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JET CONFERENCE

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For their help in organizing the conference, and the field trips, I would like to express my appreciation to:

R. J. EVANS, CO-CHAIRMAN
H. J. HANDEWITH
J. N. MURPHY
E. D. THIMONS
F. D. WANG

TEXT FOR THE OPENING COMMENTS

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THIRD U.S. WATER JET CONFERENCE,
PITTSBURGH, PA
MAY 21, 1985

On behalf of the Bureau of Mines, it is a pleasure to have the opportunity to participate in the Third U.S. Water Jet Conference, and may I commend the cosponsors for their interest and participation. Congratulations are in order for the Organizing Committee for assembling the excellent program.

All who have contributed to the phenomenal progress made in this field in recent years can be proud of their contributions. The bottom line is, of course, the manner in which and the extent to which the technology is used in the field. And, of course, the extent to which it is used is ultimately the cost benefit of these technologies to the user.

The Conference must and does consider the theoretical aspects of water jetting as well as engineering, field performance of different systems, and safety. The papers presented at the Conference will cover the use of water jets for cutting, drilling, and cleaning. The types of jets employed can be divided into three categories: jet-assisted mechanical cutting, modified jets (i.e., slurry jets, etc.), and pure (high-pressure) jets.

If you consider water-jet-assisted cutting as an example, and if you were a mining operator or a tunnelling contractor, you might find any of the following attributes of significant interest: 20-to 50-pct reduction in cutting forces; 300-to 500-pct increase in bit life; significant reduction in machine vibration, dust, and noise; and improvement in the size distribution of your product (less fines). All of these attributes are potential benefits that can be realized by water-jet-assisted cutting. In fact, we have seen some commercial realizations of this technology on roadheading machines, where the Bureau has been pleased to participate in this pioneering effort. To date, there are over 20 water-jet-assisted roadheaders in use or on order. These promising results should not be interpreted to signify that all work is done relative to water-jet-assisted cutting. There is still an essential requirement for a better theoretical understanding of the mechanisms that produce these outstanding benefits which would permit more rapid optimization and, perhaps, improvement of the performance characteristics that have already been observed. Additionally, there are many engineering details required to implement this technology, including improved rotary seals and phasing systems.

As we ask ourselves "What will the future he for water-jet-assisted cutting?" perhaps an analogy is helpful: A number of years ago if you wanted an automatic transmission for your automobile, it was necessary to special order this feature. Today, automatic transmissions are generally standard, and with some exceptions if one chooses not to have an automatic transmission, it is a special order. In 3 to 5 years, I believe water-jet-assisted cutting on mining equipment and tunnelling equipment will be a standard feature, and if you do not want it for your machinery, you will have to special order the machine without this feature.

With so many apparent benefits for water-jet-assisted cutting, one must ask "Why should anyone consider the application of the modified jets or the high-pressure jets?" While the technological hurdles are greater for these approaches, the potential rewards are commensurate with the challenges, and we must pursue these technologies as well. While, in my opinion, it may take 5 to 10 years to see extensive utilization of modified and high-pressure jets, it is imperative that we press on. Considering the analogy used above, while I am convinced that water-jet-assisted cutting will be, like the automatic transmission, on every machine, the other water jet cutting technologies will at least initially be used on special applications; to keep with the automobile analogy, perhaps this technology is like the four-wheel drive, while not used everywhere, where it is needed you cannot do without it.

For all of these areas safety must be of utmost concern; it is an essential part of the research, as well as the product engineering.

This Conference provides the forum to exchange theory, engineering, field experience, and safety considerations of these technologies.

May I extend to you my best wishes for a successful meeting and urge you to press on with the development and application of some of the most revolutionary mining and industrial technologies that we have seen in recent years.

Please Note.

This text is a scanned in version of the original. Because of some limitations in our programming the original pagination has been changed. Other than that we have tried to make the text a little more readable by increasing the spacing between paragraphs, but the text itself has been (subject to possible OCR misinterpretations) left as written.

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REMOVAL OF CEMENT LINING FROM OIL FIELD INJECTION LINES WITH WATERJETS

by

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ABSTRACT

This report covers a two phase program concerning removal of cement lining from secondary recovery water injection lines in oil production. The first phase involved cleaning 2 3/8 inch (6 cm) well tubing which was out of the ground. Phase two was a test of cleaning a section of buried 8 inch (20 cm) pipeline. The lining ranged from 3/16 inch (.5 cm) to 3/8 inch (1 cm) thick. The high pressure water system operated at 10,000 psi (690 Bar) and 15 gpm (60 L/min).

INTRODUCTION

In many oil producing areas a water induction system has been installed for secondary recovery. The injection grid recirculates water found in the oil bearing formations to stimulate oil production. Typically these injection systems operate at 3000 to 5000 psi (200 to 350 Bar). In some cases the water has sufficient H_2S levels to cause significant corrosion of the steel pipe. To extend the life of the injection system, a cement lining was installed in the pipe to protect it from corrosion.

Operators are now evaluating the options and costs associated with progressing to tertiary recovery using a CO_2 water flood injection system. One obstacle is the chemical non-compatibility of the CO_2 and the cement lining. The injection lines must be compatible to both the H_2S contaminated water and the CO_2. StoneAge's part in this program was to demonstrate the feasibility of cleaning the pipe in preparation for installation of a new plastic or fiberglass lining. The costs of in-situ cleaning and relining would be compared to the costs of right-of-way and installation of new lined-pipe.

PHASE I - ABOVE GROUND TEST

Beginning in April of 1984, StoneAge began cleaning the cement lining from 2 7/8 in (7.3 cm) Hydril tubing. During the summer months, over 400 joints of tubing wore successfully cleaned. Many different nozzle-bits were designed, fabricated and tested. The most successful cleaning configuration involved the combination of waterjets and mechanical cutting

Because of minor differences in the ID of different joints of tubing, it is difficult to clean completely to the wall with a mechanical drill. The mechanical bits would also leave layers of cement wherever the tubing changes diameter and also in the pin and coupling. At first waterjets were used to supplement the mechanical cleaning action by removing the thin layers which the rotary mechanical bit missed. However, as testing continued the relationship was reversed. The waterjets accomplished most of the cleaning and the mechanical bit scraped the wall of the tubing to complete the cleaning process.

The nozzles were both angled to one side, so by reactive force the bit was thrown against the opposite side of the tubing. This enabled a single bit to accommodate varying pipe diameters. As the drill rod was rotated the bit would scrape the wall and loosen any cement or corrosion which was left by the waterjets. A side thrust of 30 to 50 lbs (13 to 22 kg) insured that the wall would be completely cleaned. To get complete coverage of the tubing wall the advance rate must be balanced with the rotation rate

In addition to the cleaning of the cement, the forward facing waterjets created a draft of air through the lined tubing to assist with removal of cuttings. This eliminated the need for an additional compressed air chip removal system as is used on many mechanical pipe cleaners.

A custom designed pipe cleaning rig was developed for this job (Figure 1) Rotation of 0 to 200 rpm is provided by a 1 1/2 horsepower (1 kw) motor. The rotation carriage is driven along 40 foot (12 m) rails. Thrust of 200 pounds (90 kg) is provided by a chain/gearbox drive from a I1horsepower (.74 kw) air motor. A high pressure water swivel was coupled between the rotation motor and the drill rod with the nozzle/bit mounted on the end of the drill rod. The cleaning rate was found to be proportional to the horsepower of the cleaning jets. By delivering 65 hydraulic horsepower (48 kw) to the nozzle, a 32 foot (9 m) joint of tubing could be cleaned in 6 minutes. The energy was delivered through two .054 inch (14 mm) carbide orifices, requiring 15 gpm (60 L/min) at 9000 psi (620 Bar).

A positive displacement pump was used to supply the high pressure water. The pump has five plungers and is powered by a diesel engine through a clutch and belt drive. These pumps are commercially available for purchase or lease. Standard 1/2 inch (1.2 cm) waterblast hose and a foot operated dump valve connected the pump to the water swivel.

In addition to the lined tubing, several hundred joints of plugged tubing were also cleaned. The tubing was full of cement, presumably from cementing accidents during squeezing operations. Similar cleaning rates wore accomplished using a bit that was 95% of the pipe diameter. The bit was equipped with carbide spades on the face. The water-jets were forward facing, and assisted the carbide cutters. It is necessary to grind the chips small enough that the water flow will flush them from the tubing. Large chips can plug up behind the bit, creating hazardous conditions when the drill rod begins to "hydraulic" back out of the tubing under 10,000 psi (690 Bar) pressure.

