting depth
Number of passes	- Sometimes multipass cutting is more efficient than single-pass cutting
Standoff distance	- Large standoff distances generally reduce depth of cut - Approximately linear
Abrasive hardness	- Increasing abrasive hardness beyond a certain limit does not improve results

Visualization of the cutting interface in transparent materials has revealed some of the global kinematic processes associated with kerf development. The cutting process has been found to be cyclic and consists basically of two cutting mechanisms. The first is cutting wear by the impact of particles at shallow angles, while the second is deformation wear by the impact of particles on a step at large angles.

Additional research is needed to further elucidate both the site phenomena of the mechanics of cutting and the mixing process between abrasives and high velocity waterjets.

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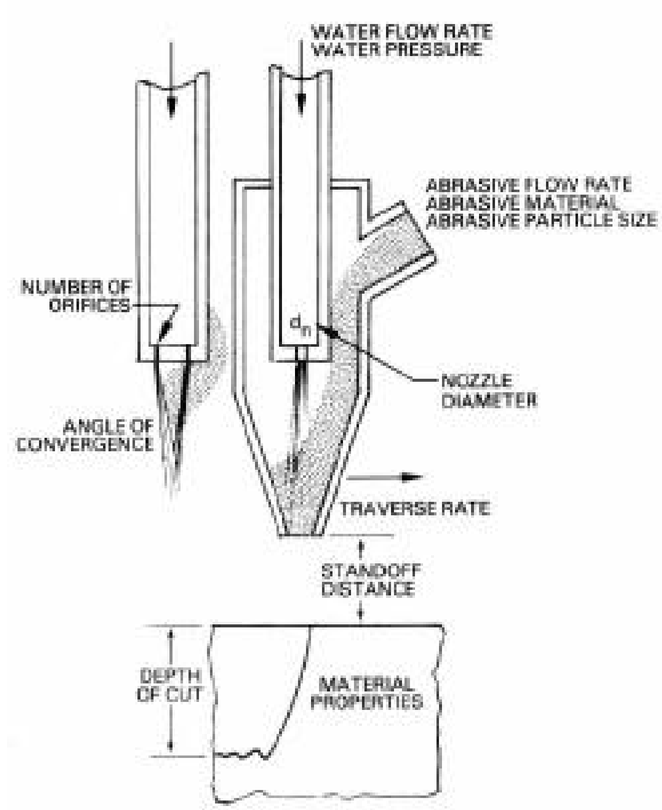


Figure 1. Abrasive waterjet nozzles and parameters.

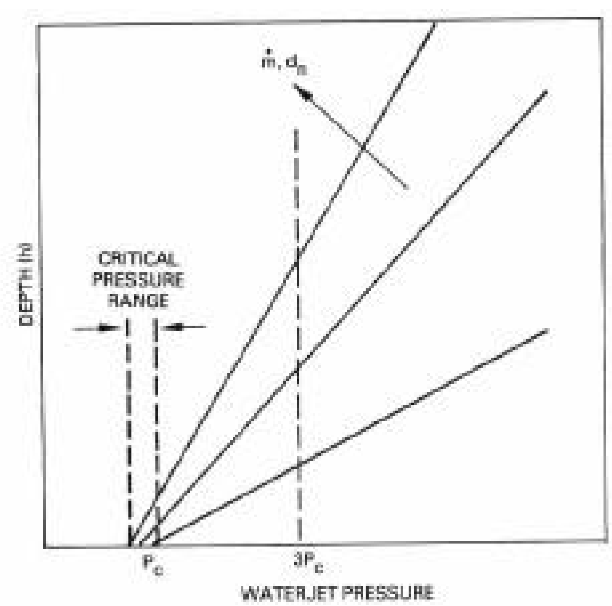


Figure 2. Effect of Pressure on depth of cut.

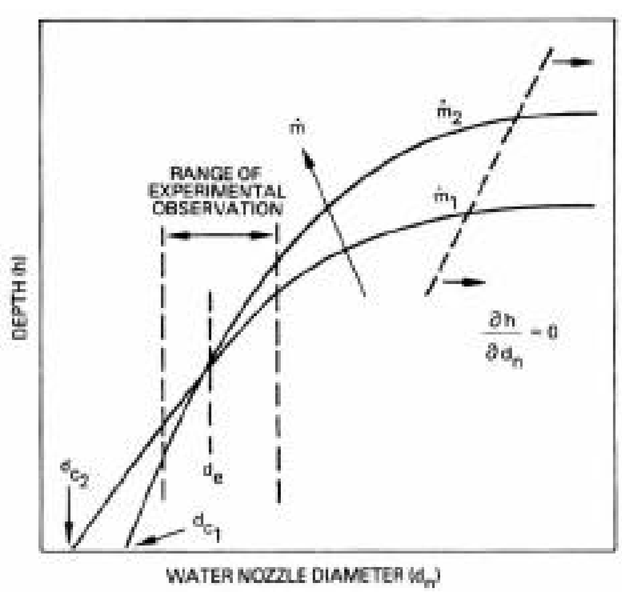
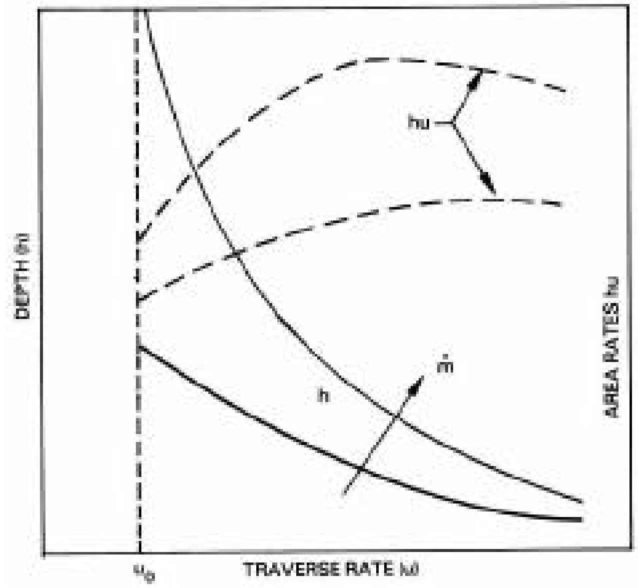


Figure 3. Effect of waterjet nozzle diameter on depth of cut.



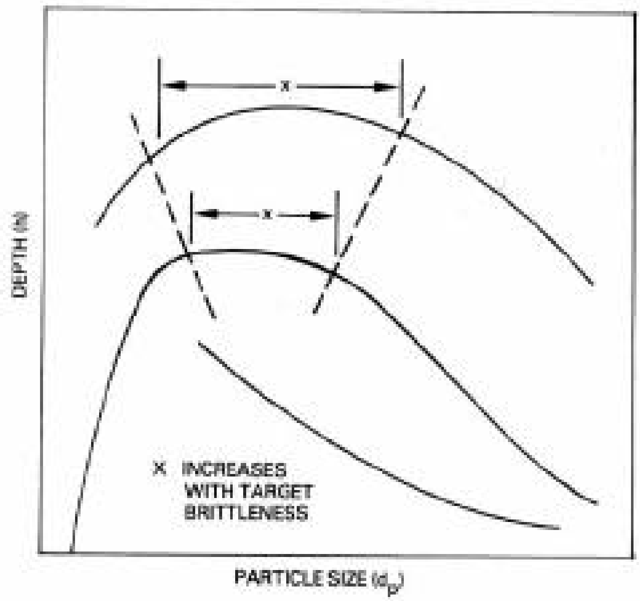
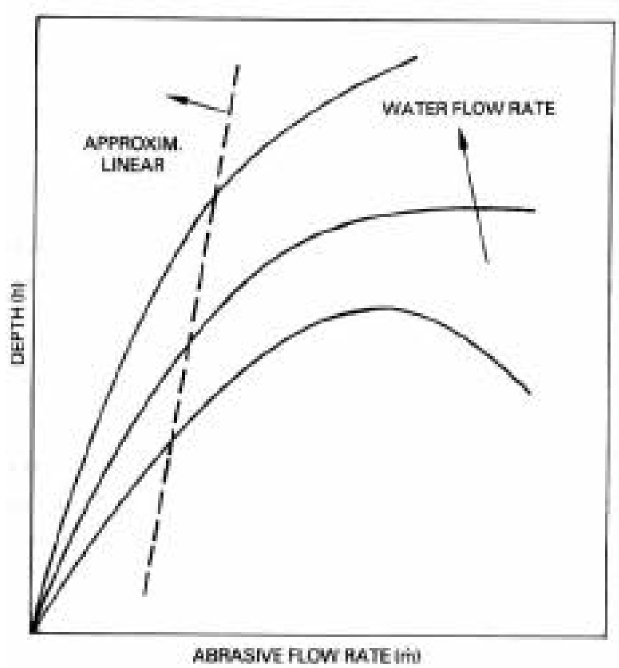


Figure 4. Effect of Traverse rate on depth of cut.

Figure 5. Effect of abrasive particle size on depth of cut.



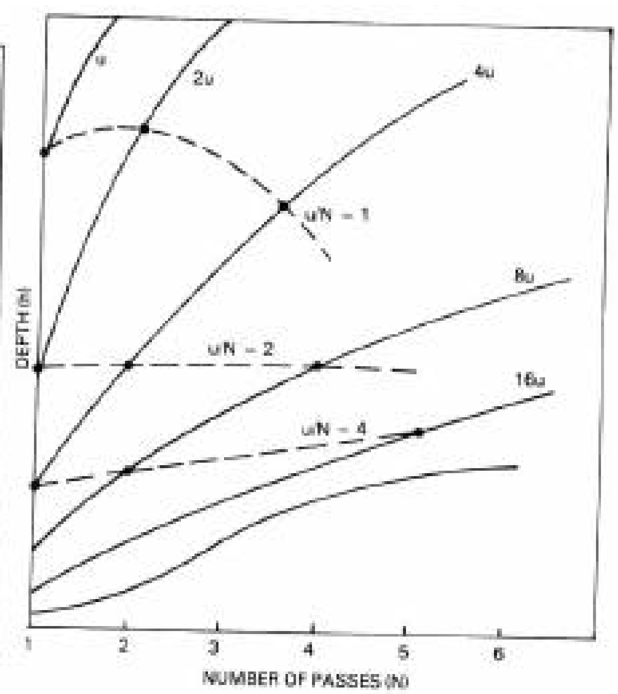
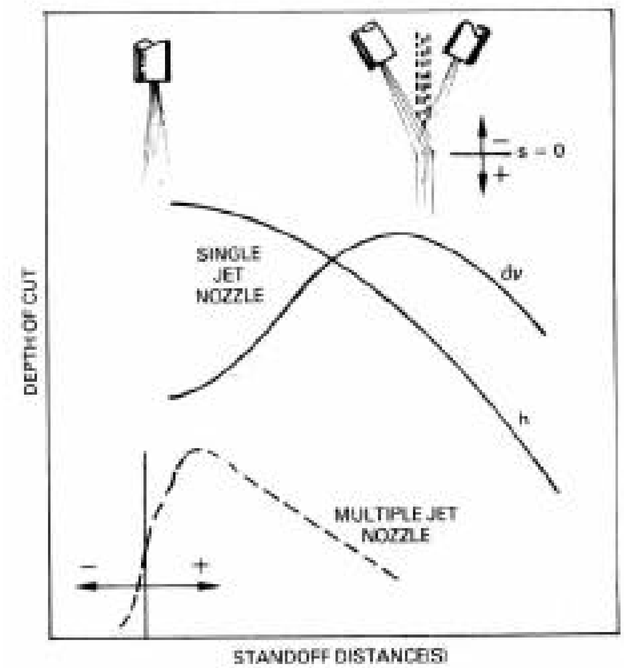


Figure 6. Effect of abrasive flow rate on of cut.

Figure 7. Effect of number of passes on depth of depth cut,



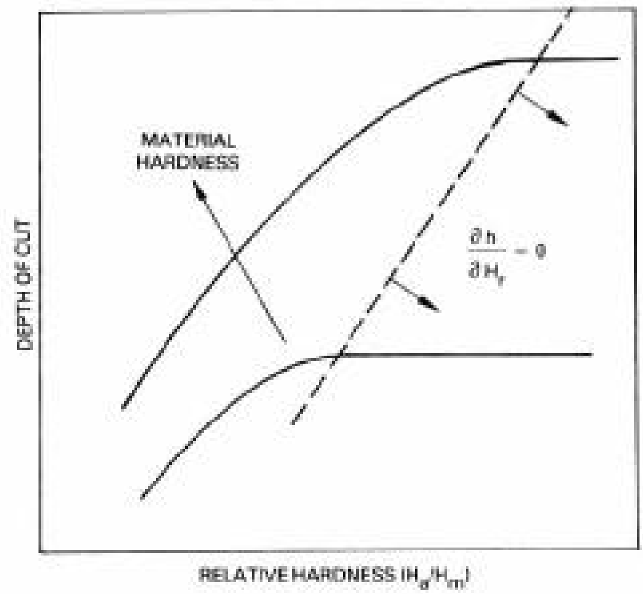
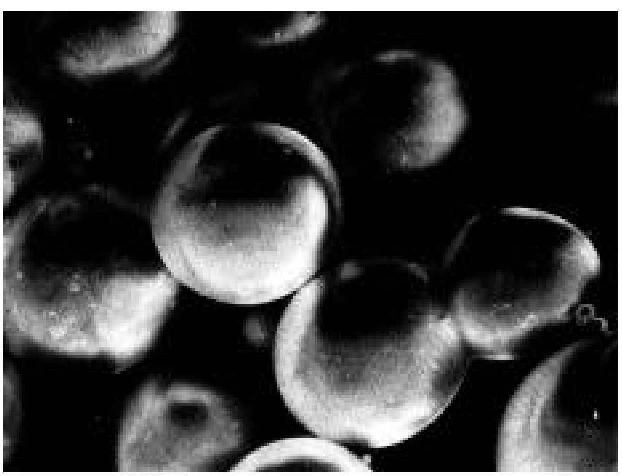


Figure 8. Effect of standoff distance on depth of cut.

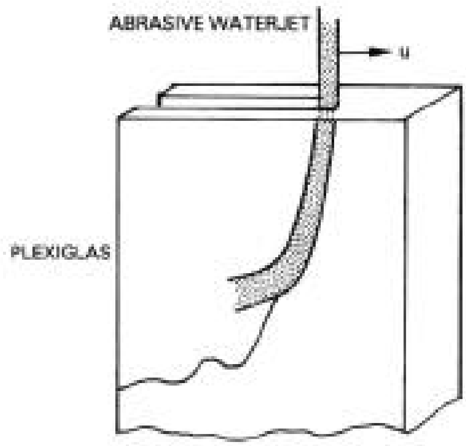
Figure 9. Effect of relative hardness on depth of cut.



a) Glass Beads

b) Silica sand.

Figure 10. Abrasive particles with different shapes.



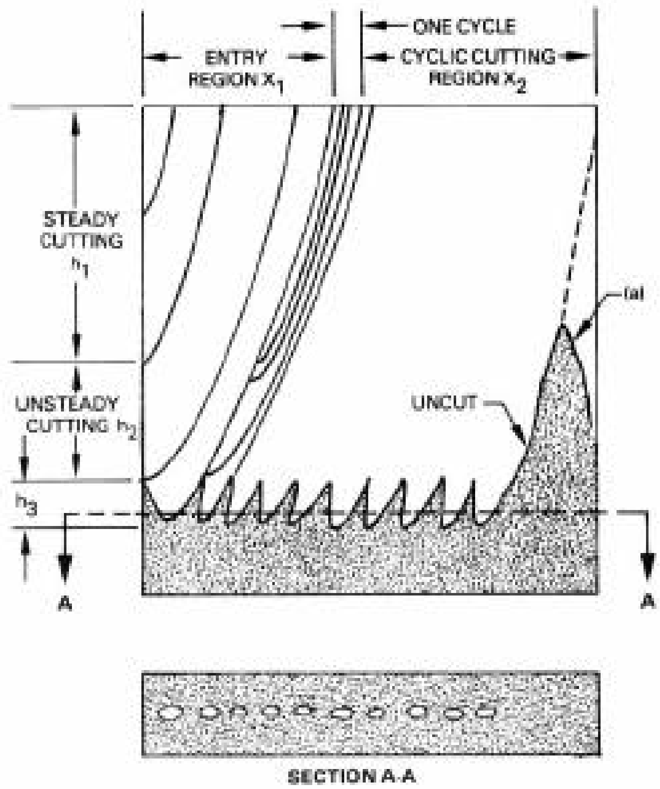


Figure 11.Visualization of cutting in Plexiglas.

Figure 12. Contours of the cutting surface at different intervals.

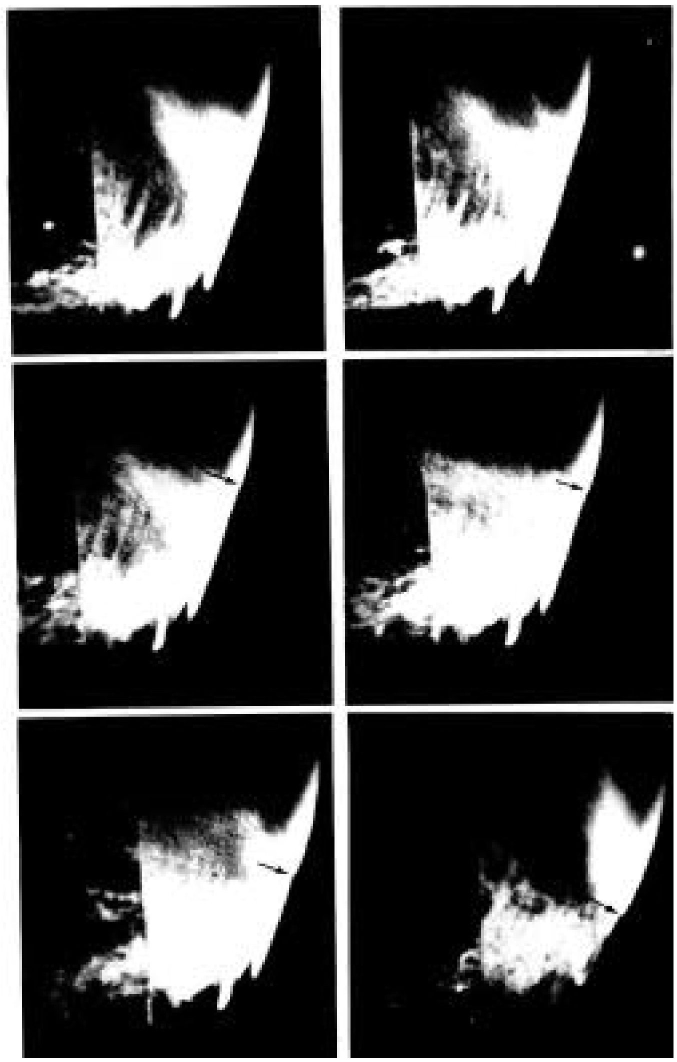
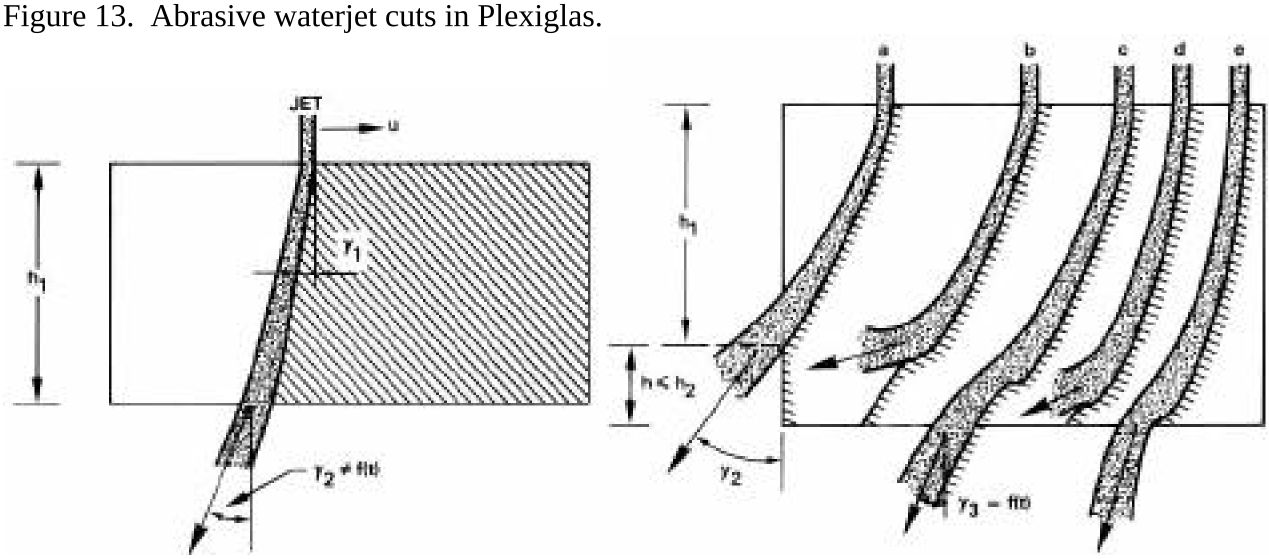
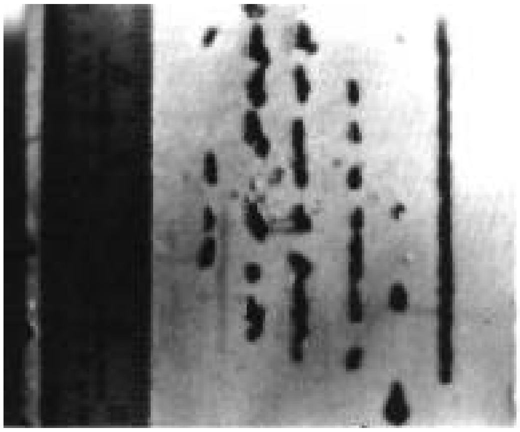


Figure 13. Abrasive waterjet cuts in Plexiglas.



a) b)
Figure 14. Jet deflection characteristics.



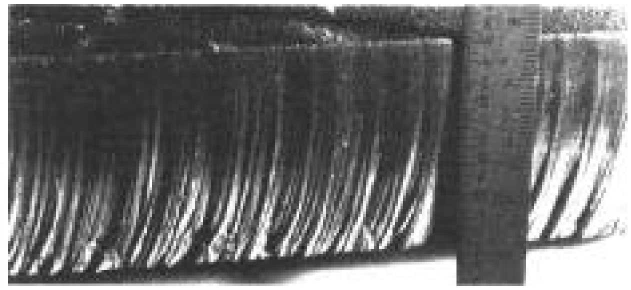


Figure 15. Bottom surface penetration Pattern for relatively thick material,

Figure 16. Variation in kerf smoothness with depth of cut.

Figure 17 Reduced width in depth of cut.

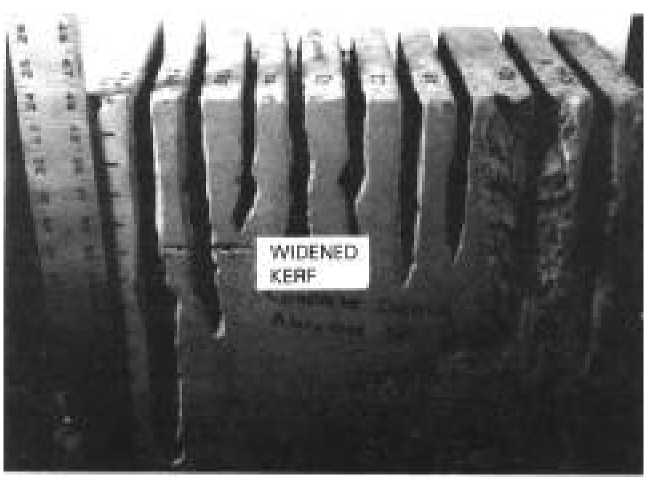


Figure 18 Effects of jet spreading on kerf widening.

CUTTING WITH ABRASIVE WATERJETS

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ABSTRACT

The ability of an abrasive waterjet to cut advanced materials is unmatched by current mechanical or thermal cutting methods. Abrasive waterjets can be used on titanium, ceramics, metallic honeycomb structures, glass, Kevlar, graphite and bonding compounds without delamination or inducement of thermal or deformation stresses along the cutting path. Processors of high-technology materials in industries such as aerospace, defense, naval, nuclear, automotive, and oil and gas could realize immediate benefits from commercial introduction of this novel cutting concept. In this paper, data are presented regarding performance, operating costs and the current hardware status of abrasivewaterjet cutting.

1. INTRODUCTION

High-pressure waterjets have been accepted as standard cutting tools in a wide variety of industries. Most nonmetals can be cut rapidly and efficiently with a waterjet. Cutting applications include paper, cloth, wood, plastics, fiberglass and rocks of medium hardness.

The advantages offered by waterjet cutting include the following:

- Minimal or no dust
- High cutting speed
- Multidirectional cutting capability
- No dulling of the cutting tool (the jet)
- No thermal or deformation stresses
- No fire hazards associated with the cutting

There are also some shortcomings of waterjet cutting systems:

- Hard materials, such as metals, ceramics, high-strength composites, and hard rocks, cannot be cut.
- High power levels are needed for reasonable cutting rates in many applications.
- Delamination or striation occasionally result in some materials.

To increase waterjet cutting capabilities, abrasives were introduced to form the abrasive-waterjet system. This greatly increased the cutting capability of waterjets and improved the cutting performance (Hashish, 1982a, 1982b, and 1982c). The advantages of abrasive-waterjet cutting systems include those mentioned above for waterjet cutting and the following:

- Any material can be cut.
- Power requirements are decreased.
- Delamination does not occur and striation is reduced.

The abrasive waterjet is easily adaptable to existing waterjet cutting systems and automation equipment. It can be readily adapted to remote control operations.

2. PRINCIPLE OF ABRASIVE WATERJETS

The heart of the abrasive-waterjet cutting system is a small abrasive-jet nozzle, as shown in Figure 1. Water is pressurized up to 60 ksi and expelled through a sapphire nozzle to form a coherent, highvelocity jet. The waterjet and a stream of solid abrasives are introduced into the specially shaped abrasive-jet nozzle from separate feedports. Here, part of the waterjet's momentum is transferred to the abrasives, whose velocities rapidly increase.

The momentum transfer between the waterjet and the abrasives is a complex phenomenon consisting of several components. One of the mechanists by which this occurs is associated with the limited dynamic stability of the high-pressure waterjet. The initially coherent waterjet breaks into droplets that accelerate the solid particles. A second mechanism corresponds to hydrodynamic drag forces imposed by the water phase on the solid particles.

As a result of momentum transfer between water and abrasives, a focused, high-velocity stream of abrasives exits the accelerator nozzle and performs the cutting action. Cutting or controlled depth penetration of the target material occurs as a result of erosion, shearing, failure under rapidly changing localized stress fields, or micro-machining effects, depending upon the specific properties of the material being cut (Hashish, 1982c). The cutting rate can be controlled by adjusting the feed rate, the standoff distance, the waterjet pressure, or the abrasive parameters.

3. ABRASIVE SYSTEM COMPONENTS

The key components of an abrasive-jet cutting system are the high-pressure pump, the waterjet, the abrasive feed system and the abrasive-jet nozzle. Secondary components include the abrasive and water catcher and accessories, such as swivels, hoses and control valves.

3.1 High-Pressure Pumps

A pressure range of 25 to 45 ksi has been found to be effective in cutting even the hardest materials with abrasive waterjets. Water flow rates up to 3 gpm are typical for operations. Pumps that offer these pressures and flow rates are well within the present state of the art. Figure 2 shows Flow Industries' high-pressure pumps for field and factory applications, which use single or dual intensifiers with full electronic control and pressure-compensating circuits. These pumps are driven by a 75-hp motor. The typical operating lifetimes and maintenance parameters for critical pump components are:

Component	Typical Lifetime	Service Time
High-Pressure Seal	250-750 hr	15 min
Check Valve	500-1000 hr	30 min

3.2 Waterjet

Figure 3 is a photograph of a high-speed waterjet. The stagnation pressure of the jet is up to 45 ksi and its speed is approximately 2500 ft/sec. The jet is formed with a sapphire orifice. Common diameters for cutting applications range between 0.003 and 0.020 in. The orifices have a typical lifetime of 250 to 500 hr and a replacement time of about 5 min.

3.3 Abrasive Feed Systems

For abrasive-waterjet cutting, precise control of the abrasive flow rate is an essential requirement, a feature that is not offered by commercial systems. Therefore, Flow Industries developed an abrasive feed system that provides both steadiness and precision of abrasive flow rate. Dry abrasives are fed into a collector just after the waterjet is formed. An orifice is used to control the abrasive flow rate. The waterjet and abrasive-jet nozzle act as an injector pump to accelerate the abrasives. For feeding abrasives in underwater applications, a pressurized, sealed tank may be required. Feeding dry abrasives over long distances needs further development.

The slurrying of abrasives shows promise for better control of their flow characteristics but introduces the need for a separate slurrying circuit. Because slurries result in less efficient mixing of the abrasives with the waterjet in the mixing chamber due to the additional mass of the carrier fluid, higher pumping flow rates for waterjets, along with increases in power, will be required. More development of slurried abrasives is necessary before this approach can match performances obtained with dry abrasive feed systems.

3.4 Abrasive-Jet Nozzles

The abrasive-waterjet cutting nozzle should provide (1) an efficient mixing of the abrasives with the water, (2) a coherent, focused abrasive stream at the nozzle exit, and (3) a reasonable service life cycle. Two basic concepts have been tested that show great promise for commercialization.

The first concept, which has been tested extensively, is shown in Figure la. High-pressure water enters the center of a mixing chamber into which abrasive particles are fed from side ports. The water entrains the particles and accelerates them to high velocities. The shape and material of this mixing chamber are critical in determining both its wear rate and the jet's performance. Typical commercial nozzle materials are tungsten carbide and boron carbide. For optimum performance, the configuration of the mixing chamber may be different for different applications.

Another nozzle design, which has been tested with encouraging results, utilizes multiple waterjets that converge to form a single stream. The abrasives are fed into the core of that stream, Figure lb. The prime advantage of this design is the increased service life of the mixing chamber or, possibly, the complete elimination of that chamber.

The angle of convergence and the mixing chamber shape are critical parameters. Investigations with multiple jet nozzles for limited water flow rates of up to 2 gpm showed that nozzle performance greatly depends upon machining accuracy to produce perfectly symmetric jets. Extensive research will be required before an optimum nozzle of this type is fully developed for commercial use.

3.5 Abrasive and Water Catcher Systems

The development of catching systems to collect the abrasives and water is of prime operational importance. The catcher design depends on whether the cutting system is for a (1) factory application with stationary nozzles, (2) factory application with moving nozzles, (3) field application with through cutting, as in cutting through walls or ship hulls, or (4) field application without through cutting, as in cutting concrete for roads.

A variety of catching systems is required. Laboratory investigations have shown that, for stationary jets, a concept similar to commercial waterjet catchers can be effective by using ceramic liners and a collecting tank for the abrasives. This system also suppresses the noise effectively. For moving nozzles, a catcher has been designed and is satisfactorily performing in our lab with a commercial optical tracing table. This catcher is basically a large collecting tank. Abrasives settle at the bottom of the tank to be disposed of periodically. For field applications, a vacuum-type catcher can be adapted with a shroud surrounding the jet to catch rebounding water and abrasives.

3.6 Supporting Accessories

Swivels and hoses are examples of accessories that are necessary for a complete cutting system. High-pressure swivels (35 ksi) rotating at 600 rpm are becoming standard commercial items with a minimum lifetime of 60 hours. These swivels are used in a wide range of applications, either as flexible joints or for continuous rotation. When used as flexible joints, their lifetimes may be extended to several hundreds of hours. Flexible hoses with a working pressure of 35 ksi are now standard items for high-pressure drilling and cleaning systems. An optimum pressure range for abrasive-jet cutting is well within the safe ranges of these hoses.

4. PERFORMANCE OF ABRASIVE WATERJETS

A discussion of the cutting capabilities of abrasive waterjets is presented in this section along with data on materials that have been cut.

4.1 Parameters of Abrasive-Waterjet Cutting

There are many independent parameters that affect the cutting performance of abrasive waterjets.

Hydraulic parameters
- Waterjet orifice diameter
- Supply pressure

Abrasive parameters
- Material (density, hardness, shape)
- Size
- Flow rate
- Feed method (force feed or suction)
- Abrasive condition (dry or slurry)

Mixing nozzle parameters
*Mixing chamber dimensions
- Nozzle material

Cutting parameters
- Traverse rate
- Number of passes
- Standoff distance
- Angle of cutting

Material to be cut

Because of the large number of influencing parameters, the optimization of cutting performance requires extensive cutting data for each application. The results presented in the next section are representative data but not necessarily optimum for a given material.

4.2 Abrasive-Jet Cutting Results

Table 1 lists some materials that have been cut successfully with abrasive waterjets. The maximum depth-of-cut limit for these materials has not been determined since penetration was always achieved. Abrasive-waterjet cutting performance results for several materials are presented next.

Concrete and reinforced concrete. Figure 4 shows a plot of the effect of the number of passes on the depth of cut (Hashish, 1982b). The top slope of the curves shown suggests that very thick concrete can be cut without allowing the nozzle to enter the kerf. Recent results with improved nozzles indicate that only 4 to 5 lb/min of abrasives are needed to match the results in Figure 4 that were obtained with 9 lb/min.

A concrete slab 14 in. thick has been cut through at a rate of approximately 1 in/nin. Reinforced concrete, 13 in. thick with 0.75-in.-diameter rebars spaced in the concrete at 3 in. and 7 in. from the top surface, has been cut at rates slightly higher than 1 in/min, with a jet stream of 50 hhp and less than 3 gpm of water. The cuts produced are of high quality, as aggregates are effectively cut rather than knocked loose (Figure 5a). Figure 5b shows the nonspalling characteristics of the cutting process.

Steel. Stainless, mild, tool and special alloy steels in different shapes, such as plates, bars, beams, tubes, and corrugated structures, have been cut; some examples are shown in Figure 6. The cutting results for mild steel (Hashish, 1982b) have been improved and extended to include other types of steel. Typical cutting results with a 45-hp jet and 3 lb/min of abrasives with the jet pressure limited to 30 ksi are given below.

Figure 6 shows examples of kerf shapes. Kerf widths observed are between 0.06 and 0.09 in. Inspection of the cut surfaces shows that the top 60 to 70 percent of the kerf is smooth, with some striation visible below that range. It should be noted that no embedded abrasives were observed when samples were viewed using an electron microscope.

It is expected that further system and nozzle refinements will improve the quality of the surfaces produced, thus keeping, if necessary, subsequent surface finishing to a minimum. As the cutting process is "cold," no thermal stresses affect the material. Comparisons of surfaces produced by abrasive waterjets and flame jets clearly indicate the superiority of the former. Figure 7 shows an oil well casing section with a 16-in.-diameter that was cut at 1 in/min and 30 ksi.

Aluminum. Abrasive waterjets can cut aluminum blocks 5 in. thick in a single pass at 1 in/min. A 0.5-in.-thick plate can be cut at 24 in/min and result in a high-quality surface. The hydraulic horsepower used for these cuts is about 20 hp. Silica sand, an inexpensive abrasive material, was used at flow rates below 3 lb/min. Also, cutting aluminum with abrasives results in only a small fraction of particle fragmentation, thus allowing recycling of these abrasives. Beveled cuts for welding are easy to produce. Figure 8 shows sample cuts in aluminum.

Other metals. Abrasive-waterjet cutting was tested briefly on hard metals and alloys such as titanium, tungsten carbide and amorphous metal, and on soft metals such as copper, copper alloys and lead. Titanium was cut as easily as steel and resulted in smooth cuts. Sandwiched honeycomb titanium structural material used in the aircraft industry was cut well at rates up to 30 in/min for 0.32-in.-thick plates. Figure 9 shows the abrasive jet cut produced. Cutting tungsten carbide requires hard abrasives and generally higher pressures. Holes can be drilled in 3/4-in.-thick tungsten carbide blanks in 30 seconds. When amorphous metal was cut, minor chipping occurred. It is believed that this may be eliminated by the proper selection of abrasive-jet parameters.

Glass. Contour cutting in glass was accomplished with power levels under 10 hp and fractions of 1 lb/min of abrasives. A l-in.-thick glass plate was cut at 20 in/min and produced a fine surface free from cracks. Twenty-five layers of 1/ 16-in.-thick plates were cut at 6 in/min. Figure 10 shows a piece of laminated glass cut by an abrasive jet. Pyrex tubes and crucibles may be cut by rotating the samples under the jet to obtain homogeneity along the cut. Careful selection of parameters, however, was found essential to avoid the graying effect that can occur due to abrasives that may be on the outside of the jet. Ceramic sheets of alumina, fire brick and building tiles, usually in thin plate forms (less than 0.5 in.), were cut without chipping and with smooth cut surfaces.

Advanced composites. A 1.5-in.-thick armor plate made of Kevlar, fiberglass and steel wire mesh was cut at speeds up to 12 in/min with a 22-hp jet, producing a smooth cut without delaminations (Figure 11). This example illustrates the abrasive waterjet's unique capability of cutting multi-material composites of high strength. A thick section of Plexiglas did not show brittle cracks when cut with abrasives. Graphite samples up to 2 in. thick were cut without fraying or delamination. Sections of 3-in.-thick phenolic were cut at a rate of 24 in/min with a jet at less than 25 hp. Figure 11 shows cuts in different samples of such materials.

Rocks. Basalt and granite are among the hardest rocks known. A 12-in.-thick piece of granite with a compressive strength of more than 45 ksi was cut in a single pass. Figure 12a shows the shape of the kerf obtained. An example of kerfs in basalt is shown in Figure 12b. Dimension stones 3-in.-thick can be cut without need for subsequent finishing operations.

Marble and quartzite in 2- to 5-in.-thick slabs require only a 25-hp jet to be cut at a rate of 15 in/min. Figure 12c shows an example of the high surface quality produced in a sample of babeete quartzite. Thick sections of softer rocks, such as limestone and sandstone, can be cut with a high-quality surface finish. Figure 12d shows a 10-in.thick sample of Indiana limestone sliced with a single pass at a traverse rate of 5 in/min.

5. ECONOMICS OF ABRASIVE-WATERJET CUTTING

The major cost elements in the operation of abrasive-waterjet systems are (1) the capital cost of the equipment, (2) the cost of power, (3) the cost of abrasives, and (4) the cost of nozzles due to wear. Although actual operating costs are not available, reasonable operating costs can be estimated since the costs of each element are well known.

5.1 Estimated Hourly Costs

The major operating cost is the cost of equipment. Based on a price of $100K, a 15-percent interest rate over a period of five years, and 10,000 hours of total operating time, the equipment cost is estimated at $15/hr. The cost of power is about $3/hr. The nozzle replacement cost is projected at $1/hr. Abrasive-jet cutting using throwaway abrasives, at a rate of 2 lb/nin, will cost $5/hr, at an average cost of 5//lb for abrasives.

The overall hourly cost will then be $25/hr. Taking into account other running costs, such as pump parts, cleaning, water, orifice jewels, etc., a cost of $27/hr becomes a reasonable estimate. Figure 13 shows estimated hourly cost figures for some applications.

5.2 Comparison with Other Cutting Methods

There are several techniques for cutting metals being used by the industry that could be replaced by abrasive-jet cutting under special circumstances. Figure 14 shows the cutting thickness range for metals by several methods along with the present, experimentally proven range for abrasive-jet cutting. These methods are oxyfuel torches, plasma torches, lasers, mechanical saws and diamond wires. The hourly operating costs for these methods are estimated to be as little as $2/hr for oxyfuel, mechanical saws and diamond wires. They increase to more than $20/hr for lasers and plasma torches.

Based on these rough estimates, it can be said that abrasive-waterjet cutting is more expensive than plasma, lasers, oxyfuel torches, etc., in their respective optimal thickness ranges. However, mechanical saws can cut only in straight lines and the other methods induce thermal effects and, in many instances, produce a poor quality surface that must be machined. Abrasive-waterjet cutting presents a totally different method for cutting advanced materials and, thus, could be economical for a number of metal cutting applications. These include cutting of titanium, ceramics and graphite-epoxy composites that can also be cut, but with difficulty, by conventional means.

For concrete cutting applications, the actual comparison of the cost of conventional concrete cutting systems with abrasive-waterjet cutting is difficult to make. It suffices to say that abrasive waterjet cutting provides smoother cuts, possibly faster cuts, minimum disturbances to adjacent structures, and is vibration free. These secondary benefits make abrasive-waterjet cutting the preferred method in a large number of applications.

5. ENVIRONMENTAL CONSIDERATIONS

The environmental aspects of materials cutting discussed here are (1) those aspects related to the nature of the abrasive-waterjet cutting process and its related equipment and (2) comparisons with other competitive techniques.

6.1 Abrasive-Waterjet Cutting Process and Its Equipment

In the abrasive-waterjet cutting process, the significant environmental concerns are:

- High pressures
- Noise levels
- Hazards due to rebounding abrasives
- Pollution with abrasives

Commercial equipment used in waterjet cutting applications, such as pumps providing high pressures up to 60 ksi, tubes, fittings, valves, swivels and other components, are well within the safety standards and codes of environmental organizations. High-pressure flexible hoses are available with a maximum safe operating pressure of 35 ksi. This is more than adequate for most abrasivewaterjet cutting applications.

Both mechanical and aerodynamic noises are associated with abrasive-jet cutting. The mechanical noise is primarily that of a diesel or gasoline engine or an electric motor. The aerodynamic noise is from free jets traveling at higher velocities than the speed of sound. This occurs when large standoff distances are used or when material penetration by a jet results. While cutting at reasonable standoff distances, the abrasive jet is quieter than most mechanical tools, especially if material is thick or if the jet is caught as it exits the kerf. Under these conditions, hearing protection devices night not be necessary, but their use is always a good practice.

The rebound of abrasives in the jet back flow, which results from cutting thick materials without penetration, nay be hazardous. However, this rebound can be prevented by shielding the nozzle with a shroud.

To prevent pollution and to adhere to strict environmental regulations, the catching of abrasives and cleaning up after cutting is essential. The cutting process does not produce airborne particles that might be hazardous to the health of an operator who is properly protected.

6.2 Comparison with Other Techniques

Most of the environmental benefits of abrasive waterjet cutting are realized in field applications. When abrasive jets are used in the field for cutting concrete, e.g., for underground utility access manholes or road and bridge repair, the noise and fatiguing characteristics of conventional tools, such as jackhammers and air operated drills, are eliminated. Because abrasive jets cut faster and require less manpower, traffic disturbances will be minimized. This is an important advantage, particularly in heavily populated areas. The remote operation capability of abrasive-jet systems will be a great advantage for operating in contaminated areas.

Compared with cutting using flames, the use of abrasive jets eliminates the danger of explosives in potentially hazardous areas (Saunders, 1982). Abrasive waterjets also require minimal support of the work piece because they produce very little force and they do not heat up a workplace to any significant amount.

7. INDUSTRIAL APPLICATIONS AND BENEFITS OF ABRASIVE WATERJETS

The potential applications of abrasive-jet cutting are numerous. Because of the technical and economic performance of abrasive waterjets, many industries could immediately benefit from this new technology. Table 2 lists some of these industries, some applications and the advantages of the abrasivejet technique.

For the construction industry, in the area of road and bridge repair, an abrasive-jet system may consist of a trailer housing a hydraulic pump, a high-pressure water intensifier, an engine, and online abrasive storage. A crew van can also be used to store abrasives, water, hoses, and applicators. This system could perform several useful functions, in addition to cutting and scarifying. For example, it could be used to sandblast corroded rebars and cut them, or it can drill holes for bolting posts.

Table 2 Abrasive-Jet Advantages

for Various Applications

Industry	Application	Abrasive-Jet Advantages
Construction	. Reinforced concrete buildings, road, . Underground work. . Hydrodams. . Pile Cutting.	. Quick set-up . 10-in.-deep cuts in one pass . No vibrations induced . No crack propagation . Same tool can cut and scarify . Fast and cost effective
. Oil and Gas Production	. Oil well casings . Pipeline repair . Platform repair . Inspection work	. Handheld or manipulator integrated Tool can cut steel and concrete in hazardous environments . Sea water operation is possible. Compact tool for downhole work

(For additional applications see Table 2a below).

In the oil and gas industry, a ship can be equipped with an abrasive-jet cutting system for offshore work. Casing cutting for decommissioning of oil wells, rescue operations, platform cutting and repair, underwater construction, and pipe cutting are only examples of what a single cutting system can offer. British Petroleum, in cooperation with the British Hydrodynamic Research Association, has already realized this potential and developed an abrasive-jet cutting system for offshore rescue operations. The system developed, however, requires high power with high flow rates of water (up to 15 gpm) and abrasives (up to 15 lb/ min).

In aerospace, glass, automotive, ceramics, and metals manufacturing industries and in general factory applications, a stationary system could be used. This system may consist of a pumping unit, a catcher, possibly a recycling unit, and a traversing device. Robots, x-y tables, and optical tracers are examples of traversing devices that can easily be equipped with abrasive-jet cutting nozzles. Specific examples of products that can be cut in these industries are heat exchanger cores, ceramic catalytic converters, thick ceramic sheets, thick plate glass, laminated glass, high-strength composite armor, composite missile cases and launch tubes, armor plates, titanium sheets, tubes, and honeycomb structures. These materials are currently cut by a variety of devices, including diamond saws, diamond wires, abrasive discs, plasmas and lasers.

However, the thermal devices may produce undesirable changes in material characteristics and, in some cases, may not be able to cut as effectively as an abrasive waterjet.

The coal mining industry can presently benefit from this technology by being able to safely cut metal structures in the potentially explosive environment underground. Coal picks can be replaced with abrasive jets for higher productivity.

Only a few examples of the potential application of the abrasive-jet cutting technology have been presented in this discussion. However, they do demonstrate the broad range of applications of abrasive-waterjet technology. The possibilities for the industrial use of abrasive jets in the immediate future seem unlimited.

8. CONCLUSIONS

The conclusions of this discussion of abrasive jet cutting in industrial applications are summarized below.

1. Cutting with abrasive waterjets has a great potential for a wide range of applications in many industries.
2. With the present state of the art of the abrasive-jet cutting technology, a number of industries can immediately benefit technically, economically and environmentally.
3. The cost of metal cutting with abrasive jets is presently higher than thermal cutting techniques, especially for thin sheets. However, the reduced costs of subsequent finishing operations and the versatility of abrasive jets in cutting a wide range of metals may justify their use.
4. Cutting with abrasive jets has many technical and environmental advantages unmatched by other techniques. Examples are the ability to cut very hard materials, the high quality of cuts produced, reduced noise levels, and increased safety.
5. Research and development efforts have improved the cutting performance of the abrasive jet. To compete with existing techniques and penetrate new markets, further efforts are needed to fully optimize cutting performance and equipment.

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Category	Material	Maximum Thickness
Concrete	Concrete	14 in. with fine and coarse aggregate
	Reinforced concrete	14 in. with 0.75 in diam. rebars
Rocks	Granite	12 in.
	Basalt	12 in. across the bedding plates
	Quartzite	12 in.
	Marble	8 in.
	Fire bricks	1.5 in.
Metals	Stainless steels	5 in. in 15-5 pH, 170-4 pH
	Mild steel	up to 4.5 in.
	Tool steel	up to 3 in.
	Aluminum	6 in. in 7075-T5
	Titanium	2 in. plates, honeycomb and tubes
	Cast iron	up to 3 in.
	Silicon	up to 4.5 in.
	Tungsten carbide	0.5 in.
	Bronze, lead, copper	4 in.
Ceramics	Alumina	1 in.
	Ceramic tiles	0.5 in
	Alumina honeycomb	1 in.
	Alumina foam	1 in.
Advanced composites	Kevlar-steel	2.5 in. armor plate
	Fiberglass	2 in.
	Graphite	2 in.
Glass	Single layer	2 in.
	Multilayers	25 layers of 1/16 in. thick plates
	Laminated glass	3 in. of glass and Lexan
	Aircraft windshield	1.5 in. sandwiched tempered glass

*Depth-of-cut data presented in this table do not present maximum cutting limitations.