PHASE I CONCLUSIONS

High velocity waterjets are very effective in removing cement from pipe. In addition, the optimum nozzle design removes corrosion from the inside of the pipe.

The most effective cleaning was done with water pressures between 7,000 and 10,000 psi (480 and 690 Bar).

On the rack, pipe can be cleaned for less than $1.00 per foot. The cost is affected by pipe size and chemical composition of the cement. If there is sufficient volume to justify the investment in more mechanized pipe handling equipment, this cost could be reduced.

PHASE II - UNDERGROUND TEST

The information gained in Phase I, was used to design prototype equipment for the Phase II test the test consisted of cleaning the cement lining out of a 500 foot (150 m) section of buried pipeline. Selected for the test was an abandoned section of 8 inch (20 cm) trunk line. The section was straight, with an even 3% slope with about 3 foot (1 m) of soil cover.

A single large rotating waterjet, was the preferred method developed for the field test. The cleaning tool consisted of an air motor, a high pressure swivel, and the rotating nozzle. These parts were encased in steel to protect them while the tool was being pulled through the pipe. A sizing ring followed the nozzle to insure complete cleaning as the nozzle proceeded along the pipe (EXHIBIT B). The cuttings were flushed out of the pipeline with water. A closed system was established to recirculate 400 gpm (1500 L/min) with a low head gasoline powered centrifugal pump. A 6 inch (I5 cm) diameter aluminum pipeline was laid above ground to return the flushing water to the upper end of the cemented pipe. Connected to the discharge end was a holding tank, with a wire basket used to separate out the cuttings. The cleaning was begun from the low end, to allow flushing of the cuttings to pass through cleaned pipes.

Preparation

We dug 6 ft by 20 ft (2 m x 6 m) pits to expose the pipe at each end of the test section. The pipe was flame cut to allow access. At the discharge end, a 9 ft (3 m) section was removed and the holding tank was put in line and attached to the pipe. This allowed the cuttings to fall into the tank and the flushing water to be recirculated. The upper end of the pipe was raised about 2 feet (. 6 m), to allow good access for introducing flushing water and hoses into the pipe.

The flushing water was used to wash a light line through the test section. This line, with friction assistance from the flushing water, was used to pull the 500 ft (150 m) of air and high pressure water supply hoses through the pipe. The cleaning tool was attached to the hoses in preparation for being pulled back up the pipe during the cleaning process. A cable was attached to the cleaning head from an air winch located at the discharge end. This cable was to ensure the tool would not get stuck, and could be backed up or retrieved at any time.

The flushing system required 1500 gallons (5700 L) of fresh water to fill the holding tank, the pipeline, and the return line. A vacuum truck of 80 barrel (13 m^3) capacity was used to control the water level in the holding tank. The truck started about half full leaving room to pick up the additional water produced by the nozzle during the cleaning operation.

The high pressure pump was located at the upper end of the section, and received water from a second water truck. The pump requires some inlet pressure, which can be easily provided by a vacuum truck. This truck should start full, or at least have enough water to complete the cleaning job.

Compressed air was used for rotation of the cleaning nozzle. The air vane motor required 30 cfm (.8 m^3 /min) at 100 psi (6.9 Bar). The hose reel was rotated by an air winch, utilizing up to 100 cfm (2.8 m^3 /min) at 100 psi (6.9 Bar) The return cable winch, located at the lower end, required the same. The air was supplied by two portable air compressors. The air drive for the hose reel was throttled to provide about 250 lb (7.4 MN) pull on the cleaning tool via the supply hoses. As the cleaning operation progressed, the 500 feet (150 m) of high pressure water hose and air hose were returned to storage on the hose reel.

Cleaning Process

The cleaning tool was attached to both the supply hoses and the return cable. The flushing circuit was started, and a constant flow of water through the pipe line was established. The cleaning head rotated with a speed of about 200 rpm. The operating pressure at the pump was 9,000 psi (620 Bar). Allowing for friction losses through the 1/2 inch (1.25 cm) hose, the water pressure at the nozzle was about 7,300 psi (500 Bar). The single .078 inch (20 mm) nozzle delivered 15 gpm (60 L/min), which resulted in 65 horsepower (50 kw) being applied to the cleaning process. Effective cleaning could not be accomplished below 5,000 psi (350 Bar) nozzle pressure.

The operating time to clean the 480 foot (146 m) section was 2.5 hours The average cleaning rate was about 3.2 feet (1 m) per minute. Assuming constant travel, the cleaning jet would complete 5 rotations every inch (2.5 cm). Inspection of the pipe end showed the spiral cleaning pattern on the wall (EXHIBIT C). The location of the cleaning head and the cleaning rate could be easily monitored by watching the return cable. Observation of the cable movement showed that the sizing ring would sometimes stop the cleaning head's progress until all the cement was removed.

PHASE II CONCLUSIONS

The pipeline cleaning test was highly successful. The field demonstration confirmed the feasibility of cleaning the lining out of the pipeline while it is in the ground. The actual cleaning out of the cement lining was quickly and easily performed. The logistics of the operation and the flushing of the cuttings from the long pipe section required the major effort.

The general set-up proved to be functional, however there are some improvements that should be instituted before beginning a full-scale operation.

1 The cleaning tool should be pulled with a cable, rather than the hoses. The control would be much more positive, by avoiding the elasticity of the hoses.

2 More care should be taken to improve the flow of flushing water around the cleaning tool. There was more drag than necessary, and some indication that the cleaning tool was plowing cement ahead of it some of the time.

3 A smaller rotation system will be needed for smaller pipe. The equipment and technology is available to accomplish this.

4 Field time for the project was about eight hours, with a four man crew The most significant improvements in the system can be made in reducing the set-up and moving time. With the addition of a jet lift pump at the discharge end of the pipeline, the holding tank could be kept on the surface of the ground. This would save 3 hours at each location and eliminate the need to remove a section of pipe.

5 Improved equipment for attaching the flushing system to the pipeline could save 1/2 hour per location.

In addition to these, some experience would improve the overall efficiency of the crew. We predict that it should be possible to complete two 500 foot (150 m) sections per day. With these improvements and a minimum of 1 to 2 mile (1.6 to 3.2 km) job size, we project a cost of $2.00 to $3.00 per foot to clean the cement lining out of the buried injection lines.

Figure 1. Cleaning cement filled pipe with high velocity waterjets in Phase 1.

Figure 2. Rotary cleaning head with sizing ring used in Phase II.

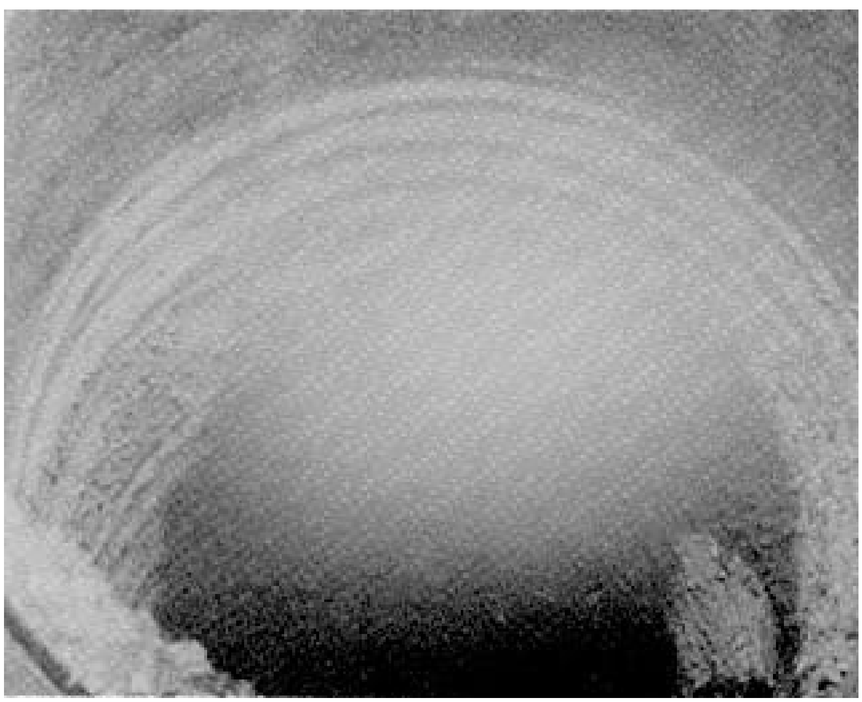

Figure 3. Spiral cleaning pattern made by water jet.