Table 1. materials cut by abrasive waterjet *

Industry	Application	Abrasive-Jet Advantages
. Naval	. Steel and special alloys . Rescue tool	. Same tool can clean, expose welds for inspection and cut, depending on operating conditions . Very low reaction forces allow divers to easily manipulate tool
. Aerospace	. Aluminum . Titanium . Special alloys . Cured composites	. No thermal exposure . No oxidation . Deformation stresses avoided . Cuts thick composites and bonded structures without delamination . Narrow, clean kerf
. Defense	. Composite armor production and repair . Rocket motors, casings, fuel tanks . Munitions disposal	. Bonded metal-ceramic ballistic fiber structures are integrally cut . Handheld repair systems . Omni-directional . Integrable with vehicle hydraulics
. Aviation	. Aluminum, titanium and composite structures	. Lightweight rescue tool . Can operate in hazardous areas . Helicopter portable
. Strategic Construction	. Heavily reinforced concrete silos, caverns, and tunnels	. No heat, fumes or sparks generated . No vibrations induced
. Nuclear	. Exotic alloys, concrete	. Cutting without stresses along cutting path . Easily integrated with manipulators . Can scarify or cut decontaminated surfaces
. Glass	. Laminated safety glass and layers	. Cuts intricate contours in laminated or solid glass
. Coal Mining	. Metal structures	. Cuts seized bolts, chains and fixtures without sparking
. Utility	. Access manholes for utility repair . Cutting of buried pipes . Deep trenching	. Versatility and quickness . Reduced noise and labor . No vibrations to neighboring structures
. Automotive	. Metal . Composites . Glass	. High quality of cuts . Can be automated . Versatile

Table 2a. Abrasive jet advantages for various applications (contd.)

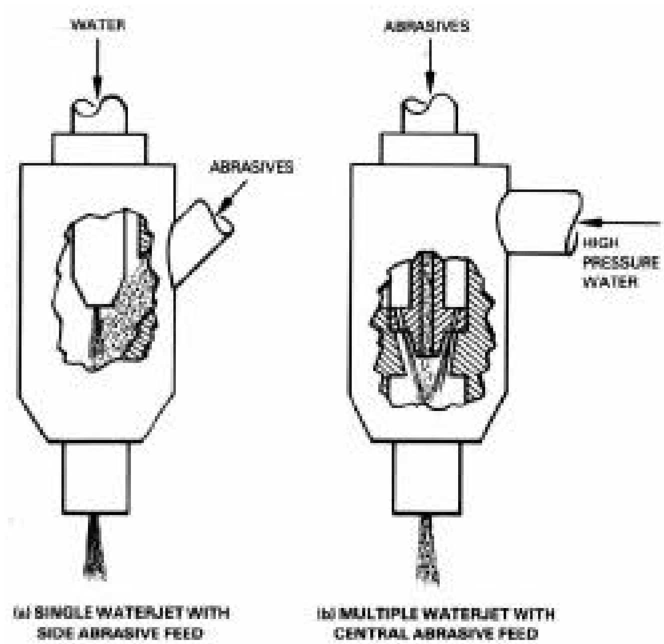


Figure 1. Abrasive waterjet nozzles.

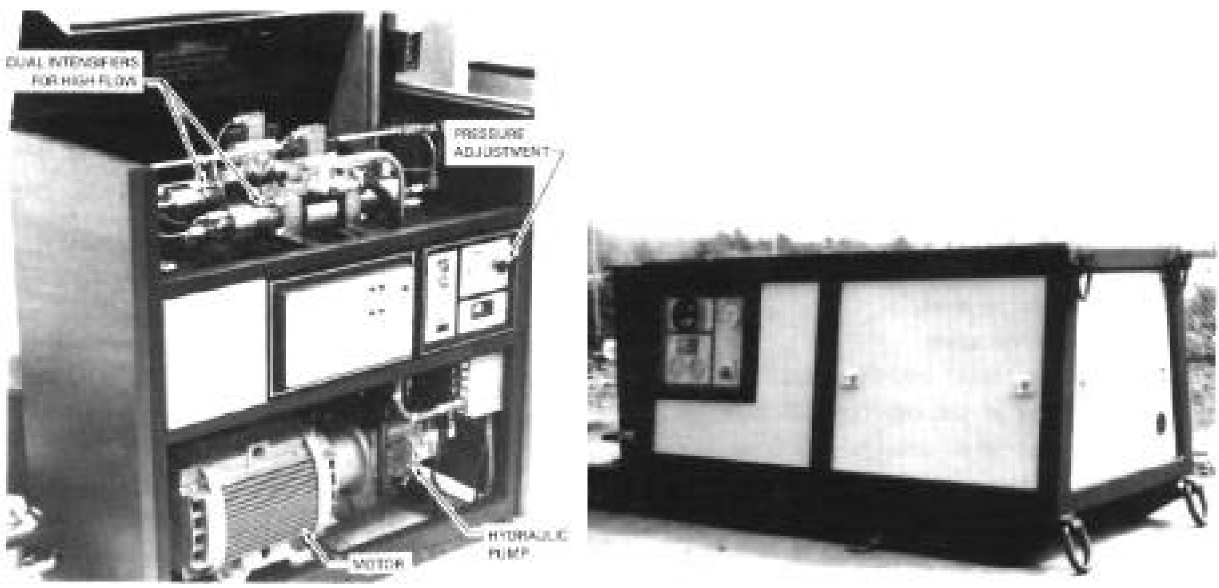


Figure 2. High-pressure pumps

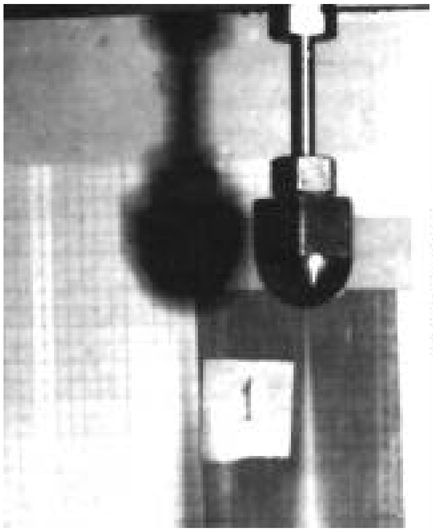


Figure 3. High-speed waterjet.

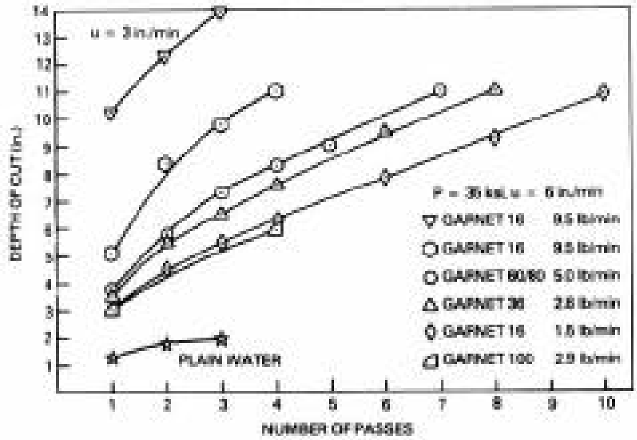
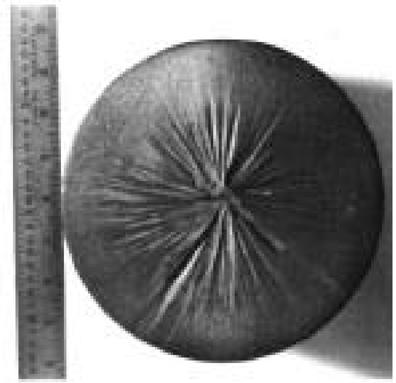


Figure 4. Effect of number of passes on depth of cut in concrete.

a) abrasive waterjet cut in concrete b) top view of kerfs obtained in concrete.

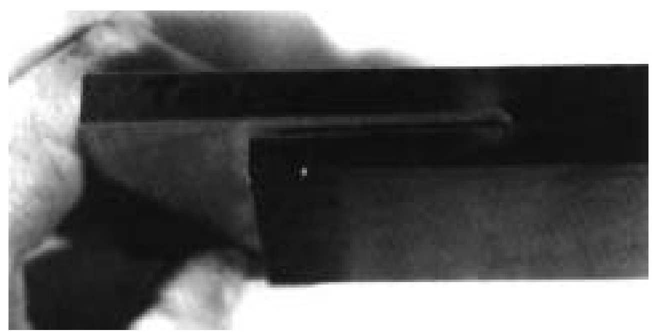
Figure 5. Samples of concrete cutting

a) kerfs in mild steel



b) S.S. 15.5 PH bar cut with abrasive waterjet

c) contour cutting in stainles steel



d0 cutting thick (3 in) tool steel

Figure 6. Samples of steel cutting

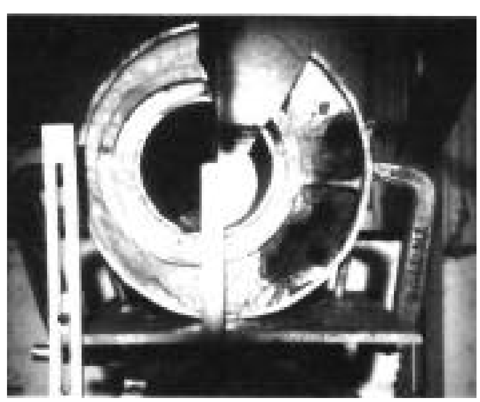
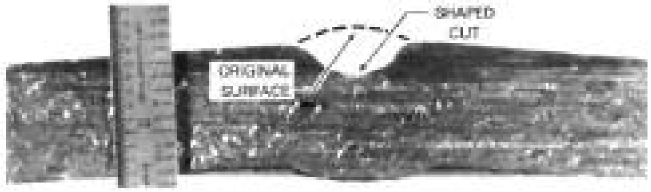
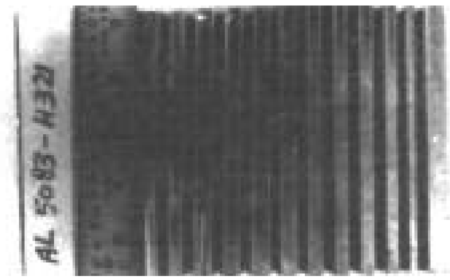


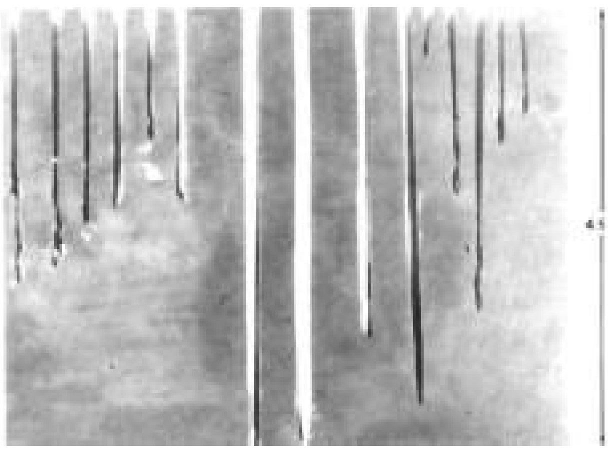
Figure 7. Linear cut in an oil well casing section.

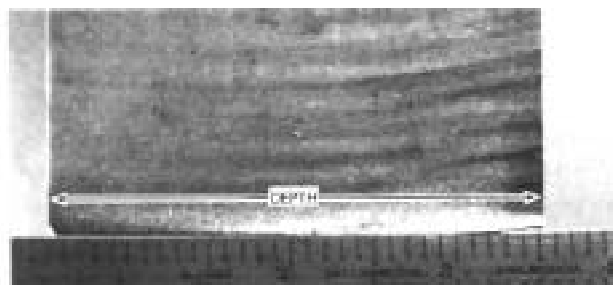




a) shaped cutting

b) top view of kerfs in aluminum

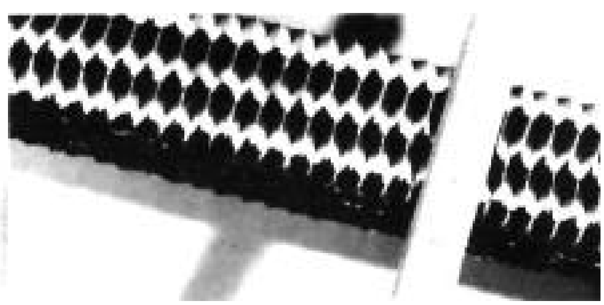




c) side view of kerfs in aluminum

d0 cutting surface in thick aluminum

Figure 8. Samples of aluminum cutting.



a) cutting thick titanium

b) bevelled cut in titanium honeycomb

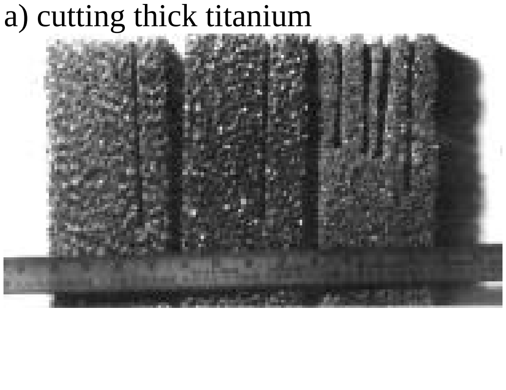
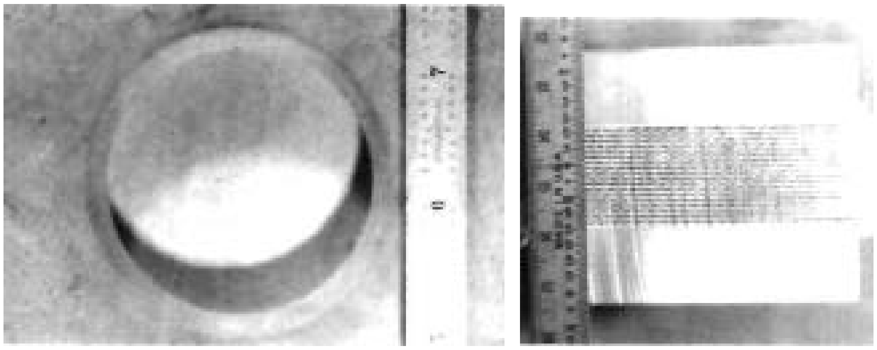
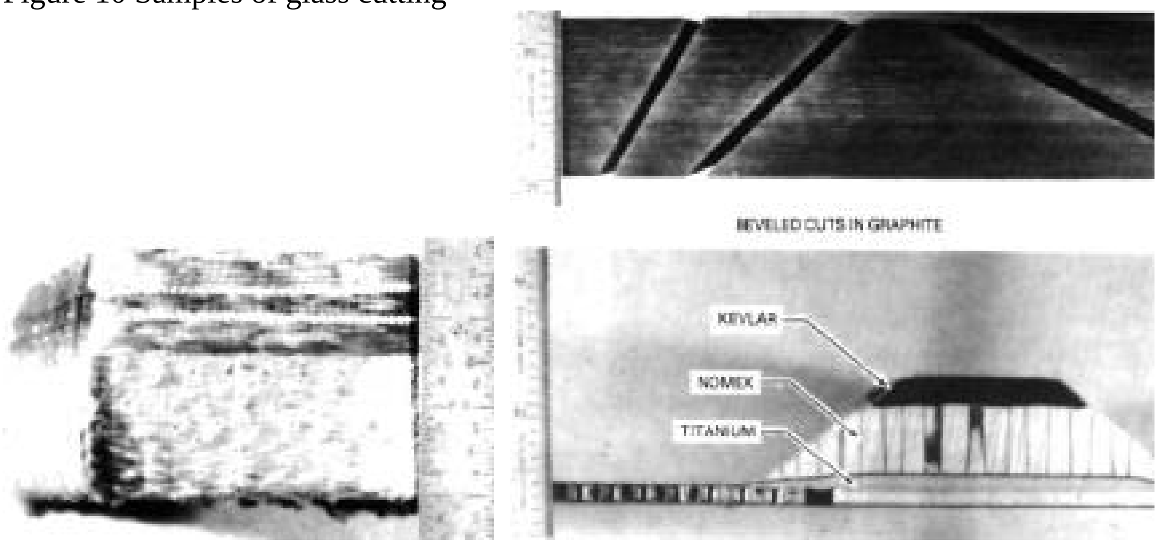


Figure 9. Samples of titanium cutting.



a) circle cutting in glass b) laminated glass

Figure 10 Samples of glass cutting



a) kevlar armor plate with steel wire mesh Cutting in multi-element structures

Figure 11. Cutting of advanced composites.

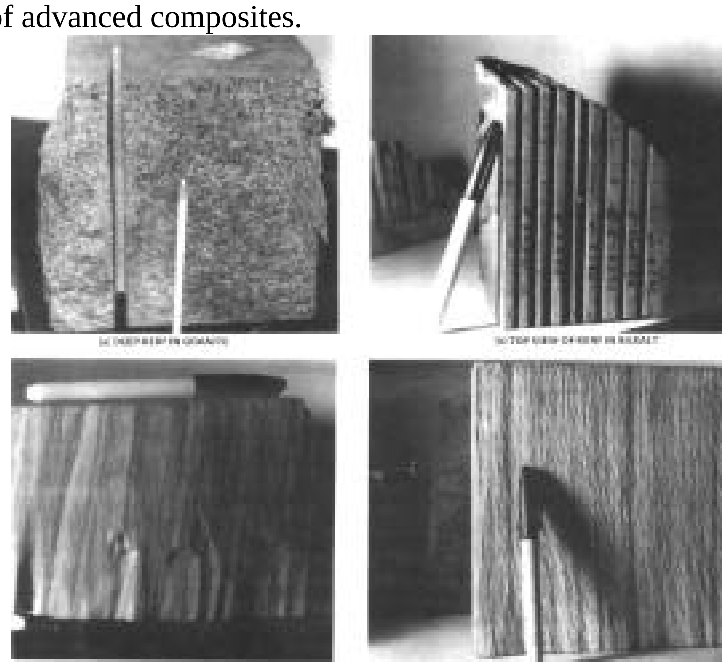


Figure 12 Single pass cuts in different rocks (granite, basalt, quartzite and limestone)

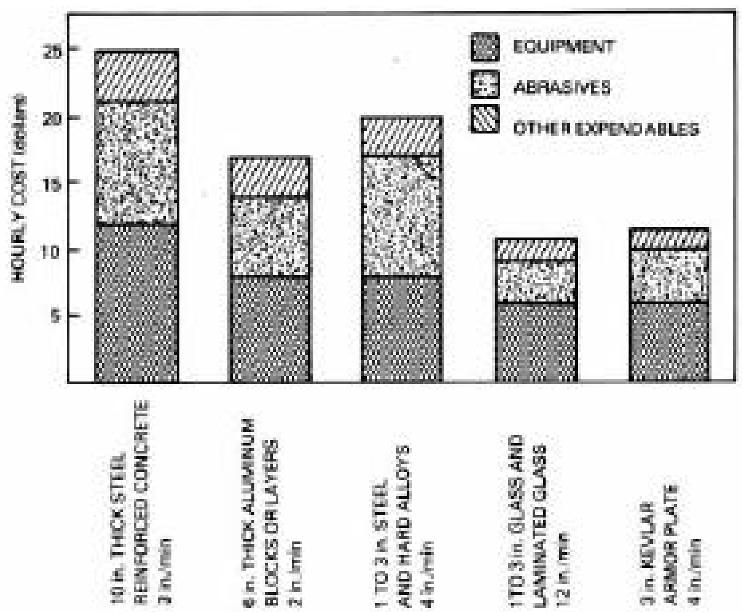


Figure 13 Estimated hourly costs of abrasive waterjets for some applications

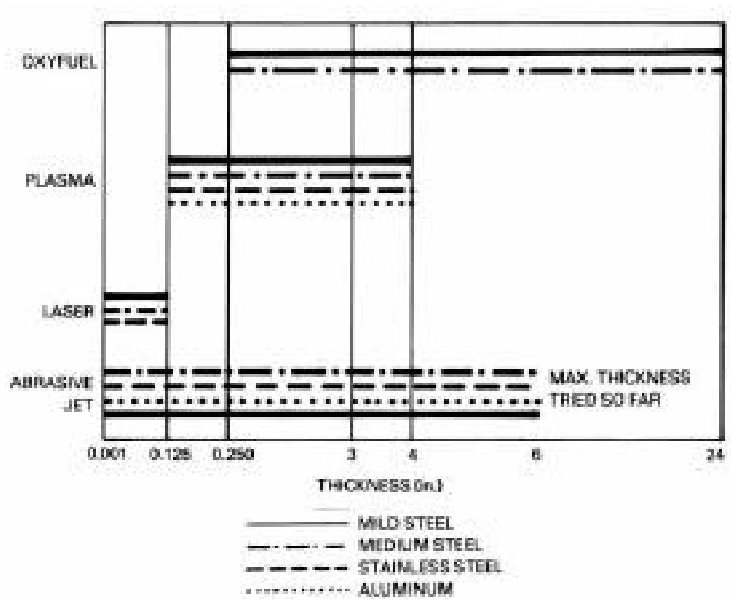


Figure 14. Cutting thickness ranges for various cutting processes.

DISCUSSION

NAME: Andrew F. Conn
COMPANY: Tracor Hydronautics

QUESTION: "Since the main erosive action in your jets is impact of the solid particles, why use water? Why not just use air to drive your particle against the target?"

ANSWER: The advantages of using water over air are:
a. Water accelerates a reasonable quantity of abrasives (1 to 5 lb/min) to effective butting speeds more efficiently than air jets.
b. Water produces more coherent jets than air, so the impact energy of particles will be more localized when water is used.
c. Water may he involved in the cutting process itself by superimposing hydrodynamic forces on the impacting particles. However, further research is needed to substantiate this phenomena. Preliminary investigation* indicates that these hydrodynamic forces are not significant.

*Hashish, M., "Modeling of Metal Cutting with Abrasive-Waterjets" sent for publication in ASME Trans. J. Mat. Tech, WAM Nov. 1983.

NAME: Dr. M. Vijay
COMPANY: N. R. C., Canada

QUESTION: "The important property in cutting rocks is the permeability/porosity. Would not the abrasives plug the pores and destroy the effectiveness of cutting?"

ANSWER: Although physical properties such as rock porosity, permeability and grain size are significant when cutting with plain water jets, they are not likely to be as significant when cutting with abrasive jets. The reasons for this might be:

a. Rock grains are removed or broken by the impact of solid particles in less time than it takes for pore pressures to develop.

b. The significant increase in cutting results with abrasive jets over those with plain water jets suggest an entirely different cutting mechanism.

c. Observation produced surfaces suggests that grains are cut rather than knocked loose.

d. If particles are forced between grains, material removed is likely to increase, due to additional stresses that weaken the rock, rather than decrease because of reduced porosity.

An investigation is required to identify the role of the rock properties relevant to the abrasive jet cutting process.

NAME: W. G. Howells
COMPANY: Berkeley Chemical Research

QUESTION: "What is the general effect of inclusion of abrasives in water jets from the standpoint of jet collimation?"

ANSWER: We do not have quantitative measurements on the (l/d) ratio over which the abrasive jet is coherent. However, we have observed that coherency depends on many parameters, as listed in the paper. The proper combination of these parameters produces a jet which retains reasonable coherence over long distances, and can cut up to 2 feet deep in hard rock.

NAME: James L. Evers
COMPANY: Southern Illinois University - Carbondale

QUESTION: "Is nozzle wear a large enough problem to warrant studies of techniques to protect the nozzle surface (e.g. water infection from wear surface)? Ilave you studied nozzle protection?

ANSWER: Nozzle wear certainly is a problem that requires further investigations. The use of porous materials with water injected from outside into the mixing zone is an attractive idea that has not been tried yet. However, with the present state-ofthe-art in materials technology, I believe that "tailoring" materials to offer long service life is quite possible.

We have tried to protect the nozzle by using multiple convergent water jets with central abrasive feed. This approach, however, is critically dependent on the precision of nozzle machining especially for low water flow rates similar to what we are using.

CUTTING HARD ROCK WITH ABRASIVE-ENTRAINED WATERJET AT MODERATE PRESSURES

Gene G. Vie
Fluidyne Corporation
Auburn, Washington 98002

ABSTRACT

It has been known that waterjet's ability in cutting hard materials can be significantly improved if suitable abrasives could be incorporated into the jet stream. Fluidyne Corporation has developed unique nozzles to achieve this objective without sacrificing the quality of waterjet or creating severe wear problems. With such nozzles, waterjets of less than 20,000 psi (138 MPa) in pressure can now be used to cut very hard rock, such as granite, quartzite and basalt, at practical speed and minimum consumption ot water and abrasives.