OPERATIONAL AND MAINTENANCE MISCONCEPTIONS
OF HIGH PRESSURE POWER PUMPS

by

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NLB Corp.

ABSTRACT

This paper presents common misconceptions that exist regarding the operation and maintenance of high pressure power pumps and describes how the misconceptions can result in serious safety and pump reliability problems. As a prerequisite and preceding the misconceptions, a practical description of what a power pump is, along with its operating principle and some design factors will be presented.

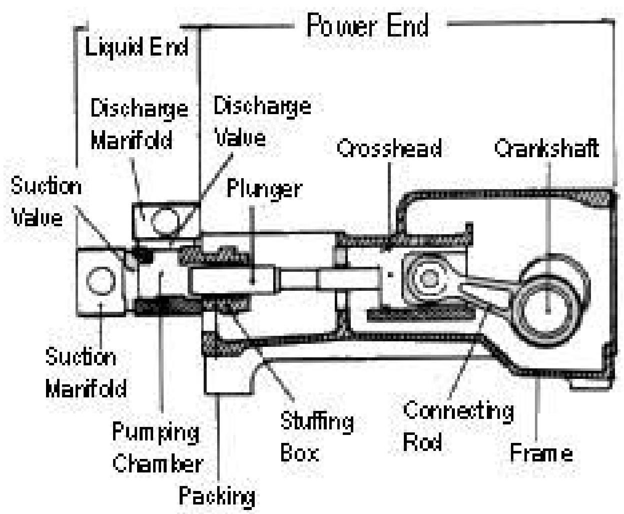


Figure 1

INTRODUCTION

In the water jetting industry, the majority of high pressure pumps used are single acting triplex and quintuplex power pumps. The safe operation and design performance integrity of a water blasting system is largely dependent on the proper operation and maintenance of those pumps. It is the intent of this paper to highlight some common misconceptions that many people have regarding the operation and maintenance of power pumps, and to describe how those misconceptions can cause serious safety and pump reliability problems. To accomplish this, it is first necessary that we have a basic understanding of what a power pump is, its operating principle, and some preliminary design factors associated with it.

POWER PUMP DESCRIPTION

A power pump is an energy transmitting, positive displacement reciprocating machine, which consists of two major subassemblies; the power-end and the liquid-end, as illustrated in Figure 1. In the simplest form, the power end consists of the frame, crankshaft, connecting rod and crosshead. The liquid-end incorporates the plunger, stuffing box, packing, pumping chamber, suction and discharge check valves, and suction and discharge manifolds.

OPERATING PRINCIPLE

The operating principle of a power pump is not as complex as it may first appear. A prime mover such as an electric motor, engine or turbine is connected to the crankshaft of the pump. When the prime mover is energized, the pump crankshaft will rotate. One end of the connecting rod is attached to the throw of the crank and the other end is attached to the backside of the crosshead, as shown in Figure 2. When the crankshaft turns through one half revolution, by means of the connecting rod, it drags the crosshead in a linear motion from position one to position two, as illustrated in Figure 3. On the second 180 degrees of its rotating cycle, the crank, again by means of the connecting rod, pushes the crosshead back to position one, as depicted in Figure 4. The distance between positions one and two is the stroke length of the pump. The major functions of the connecting rod are: (1) It serves as an energy transmitting device which transfers the energy imparted to the crankshaft by the prime mover, to the liquid being pumped; and (2) serves as a mechanical linkage between the crankshaft and crosshead, converting the rotating motion of the crankshaft into linear reciprocating motion of the crosshead.

In describing the liquid-end, the back end of the plunger is attached to the face or front side of the crosshead, as depicted in Figure 2. The plunger hangs out from the crosshead with a cantilever effect. The free end of the plunger protrudes through the stuffing box into the pumping chamber. Packing is installed in the stuffing box which surrounds the plunger with zero clearance, (ref. 1), forming a seal between the pumping chamber and the atmospheric side of the stuffing box. Since the plunger is attached to the crosshead, the motion of the plunger will also be in a linear reciprocating fashion.

When the plunger is on the suction stroke and moving out of the pumping chamber, a vacuum is created in the pumping chamber due to the absence of the plunger. Because of that vacuum, the differential pressure between the pumping chamber and suction manifold becomes great enough for the liquid in the suction manifold to force the spring loaded suction check valve to open against its spring force. When this occurs, the liquid in the suction manifold then travels through the suction valve into the pumping chamber, as illustrated in Figure 3. When the plunger completes its suction stroke and stops to reverse its direction, the spring loaded check valve closes, trapping a set volume of liquid in the pumping chamber.

As the plunger starts on the power stroke and moves deeper into the pumping chamber, it quickly pressurizes the trapped liquid to the point where the pressure is high enough to open the spring loaded discharge check valve against its spring force and against discharge manifold pressure. When that occurs, the plunger then displaces the liquid from the pumping chamber which travels through the discharge valve into the discharge manifold, where it is collected from all other cylinders and exits the pump, as shown in Figure 4. In a triplex pump, this hydraulic and mechanical action occurs three

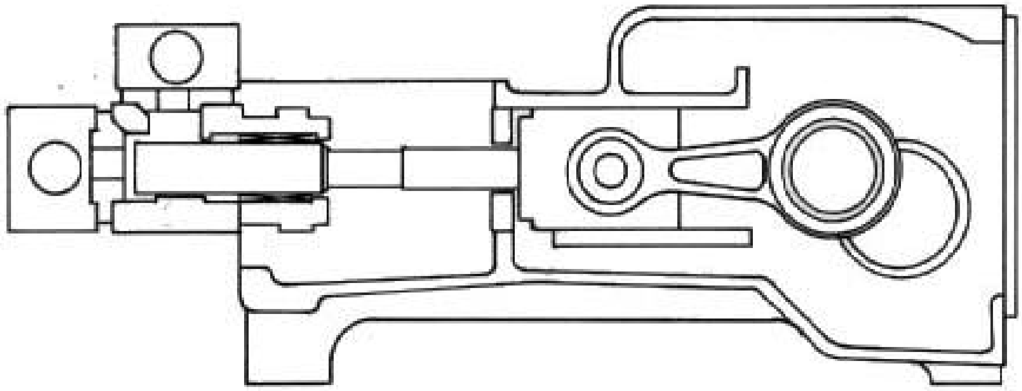


Figure 2

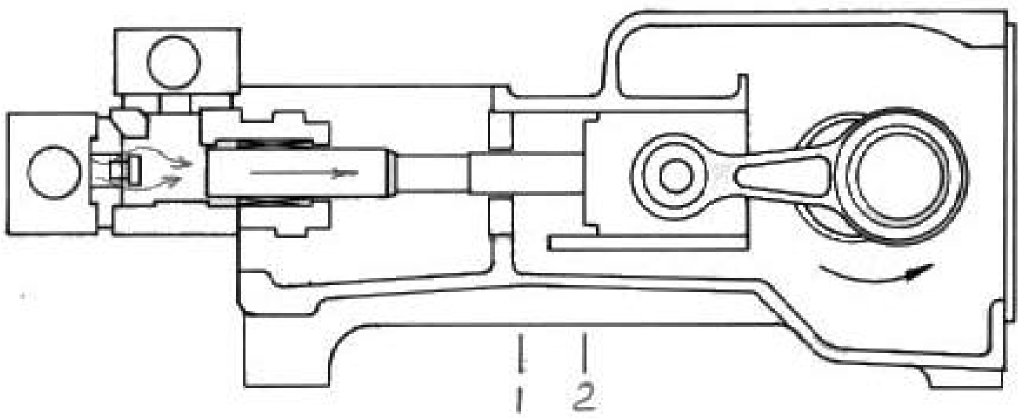


Figure 3

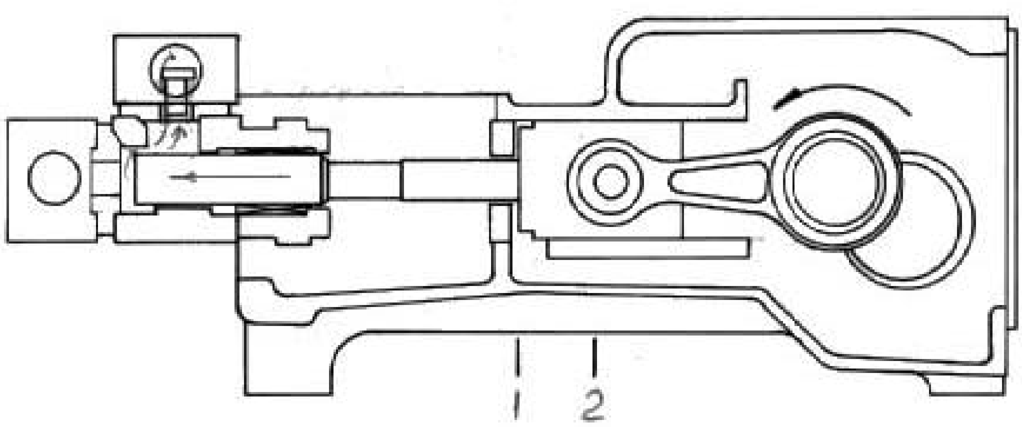


Figure 4

times for each revolution of the crankshaft. In a quintuplex pump, it occurs five times per revolution of the crank.