INTRODUCTION

Waterjet Technology

High-pressure waterjets have been evaluated extensively for cutting and drilling concrete, rock and minerals; numerous publications on this subject can be found among those presented at the International Symposium on Jet butting Technology (ISJCT) and in other journals. It has been observed that there exists a "threshold pressure" for a given rock that below which waterjet cannot cut the rock to any depth within practical dwelling time. This "threshold pressure" was believed to be a function of rock's compressive strength, permeability, crystalline structure and other properties; the exact relationship, however, is not clearly understood. Some typical values of such "threshold pressure" are 5,000 psi (35 MPa) for sandstone, 7,000 psi (48 MPa) for limestone, 14,500 psi (100 MPa) for granite, and over 20,000 psi (138 MPa) for quartzite and basalt. However, to obtain significant cut depth and speed requires waterjets of a pressure level considerably higher than the "threshold pressure" of a given rock. Thus, most of the past investigations on cutting or drilling hard rock with waterjets involved pressure in excess of 40,000 psi (276 MPa). To obtain such high pressure requires the use of special pressure intensifiers that are known to be costly and have limited flow capacity. There are also other constraints in applying such high-pressure waterjets in the field such that attempts to cut concrete, rock and other hard materials at very high water pressure have not been successful to date.

Abrasive-Entrained Waterjet

It has been known for some time that waterjet's ability in cutting hard materials can be drastically improved if hard particles could be incorporated into the high-speed waterjet. Unfortunately, this scheme is difficult to implement because of entrainment and wear problems. However, efforts have been devoted to this task in recent years and Fluidyne Corp. (Fluidyne) is one of the organizations in the world that have investigated suitable techniques. This paper summarizes an investigative effort funded by the National Science Foundation (NSF Project CEE8260224) to cut hard rock with Fluidyne's proprietary Abrasion-Jet technique at moderate water pressures. Because of the interruption from an unforseen event, this project was not completed and the data presented here are only a portion of that planned originally .

L ITERATURE REVIEW

Abrasive Fluid Jet Drilling

During the 1960's, drilling fluids were pressurized to 10,000 to 15,000 psi (69 to 103 MPa) to generate fluid jets by oil companies for augmenting rock drilling; significant improvement in drilling rate and reduction in bit wear were reported (9 and 10). Later in 1970's, abrasives such as sand and steel shots were added into the drilling fluids to produce abrasive fluid jets to augment conventional drilling; further improvement in drilling rate was observed, particularly in drilling through hard rock (3 and 13). This effort was eventually discontinued because of technical difficulties associated with the wear of pump parts, swivels and nozzles. Nevertheless, it was clearly demonstrated that the addition of abrasives into the fluid jets reduced the downhole pressure differential required to cut rock.

Post-Orifice Entrainment of Abrasives

The wear of pump parts and swivels observed earlier could be avoided if the abrasives are added after the fluid jets have already been formed by utilizing the so-called "jet-pump" principle. This approach has been applied by many water blasting processes (2, 5, 8 and 12). However, these processes have the common shortcoming of having to sacrifice the quality of waterjet in order to entrain the abrasives; thus their usefulness has been limited to cleaning and blasting applications.

Recently, the use of abrasive-entrained waterjets for cutting hard materials has been reported by several organizations. One of them is the British Hydromechanics Research Association (BHRA), which has been studying abrasive waterjet for some time (1, 2 and 11). The published papers revealed that most of BHRA's work was performed at water pressure in the range of 10,000 to 14,500 psi, water flow rate of about 12 gallon per minute (about 80 hp in pump power), and abrasive (copper slag) feed rate of 16.5 pound per minute. The detailed design of the nozzle has not been revealed but the water orifice used by BHRA was reported to be 0.07 inch (1.8 mm) in diameter. The bore of the slurry nozzle was not revealed but is likely to be about 0.5 inches, judging by the width of cuts made on concrete. BHRA reported cutting a wide variety of materials, including some types of rock, with its abrasive waterjet; some published test results are:

Concrete:	2.5 inches in depth at nozzle traverse speed of 6 inch per minute on concrete of 6,500-psi compressive strength (type of aggregates not revealed).
Slate:	7.9 inches in depth at nozzle traverse speed of 1.5 inch per minute.
Sandstone:	2.75 inches in depth at nozzle traverse speed of 6 inch per minute.
Mild Steel:	0.5 inches in depth at nozzle traverse speed of 4.5 inch per minute.

The ability in cutting mild steel indicates that BHRA's abrasive waterjet can cut hard rock as well.

Flow Industries, Inc. (Flow) is another organization that reported development effort in abrasive waterjet (6 and 7). The nozzle used by Flow was not revealed but the waterjet orifice was reported to be 0.025 inch (0.64 mm) in diameter. Thus, it appears that Flow's nozzle is basically similar to that used by BHRA, involving a waterjet orifice, a mixing chamber and a slurry nozzle. Testing at water pressure of 35,000 psi (240 MPa), garnet abrasive feed rate of 9.3 pound per minute, and nozzle traverse speed of 6 inch per minute, Flow reported a depth of cut of 5 inches in one pass on concrete of 5,000-psi compressive strength. Increasing the nozzle traverse speed to 9 inch per minute reduced the depth of one-pass cut to about 2 inches. Reducing the water pressure to 15,000 psi (103 MPa), the depth of one-pass cut at nozzle traverse speed of 9 inch per minute was reduced to about 1.3 inches. Although rock cutting was not reported, it is believed that Flow's abrasive waterjet would cut hard rock at a rate less than that observed with concrete.

Neither BHRA nor Flow revealed anything about the nozzle's performance in entraining abrasives and the life of the slurry nozzle. Judging from the single-jet basic nozzle design and the amount of abrasives consumed in respect to the water flow rate, it is reasonable to speculate that only a portion of the abrasives was actually entrained into the waterjet and that the slurry nozzle will have a short life even if it is made of very hard materials. The use of a single waterjet implies that the slurry nozzle must have a reasonably small and long bore if the abrasives are to be forced into the waterjet. The un-entrained abrasives would flow out of the slurry nozzle around the waterjet, thus wearing out the nozzle. The wear of slurry nozzle would accelerate and the abrasive entrainment would worsen as more abrasives would flow out around the waterjet. Therefore, the key to the practicality of the post-orifice entrainment approach of generating abrasive waterjet is the aspect of abrasive entrainment and nozzle wear.

EXPERIMENTAL SETUP

Abrasion-Jet Nozzle

Fluidyne's abrasive waterjet nozzles were developed to achieve

(1) highest quality waterjet,
(2) high efficiency in entraining abrasives, and
(3) minimum wear of nozzle parts.

Fluidyne also utilized the so-called post-orifice entrainment approach for introducing abrasives into waterjet. The basic design of the nozzle is illustrated in a schematic drawing presented in Figure 1. It differs from the others primarily in the use of multiple waterjets to generate strong fluid actions for entraining abrasives naturally. The orifice cone is a disk that has multiple, high-precision orifices arranged in a circular pattern to generate multiple waterjets of desired jet configuration; as many as nine waterjets could be accomodated in such arrangernent. A mixing cavity is formed between the orifice cone and a nozzle cone, which is made of wear-resistant materials and has a precision bore that fits the diameter of the jet bundle. The position of the nozzle cone can be adjusted to obtain the desired fit. The abrasives enter into the mixing cavity either from the top or from a side inlet, and can enter into the central area of waterjet bundle through the space between the waterjets. The outside view of one of the test nozzles is presented in Figure 2. Such nozzles can generate very strong negative pressure in the range of 25 to 30 inches Hg inside the mixing cavity at water pressure of 15,000 psi with a bore of nozzle cone as large as 0.4 inches.

Because of the shielding provided by the multiple waterjets, a tungsten carbide nozzle cone can last more than a day's operation without affecting the level of negative pressure inside the mixing cavity or the performance of abrasive waterjet. Fluidyne termed the abrasive-entrained waterjet generated by it patent-pending nozzles the Abrasion Jet to denote the abrasions and erosion actions involved in cutting materials.

The multiple waterjets are currently arranged in two basic configurations parallel or converged. Thus, the diameter of the resultant abrasive waterjet can be changed to suit the applications. Since the orifice cone is interchangeable, the same nozzle body can be used for a wide range of water pressure and flow rate, and a wide variety of abrasive powder or slurry. In this project, testing of both parallel- and converged-jet configurations was planned but only a 5-parallel-jet nozzle was actually tested over a range of parameters during the time period. The testing of several other jet configurations was curtailed.

Experimental Setup

The test equipment consist of a water pump system, an abrasive feed system, a water supply system, and a nozzle traverse system. A triplex crankshaft pump powered by a 60-hp motor was used to supply the water at a peak pressure of 17,000 psi. The abrasive feed system is simply a tank-valve-hose arrangement in which the abrasive gravity flow is controlled with an interchangeable orifice. A plastic hose connects the abrasive tank to the nozzle. The water supply system consists of a solenoid valve, 100-micron filter cartridges, an in-line flow meter, a 40-gallon surge tank and necessary hoses. The linear nozzle traverse movement is provided by a gear motor-drive screw-sliding rods arrangement; the nozzle traverse speed could be adjusted from less than 1.0 inch per minute to more than 10 feet per minute.

Abrasives

The abrasives used in this project were mostly Idaho garnet of several grit sizes. Some Texas "Green Diamond" abrasives and copper slag were also tested briefly for comparison. The testing of other types of abrasives was planned but not carried out. The abrasive feed rate was measured during each run of linear cutting by pre-weighing a given amount of selected abrasives that was subsequently poured into the tank and sucked to the nozzle. The time involved in consuming this amount of abrasives was recorded with a stopwatch. Dividing the weight of abrasives by the time yields the abrasive feed rate for that particular run. Accurate measurement of abrasive feed rate is difficult as it is affected by several factors; further discussion on this subject will follow.

Rock Specimens

A variety of rock specimens were procured for this project from local sources. The collection included soft sandstone, slate, hard sandstone, gray granite, quartz marble, rhyolite, quartzite and basalt. Two types of cast concrete specimens of 6,000-psi minimum compressive strength were also available for comparison; one has crushed Georgian granite aggregates and the other has small very hard igneous pebble aggregates. The properties of the collected rock specimens are not known. However, straight waterjet was applied during the testing to provide clue for the hardness of the rock involved.

DISCUSSION OF TEST RESULTS

Jet Configuration

In the case of post-orifice entrainment of abrasives to generate high-capability abrasive waterjet, the aspect of abrasive entrainment is very important as the coherence of waterjet and abrasive entrainment are conflicting issues. With Fluidyne's nozzles, the issue of abrasive entrainement is related to the jet configuration, which denotes the number of waterjets, size and geometries of orifices, and the physical arrangement of orifices. In order to study the aspects of abrasive entrainment, Fluidyne prepared orifice cones of many configurations. The number of orifices on each orifice cone varied from one to nine. Some were designed to issue parallel jets while others were designed to produce converged jets. Orifice cones of similar configuration also have orifices of different sizes for operating at different pressure levels and flow rates. Testing these orifice cones on concrete revealed that abrasive entrainment is significantly better with orifice cones having five or more waterjets, as judged by the amount of abrasives introduced without choking and the depth of cuts produced. The 5-parallel-jet nozzle, for example, was also found to be superior than a single-jet nozzle of similar output in cutting concrete without abrasives. This observation may be explained by the cavitating actions of the 5-jet bundle as the five individual waterjets will form a single cavitating jet when they reached the target material. In comparison, a 5-parallel-jet nozzle can entrain more abrasives than its converging-jet counterpart. The cuts made by parallel-jet nozzles are wider. On the other hand, converging-jet nozzles can make narrower and deeper cuts at reduced abrasive consumption rates than their parallel-jet counterparts.

Rock Specimens

The test results showed that the performance of Abrasion Jet in cutting rock is much influenced by the type of rock involved. Figure 3 presents a comparison of cut depth obtained with various rock specimens, with or without abrasives. The benefit of adding abrasives into the waterjet is clearly demonstrated and the effect of rock properties on depth of cut is also very well indicated in this data. The ratio of depth of cut obtained with and without abrasives can vary from 5 for soft sandstone to more than 15 for hard quartzite and basalt. This depth ratio on hard rock is even greater when the Abrasion Jet was applied at slow nozzle traverse speed or when multiple passes were applied.

It is believed that the depth of cut on rock with Abrasion Jet may indicate the hardness and/or grain structure of the rock involved if one can precisely control the flow of a chosen abrasive. It could be a useful method for studying and classifying various rock types as it is easier to perform than conventional compressive-strength tests. However, more understanding on the cutting mechanism of Abrasion Jet is necessary before this process could be standardized.

Effect of Abrasives

The type of abrasives used, its grain size and shape, and the mass flow rate all intimately influence the rock cutting capability of the abrasion Jet. This aspect, unfortunately, has not been studied in depth to date in this project. Garnet abrasives have shown to be far superior to silica sand, copper slag and "Green Diamond" abrasives in cutting concrete and rock. This observation can be explained if one compares the relative hardness of these abrasives and observe the shape of the grains under a microscope. Garnet grains are sharper and have several cutting edges owing to their crystalline structure. The grains of some beach sands, on the other hand, are well polished and have lost their sharp edges. The presence or absence of natural fractures on the abrasive grains is also a related factor.

The benefit of using hard and sharp abrasives was found to be influenced by the rock involved. For cutting sandstone, the difference in cut depth between garnet and copper slag was not significant. In cutting quartzite, garnet was far superior to copper slag. For the same reason, garnet and silicon carbide probably produce similar cutting results on quartzite but can be significantly different when they are used to cut glass or steel. The relative hardness between the abrasives and the target material is a subject relevant to Abrasion Jet cutting of materials. This subject is believed to be related to some other factors, such as water pressure.

Using garnet as abrasives, the grain size was found to affect the cutting rate of Abrasion Jet, as shown in Figures 4 and 5. Garnet of Grit #36 and #60 were superior to #100 in cutting quartzite specimens. Grit #60 was found to be slightly superior to Grit #36 in cutting quartzite but the reverse was observed with other types of rock. The study of abrasive grain size's effect on Abrasion-Jet cutting was complicated by the fact that changing the abrasive grain size can change the amount of abrasives entrained into the multiple waterjets and thus the gravity flow of abrasives at the tank. A slight change in the abrasive flow rate can affect the depth of cut such that the observation of the effect of other factors becomes difficult.

With #100 garnet, choked abrasive flow occurred early and a maximum amount of only 3 pounds per minute was able to be introduced into the nozzle. Also, the benefit in cutting with #100 garnet was seen to level off early at about 2.0 pound per minute. With the same nozzle but #36 and #60 garnet, the abrasive feed rate could be increased up to 4.0 pound per minute and the benefit in cutting increased with the increase in feed rate until abrasive flow was choked. The reason for the occurrence of choked abrasive flow is not yet clearly understood but is believed to be related to the water flow rate and jet configurations. It has been observed that the abrasive feed rate could be increased beyond 5 pound per minute with the same abrasives and abrasive orifice when a different 5-parallel jet orifice cone was used. The only difference in these two cases was the water flow rate. Obviously, the abrasives entered into the mixing cavity must be carried away immediately by the waterjets if accumulation and clogging of abrasives are to be avoided. How does the abrasive grain size affects the level of negative pressure generated in the mixing cavity is not clear at present.

Garnet abrasives of different grit sizes produced cuts of different profile on rock, as shown in Figure 6, which presents a photograph of cuts made on a quartzite specimen. The cuts on the left were made with #60 garnet while those on the right were made with #36 garnet. The sharp taper exhibited by the cuts is associated with cutting very hard rock. The influence of spent jet is also visible on the rock specimen and is indicated by the bottom portion of the cuts. This aspect is associated with the impingement angle of Abrasion Jet, which can affect the depth of cut but has not been studied in this project.

Nozzle Standoff Distance

Within a range of a few inches, the nozzle standoff distance of Abrasion Jet does not have significant effect on the depth of cut on rock as it does in high-pressure waterjet cutting of materials. The flexibility in nozzle standoff distance is affected, however, by the hardness of rock involved. With basalt, for example, the nozzle standoff distance of Abrasion Jet can be changed within one inch without producing appreciable difference in depth of cut. The width of cut at surface would be increased as the nozzle was raised. With softer rock, the nozzle standoff distance could be changed over two inches without affecting the depth of cut. With the nozzle used in this project, a 0.25-inch nozzle standoff distance indicates an actual distance between the waterjet orifices and the target material of about 2 inches.

The effective cutting length of Abrasion Jet can be indicated by the accumulated depth of cut in multiple traverses of the nozzle. One multiple-pass data with quartzite is presented in Figure 7, which shows that Abrasion Jet was still effective after penetrating into the rock to a depth beyond 4 inches, although its effectiveness started to level off. The quartzite specimen with which this data were collected is shown in a photograph presented in Figure 8. The black arrow shown in Figure 8 points at a shallow groove that was produced by traversing a straight 15,000-psi waterjet four times at 12-inch-per-minute nozzle traverse speed. The depth ratio obtained with and without abrasives was found to be more than 25.

The shape of the curve of accumulated depth of cut vs. number of pass was found to vary with the hardness of rock; the slope of this curve decreases more rapidly in cutting hard rock than in cutting soft rock. In cutting concrete, for example, a 15,000-psi Abrasion Jet was still effective after cutting to a depth of more than 20 inches, as shown by a data presented in Figure 9. It is believed that the presence of a groove enhances the performance of Abrasion Jet. Thus, the initial nozzle standoff distance has a different effect on depth of cut as compared to the same distance after a cut has already been formed. In Figure 7, for example, the second pass of Abrasion Jet had a nozzle standoff distance of 1.75 inches and created a cut in the quartzite specimen of 1.2 inches in depth. However, if the initial nozzle standoff distance was set at 1.75 inches, the Abrasion Jet will not make a 1.2-inch-deep cut in one pass.

<u>Nozzle Traverse Speed</u>

The speed at which an Abrasion Jet is moved over the surface of a target material has pronounced effect on the depth of cut; the depth is increased as the nozzle traverse speed is reduced. Figure 10 presents a plot of depth of cut vs. nozzle traverse speed obtained in Abrasion-Jet cutting of a granite specimen at 12,000- and 15,000-psi pressure, together with two data points stained with straight waterjet. Figure 11 presents a photograph of this particular specimen used in this test; the dark arrow points at a groove made by the 15,000-psi waterjet traversing at 27 inch per minute. The wide and shallow groove in the center of the granite specimen was made by two adjacent traverses of a 15,000psi waterjet at 6-inch-per-minute traverse speed.

Judging from the depth of cuts made by the straight waterjet, the granite specimen G-1 is relatively soft as compared to quartzite and basalt specimens. As a result, the data obtained at 12,000-psi water pressure are quite similar to that obtained at 15,000 psi; the slightly increased abrasive feed rate in 12,000-psi cutting compensated for the reduced water pressure. The cuts made by Abrasion Jet in granite were found to be quite uniform in depth, having a deviation of less than 10 percent. With the Abrasion Jet, the cuts could be spaced close to each other without spalling. On the other hand, straight waterjet has a tendency of causing rock to spall, which makes it impossible to space two cuts close together. The waterjet's tendency to cause rock or concrete to span disappears almost immediately when a small amount of abrasives is added.

Figure 12 presents a plot of depth of cut vs. nozzle traverse speed obtained in cutting a quartzite specimen at two levels of water pressure. The 12,000-psi data were obtained at increased abrasive feed rate and, as a result, showed greater depth of cut than the 15,000-psi data. Figure 13 presents a photograph of a quartzite specimen sliced with Abrasion Jet at 15,000-psi water pressure, 4.0 pound-per-minute abrasive feed rate and 12-inch-per minute nozzle traverse speed. This photograph reveals that Abrasion Jet was sharply deflected by the hard rock during the traverse and the curvature of the jet trajectories showed that Abrasion Jet had a long dwell time. It is believed that a study of these trajectories can lead to information that will be useful for optimizing the performance of Abrasion Jet.

Figure 14 presents a plot of depth of cut vs. nozzle traverse speed obtained in cutting a hard sandstone specimen, which is quite hard and has a reported compressive strength of 20,000 psi.

Water Pressure

A very limited range of water pressure was investigated in this work. The lowest water pressure attempted was 8,000 psi and a 5-parallel-jet Abrasion Jet was able to cut into a quartzite specimen cleanly, albeit to a depth reduced from that obtained at higher pressures. The maximum water pressure planned for this project was 20,000 psi but tests at this pressure were curtailed. Thus, a detailed discussion on the effect of water pressure cannot be made at this time. Still, the test data collected to date do show some general trends .

For a fixed pump power input, increasing the water pressure is accompanied by reduced water flow rate, which, in Abrasion-Jet case, can affect the entrainment of abrasives and thus the performance in cutting. The lower limit of water flow rate for achieving good abrasive entrainment is related to the jet configuration and nozzle design and is not clearly known at this time.

For a fixed water flow rate and jet configuration, increased water pressure is without doubt beneficial to Abrasion-Jet cutting of rock, particularly hard rock and in cutting to greater depth. The greater velocity of abrasive particles at higher water pressure can mean greater effective length of Abrasion Jet, which is important if the rock involved is very hard. On the other hand, the influence of water pressure has been overshadowed by the effect of abrasives in Abrasion-Jet cutting. The presence of suitable abrasives in waterjet may have obliterated the term of "threshold pressure" mentioned frequently in cutting rock with high-pressure waterjets. With hard abrasives, it is believed that Abrasion Jet can cut any rock even at quite low water pressures if sufficient dwell time is provided. The resultant cuts at low water pressures will be quite similar to that obtained at higher water pressures with the only difference being the rate of cutting. It is highly unlikely that a definitive water pressure exists with a given rock such that significant improvement in Abrasion-Jet cutting can be observed when the water pressure is increased beyond this level. The benefit of increased water pressure with Abrasion Jet will be gradual and is more observable in cutting hard rock to greater depth. The benefit of higher water pressure can be wiped out if difficulties in abrasive entrainment start to appear because of reduced water flow rate.

Opposite to the effect of water pressure, there may exist a "threshold hardness" in the selection of abrasives for cutting rock or other hard materials with Abrasion Jet. The effectiveness of Abrasion Jet in cutting a given material may show sudden improvement when a softer abrasive is replaced with one that is harder than the target material. The effectiveness of an abrasive may be related also to the fragility in impacting a given target material. The shape of abrasive grains in relationship to the structure of the target material could also be important. Thus, the "hardness" alone may not be adequate in describing the relationship between the abrasives and rock in Abrasion-Jet cutting

<u>Jet Exposure Time</u>

The data obtained in this work showed that the effect of nozzle traverse speed on the depth of Abrasion-Jet cutting of rock may follow a simple relationship, involving the length of time that rock was exposed to jet impingement, which may be called Exposure Time and may be estimated by:

$$T = n\, d / v \tag{1}$$

where

T = Abrasion Jet exposure time, second

n = number of pass

d = diameter of Abrasion Jet, inch

v = speed of nozzle traverse, inch per second.

Thus, multiple-traverse cutting data could be lumped together with single-pass cutting data obtained at various nozzle traverse speeds. When the depth of cut is plotted against the exposure time on logarithmic scale, straight lines can be expected. It is possible that a family of lines can be obtained for a given rock in that changes in the abrasive feed rate will change the slope of the lines while changes in water pressure produce a group of parallel lines. Rock of different types will have family of lines situated at different parts of this chart.

In other words, the term of "jet interference" mentioned in high-pressure waterjet cutting of materials may not have any meaning in Abrasion-Jet cutting as the effect of abrasive particles grossly overshadowed that of the waterjets such that the cutting of rock is reduced basically to an abrasion process. In such process, the number of impacts made by the high-speed abrasive particles within a unit time basically determines the rate of material removal. The kinetic energy of the impacting abrasive particles and the characteristics of the impact are two finer points of this abrasion process. Nevertheless, the ability of waterjet in exploiting fractures created by the impact of abrasive particles cannot be ignored. The presence of the water stream also allows the abrasive particles to impact the rock more than once and prevents the particles from rebounding after impact.

SUMMARY AND CONCLUSIONS

The data collected in this study showed that waterjets of moderate pressure can effectively cut very hard rock if suitable abrasives are entrained into the waterjet without sacrificing the quality of the waterjet. It also showed that Fluidyne's Abrasion-Jet nozzles can provide the desired abrasive entrainment without severe wear problems of nozzle parts.

This study also showed that entrainment of abrasives into a group of waterjets is a complex process, involving the interactions of several factors. Good abrasive entrainment has been shown to be a prerequisite of generating high-performance abrasive waterjet. With the use of hard and sharp abrasives and an effective nozzle, water pressure was found to be of secondary importance in cutting hard rock .

The process of Abrasion-Jet cutting of rock is believed to be basically an abrasion process in which the number of impacts made by the high-speed abrasive particles within unit time determines the cutting rate. This process can be optimized by considering the following aspects listed in the order of decreasing priority:

- Entrainment of abrasives into waterjets
- Characterization of abrasives in respect to properties of rock involved
- Water flow rate and abrasive feed rate
- Water pressure
- Nozzle traverse speed
- Nozzle standoff distance
- Jet impingement angle.

Further studies are required to understand the finer points involved in each of these aspects.

ACKNOWLEDGMENT

The author is indebted to the support provided by the National Science Foundation for this study.

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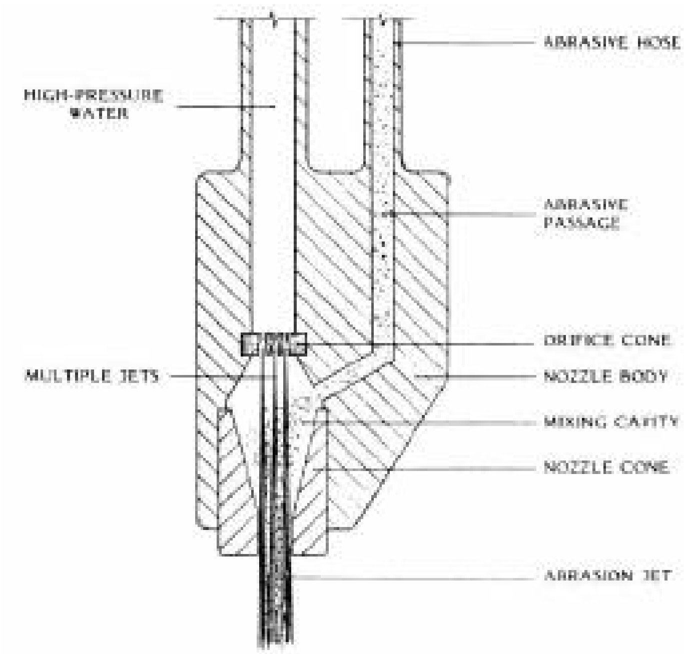


Figure 1. Schematic illustration of Fluidyne's abrasion-jet nozzle.

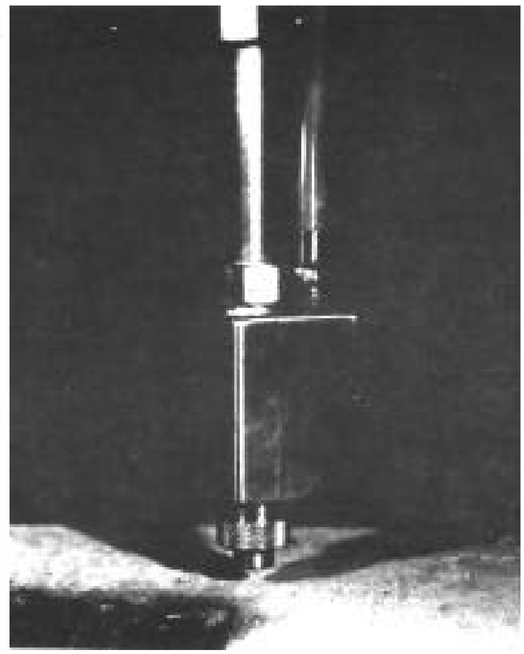


Figure 2. Exterior view of an abrasion-jet Nozzle assembly.

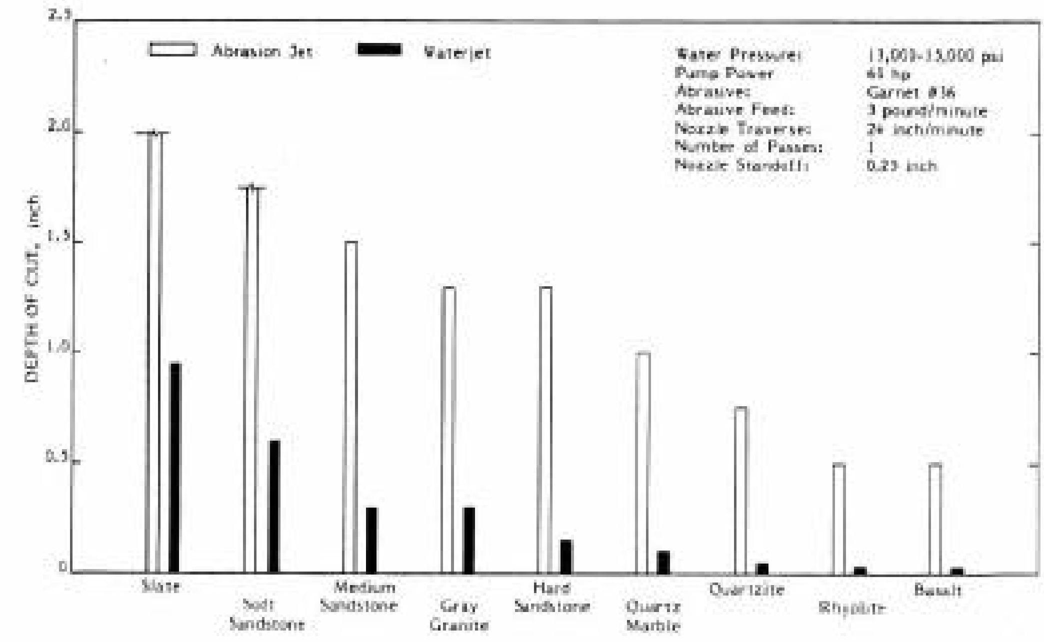


Figure 3. Comparison of depth of cut made on various rock specimens.

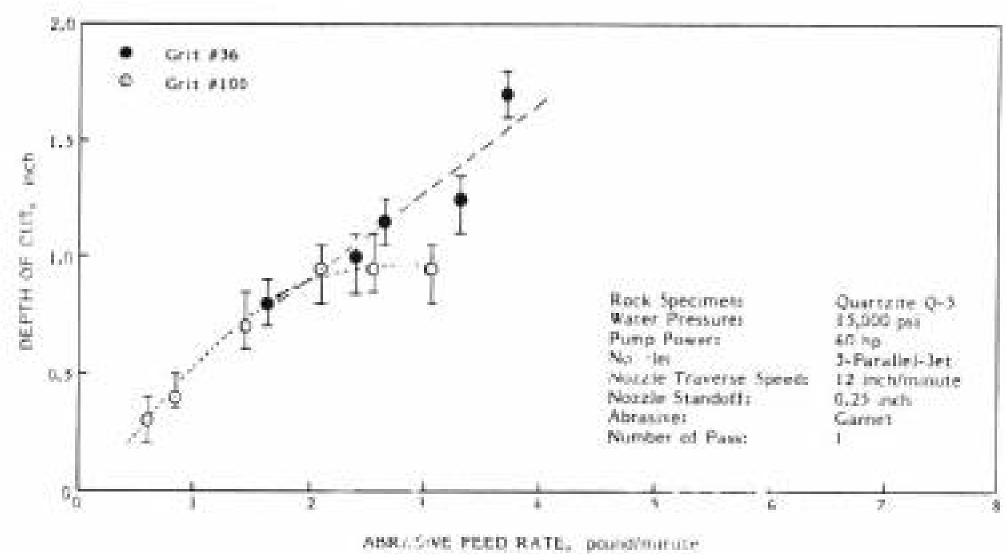


Figure 4. Effect of abrasive feed rate and grain size on depth of cut.

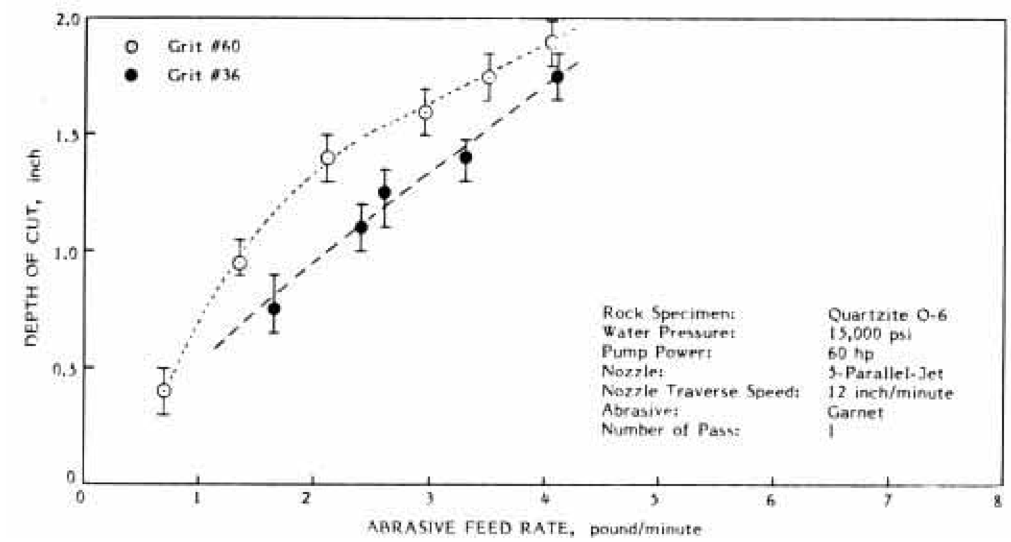


Figure 5. Effect of abrasive feed rate and grain size on depth of cut.

Figure 6. Cuts made with garnet abrasives of two grit sizes.

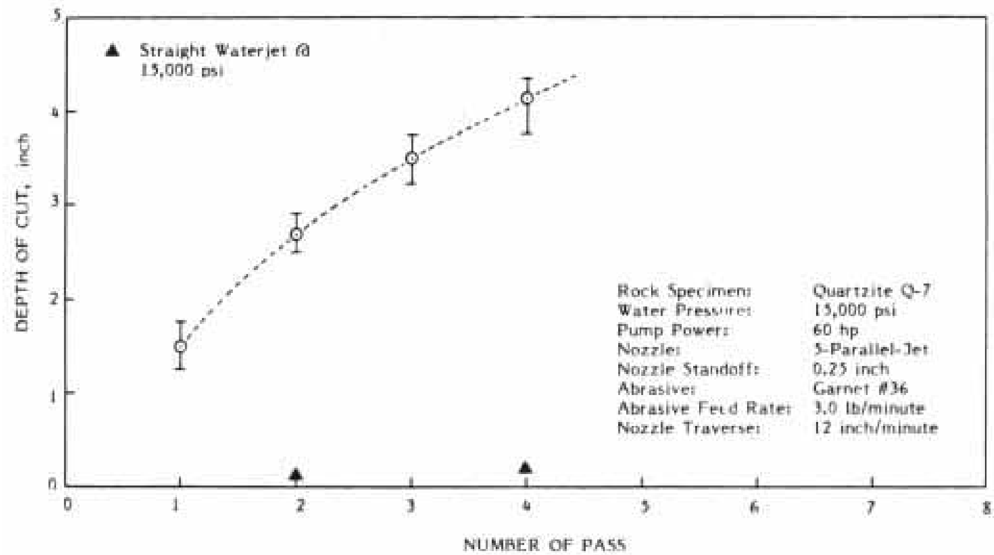


Figure 7. Effect of multiple passes on nozzle on depth of cut.

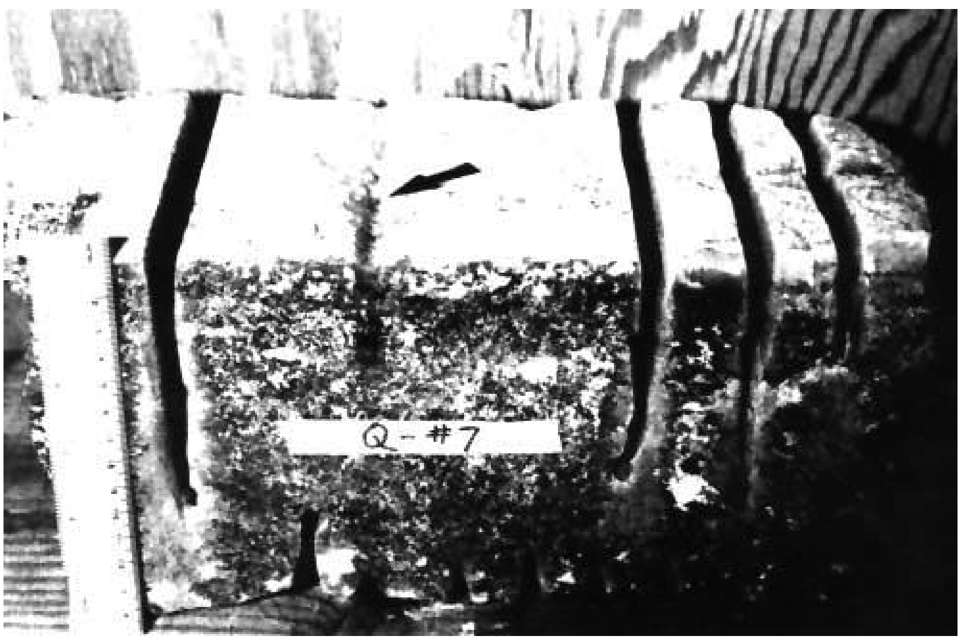


Figure 8. Cuts made in multiple passes of abrasion jet.

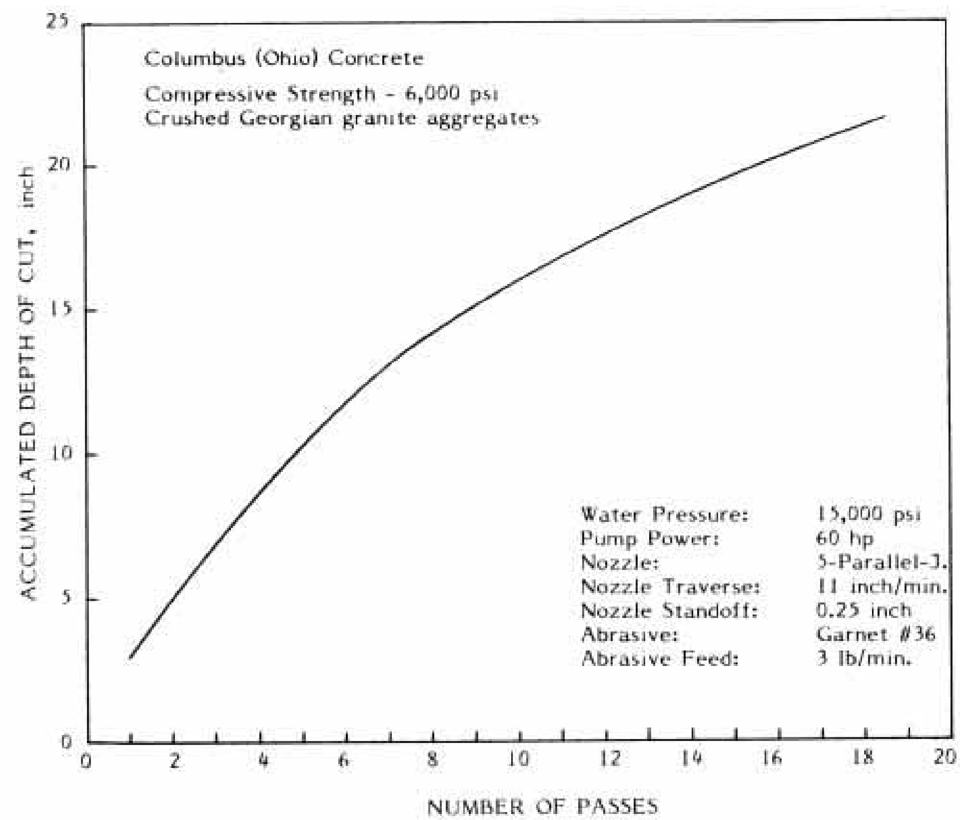


Figure 9. Accumulated depth of multiple pass cut on concrete.

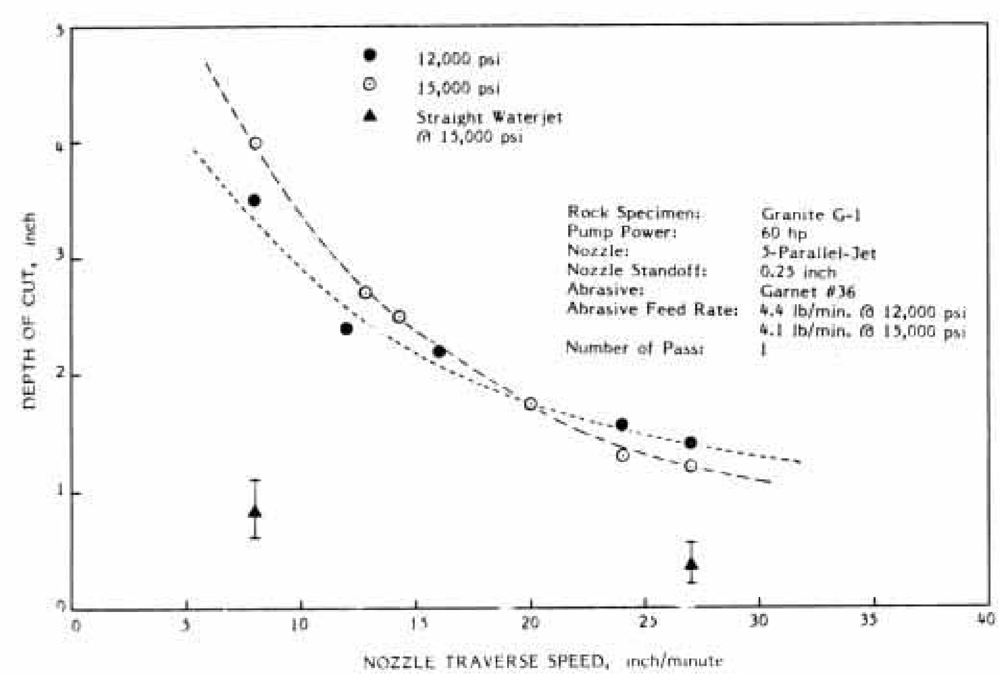


Figure 10. Depth of cuts made on a granite specimen at various nozzle traverse speeds.

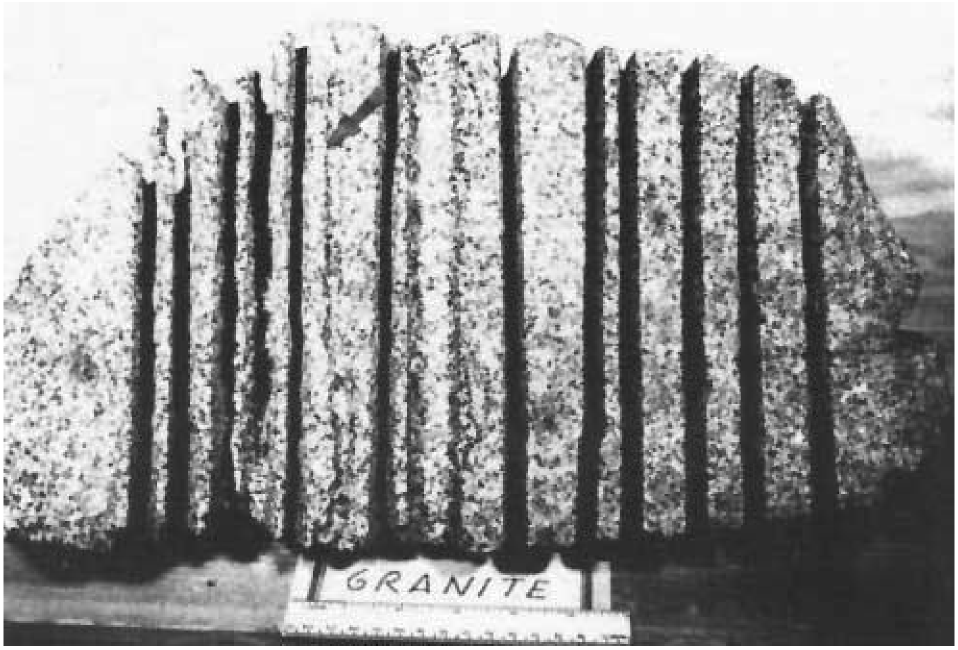


Figure 11. Granite specimen cut with abrasion jet at various nozzle traverse speeds,

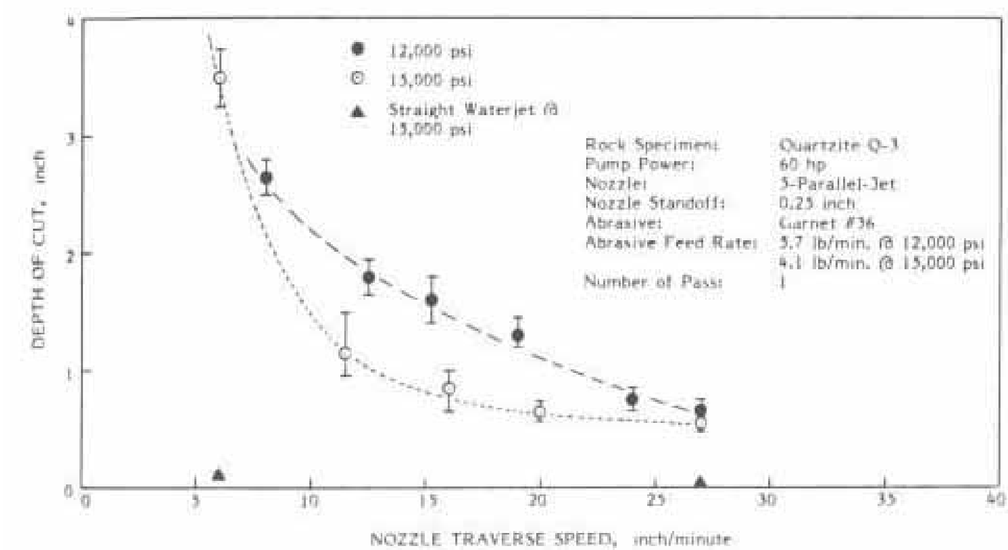


Figure 12. Depth of cuts made on a quartzite specimen at various nozzle traverse speeds.

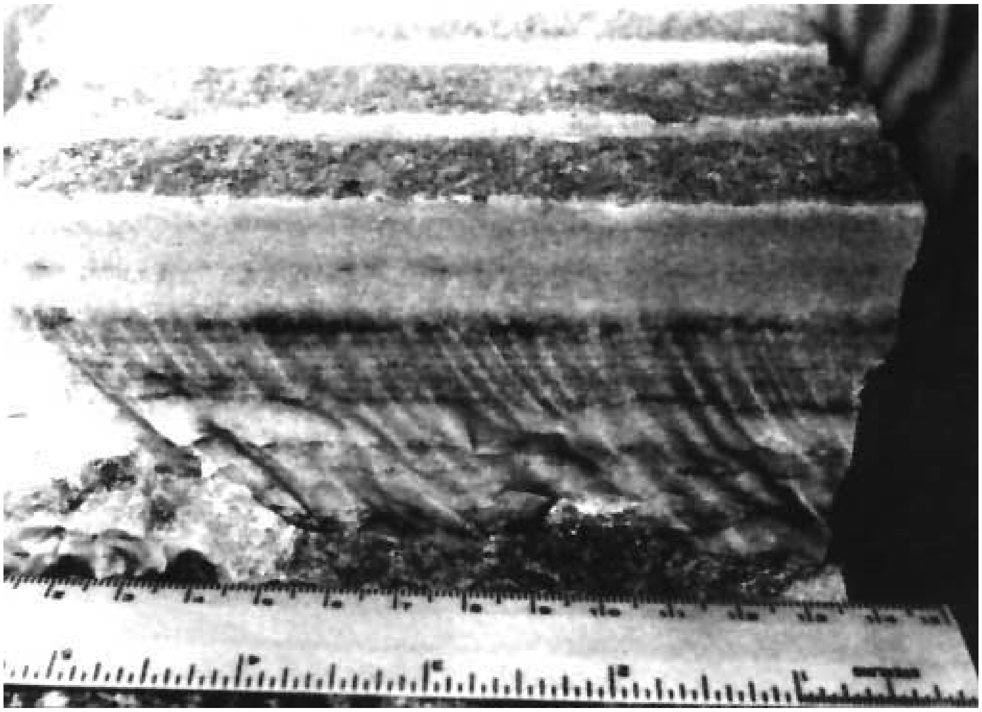


Figure 13. Close-up view of quartzite surface obtained in abrasion jet cutting.