DESIGN FEATURES AND FACTORS

The crankshaft in a triplex pump is designed with three throws positioned 120 degrees apart, as illustrated in Figure 5A. The quintuplex crank shown in Figure 5B, is designed with five throws which are positioned 72 degrees apart. For each throw of the crank, the pump will be equipped with one plunger. Therefore, a triplex pump will have three plungers or pumping elements, (Ref.1), while the quintuplex pump will incorporate five pumping elements. Since the crankshafts between the triplex and quintuplex pumps are designed differently, and the quintuplex unit has two more pumping elements than the triplex pump, it is reasonable to expect that the flow profiles they exhibit will be different as well. In fact, when viewing the flow characteristic curves shown in Figure 6, it is easy to see that the quintuplex pump has a smoother flow characteristic than the triplex unit. There are several reasons for that. In the triplex pump, the three pumping elements are attached to the crankshaft via the crosshead and connecting rod and are offset from each other by 120 degrees. This results in a 60 degree overlapping of the plungers on each half cycle of crank rotation, (Ref.1). Because of that, there are only two plungers either taking in or discharging liquid at the same time, (Ref-l), which creates a flow variation. The triplex pump has a total flow variation from the average flow of 25.06 percent, with the peak flow being 6.64 percent above the average flow and the minimum flow 18.42 percent below the average flow, (Ref-1, Ref-2). What this means, if a triplex pump was pumping an average capacity of 20 GPM, the suction and discharge lines and the manifolds of the pump will experience a high flow of 21.3 GPM and a low flow of 16.3 GPM, (Ref.1).

In the quintuplex pump, the pumping elements are attached to the crankshaft and offset by 72 degrees from each other. This results in two to three plungers either taking in or discharging liquid simultaneously. Because of that, the quintuplex pump has a total flow variation of only 7.51 percent of the average flow, with the peak flow being 1.88 percent above the average flow and the minimum flow 5.63 percent below the average flow, (Ref.1, Ref.2). Using the same example as the triplex pump, that is, 20 GPM, the suction and discharge piping as well as the manifolds of the quintuplex pump will experience a high flow of 20.4 GPM and a low flow of 18.9 GPM. In general, a power pump operating at a constant speed is considered to be a constant capacity machine. However, that is only true for a period of time. For any instant in time, the capacity is constantly changing between the parameters previously stated. Since the capacity of a power pump is measured over a period of time, normally sixty seconds, it can be difficult to relate to the capacity as constantly changing throughout that period of time. However, the pressure pulsations that are inherent to these pumps is indicative of the constantly changing flow rate. The reason for that is the liquid in the piping and manifolds of the pump is constantly accelerating and decelerating. In further examining the flow characteristic curves shown in Figure 6, the humps in the curves indicates a peak pulse. Note that the triplex pump will have six peaks pulses in one revolution of the crankshaft, while the quintuplex pump will pulse ten times in one revolution of the crank. If both these pumps are operating with a crankshaft speed of 400 RPM, the triplex pump would pulse 40 times per second, while the quintuplex pump would pulse 66.6 times per second.

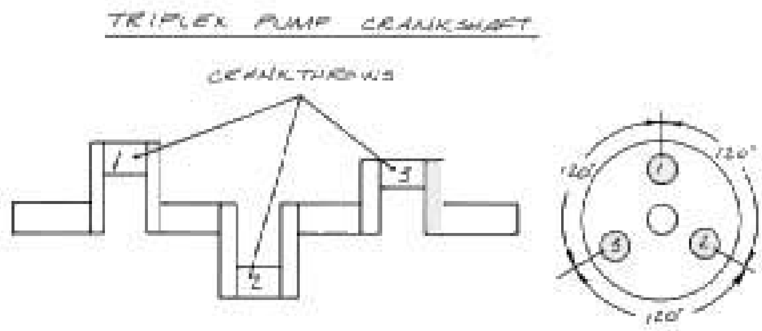


Figure 5A

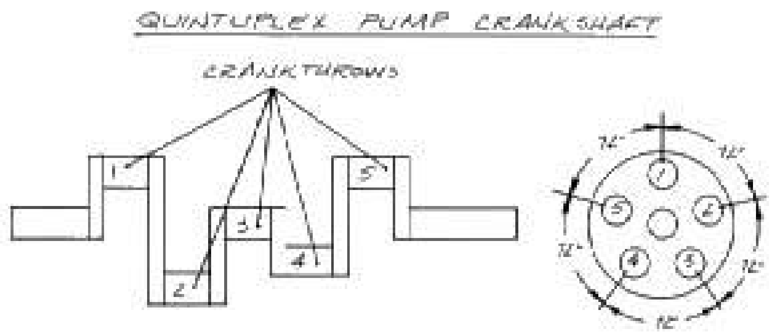


Figure 5B

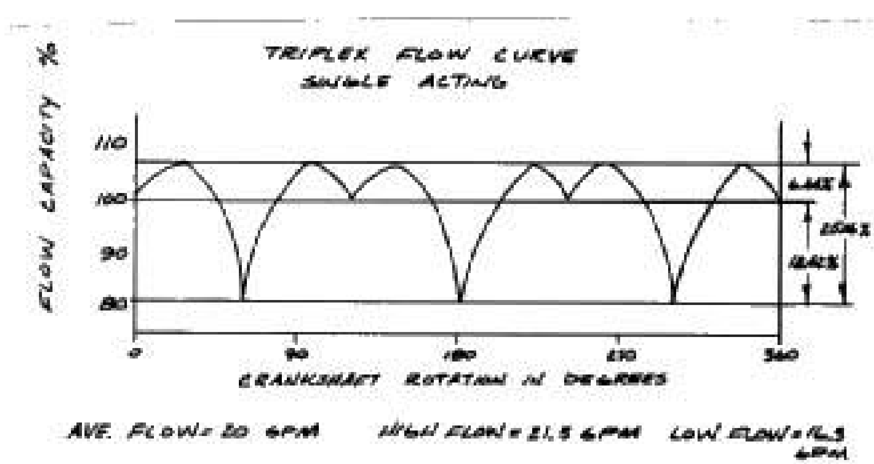


Figure 6A

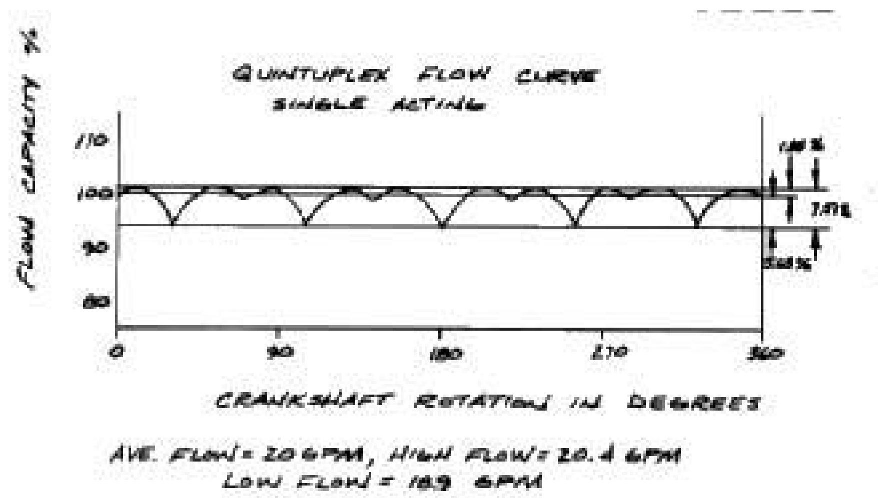


Figure 6 B

Since the flow in the piping and manifolds of the pump is constantly accelerating and decelerating, special attention must be given to the piping associated with power pumps. This is particularly true on the suction side of the pump where the liquid in the suction line must accelerate at the proper instant the pump demands. If it does not, cavitation will occur which can have damaging effects on the piping, valves, and the pump itself. The energy required to accelerate the liquid at the moment the pump requires it, is referred to as acceleration head. Figure 6C shows the acceleration head formula along with example calculations. Note that as the pump speed increases, both V and R increase, and the acceleration head will vary as the square of the pump speed. The acceleration head will also vary directly with the length of the suction line, L. (Ref.3). Note in the example calculations, for a speed increase of 100 RPM, the acceleration head increased by 6.31 feet, which is equal to 2.73 PSI of energy. Acceleration head should not be confused with NPSH. However, it is a component in the NPSHA calculation.