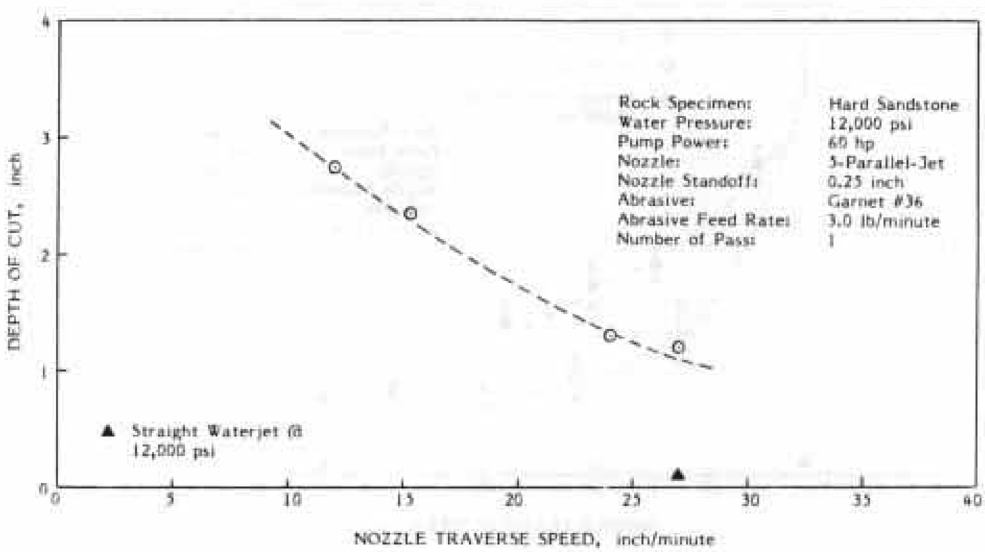


Figure 14. Depth of cuts made on a hard sandstone specimen at various nozzle traverse speeds.

DISCUSSION

NAME: Fun-Den Wang
COMPANY: Colorado School of Mines

QUESTION: "What is the abrasive feed rate and cost?"

ANSWER: Idaho garnet was used as the main abrasive in this study. The abrasive feed rate ranged from 1 to 5 pounds per minute. The retail cost of garnet was 15 cents per pound in 100-pound bags. However, the cost would be reduced by 50 percent if the garnet was purchased directly from the producer in large quantity.

NAME: Tom Brunsing
COMPANY: Foster-Miller

QUESTION:"Does rebound build-up of material in a blind hole reduce effect of jet or abrasive?"

ANSWER: The rebound of spent jet has not been observed to interfere with the cutting of rock with the Abrasion Jet. In fact, a stationary Abrasion Jet applied to a hard rock in a vertical position can drill a deep hole in seconds. The diameter of the hole is determined basically by the amount of water and abrasives consumed and the jet configuration of the nozzle involved. A hole of about 0.5 inches in diameter was made in rock when a 6-parallel-jet nozzle was used at 15,000 psi water pressure and 60 hp pump power output.

NAME: John E. Wolgamott
COMPANY: Stone Age, Inc.

QUESTION: '!What can you tell us about the wear rate of your nozzles?"

ANSWER: The water orifices of my nozzles are made of sapphire or ruby; thus, they last many hundreds of hours at a water pressure of 20^000 psi or less, and if water filters are used to remove the particles. The abrasive inlet passages of my nozzles have not shown much wear because they are designed and sized to minimize direct impingement and wear. The only item that is subject to severe wear is the slurry nozzle through which the waterjets and abrasives exit the nozzle even if it is made of tungsten carbide. Since I utilized multiple water jets that are arranged to form a "sleeve" around the throat of the slurry nozzle, the wear of this critical item is also minimized if one compares its life against that of dry or wet sand blasting. Available data have shown that a good tungsten carbide slurry nozzle should last more than a day's work before the bore becomes too large for entraining abrasives effectively. Even then the cutting ability of my abrasive water jet will not be reduced drastically as some abrasives will always be entrained into the multiple water jets without the presence of the slurry nozzle. The exact wear rate of the slurry nozzle is dependent on many factors, including the quality of the nozzle material (grade of tungsten carbide and method of manufacturing), initial bore size, abrasive type and flow rate, water pressure, and the configuration of the multiple waterjets.

ABRASIVE INJECTION USAGE IN THE UNITED KINGDOM.

N.J. Griffiths M.A. Cantab.
Sheldon Industrial Equipment,,
Birmingham, England.

ABSTRACT:

This paper follows the development of systems using high pressure water and abrasives from the early 1960's when the technique was purely used for cleaning vehicles, metal panels and some buildings as a contracting tool; to continuously running systems with wear resistant components. It explores the possibilities of factory bared installations for process industries and covers special equipment for abrasive cleaning of pipes and metal cutting systems for use in hazardous environments.

INTRODUCTION AND HISTORICAL BACKGROUND:

In the early/mid 1960's water jetting was becoming recognized as a useful tool for industrial cleaning and descaling, as well as an efficient method of drain cleaning and unblocking. However, because of the pressure limitation of existing pumps and ancillaries, particularly hose [maximum pressure usually 5 - 8000 psi], a method of enhancing the cleaning power of the water jet was sought, and the answer appeared to be the addition of abrasives into the water jet stream. The early sand guns were usually of the form shown in Figure 2.

While being a form of jet pump and efficiently conveying abrasive and accelerating it, they suffered from very rapid wear on the central abrasive passage, the angled water jets in the spool and the gas pipe extension.

ABRASIVE FEED.

The earliest sand guns had a hose from the sand gun which was led to a second operator having a bag of abrasive, into which the hose was fed. Unless this second operator was very careful as to how he skimmed the hose over the surface of the abrasive, he tended to have an intermittent feed, ranging from no abrasive to an excess, which clogged the hose and gun. This led obviously to very intermittent cleaning performance. The next stage in the development of an abrasive feed system, was the introduction of hoppers with variable orifices [see Fig.1] at the bottom, to change feed rate. These still suffered from blockages from damp abrasive and from water flowing back from the gun and clogging feed lines with wet abrasive. Pressurization of the hopper improved matters [especially when being used under water], but what was needed was a positive abrasive feed system which could be separately controlled, independent from the high pressure gun. This could alter abrasive feed rate, and be switched off separately to enable feed lines to be sucked clear of residual abrasive.

CONCRETE CUTTING GUNS.

The earliest attempts of the use of high pressure water as a cutting tool was for enhanced cutting of concrete, where the addition of abrasive could cut [be it slowly] the reinforcing bars. While the earliest attempts at such cutting systems were rather crude and subject to very rapid wear, they showed the potential of what water and abrasive could do, and most cutting heads are a development from this early equipment.

SLURRY FEEDS.

In order to get around the problem of clogging and the need to feed dry abrasive, there has been considerable development of wet slurry systems. However, these in themselves had further problems in that there is the need for a wear resistant feed pump, the abrasive has a tendency to settle out in feed lines and the mixture must be kept agitated. However, the main disadvantage is that under similar operating parameters a slurry system can be 50% less efficient than with a dry feed system. This is almost entirely due to the fact that it is very nearly impossible to feed slurry with greater than 50% by volume of abrasive. Thus, nearly half the power of the water jet is being used to accelerate water, which has next to no cleaning effect. The following graph (Figure 3) illustrates the relationship between abrasive feed and cleaning rate for typical wet and dry feed systems.

While most abrasive devices are of the type described previously, other devices have been used. They all work on the principle of the velocity of the water jet creating a venturi to entrain the abrasive and accomplish mixing.

ENGINEERING CONSIDERATIONS OF CLEANING HEADS:

Abrasive cleaning heads must have the following characteristics in order to be usefully employed :

1. Maximum cleaning speed for minimum abrasive usage.
2. Maximum band width to translate into slow linear speed when held on the end of a gun.
3. Minimum wear of internals, in order to be used reliably for long periods of time.
4. Easy replacement of wearing parts, as these must Often be replaced on site under difficult conditions.
5. Light weight when used manually to reduce operator fatigue.

Because early sand guns were made from normal steels it was found in practice, that spools and discharge pipes were lasting less than 2 hours. Wear on these units occurred in three main areas

1. Abrasive passage in centre of spool due to high velocity within venturi system.
2. Water jets wear outwards so that they no longer intersect at a fixed point, so they quickly wear through discharge tube.
3. Discharge tubes normally made from @" gas pipe wear out at best in approximately 2 hours.

In order to minimize wear, modern cleaning heads are of a different design (Figure 5).

RESISTANT MATERIALS: BARREL LIFE.

While normal steel barrel life of 1-2 hours is satisfactory in a contracting hand held situation it is impractiacal in a continuous blasting operation or process machine. It was necessary to design a wear resistant outlet tube (Figure 6).

Trials carried out early in the 1970's showed that the cleaning rate was proportional to the amount of abrasive up to a certain point. [see Fig 4] As the amount of abrasive increased beyond this point the performance began to tail off. This was because the abrasive head could entrain more abrasive than it could accelerate to optional velocity. At the same time, it was found that sudden changes in the feed rate effected the stability of the jet and hence cleaning performance. The best solution to these problems to date, has been to use a vertical hopper above a screw conveyor feeding into a secondary downstream chamber, to help dampen pulsation caused by the nature of the screw feed. This patented system has been in use with several customers for about 7 years, and has had very little in the way of either maintenance, wear or spares and has been trouble free for this period. (Figure 7.)

ABRASIVES:

Types of Abrasives.

The relative cutting power of various abrasives was determined on a range of applications, and the Figure 8 shows the results :

The interesting point as shown by this chart (Figure 8), is that a relatively low cost abrasive [approx $35 per ton] when compared with a high cost abrasive Silicon Carbide [approx' $500 per ton] has very little difference in performance.

PARTICLE SIZE.

While it is a general fact that the greater the particle size, the greater the cleaning power; above 2mm. the performance begins to tail off, while with particles above 4mm. very little cleaning is accomplished, due to the inability of the water jet to accelerate such large masses. (Figure 9).

CLEANING EFFICIENCY: SOFT ABRASIVES.

In certain circumstances, such as where the workpiece is relatively soft, easily damaged, or where a very light shot finish is required, it is necessary to use relatively soft abrasives rather than abrasives of the hard type with sharp edges, such as copper slag or Silicon Carbide. In this category are crushed walnut shells and crushed dried peach stones as used for the removal of carbon from cylinder heads in the Engine re-manufacturing Industry. Another interesting example in the use of special abrasive is 'Solugrit' this though a tradename represente s a class of crystalline soluble abrasives which after impact on the workpiece, dissolved into the water leaving no debris. However, being of a delequescent nature they tended to absorb water vapour and clog the abrasive feed system. Better results were given when this class of abrasives were fed by a positive metered system.

THE USE OF ABRASIVES IN A FACTORY ENVIRONMENT:

The use of in house factory installations for the cleaning of castings and mechanical components, b high pressure water has become farily common. These installations having an enclosed blast box 3with turntables and gimbal mounted guns, are an ideal way of manually cleaning components. With the addition of an abrasive cleaning head and metering system, the range and capability of the operations which may be carried out are greatly increased.

IMPROVEMENT OF SURFACE FINISH.

Abrasive water Jets may be used to improve the surface finish of a machined component or for the, removal of Finishing marks. In the same way burrs and sharp edges may be removed and radiused, Fig 10 shows a typical component where Finishing marks have been blended out prior to electroplating.

ELIMINATION OF SOME STAGES OF MULTIPLE LINISHING OPERATIONS.

A number of components are finished by several stages of finishing, by rotary bands of bonded abrasive paper. The use of high-pressure water abrasive can take surfaces from a coarse to a relatively fine finish, sometimes eliminating up to four stages of finishing, by progressively finer papers. Some examples of where this has been done are :Helicoptor main rotors, which are machined and Finished in a very labour intensive process, the use of water abrasive at two points in the process can eliminate up to 6 hours per rotor blade.

Wing Spar Milling :

The wing spar of most modern aeroplanes are milled on very large machines, and take up to 100 hours to machine a spar from a solid aluminium billet to its final state. At the end of this period a further time of up to 50 hours is needed to remove sharp edges, corners and burrs. If the mill is followed by an abrasive jet controlled by the same robot and N.C. programme most of this finishing time can be eliminated.

ABRASIVE RECIRCULATION MACHINES:

With water abrasive cleaning heads, taking up to 30 lbs of abrasive per minute, it can easily be seen that for a continuous process it is necessary to recirculate the abrasive and the water. Taking the water first, in order to recirculate this through high pressure pumps, it is necessary to recirculate this through high pressure pumps, it is necessary to filter it very efficiently. Multi stage filtration by a combination of sedimentation, cyclones, sand filters and final media filters, can reduce the contamination to a level of less than 10 mg. per litre and particle sizes of below 5 micron. However, even with this degree of filtration, it may be necessary to fit wear resistant materials to certain parts of the pump. This is mainly due to the hardness of the particles, 9 on the Moh scale for certain types of silicon carbide and aluminium oxide. These can travel over some critical pump components, such as valves and seats at speeds up to twice the speed of sound causing extensive erosion. The abrasive may be recirculated provided the following principles are observed :The abrasive must be separated from deposits off the workpiece, such as burrs, metal particles and flakes of rust or paint. That broken down abrasive is continuously discarded as the particles become very small and round losing their sharp cutting edge. The relative merits of different abrasives when recirculated, are shown in Figure 12.:

BREAKDOWN OF COMMON ABRASIVES

From this chart is can be seen that for recirculation the more expensive abrasives such as aluminium oxide and silicon carbide are by far the best in service. Copper slag which has very good cleaning and cutting power is totally unsuitable for recirculation as the particles shatter and breakdown on impact.

ABRASIVE USAGE:

With a device for collecting and concentrating the abrasive, such as an abrasive intensifier, [Patented Device] abrasive is broken down within a typical system at the rate of approximately 15 lbs per gun, per hour. Some device must be fitted in order to top up the system, or the abrasive must be manually topped up on a regular basis. System capacity is usually of the order of 50 kg. per gun, depending on the design of the machine and the amount 'lost' by resting on flat areas within the machine.

Aglomeration of Abrasive.

If the recirculation machine is used as part of a degreasing process, or where an aqueous rust inhibitor is added to the water, care must be taken to prevent the abrasive gelling into large amorphous masses. Similarly no additives which have a tendancy to foam must be used, otherwise the machine may dissappear under a cloud of foam. The design of each individual machine must be very carefully made to eliminate flat areas where abrasive can sit, and the conical section of tanks are sloped greater than the natural piping angle of the specific abrasive chosen (Figure 13).

SPECIAL ENTRAINING DEVICES:

Tube Cleaning Heads for Rotating Tube Systems.

Normal cleaning heads have the abrasive inlet, spool and discharge tube along a single axis However, they may be angled [see Fig. 14]

This head is mounted on a rigid lance, which can slowly be inserted into the tube, while the latter is rotated. If the speed of entry and rotation are correctly coupled together the internal pipe surface will be cleaned by a screw cutting action.

360° Tube Cleaning Devices.

There has been a need for an abrasive internal pipe cleaning device, which can give complete coverage without the need for the pipe to be rotated. Such a device for the cleaning of 50ft. 7" diameter drill pipe,is shown in Fig. 15.

This unit has many special features. The unit has two jetting heads combined on the carrier, the rear one being for abrasive jetting while the front one is for washing and the removal of debris. The rear abrasive jets are angled backward, so the unit is self propelled up the pipe. The speed of advance is controlled by the speed at which the high pressure hose is fed off a hydraulic hose reel. Abrasive is fed into the unit via the two inlets at the rear, while high pressure water is fed via a single central entry at the rear. The unit progresses up the pipe at a predetermined rate, until on reaching the far end, the front valve spindle is moved by contact with a blanking plug. This has the effect of opening a second circular jetting orifice at the front of the unit. Because the pump is new feeding two sets of jets; the pressure falls. This fall in pressure is detected at the hose reel, which reverses, thus pulling the cleaning head debris and abrasive back to the entry end. At the same time the abrasive feed is turned off and on return all debris is flushed out of the pipe. This unit is designed to run at a pressure of 10,000 psi with a flow rate of 40 gallons U.S. per minute. Figure 16 shows the relationship between cleaning head advance rate and abrasive feed for varying pump horsepowers at a constant pressure of 7000 psi.

Quality of the Jet.

In a normal cleaning head the jet quality is not of primary importance, as the flow from the discharge tube is a mixture of abrasive and water. (The better the mix the greater the cleaning efficiency of the head). In a cutting gun the jet needs to be of the best quality available with the maximum plain portion of the jet prior to breakup. The abrasive should be introduced around the jet in an even manner, and not on one side as was the case in early concrete cutting guns. While a good jet stream will travel down the centre of the discharge tube without touching the sides, a poor jet will engage the sides prior to the end of the discharge tube. In effect this will mean that the vacuum generated in the plenum chamber, will be considerably less in the case of an. efficient cutting head. (The discharge tubes remaining the same size). In order to increase the coherence of the water jet, it is usually necessary to use a polished carbide or ceramic insert.

Abrasive Feed.

Because the integrity of a cutting jet depends on the stability of a pattern of abrasive around a water jet, any change in the amount of the abrasive will be reflected in the depth and quality of the cut.

Even the rotation of a feed screw conveyor, with each time the screw goes over top dead centre of the pitch, will be shown on. the workpiece. Normal screws with a feed rate of approximately 11b of abrasive per revolution, quite satisfactory for cleaning heads,are too coarse for cutting systems. Either a high speed screw is necessary to reduce pulsation or a metering device with a completely steady feed is needed. Even the band spread within a range of graded abrasive, can have an adverse effect on cutting capability with large particles upsetting jet stability.

Plenum Chamber.

This is the chamber where the abrasive mixes and surrounds the water jet, and will usually taper at one end to the discharge tube diameter. As this chamber is where the abrasive is introduced and accelerated to the speed of the water jet, sometimes changing direction, it is necessary that this chamber be lined with a wear resistant material such as tungsten carbide. The diameter of the chamber is in the order of 10 - 20 times that of the water jet and 2 - 4 times that of the discharge tube.

Discharge Tubes.

Because of the self destructive nature of abrasive cutting equipment, and the fact that the abrasive normally surrounds the water jet, this item is subject to extremely rapid erosion. A hardened steel 1/4" diameter discharge tube can wear in size up to 3/8" in less than 15 minutes operation. Special grades of tungsten carbide is the only material which has proved to have any sort of life for this component. Average life of a discharge tube is in the order of 25 - 50 hours when used with a pressure of 10,000 psi, 15 lbs per minute of copper slag abrasive, .080" water nozzle and '1/4" discharge tube diameter.

The diameter of the discharge tube has a very marked effect on cutting performance. Maximum rate of cut of 1/2 " mild steel plate, varies from 2" per minute with a .310" discharge tube, to over 91/2” per minute with a 0.178" discharge tube. Generally the smaller the tube the greater the speed or depth of cut. However, if this diameter is reduced too much, the head will clog; there not being sufficient area to allow the water and abrasive to escape without back pressure.

The length of the discharge tube is not of such a critical nature and is usually in the order of 21/2" to 4" from the plenum chamber to discharge point into free air.

CUTTING PARAMETERS.

As well as factors which have already been discussed, such as outlet tube diameter, the rate of cut is also effected by the following

Pressure:

Generally the rate of cut is increased the higher the pressure is (Figure 17). Flow and nozzle size will also have their effect on the final performance.

Standoff Distance:

While it is generally true that the closer a cutting head is to the work piece the more efficient the cut, this cannot be carried to extremes. In the same way that too small a discharge tube will cause the head to back pressure and clog, the same thing will happen if the discharge tube is too close. This is particularly true if the head is at right angles to the work piece. For all practical purposes this means that the head should normally not be used closer than a 1/4” from the work piece. The maximum distance at which effective cutting can take place is determined by the operating pressure, coherance of the water jet and the horsepower of the power unit. As an example a 75 horsepower unit is capable of cutting through 4" steel plate, 18" of reinforced concrete and 10 X 8 'I' Beams in a single slow pass.

Quantity of Abrasive.

The rate and depth of cut are proportional to the amount of abrasive within certain limits. Above a certain point the performance drops off in the manner shown in Fig 4 as for abrasive cleaning heads.

Abrasive Type.

The cutting performance of various abrasives is shown in Fig.18. Again the cutting effect of copper slag and silicon carbide is similar with sand being a very inferior abrasive.

CUTTING DATA:

Steel Cutting. Figure '19' shows a .5" mild steel plate which has been slotted using a water abrasive cutting head. As in most cutting operations the head was angled in the direction of travel approximately 30°, so the deflected jet was carrying out a preliminary grooving cut. The slot tapered from approximately 1/4" on the entry side to 7/32" on the exit side of the cut. The quality of the cut with the absence of rough edges and burrs, as formed by flame cutting and abrasive wheels, is an ideal weld preparation for joining pipes and plates together.

Stone Cutting.

Fig.'20' shows a piece of Darley Dale Limestone slotted using water and abrasive. This sample was cut under the following conditions :2" thick Darley Dale Limestone cut at 36" per minute, 8000 psi and 12 lbs copper slag per minute. Similar performance has been achieved in other soft rocks, such as sandstone while up to 2/3 this performance has been achieved in hard rocks such as Dakota Granite.

Concrete Cutting.

Figure '21' shows a piece of 2" thick concrete slab, with 1/4” reinforcing bars at 1" intervals. Cutting conditions were the same as for limestone except for traverse rate which was 15" per minute.

Deflashing of Castings.

The water abrasive cutting head may be used to remove flash, risers and fillers from complex castings. It leaves a finish free of burns and burnt on sand. It is particularly appropriate to the removal of mould bosses from investment casting in hard materials, such as high temperature aircraft alloys.

Cutting of Hard Materials.

Some work has been carried out on the cutting of hard materials, which are very difficult and expensive to cut using conventional techniques. 3/8" Chobham Armour plate may be cut at speeds up to 5" per minute using a 75 horsepower cutting head at 8000 psi. Similarly bars of nimonics, waspalloys and titanium alloys may be cut at speed similar to and in some cases faster than existing cut off machines. In these cases there is a viable economic justification once the cut width is reduced. The present cut width of just under 1/4” wastes too much of these expensive materials.

Cutting in Hazardous Environments.

Considerable work has been carried out by BHRA Fluid Engineering' into the use of water abrasive cutting heads in highly inflammable atmospheres. The cutting of steel and stone has been carried out in atmospheres containing explosive mixtures of methane, propane, acetylene, hydrogen and other highly inflammable gases. This work was carried out because cutting with a water jet and abrasive can generate sparks, which can be seen in a darkened environment. As they have never caused an ignition, it is thought that they were of too low an intensity to be of danger. Further works showed that far fewer sparks were generated when the abrasive was supplied as an aqueous slurry. The work was carried out in conjunction with the 'Safety in Mines Research Establishment' of the 'Health and Safety Executive'. While not giving a categorical guarantee of the inherent safety of such equipment, they are satisfied enough to wish for full in house tests within a mine. The equipment was also designed by 'BHRA' in conjunction with 'BP' for use on off shore oilwells in emergency cutting situations. The first set of equipment is now under construction for use on an offshore rescue vessel. 'BHRA's contribution to the development of abrasive systems in Europe has been considerable, in the form of wide angle clearing heads, slurry feeds, underwater systems, cutting systems in air and in explosive environments.

CODE OF PRACTICE FOR THE USE OF ABRASIVE ENTRAINING CLEANING AND CUTTING HEADS:

The code of practice recently published by 'The Association of High Pressure Water Jetting Contractors' is now being extended into the use of abrasive cutting and cleaning systems. It will cover such aspects as :

Power Units:
Cleaning Heads:
Abrasive Feeds [Wet & Dry]:
Handles:
Trigger Guns and Shut Off Devices:
Discharge Tubes:
Abrasives - [Grades-Contamination-Storage arid Disposal]:
Site Marking:
Cutting Systems:
Fire Risk Environments:
Contaminants of Debris:
Protective Clothing:
Permanent Abrasive Blast Areas:

This code is being produced by a committeee consistiing of members of 'the association of High Pressure Water Jetting Contractors', BHRA Fluid Engineering and Manufacturers in conjunction with the 'Factory Inspectorate' and the 'Health and Safety Executive'.

ACKNOWLEDGEMENTS:

The Author would like to thank the help and assistance given in the production of this paper by the following organisations:

1. Association of High Pressure Water Jetting Contractors, 33 Catherine Place, LONDON SW1E 6DY.
2. BHRA Fluid Engineering, Cranfield, Bedford, MK43 OAJ. [3]
3. Paul Hammelmann Maschinenfabrik GmbH & Co. Kg. Postfach 3309, D47-40, Zum Sundern 13-17, OELDE 1, WEST GERMANY.
4. Reliance Hydrotech Limited, 7 Ashlyn Road, Meadows Industrial Estate, Derby DE2 6XE.
5. Sheldon Descaling Contractors Limited, 117-122 High Street, Bordesley, BIRMINGHAM B12 OJU.