<u>Acceleration Head [h(a)] Calculation</u>

$$h(a) = L.V.R.C/K.g$$

where

h(a) = acceleration head (ft)
L = length of suction line (ft)
V = velocity in suction line (fps)
R = pump speed (rpm)
C = constant
K = 1.4 (for water)
g = gravitational constant (32.2 ft/sec^2)

For a single acting triplex pump C = 0.066
For a single acting quintuplex pump C = 0.040

Note: the above information is from the Hydraulic Institute Standards 12th Edition, N.Y., N.Y.

Example calculations

Triplex pump @ 400 rpm, 25 gpm
Triplex pump @ 500 rpm 31.6 gpm
Pumping though a 3 ft long line 1/25" internal diameter.

At 400 rpm   h(a) = (3) (6.52) (400) (0.066)/(32.2) (1.4)   = 11.49 ft.
At 500 rpm   h(a) = (3) (8.11) (500) (0.066)/(32.2) (1.4)   = 17.8 ft.

For 100 rpm pump speed increase h(a) increases by 6.31 ft which is equal to 2.73 psi.

Figure 6C

NPSH is an abbreviation for the "NET POSITIVE SUCTION HEAD", and it denotes the amount of energy in the liquid being pumped over and above the liquid's vapor pressure when measured at the suction connection of the pump. When choosing a pump for your jetting application, there are two forms of NPSH that must be considered. There is NPSHR, which stands for "NET POSITIVE SUCTION HEAD REQUIRED", and it is a characteristic of the pump design. The second is NPSHA, which means "NET POSITIVE SUCTION HEAD AVAILABLE", and that is a characteristic of the system the pump will operate in. For the pump to operate properly, NPSHA must always be greater than NPSHR. If it is not, again, cavitation will occur.

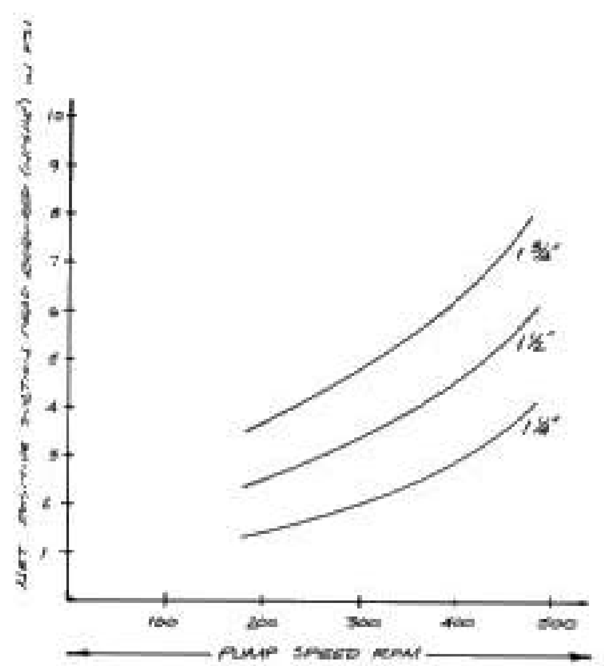


Figure 7 – NPSHR Curves

The manufacturer of the pump determines the NPSHR by calculations and by actually testing the prototype pump. The test is carried out by installing the pump into a system in which the NPSHA is known. The NPSHA is then slowly reduced until the flow capacity of the pump drops off by three percent. The NPSHA of the system at the three percent capacity reduction is then assigned to the pump as the NPSHR at the plunger diameter and speed at which the pump was tested. The same test is performed with different plunger diameters and at different speeds. The information is collected and put into graph form with NPSHR plotted against operating speed in RPM, similar to the curve illustrated in Figure 7. Note that as the pump speed increases, the NPSHR also increases.

The NPSHA is determined by the designer of the system that the pump will operate in, and takes into account the system configuration. Figure 8 illustrates a typical system. If the water supply tank is vented, then the value of "P" in the NPSHA equation is equal to barometric pressure. Although figure-8 illustrates a tank system, the tank can be substituted with a pressure line and the NPSHA equation shown can be used. The acceleration head stated earlier, is represented by the notation h(a). Note in the NPSHA formula, as the pump speed increases, the value of h(f) increases due to the velocity increase of the liquid in the suction line. From the information shown in the three previous illustrations, the importance of pump speed is obvious for the proper operation of the total system. Remember, if the pump is to operate cavitation free, NPSHA must be greater than NPSHR.

$$NPSHA = \frac{\left[\left(P + L_H \right) - \left(V_p + h_f \right) - h(a) \right]}{2.31}$$

where

P=pressure on surface of the liquid
$L_{(H)}$= Minimum static suction head in feet.
V_p = Vapor pressure of the liquid at the pumping temperature
h_f = friction loss in feet in suction line at the pumped capacity
SG = specific gravity of liquid at pumping temperature
h(a) = acceleration head
L = suction line length in feet

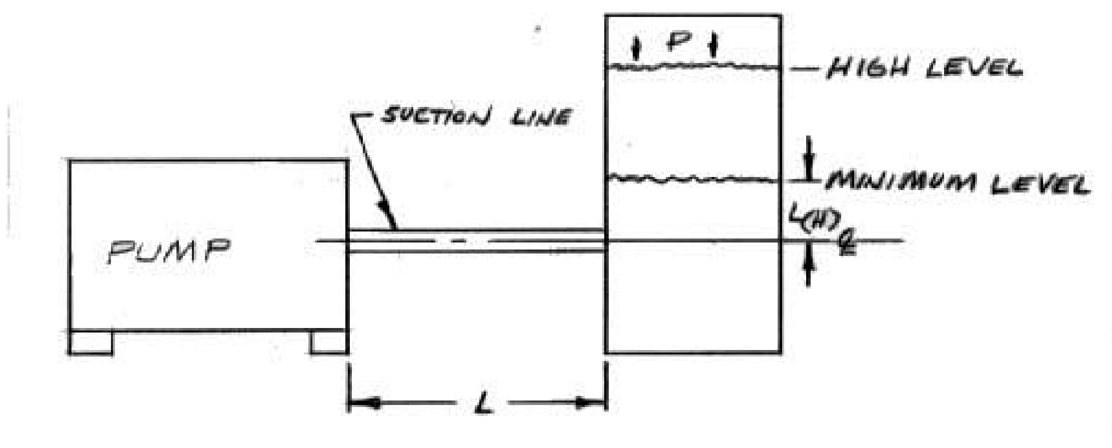


Figure 8 NPSHA

Let's take a look at another important design factor which is referred to as plunger or frame loading. When a power pump is designed, it is engineered around certain conditions that dictate the plunger loading the design will be based upon. To determine the plunger or frame loading of a pump, calculate the face area of the plunger and multiply that value by the maximum permissible discharge pressure for that plunger diameter. The product of that multiplication is the frame loading the components in the power frame were designed around. Since the various components may have different safety factors assigned to them when they were designed, it is important not to operate the pump above the maximum frame load rating. if you do, it is possible that a crack will initiate in one or more of the power frame components which will create an unsafe operating condition.

Now that we have a basic understanding of what a power pump is, how it operates, and some design factors associated with it, we can objectively view the most common operational and maintenance misconceptions that many people have regarding power pumps, and relate back to the preceding paragraphs as to how those misconceptions can cause serious safety and pump reliability problems.

COMMON MISCONCEPTIONS

The following is a list of the four most common power pump operational and maintenance misconceptions:

1. Operating the pump above the manufacturers maximum recommended speed is not really harmful to the pump.
2. The faster the pump operates, the smoother the flow becomes.
3. Operating the pump for brief periods of time above the maximum pressure rating of the pump is okay.
4. Using less expensive parts in the maintenance of the pump saves money.

We will go through the above list one at a time and based on the previous statements, determine why they are misconceptions.

The first misconception is that operating the pump above the manufactures maximum recommended speeds is not really harmful to the pump. There are several reasons why that is a misconception.

The first and most obvious is NPSH. We have learned that as the pump speed increases the NPSHR increases, and as the flow from the system increases the NPSHA decreases. We also know that for the pump to operate properly the NPSHA must be greater than the NPSHR. Using this information, let's examine a pump system to determine the combined effect of NPSH with increased pump speed.

Figure 9 illustrates a pump NPSHR curve along with the NPSHA calculations of a pumping system. The system consists of a vented suction tank with a minimum static suction head of three feet, similar to Figure 8. Since the tank is vented, the "P" valve in the NPSHA equation will be equal to barometric pressure. Let's assume the barometric pressure is at sea level which is equal to 33.96 ft. of head. The suction line is a three foot straight run of 1.25 inch nominal steel pipe, schedule - 40, with a 1.38 inch l.D. The pump is a triplex unit with 1.25 inch diameter plungers. Since the pump is a triplex, the "C" valve in the acceleration head equation is equal to 0.066. At 400 RPM the pump will produce 25 GPM and at 500 RPM it will pump 31 GPM.

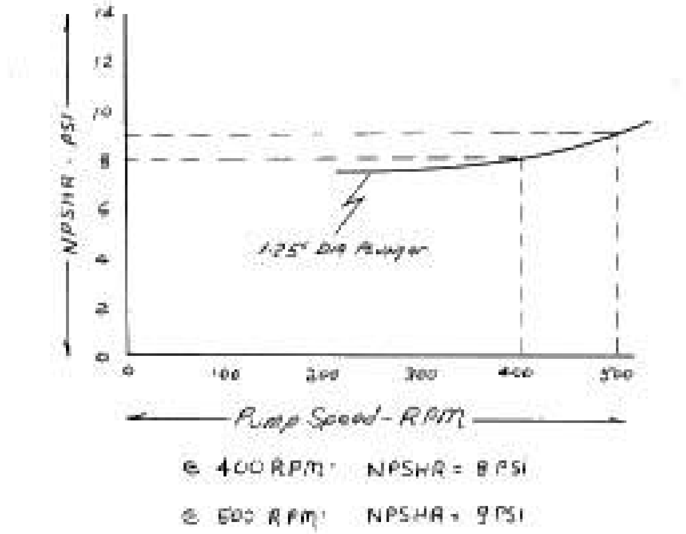
From the pump NPSHR curve at 400 RPM, the NPSHR is 8 PSI, and at 500 RPM speed, the NPSHR is 9 PSI. This is an increase of only 12.5 percent.

The NPSHA calculation at 25 GPM supply, reveals the NPSHA to be 11.5 PSI. While the calculation at 31 GPM supply shows the NPSHA to be 7.92 PSI. This is a decrease in NPSHA of 31 percent. When comparing the NPSHA to the NPSHR at the 400 RPM speed, it shows the NPSHA to be 44 percent above the NPSHR. This means the pump will operate cavitation free. However, when comparing the NPSHA to the NPSHR at the 500 RPM speed and increased system flow requirement, it shows the NPSHA to be 12 percent below the NPSHR, which means cavitation will occur with all the side effects of shock loading which will cause damage to the pump components. In fact, even if the pump NPSHR did not increase, there is enough decrease in the system NPSHA to cause cavitation.

The argument can be made here that if the vented tank was a pressurized tank, the NPSH problem illustrated in Figure 9 could be overcome. This is correct! However, NPSH is not the only limiting factor on pump speed. Let's take a look at what happens to the plunger and valve frequency.

Earlier in this paper I described how a power pump operates. In that section, it was illustrated that when the plunger is on the suction stroke, the suction valve opens, permitting the water to enter the pumping chamber. When the plunger stops to reverse its direction, the suction valve closes. Depending on the valve and valve spring design, the chances are good that if you violate the maximum permissible operating speed of the pump, the plunger and valve frequency will get out of phase with each other, resulting in a late closure of the suction valve. When this occurs, the suction valve is partially open as the plunger is rapidly increasing in acceleration on the power stroke, pushing liquid back through the suction valve. When the suction valve finally seats, the plunger will be close to its maximum velocity, and the liquid being pushed by the plunger then slams the discharge valve fully open, sending shock waves through the pump and discharge manifold. Those shock waves can be ten times or greater in magnitude than the normal operating pressure developed by the pump, which will create a very serious and dangerous operating condition. If you think the solution to this problem is to simply put in a heavier valve spring in the suction valve, think again! Once you do that, the pump NPSHR will go up, because a major portion of the NPSHR is directly related to the opening of the suction valve. It becomes a catch-22 situation. That's why pump manufacturers have maximum speed limitations on their equipment.

Another factor effected by the pump speed is heat generation in the power-end and stuffing box areas of the pump. The power-end oil reservoir is normally designed to dissipate the heat generated at the design speed and frame loading of the unit. Operating above the design speed of the pump will add additional heat into the power-end, causing overheating to occur which will result in premature failure of those components. In the stuffing box area, there is friction between the plunger and packing, and the higher velocity of the plunger will cause additional heat generation which causes premature failure of those components also.



$$NPSHA_{(400)} = \frac{[(33.96) + (3) - (0.59 + 0.24) - 9.434](0.998)}{2.31} = 11.5$$

$$NPSHA_{(500)} = \frac{[(33.96) + (3) - (0.59 + 0.43) - 14.60](0.998)}{2.31} = 7.92$$

Figure 9

$$PL = (A_p)(P)$$

$PL =$ Plunger loading; (Lbs.)

Where: $A_p =$ Plunger Face Area; (IN^2)

$P =$ Operating Press. (LBS/IN^2)

With a 1.25" DIA. Plunger, P=10,000 psi

$$A = (1.25)^2(.785) = 1.227\ IN^2$$

$$PL = (1.227\ IN^2)(10{,}000\ LBS/IN^2) = 12{,}270\ LBS$$

With a 1.50" DIA. Plunger, P= 6900 psi

$$A = (1.50)^2(.785) = 1.77\ IN^2$$

$$PL = (1.77\ IN^2)(6900\ LBS/IN^2) = 12{,}213\ LBS$$

With a 1.50" DIA. Plunger, P=10,000 psi

$$PL = (1.77\ IN^2)(10{,}000\ LBS/IN^2) = 17{,}700\ LBS$$

17,700 LBS is 44% above the 12,270 LB rating of the power-end.

Figure 10

The three explanations just presented should cast no doubt that the statement "operating the pump above the manufacturers maximum recommended speed is not really harmful to the pump", is a misconception.

Let's examine the second misconception which is: "the faster the pump operates, the smoother the flow becomes". It is surprising how many people think that the faster the speed is, the closer the pulsation becomes and that a smoothing effect occurs. The fact is the only thing that occurs with regard to pulsations when the pump operates faster is that more pulsations occur in a shorter period of time. The reason is, the magnitude of the pulse is a function of the crankshaft design and the degree the plungers are offset from each other, rather than speed. If you examine the flow characteristic curves illustrated in figure-6, it becomes clear that the idea of running the pump faster smoothens the pulsations is a misconception.

Of all the misconceptions, the third, "operating the pump for brief periods of time above the maximum pressure rating of the pump is okay", is probably the most widely accepted notion. Perhaps the reason for this is that many people associate the capabilities of the pump with the liquid-end pressure rating only. And where they normally get into trouble is when they are operating with a plunger diameter greater than they normally use. For instance, say a particular pump is suitable to operate at 10,000 PSI using a 1.25 inch diameter plunger. And, the pump is equipped with 1.5 inch diameter plungers to operate at 6,900 PSI pressure. The equipment is on the job site and not cleaning fast enough at 6,900 PSI. The first thought seems to be, the liquid-end is suitable for 10,000 PSI operation, let's crank it up to 10,000 PSI and clean faster. The assumption is correct, the liquid-end is suitable for 10,000 PSI operation either with a 1.25 inch or 1.5 inch diameter plungers. But what about the other pump components? Let's take a look at what's happening in the power frame when that occurs. To do this, we need to determine the frame or plunger loading the power-end components where designed around. Figure-10 illustrates those calculations. Note, that with a 1.25 inch diameter plunger, the face area of the plunger is 1.227 sq. inches, and when used in the plunger loading formula at the appropriate pressure of 10,000 PSI, the frame loading is 12,270 lbs.