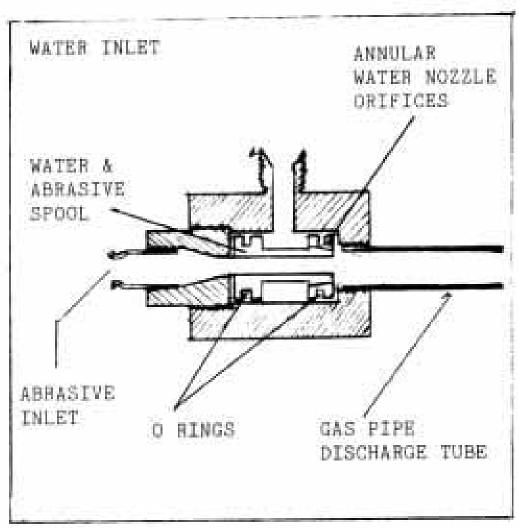
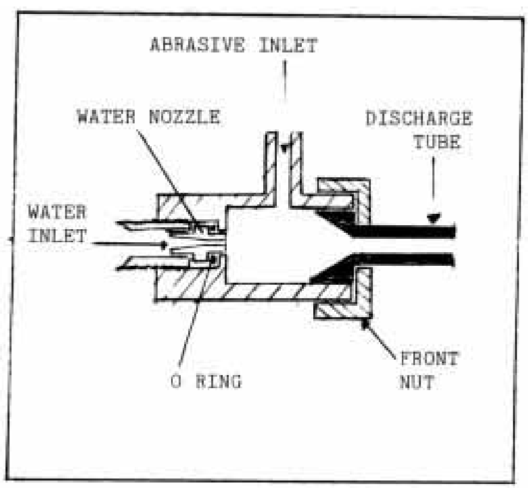


Figure 1. General cleaning with abrasive head and gravity feed hopper.

Figure 2. Early spool type abrasive cleaning head



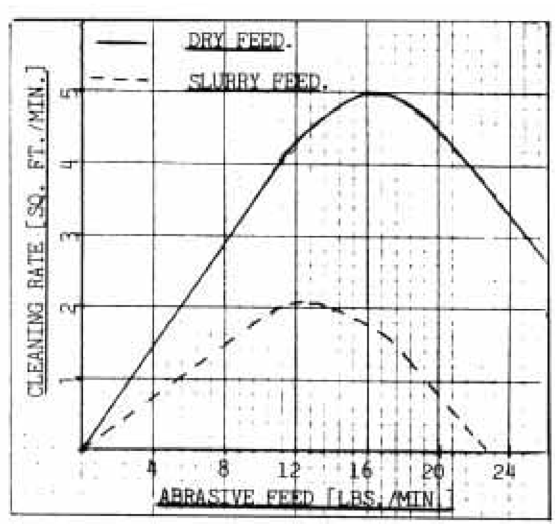
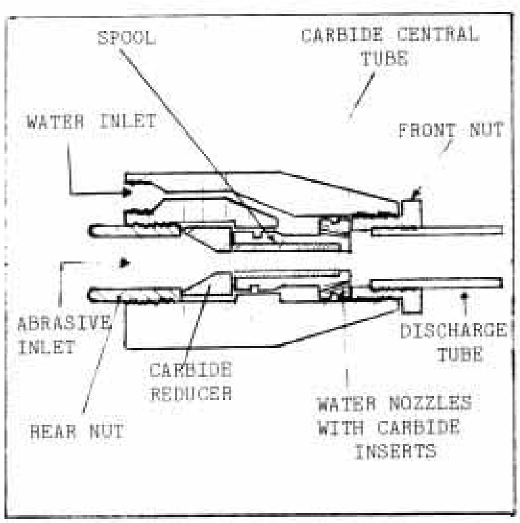


Figure 3. Early water abrasive cutting head

Figure 4. Relationship between abrasive feed rate and cleaning rate for wet and dry feeds.



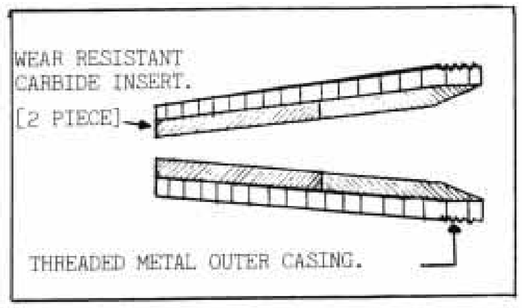


Figure 5. Water abrasive cleaning head with wear resistant materials.

Figure 6. Wear resistant constrictor outlet tube.

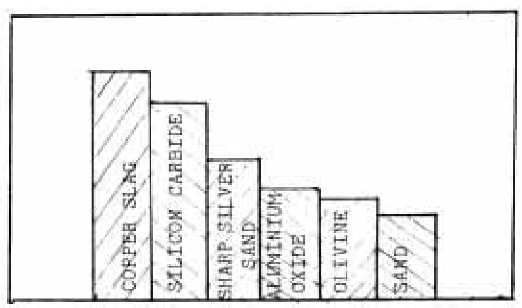


Figure 7. Abrasive metering unit.

Figure 8. Abrasive cleaning efficiencies for different abrasives.

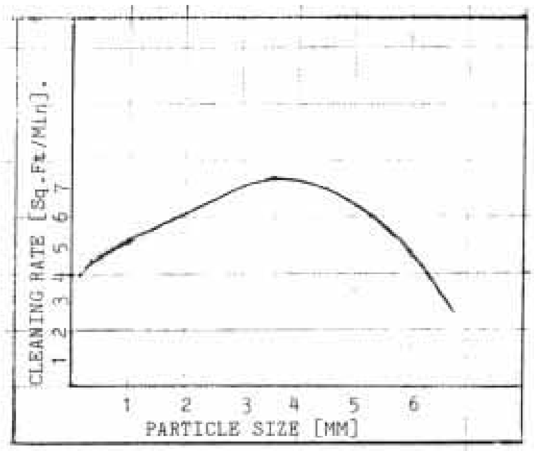


Figure 9. Relationship between particle size and cleaning efficiency.

Figure 10. Components prefinished by abrasive blasting.

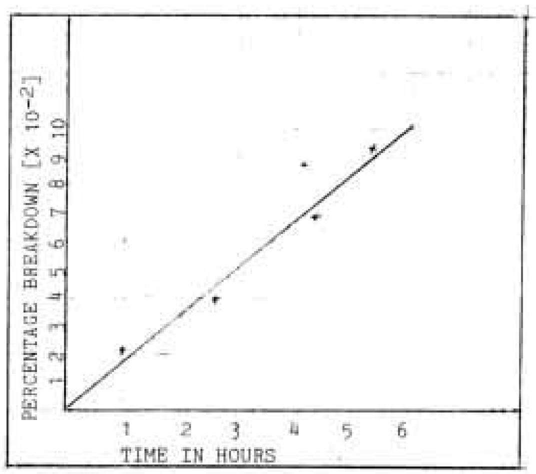


Figure 11. Relationship between abrasive breakdown and running time.

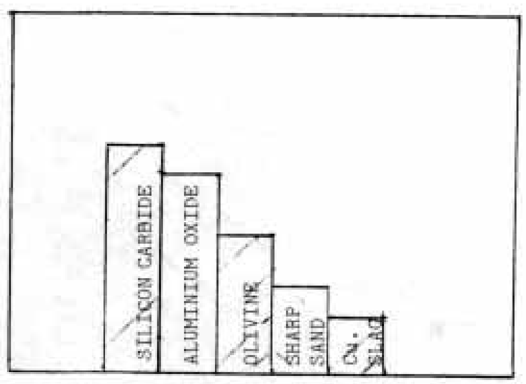


Figure 12. Bar chart showing the resistance to breakdown of common abrasives.

Figure 13. Typical factory based installation

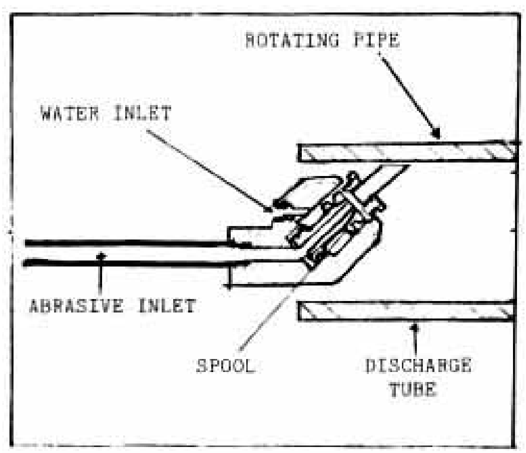
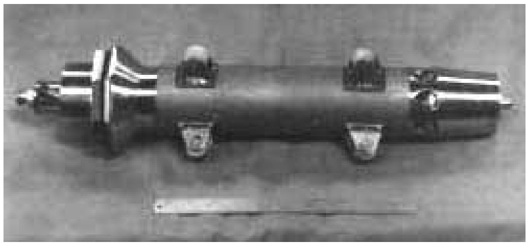


Figure 14. Angled cleaning head for internal pipe cleaning.



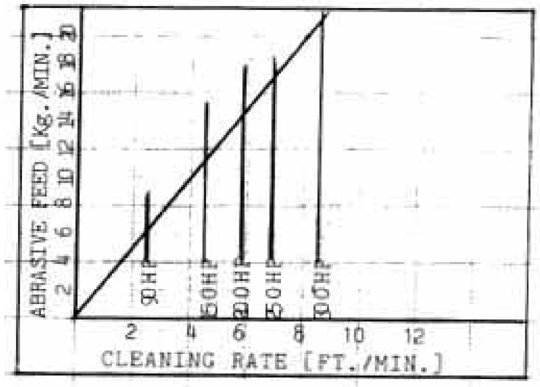
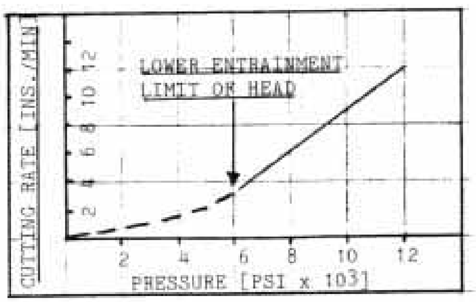


Figure 15. 360^0 abrasive pipe cleaning head.

Figure 16. Parameters effecting the performance of a 360^0 pipe cleaning head.



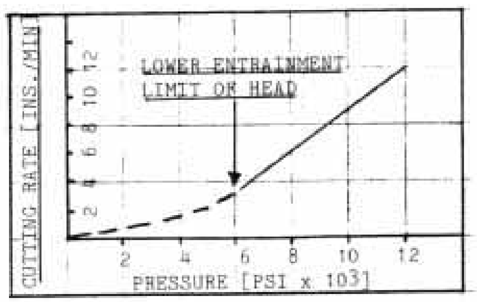
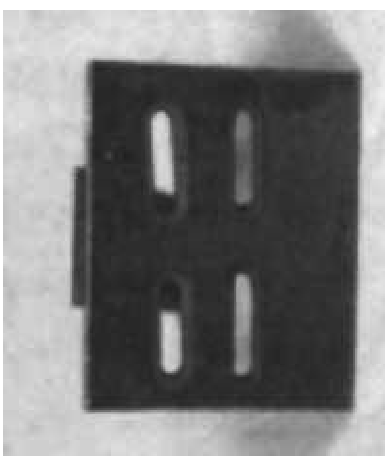


Figure 17. Relationship between cutting rate and pressure.

Figure 18. Cutting performance of different abrasives under similar operating conditions



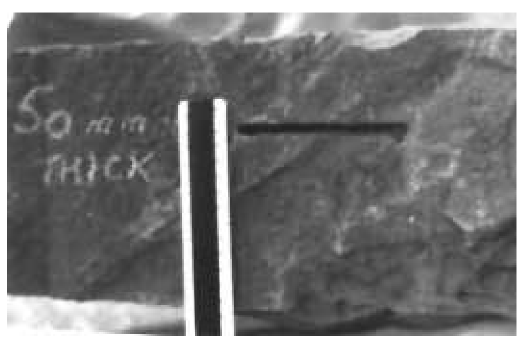


Figure 19. Steel plate cut at 8,000 psi 6” per minute and 12 lbs of copper slag per minute.

Figure 20. Limesotne cut bu a water abrasive jet.

Figure 21. Reinforced concrete cut by a water abrasive jet.

ECONOMIC CONSIDERATIONS IN WATER JET CLEANING

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ABSTRACT

Some of the economic factors acting on water jet cleaning are investigated.

The Economics of Equipment are discussed including capital costs, running costs and effective use. The effects of increase in pressure and decreasing water quality are described. The economic effect of Personnel and training and some examples of the learner curve are also outlined. Investigations as to the effects of experience showing increasing effectiveness over increasing time are described. Manual and Automated systems are compared and the improvement gained from quite minor increases in automation are indicated. Pricing and the various factors acting on price determination for the contractor are shown including effects of competition, operator experience and motivation and the differences between day work and fixed price contracts are discussed.

INTRODUCTION

The use of high pressure water as a tool for the removal of deposits from surfaces, tubes and pipes has only recently received the attention which it has merited. However, to date, this attention has focussed on applications, theory and the mechanism of water jet cleaning and papers and information dealing with practical determinants in this area are indeed rare.

In this paper I shall attempt to outline some of the economic factors which have been shown to act within the industry,

I consider these factors to fall into four main groups.

1. Economics of Equipment.
2. Personnel and Training,
3. Manual methods and Automation.
4. Pricing - The Company/Contractor Interface,

In all these sections I have used UK prices transposed into US dollars for ease of understanding.

1. ECONOMICS OF EQUIPMENT

The equipment necessary for carrying out general water jetting operations is a prime mover, a high pressure pump and ancillary equipment such as hose, guns, lances, foot valves and nozzles.

As a general rule for diesel driven prime movers and non specialised applications the market price at present is approximately $300-500 per horse power of prime mover for a total equipment package. Since all manufacturers will claim that their pump has obvious advantages over its rivals in reliability, ease of maintenance and running costs, a critical assessment should be undertaken prior to purchase of any system.

An examination of data gathered over a ten year period yielded the following results for comparable pumps of different manufacture but of similar nature ie Positive displacement plunger pumps

TABLE 1 Cost of running spares per hour for pumps
from four manufacturers @ 6000 psi (Reliance Hydrotech 1977-80)

Horse power of		Manufacturer		
Prime Mover	A	B	C	D
40 HP	2.00	5.00	6.00	4.50
75 HP	2.50	6.00	6.50	5.00
150 HP	3.00	9.0O	ll.00	6.00
220 HP	5.00			

Thus running costs are dependent on both the manufacturer of the pump and the horsepower of the prime mover. But there are other variables to be considered. It is well understood (Paul Hammelmann GmbH 1979) that in manual, surface cleaning operations an operator should not be required to take more than a 55 lbs thrust.

This can be discovered by the approximation

$$R = \frac{66xHP}{\sqrt{P}}$$

where

R = reaction force in lbs thrust
HP = horse power of prime mover
P= pressure in psi.

In the above circumstances ie using 55 lbs thrust as the limit of manual use and utilising the full horse power of the prime mover, fuel usage for a diesel engine has been shown to have the following approximations

TABLE II Fuel usage in jetting operations using single and multiple outlets with a 55 lbs thrust limit placed on each outlet. Determined over 8 hour jetting shift. (Reliance Hydrotech 1980)

HP of Prime Mover	Fuel Usage imp/gal/hr
40	1,5
75	2.5
150	5.0
220	10.0

Other criteria to be considered in outlining the economic factors acting on equipment choice and usage are

(i) the quality of the water delivered to the pump

(ii) the pressure at which the pump is used.

Moodie (1983) gives the approximation for the escape velocity of water from a high pressure system as

$$V = 12\sqrt{P}$$

. Where

V = velocity of water feet/sec. and

P = pressure in psi.

Thus, an increase in pressure leads to an increase in wear within any high pressure system

Figure I Shows the increasing running costs for a 75 HP pump with increasing pressure (1979) 1000 to 14500 psi

Increasing debris within a system also leads to an increase in running costs and these increases can sometimes be of an horrifically steep nature and do vary from manufacturer to manufacturer.

From Fig II it can be seen that unacceptably high levels of wear due to suspended solids (A) can be ameliorated by modification of pump parts (B) by the addition of wear resistant tungsten carbide this adds some ,$40-60 per HP to the capital cost of the equipment but can prove to be very cost effective.

2. PERSONNEL AND TRAINING

Since the inception of water jetting in Europe after the Second World liar there has been a steady move towards professionalisation in the methodology. This drive has been reflected in the establishment of Codes of Practice - Association of High Pressure Water Jetting Contractors (1982)-which are now mandatory in most EEC countries. The current Code of practice in force in the UK together with Health and Safety Legislation - Health & Safety at Work Act (1974) makes it a duty upon the employer to utilise only <u>trained personnel,</u> though "trained" is loosely defined.

However, from the point of view of consideration of costs it is true to say that the learner curve is a potent criterion in choice of personnel to carry out a particular task. A series of ferrous investment castings which the operator had not previously seen were cleaned using a 125 HP pump delivering 6500 psi @ 22 gpm by two operators. The castings were mounted centrally on a tilt/rotate turntable within a cabinet and water was delivered through a gimbal mounted gun to take the reaction force. A series of castings from the same batch were cleaned by two operators. Operator A had some ten years' experience and was told that this exercise was part of a training scheme, and that he was to complete a batch of castings as quickly as possible and to train

Operator B. who had approximately one year's experience on general water jetting, but no experience on either the use of a jetting cabinet nor the cleaning of investment castings.

The results obtained in Fig III clearly show both the learner curve for both Operator A - the more experienced man, and for Operator B. It is interesting to note the considerable disparity between the two operators' final times, which is much greater than the disparity shown at the beginning of the operation.

Similar figures showing the effect of experience on the cleaning operation can be extracted from engineers log books,and other variations in operator performance which have been suspected can be more clearly identified for investigation, Table III shows an effect similar to that previously described in Fig III. For the same piece of process plant a series of boiler heaters which were cleaned by the same crew over a two day period - 13/14 July 1973.

TABLE III Boiler heater cleaned using 5000 psi 12 gpm (Galinski 1973)

Heater clean Number	Time taken to clean
1	52 mins
2	50 mins
3	43 mins
4	36 mins
5	34 mins
6	31 mins
7	30 mins
8	29 mins
9	27.5 mine
10	28.25 mine

Operator experience can be all important are the operator often must work away from base, and therefore he must solve his own problems and be able to decide on equipment required, nozzle type and size and be able to fault find accurately on his own equipment.

Fig IV shows the effect of experience on a group of operators taken from the same social class on the same job. Operators with differing levels of experience were shown the job. This was 12" pipe work with heavy lime scale, with partial blocking. The operators were given the same equipment, a 150 HP pump performing @ 9500 psi @ 12 imp gpm, and a range of ancillary equipment with the instruction to clean as much of the pipe work as possible during an 8 hour shift

From Fig IV it can be seen that increased experience does relate to increased work rates. A university graduate with approximately two years' experience in water jetting was given the same task as that described above and cleaned some 85 m of pipe work in a single shift (shown X in Fig IV) thus indicating that whilst experience and training is of paramount importance it should not always be equated with expertise.

3 MANUAL METHODS OF JET CLEANING & AUTOMATED METHODS

It is true to say that over the past few years automation has come to the fore as a method of jet cleaning. This has obvious advantages in that the operator is removed from a dirty, uncomfortable and all too often dangerous situation, Where it is not absolutely necessary to clean automatically it is also not always preferable to clean automatically. Table IV is a composite table comparing the costs of cleaning condenser/tube bundles of varying lengths with varying numbers of tubes using existing automated equipment and conventional manual methods. Griffiths (1979).

The tubes in the bundles are assumed to have a soft/medium water scale and the tubes are assumed to be unblocked.

The running costs for the systems are US$ and include amortisation of equipment over a 6 year period assuming 80% use, fuel, spares and consumables and labour,

The fuel component is 17 gph for the Automated system and 10 gph for the manual system.

The spares/consumable element is $12.00 for the Automated System and $8.00 for the manual, Labour has been estimated at $12,00 for automated and $18.00 for the manual.

TABLE IV

Length of Bundle	5ft		l0ft		15ft		25ft	
Cost per Bundle	M	A	M	A	M	A	M	A
Tube No,								
100	25.	78.	51.	78.	102.	78.	155.	78.
500	102.	117.	357.	117.	459.	117.	561.	156.
1000	204.	195.	876.	195.	969.	195.	1173.	234.
2000	408.	312.	1530.	312.	1836.	390.	2240.	429.

From Table IV it will be seen that small tube bundles do not require automation, though this can become a critical necessity when long bundles with large numbers of tubes are tackled, especially at shutdown periods.

The rate at which impellers have been cleaned has been measured and costed Swan (195^8) using (1) manual methods ie casting cleaned in open space outside (2) a casting cleaning machine, with a method of clamping and the facility to approach the casting through 360° in the horizontal with a tilt mechanism through 115°, thus allowing the casting to be approached from all angles - See Plate I Photograph of casting cleaning machine showing gimbal gun and turntable.

TABLE V Casting cleaning impellors

Batch of 100 castings cleaned using
(a) Manual - 75 HP pump @ 9500 psi 10 gym
(b) Machine - 75 HP pump @ 9500 psi 10 gym

	Manual costs $	Machine costs $
Capital equipment	30,000	50,000
Time taken over 100 impellors	100 hours	25 hours
Running costs	500.00	125.00
Spares/ Ancillaries	250.00	62.50
Labour (2 men)	1200.00	(1 man)150.00
Amortisation	300.00	125.00
Total	2250.00	462.50
Cost per unit	$22.50	$4.62

From Table V it can be seen that for quite moderate degrees of automation considerable savings can be made, not only in time and money but in a more attractive and safer working environment.

The above Tables TV and V can be replicated for many applications but automation is often not cost effective in certain situations, for example:Where access is limited - eg in situ condensers in many fossil fueled power station applications or Where fine movements and manipulation are required - eg small bore passages. Other circumstances may also intervene, for example heavy and intermittent blockages in tubes can cause damage to fixed motion lances.

4. PRICING - THE CONTRACTOR/CUSTOMER/OPERATOR INTERFACE.

With the economic recession from which we have suffered in the UK over the past almost ten years we have seen competition grow fiercer, prices become keener and the customer put more and more pressure on the contractor to produce ever more for less return,

This has in turn been exacerbated by the grossing requirement of the operator for more pay to survive in a continuing inflationary spiral. Fig V shows hourly charges from a contractor (Reliance Descaling 1967-82) to a customer, and wages paid by the contractor to operators over the same period,

From Fig V it will be seen that wages have increased by a factor of X5 whilst charges to the customer have increased by a factor of X2,3, thus profit margins have been steadily eroded. It is also interesting to note the considerable decline in the rate of increase of cost to the customer after the introduction of competition in 1972.

This is now an active policy pursued by many major customers of water jetting contractors in the UK, That is to say, two contractors are used. In its most unpalatable form one contractor with old, unsafe, equipment, untrained operators and low overheads may be seen as a "good buy" because of lower rates. This is often due to lack of knowledge on the part of the customer.

Alternatively the above facts may be recognised, albeit tacitly, by the customer but the cheaper contractor may be simply kept as a bargaining ploy by the customer to depress prices.

The customer may demand priced jobs as opposed to hourly paid or day work, In these circumstances the contractor will use the following "rule of thumb", or an adaptation thereof, depending on overhead expenses

(Estimate hours/days completion) X (Hourly/date rate) + (Expenses, travel & subsistence) + 50%, Contingency = Price for job,

The contractor will also sometimes overestimate the time required because he knows that customers will sometimes misdescribe a job or provide optimistic information as to the condition of pipework etc. which he often has to take at face value, and with a shutdown budgeted if he makes a loss then usually that is his misfortune.

If the contractor does take priced work then usually he will put his operators on a bonus. Fig VI shows the comparison between operators of differing levels of experience working on the same job at hourly paid rates and bonus rates for priced work

Obviously, operators will work more diligently if they are paid more and the job will be completed much more quickly but since the customer may not wish to pay the premium for speedy completion, and this is always a matter of negotiation, the contractor in the UK is more likely than not to work on an hourly or daywork basis.

Customers are now becoming more sophisticated, they are sending engineers for training, in what is for most of them, a new technology. This can only be of benefit to the contractor and thus the operator in the longer term.

With the increasing number of applications being found every day the vendor of service or equipment is having to become more expert in a wide range of applications, but no matter how expert one may become One can always learn more.