The 1.5 inch diameter plunger has a face area of 1.77 sq. inches, and when used in the plunger loading formula at the appropriate pressure of 6,900 PSI, the frame loading is 12,213 Ibs, which is within the design rating of the pump. However, when the 1.5 inch diameter plunger is operated at 10,000 PSI pressure, the frame loading is 17,700 Ibs., or 44 percent above the design rating of the power-end. This results in over-stressing of the power-end components which are subjected to cyclic loads, which are extremely dangerous. Over-stressing components subject to cyclic loads will cause fatigue cracking to occur. The component may not fail instantaneously but as the cyclic loads accumulate, and this can be hours, days or weeks, the cracks will propagate until the point that a catastrophic failure occurs. The failure may even occur when you are operating well below the maximum frame loading of the pump, but remember, the initial damage would have occurred earlier in time when the parts were over-stressed.

At what frame loading the cracks will occur in a particular pump depends on how conservative the engineer was when the pump was designed. Some pumps will have cracks start with only a slight increase above the maximum frame load rating, while others may take a considerable overload before damage occurs. The fact is, overloading parts that are subjected to cyclic loads will shorten the life of power-end components. That's why the notion of "operating the pump for brief periods of time above the maximum pressure rating of the pump is okay", is really a misconception.

The last misconception, "using less expensive parts in the maintenance of the pump saves money". Before this misconception is discussed in detail, I want to point out that it is not my intent to imply that every organization that manufactures pump components, other than the original equipment manufacturer, does not know what they are doing. On the contrary. There are people that produce fine quality components. However, there are many that manufacture and market pump components that really don't know the design aspects of the parts they manufacture. Those people can dimensionally copy the OEM's parts and even make the part from the same material. The part may look like an original, may feel like an original and often are dimensionally the same as the original OEM part, but they can be as different as night and day. I will illustrate two such examples.

The valve parts illustrated in Figure11 look the same, except the valve on the left is broken. The broken valve is not an original OEM part and failed after 430 hours of operation, or about 9 x 10 to the 6th cycles. The valve on the right is an OEM original and has over 2,200 hours of operation or 46.2 x 10 to the 6th cycles, and can still be used. Both parts are made of the same material which is a 17-4 ph grade steel.

Why then did the valve on the left fail after only 430 hours of operation? An analysis of the part indicated that the material was indeed a 17-4 ph grade steel. However, it also revealed that the part was not heat treated. There are several commercial heat treatments available for 17-4 ph material, not to mention the confidential heat treatments that make the part suitable for the intended service. And yet, whoever made that part obviously had no concept of the complexity of the design and that heat treatment was necessary.

The unfortunate part of this was that the purchaser of the part thought he was saving $16.00 per valve or a total of $96.00 for six valves. In reality, it cost him more than $4,000, not to mention lost revenue due to downtime, because when the valve failed, it caused shock loading to occur which damaged other pump components.

The second example is illustrated in Figure 12, which shows two plungers. The plunger on the left is marred and scratched and is not an OEM original. The part on the right is an original and still in fairly good shape. The plunger on the left had been in operation only three months while the plunger on the right has more than 16 months of service. The analysis report of the plunger on the left indicated several problems which include: (1) the plunger was out of tolerance, which explains why the user could not get any reasonable packing life. Also, (2) the plunger did not meet the OEM's specification of a hard coating (which is proprietary) and was too soft for the service.

Again, the purchaser thought he was saving over $375.00 for three plungers. And again, it cost him over $1,800 because the plungers only gave a troublesome three months service and could no longer be utilized.

There are many more examples that can be illustrated. However, the two shown in figures 11 and 12 should be sufficient to get the point across that the general statement, "using less expensive parts in the maintenance of the pump saves money", is a misconception.

As a word of caution, most pump manufacturers will void the pump warranty if they discover that unproven parts are used in the pump. The reason why this is done is not because they did not get the parts order they would like, but instead, they don't know the effect the unproven part will have on the other pump components.

When it comes to the maintenance of power pumps, especially high pressure pumps, using unproven parts is like playing RUSSIAN ROULETTE. The best suggestion I can give on this particular misconception, is to quote an old Latin saying, "CAVEAT EMPTOR", which means, let the buyer beware.

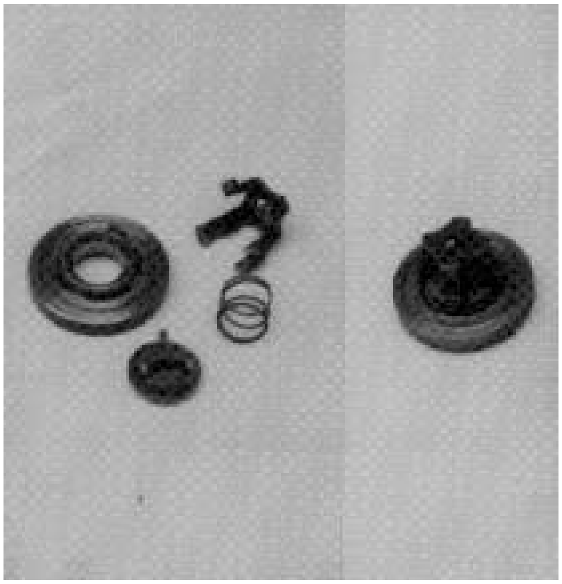


Figure ll.

Figure 12

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AN EXPERIMENTAL COMPARISON OF COMMERCIALLY AVAILABLE STEADY STRAIGHT-PATTERN WATER JETTING NOZZLES

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ABSTRACT

There are currently fifty (50) or more commercially available steady straight pattern water jetting nozzles available, nominally rated at 10,000 psi at 10 gpm (690 bars @ 38 liters/minute). Various vendors make claims as to the relative benefits of their nozzles such as "hard hitting", "long lasting", "inexpensive", "tough and rugged", etc., etc. However, to this author's knowledge, no one had ever compared the relative advantages of one nozzle versus another. For this reason, a test program was set up to compare, under laboratory conditions, the relative benefits of eight of the more commonly used straight-pattern nozzles.

This paper describes the test procedure used to evaluate the pressure profile of each of the eight nozzles at standoff distances ranging from one foot to eight feet. It also compares life expectancy of various nozzles (40-600 hours), actual flow rates of various nozzles 9.2 through 11.8 gpm (34.87 liters/minute through 44.66 liters/minute) and examines other important parameters that need to be taken into account such as cost, size and weight, when selecting nozzles.

INTRODUCTION

Many researchers in the past, e.g., Leach & Walker (Ref. 2), Barker & Selberg (Ref. 3) and Lohn and Brent (Ref. 4), have attempted to design the "best possible nozzle configuration, examining the effects of surface finish and internal design etc. on the performance of high speed water jets. However, to date, the results of these types of tests have not manifested themselves in commonly used, commercially available, nozzles. The reasons for these phenomena, in this author's opinion, are twofold. Firstly, a general lack of knowledge of the existence of this type of research work throughout the industry, and secondly, the manufacturing costs incurred in producing such "ideal nozzles". For these reasons, there exists today, a wide range of commercially available steady straight-pattern water jetting nozzles.

Having been intimately involved with users of these nozzles over the past four years, this author has found that everyone believes the nozzles he uses to be "the best", without substantiation for his opinion. For this reason, it was decided to compare eight of the more commonly used nozzles on the basis of cost, size, weight, performance and life expectancy. Hereafter, the nozzles examined will be referred to as nozzle (1) through eight (8), as delineated in Figure 1.

NOZZLES

a) Cost

The cost of each of the nozzles examined was the list price at the time of writing, as supplied by the respective manufacturer (Table 1).

b) Size

The size of each of the nozzles can be seen by the sectional drawings shown in Figure 2.

c) Weight

The weight of each of the individual nozzles was measured on a Pitney Bowes letter scale (Model S-104, Serial #41250) shown in Table 2.

d) Flow rate

Nozzle flow rates were measured by discharging the nozzles into a calibrated 50 gallon tank for a period of two minutes and then calculating the flow rate per minute by dividing by two.

e) Life Expectancy

The life expectancy of various nozzle designs was determined from actual field trials over the previous three years and are shown in Table 3.

TEST APPARATUS

a) Background

The general shape of pressure profiles obtained with steady straight-pattern water jets are well known, and documented by Dr. D. Summers (Ref. 1). The effect of the internal geometry and surface finish of nozzles, is limited to effective path width and standoff distance. For this reason, the test apparatus shown in Figures 3 and 4 was designed and built, rather than the more elaborate test apparatus used earlier by Summers.

b) Design

The apparatus consisted of a base with a nozzle retainer and eight target holders spaced 1 ft. (0.304 meters) apart, into which 1/4" (6 my thick, 3-ply plywood targets were placed. The threshold pressure of the plywood was 2700 +/- 200 psi (186 + 14 bars).