As a cautionary tale for those who price based on incomplete knowledge, we used to like to think that we knew just about everything there was to know about the cleaning of aero engine turbine blades manufactured for or on behalf of a certain well known UK jet engine manufacturer. About six months ago one of their major sub contractors asked us if we could attempt to clear some 14-16 thou cooling passages in some of their scrap blades. We attempted the job and were successful at about the 50%, level. The manufacturer was very excited and immediately ordered a machine for this purpose from use asking us in the meantime to continue to clear cores from blades which would otherwise be scrapped, and to quote a price.

Having seen many blades of a similar nature, and knowing, as we thought, the value, we undertook the operation on a "what the job will bear" basis for a price of some $3 per blade, which since we could process 60 blades per hour we were not unhappy with. After we had reclaimed some 100 scrap blades the customer proudly announced that their machine, as yet incomplete, was now more or less amortised. The blades we had processed were high integrity, experimental, single crystal castings worth approximately $5-600 each.

The moral of this story, and indeed this paper, is that the primary determinant of any successful water jetting operation is expertise, which is a function of experience.

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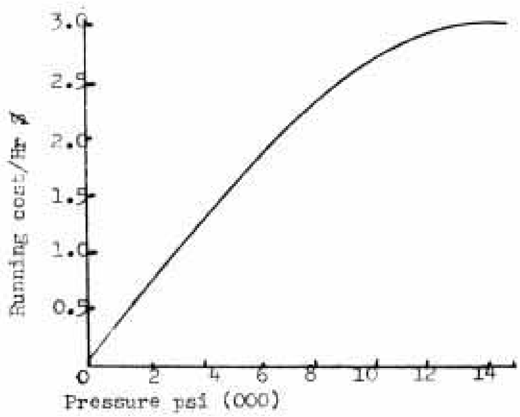


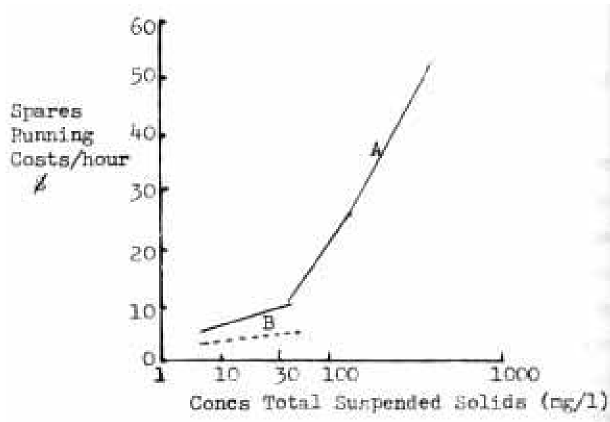
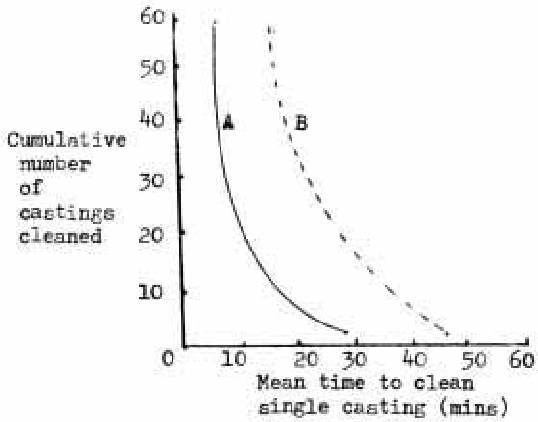


Figure 1. Showing the increased costs of running a pump with increased pressure 1979) 1,000 to 14,500 psi..

Figure 2. Running costs per hour for 75 HP with varying amounts of alumino silicate in suspension at 10,000 psi (Swan et al 1978)



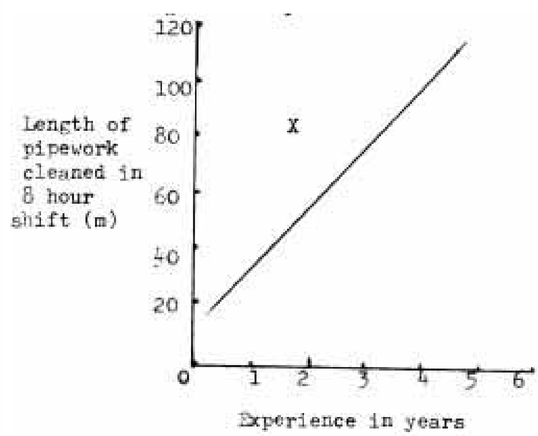


Figure 3. Learner curves for operator s cleaning castings (Swan 1980)

Figure 4. 12" pipework with heavy lime scale and partial blockage length (M) cleaned @ 9500 psi @ 12gpm (Johnson 1980) by operators of differing lengths of experience (X = graduate operator).

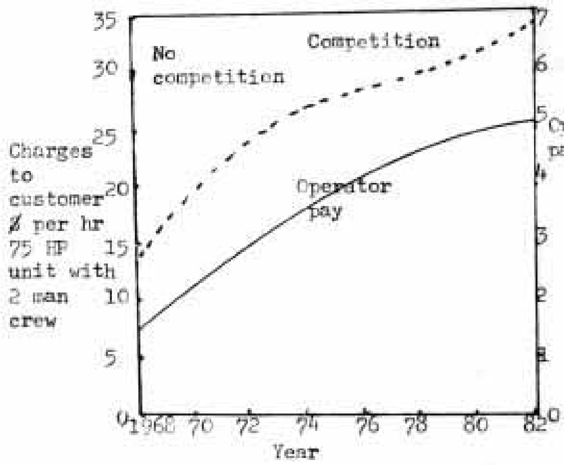


Figure 5. Comparison of charges and costs (1967-82)

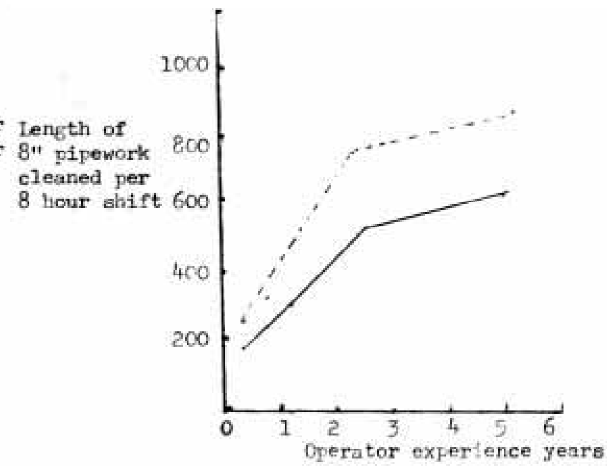


Figure 6. Length of 8" pipe containing soft deposits cleaned in m/8 hr day by operators of differing levels of experience on hourly paid work (continuous line), and on bonus (dotted line).

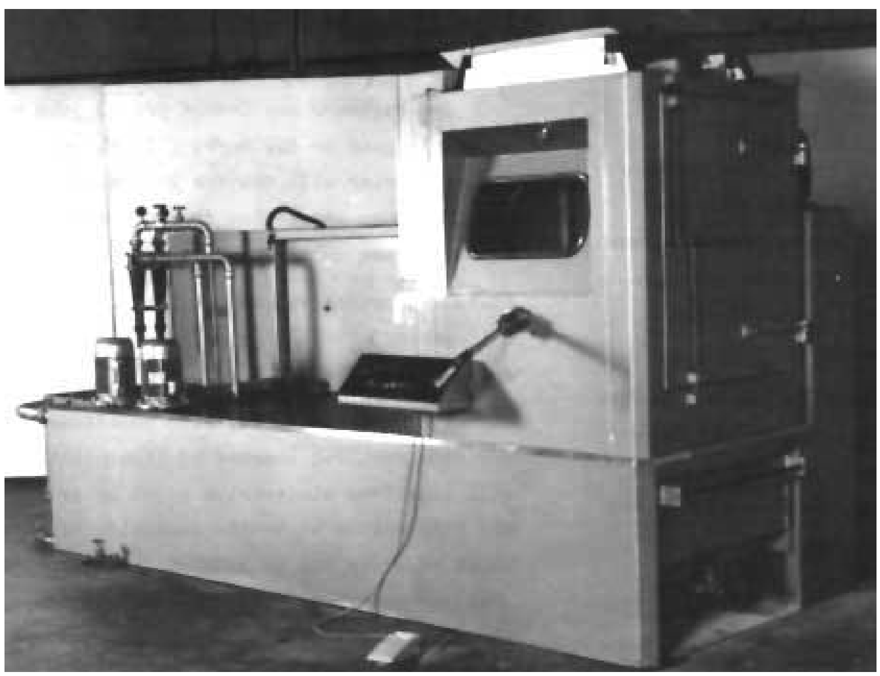


Figure 7. Minimal automation for casting/small part cleaning with water recirculation (see Table V)

Figure 8. Automatic tube bundle cleaner (see Table V)

POLYMERBLASTING - A CHEMIST'S POINT OF VIEW

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ABSTRACT

The purpose of this paper is to describe the background applications, chemical and other important considerations of the use of SUPER-WATER concentrated industrial water blasting additive. Because SUPERWATER is a polymer, the process in which it is used is often called polymerblasting although water is still the principal blasting medium. The polymer is used at a concentration of only 0.3% but even at this low concentration the distinctive characteristics of the resulting jet flow are quite remarkable.

BACKGROUND

For many years it has been known that the inclusion of low concentrations of various additives in water bring about dramatic changes in flow. These additives can be soluble or insoluble in water and although polymerblasting in the context of this discussion centers on a soluble polymer, it is worthwhile noting some of the generally important historical landmarks. Initially, the additives were not intentionally added. Exactly 100 years ago (1883), there existed records of increased flow rate in silt laden rivers (1): the term "drag reduction" to describe this general phenomenon was not coined until 80 years later by Savins (2). Credit for the first drag reduction experiment with an additive should probably be assigned to Hele-Shaw (3). He was interested in the skin friction on marine animals and was able to show that the addition of fresh bile, a somewhat bizarre additive, to water appeared to bring about reduced flow resistance (or drag reduction). Since those days, a large number of publications has appeared on drag reduction and in the period from 1956 to 1976, it is estimated that nearly 1,000 publications (4) addressed themselves to this topic. Despite these extensive research activities, however, no totally satisfactory theory has yet been developed to explain drag reduction (5).

A second characteristic exhibited by water containing additives is that of jet focusing. In 1964 (6) an article was published describing fire fighting with (so-called) viscous water. It was noted that "once the water using this viscous water (actually a solution of sodium carboxymethylcellulose) has left the nozzle of the fire fighting hose, it is less likely to be dispersed into a spray by the wind; thus, the jet travels further than ordinary water would". Other articles describing jet cohesion appeared in Chemical Engineering (7) and Fire Engineering (8). Within the same general time frame that these observations of water jet cohesion were published, Gadd (9) reported parallel findings for jets (containing dye) submerged in water. Of even greater significance to this discussion, however, is that Summers (10) described the use of polymeric additives in high pressure waterblasting.

In 1973, Russian workers (11) reported the "Destruction of metallic obstacles ("objects" would have been a better term) by a jet of dilute polymer solution".

The above is a brief and far from complete account of the knowledge that was available in 1974 on the use of polymer additives in aqueous solution. It did, however, provide the background to the subsequent cooperative project at the Chevron U.S.A. Richmond Refinery in California. This involved members of the Chemical Cleaning Department, Berkeley Chemical Research, Inc. (supplier of SUPER-WATER) and Mr. Douglas E. O'Connell now President of D.E. O'Connell and Associates, Martinez, California 94553. The experimental work started in late 1974. It involved studies of different polymeric additives, various methods of injecting the selected additive into the water used for blasting and many applications. The results of these studies logically lead to the initial report by Chevron U.S.A.'s Richmond Refinery entitled "Hydroblasting Exchangers - Less Time, Better Cleaning". This report was published in The Exchanger in 1978 (12).

APPLICATIONS OF POLYMERBLASTING

In the Exchanger article (12), some details were given of the extent to which "less time, better cleaning" of exchangers was achieved. In relation to hydroblasting, it was noted that a 30% reduction in manhours could be obtained by polymerblasting. Additionally, bundles that would previously have require retubing because of the extent of fouling were put back into service after polymerblasting. In many instances, such bundles fouled with heavy coke or hard deposits, if not requiring complete retubing, would formerly have been mechanically reamed: now they proved amenable to polymerblasting. At the February 1982 NPRA Meeting (13), during the Conference's Workshop on Chemical Cleaning, participants related to the use of polymerblasting for the following applications (14):

1. Heat exchangers: shell-side (more effective stand-off distance) and plugged tube-side (rigid lance with nozzle small enough to allow debris to be flushed backwards).
2. Removal of epoxy-bound rubber from various vessels/containers.
3. Internal (large-diameter) pipeline cleaning.
4. Vacuum filter piping "L's" and T's".
5. Falling film evaporators in soybean processing plants.
6. Removal of very hard coke deposits after refinery fire.
7. Plugged second-stage urea decomposer tubes.
8. Coke from cokers and gas effluent overhead.
9. Coke from discharge headers of coke gas compressors.
10. Calcium carbonate and slurry deposits from boiler slagging slurry lines.
11. Dense magnesium salts from boiler tubes.
12. Calcium carbonate from boiler tubes.
13. Calcium and iron fluorides from hydrofluoric acid alkylation plants.
14. Hard scale from sulfur condensers in (Benson)sulfur recovery systems
15. Removal of coke from vapor lines.
16. Removal of coke from coke heater barrels.
17. Used in automated system (cf. Butterworth) for removing plastic, at large stand-off distances, from batch reactors.
18. Saved bundle retubing by removal of otherwise intractable shell-side deposits.
19. Sewer cleaning.

The participants, all users of SUPER-WATER , came from throughout the United States, including Hawaii, and here it is appropriate to note that the product is now in use in Canada, Europe, Israel, and the Caribbean.

An additional application for polymer takes advantage of the findings of Gadd (9). Thus as delineated in the brochure on SUPER-WATER by Berkeley Chemical Research, Inc. (15) it was experimentally determined that a 15 to 30 fold increase in effective stand-off distance was achieved when plugged slots in watersubmerged oil well liners were cleaned with polymer. This finding is now being applied industrially with great success.

The wide diversity of the applications for which polymer has been used, the reduced job time and the resulting savings of maintenance and production costs, strongly suggest that additional applications merit consideration.

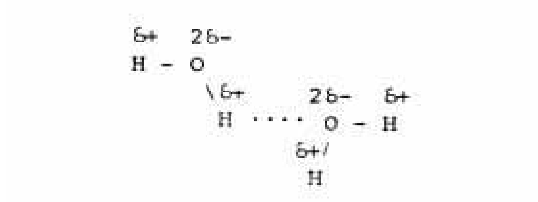
THEORETICAL CONSIDERATIONS OF POLYMERBLASTING

In the background on polymerblasting three distinct aspects were discussed namely drag reduction (1-5), jet focusing (6-10) and "the destruction of metallic objects by a jet of dilute polymer solution" (11). For this last listed phenomenon the term "macromolecular polymer bombardment" is now being used (15) although Summers and Zakin (16) have refuted the underlying hypothesis. Citing previous studies (10 and 17) these authors pointed out that filtration at 10 microns of polymer solutions before entering pumping systems still exhibited improvement over plain water jets in impact erosion of rock. They reasoned that large aggregates acting as solid particles would be held back by a 10 micron filter. It is known from simulated enhanced oil recovery studies (18) that synthetic high molecular weight polymers in aqueous solution undergo hydrodynamic degradation when passing through five closely-packed disks of 100 mesh. A 100 mesh screen has an opening of 150 microns so five closely-packed discs of 100 mesh, which inevitably would be offset or staggered, could have apertures approximately 10 microns. On the other hand Herr and Routson (19) had previously conducted very elegant studies on polymer aggregate stability using high resolution electron microscopy. They determined that a 30% hydrolyzed polyacrylamide of 4×10^6 molecular weight in aqueous solution retained its "supermolecular structure" when passed through an 8 micron and even a 3 micron filter. The onset of disaggregation only occurred on passage through a 1.2 micron filter. Herr and Routson pointed out that a well solvated molecule of 4×10^6 has a length when fully extended of 14 microns and a molecular diameter of 7 - 27 angstroms. This molecular diameter is some 4 orders of magnitude less than the 10 micron filter used by Summers and Zakin. Therefore it would appear, quite conceivable, depending upon flow rates and other parameters, that a 4×10^6 molecular weight polymer could pass a 10 micron filter prior to a high pressure pumping system and still retain its structure. A good test of the Russian hypothesis would involve examination of high pressure water blasting flow characteristics after passage of a polymer solution at high flow rates and high pressures through filters of various appropriate dimensions.

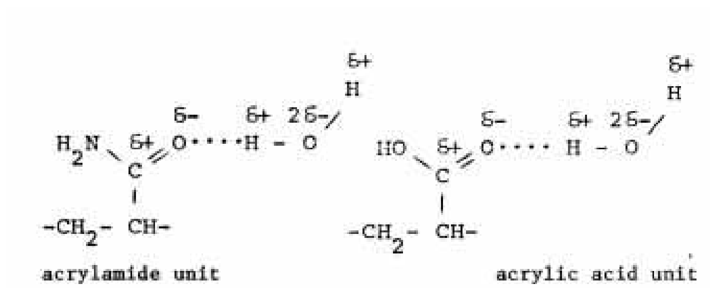
As noted previously the mechanism of drag reduction has not clearly been defined (5) and it is not unreasonable to conclude that jet focusing and particularly macromolecular polymer bombardment still require more studies for ellucidation of their total mechanisms. However, use of elementary organic chemistry, specifically consideration of hydrogen bonding and macromolecular structure, does offer some important insights into plausible mechanisms for the overall process of polymerblasting.

The molecules of SUPER-WATER possess acrylic acid and acrylamide units and in each instance a dipole occurs on the carbonyl group such that a partial positive charge is located on the carbon atom and a partial negative charge is on the oxygen atom.

In molecules of water, dipoles exist such that the hydrogen atoms carry partial positive charges which are balanced by a (double) partial negative charge on oxygen. Just as liquid water is intermoleculary hydrogen bonded (dotted line) by interaction of these dipoles,



so in aqueous solutions of acrylamide/aCrYlic acid copolymers, hydrogen bonding occurs between water and the polymer.



This hydrogen bonding of water molecules to the polymer units does not terminate after the bonding of one water molecule: hydration continues. In the case of polyacrylamide and polyacrylic acid the number of water molecules attached or bonded to each repeating unit are 13 and 14 respectively (20). As a consequence the molecules of SUPER-WATER are capable of bonding water to yield, if the polymer backbone is regarded as essentially linear, a longitudinal structure with a cylindrically oriented "water sheath". It is apparent, at a concentration of 0.3% of SUPER-WATER where each polymer unit can bond 13 or 14 molecules of water, that an overall structure will result of macromolecules and their associated aggregates of water. Such extended or macrostructures would be expected to promote or stabilize laminar flow (4) and decrease turbulence or the formation of vortices in boundary layers: this could be the basis for drag reduction.

The hydrogen bonding of water molecules to the polymer and, on a molecular level, an extended macrostructure, is apparently retained after emergence from the high pressure nozzle. This is manifested as a well collimated or focused jet (15).

Under polymerblasting conditions where the jets are traveling at supersonic speeds, it is tempting to regard the SUPER-WATERR macromolecules and their associated aggregates of water as being linearly aligned in the direction of flow. Such an assumption, however, is not altogether necessary because on a molecular level, condoned (2l), helical (22) or any ordered and extended macrostructure could bring about the observed focusing. The important feature is that structure is imparted to the water.

An additional point is worth noting in this discussion of polymerblasting jets. The water is bound to the polymer and thus the vapor pressure of the water is reduced. This leads to a marked reduction in water vapor pockets within the jet stream Cavitation (the appearance and collapse of vapor bubbles) in a fast flowing water stream is likely to result in random bombardment and random damage to the target. The use of SUPER-WATER therefore realizes a secondary benefit by diminishing formation of water-vapor pockets: cavitation is reduced. Using similar polymers experimental findings paralleling these theoretical predictions have been published (23, 24) although in both instances submerged jets were being considered. This attenuation of cavitation could be an additional factor responsible for the dramatic 15 to 30 fold increase in effective stand-off distance that is achieved in cleaning plugged slots in submerged oil well liners (15).

In the third step of polymerblasting the target material experiences macromolecular polymer bombardment (15). All molecules undergo varying degrees of intramolecular vibration. It is readily possible to envisage molecular stretching, bending or twisting and to appreciate that the larger the molecule the greater will be the time frames of these motions. During high pressure polymerblasting with jets impacting at supersonic speeds the SUPER-WATER macromolecules and their associated aggregates of water do not have sufficient time for complete stretching, bending or twisting. As a result the jets effectiveness on the target is enhanced by bombardment with molecules that are "rigid" (15). The Deborah number D is defined as the ratio between polymer aggregate relaxation time and the characteristic time scale during which the fluid stream undergoes deceleration. It is predicted that when D is greater than unity "solid-like" behavior is exhibited because the material has inadequate opportunity to adjust (relax) to its changing environment (25).

Superimposing this rigid bombardment effect upon a tightly focused noncavitating jet results in an inclusive action that water alone can not duplicate. The differences between polymerblasting and waterblasting are readily apparent in carefully conducted experiments (15) and in the actual applications described above.

OTHER CONSIDERATIONS

1. SUPER-WATER

The liquid product is available in 5-gallon plastic pails containing 35 lb net weight. The 5gallon pail is not completely filled in order to provide a headspace for ease of mixing just prior to use. SUPER-WATER is available in the form of a water-inoil emulsion with the polyacrylamide/polyacrylic acid copolymer in the internal water phase. Upon receipt, users should store the product upside down and between 40° and 90°F. Allowing the product to be stored outside this temperature range will cause the emulsion to break and severely reduce performance. For maximum efficiency the product should be stored upside down and vigorously shaken just before use. SUPERWATER has been used successfully at temperatures from -15°F (Massachusetts) to 100°F (Utah).

2. REDUCTION OF METAL EROSION

A concern that polymerblasting might increase erosion of metal parts was allayed when it was determined experimentally that, in fact, erosion was reduced. In the summary to a Materials Laboratory Richmond report (26) the following points (quoted verbatim) were made. "The influence of SUPERWATER additions on metal erosion was studied in an experiment conducted on carbon steel heat exchanger tubes by the Richmond Refinery Chemical Cleaning Specialist. SUPERWATER additive was found to increase the incubation period for erosion by a factor of nine over plain waterblasting. This incubation period is the length of time that a metal surface can be directly exposed to the water jet without significant removal of metal. In addition, the initial surface metal erosion rate using a SUPER-WATER solution was found to be half of that for plain waterblasting." "We expect the addition of 3000 ppm SUPER-WATER polymer to hydroblast water will minimize erosive removal of metal surfaces and reduce the chance of damage to items being cleaned."

3. INJECTION SYSTEMS

The use of an appropriate system for injecting SUPER-WATER into the waterblast stream is critical to ensure optimum performance. Berkeley Chemical Research Inc. recommends injection systems specifically designed to proportionally meter SUPER-WATER at the optimum concentration of 0.3%. The polymer upon injection undergoes emulsion breaking and then hydration as discussed above. Because this hydration is not instantaneous and takes 2 to 3 minutes to complete the polymer must always be injected ahead of the holding tank fitted to most high pressure waterblasting pumps.

4. SAFETY CONSIDERATIONS

The usual safety precautions employed during hydroblasting are appropriate for polymerblasting: see the OSHA Material Safety Data Sheet for SUPER-WATER . It should be noted the dilution of properties, apart from the flow characteristics, approach that of water. Neither polyacrylic acid nor polyacrylamide, or combinations, are listed in the so-called EPA Consent Decree (27) and neither are they included in the list of chemicals described as being carcinogenic in the December 1981 Second Annual Report on Carcinogens (28).

At the Chevron U.S.A. Richmond Refinery it has been found that SUPER-WATER , which is biodegradeable, does not foul the oxidation ponds (29).

SUMMARY

A description has been given of polymerblasting using SUPER-WATER concentrated industrial water blasting additive. The theory of enhanced efficiency and wide range of applications have been discussed. Also given were appropriate storage conditions, methods for mixing prior to use and the method of introducing the additive into high pressure waterblasting systems. ACKNOWLEDGEMENTS The author wishes to express sincere appreciation to Prof. Jack W. Hoyt (San Diego State University, California), Dr. Douglas A. McCombs (Western Industrial, Inter-mountain Division, Salt Lake City, Utah), Prof. David A Summers (University

of Missouri-Rolla), and Mr. Casper W. Zublin (Zublin and Company, Bakersfield, California) who were kind enough to review this paper and make valuable comments and suggestions.

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