TEST PROCEDURE

A Weatherford FE 145 pump was utilized to produce 10,000 psi @ 20 gpm (690 bar @ 76 liters/minute). Also, a Weatherford MSSV 13800 unloading valve was used to allow water flow not consumed by the nozzle to return to the pump suction manifold.

Each nozzle was set up on the apparatus with eight (8) new plywood targets (Figures 3 and 4). The targets were subjected to 10,000 psi (690 bar) for a 10 second duration. The width of hole developed in each target was measured using a steel rule. In turn, all eight nozzles were tested in this manner; thereby providing a direct comparison of path width and effective standoff distance.

TEST RESULTS

Figure 5 shows the pressure profile of each nozzle tested, as can be seen the effective standoff distance of the nozzles tested varies considerably from 12 ins. to 84 ins. It should also be noted that the "cleaning efficiency" of the various nozzles also varies from .0014 cu. ins./HP to .0062 cu. ins./HP as shown in Table 4.

CONSIDERATIONS IN CHOOSING NOZZLES

a) <u>Cost</u>

There is a wide variation in the cost of nozzles examined (Table 1). The choice of nozzle and cost justification are dependent upon the requirement of the application. For example, when cleaning the shell side of a heat exchanger with a "shell side machine", where the weight and cost are not of overriding importance, a nozzle type 1 would be used in an attempt to penetrate the inner tube walls of the exchanger. In an application where cost is more critical, a nozzle type 3 might be a better choice.

b) <u>Size</u>

Some automatic and manual cleaning applications have space limitations; therefore, in these cases the most effective nozzle that will physically fit is the obvious choice.

c). <u>Weight</u>

This criteria is usually only a factor in manual cleaning operations where the addition of a heavy nozzle to a "shotgun" may make the weight of the gun so heavy as to be extremely tiring to the gun operator.

d) <u>Performance</u>

For long reaching cleaning and cutting, nozzle 1 is the obvious choice, bearing in mind the other criteria discussed in this paper. However, if the object to be cleaned is readily accessible to the nozzle, then nozzle 3 might be a better choice based on the wider pressure profile. However, it should be noted that nozzle type 3 was uniquely inconsistent among the eight nozzles in its pressure profile. The hole produced was not always round. Examining the "cleaning efficiency", nozzle number 5 is the obvious choice and in this author's opinion is the best nozzle for contractors general hand gun use that was examined.

e) <u>Life</u>

When comparing the costs of nozzles, its life expectancy should also be considered. When determining the life expectancy of carbide insert nozzles, the location of the insert within the holder must be considered due to potential failure caused by impact fractures from flying debris. These impact fractures can alter the configuration of the exit side of the orifice, which can dramatically affect the performance of the nozzle causing premature failure.

CONCLUSION

When selecting a commercially available nozzle for use in water jetting, the user should not only consider the initial cost of the nozzle, but also its weight, size, life and performance as compared with other nozzles.

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NOZZLE NUMBER	PRICE OF COMPLETE NOZZLE	PRICE OF REPLACEABLE INSERT
1	$250.00	N/A
2	$195.00	N/A
3	$ 10.00	N/A
4	$ 55.00	$ 2.78
5	$ 33.00	$22.00
6	$ 28.00	N/A
7	$ 58.00	N/A
8	$ 15.00	N/A

TABLE I PRICE COMPARISON OF NOZZLES (As of March 7, 1985)

NOZZLE NUMBER	WEIGHT OF NOZZLE
1	3 Lbs. 12 oz. (1.701 Kg.)
2	2 Lbs. 00 oz. (0.907 Kg.)
3	0 Lbs. .5 oz. (0.001 Kg.)
4	0 Lbs. 6 oz. (0.170 Kg.)
5	0 Lbs. 1 oz. (0.028 Kg.)
6	0 Lbs. .5 oz. (0.001 Kg.)
7	0 Lbs. .5 oz. (0.001 Kg.)
8	0 Lbs. 1 oz. (0.028 Kg.)

TABLE 2 WEIGHT COMPARISON OF NOZZLES

NOZZLE NUMBER	LIFE (HRS.)
1	400
2	400
3	40
4	400
5	400
6	300
7	300
8	60

TABLE 3 LIFE COMPARISON OF NOZZLES

NOZZLE NUMBER	HORSE POWER PRESSURE(PSI) x FLOW (GPM) / 1714	MATERIAL REMOVED (CU. INS.)	MATERIAL REMOVAL PER HORSE POWER CU. INS / HP
1	69	0.42	.0060
2	57	0.12	.0021
3	57	0.14	.0024
4	55	0.08	.0014
5	69	0.43	.0062
6	54	0.24	.0044
7	57	0.34	.0059
8	61	0.34	.0055

TABLE 4 MATERIAL REMOVAL PER HORSE POWER

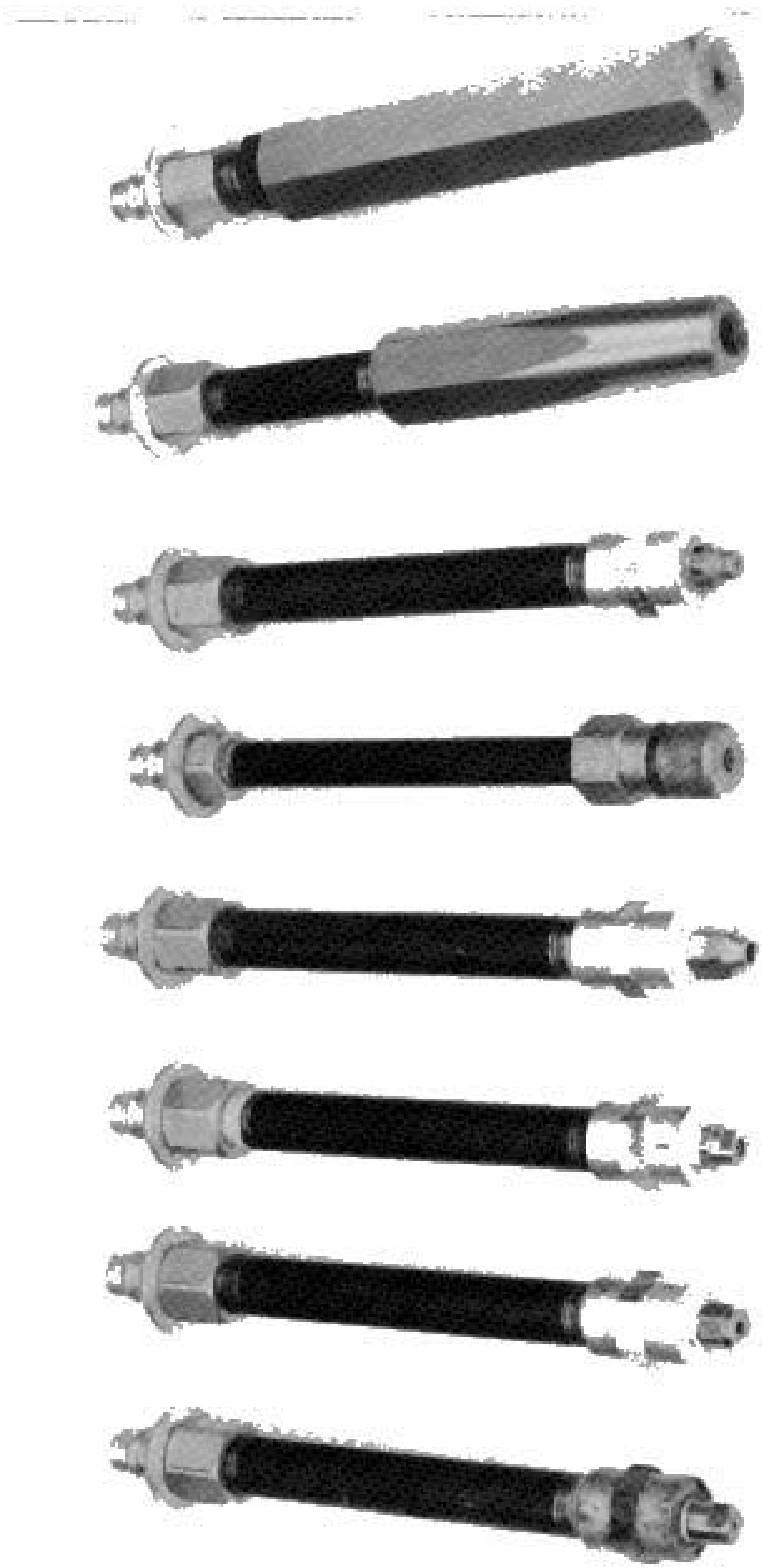

Figure 1. Nozzles Examined

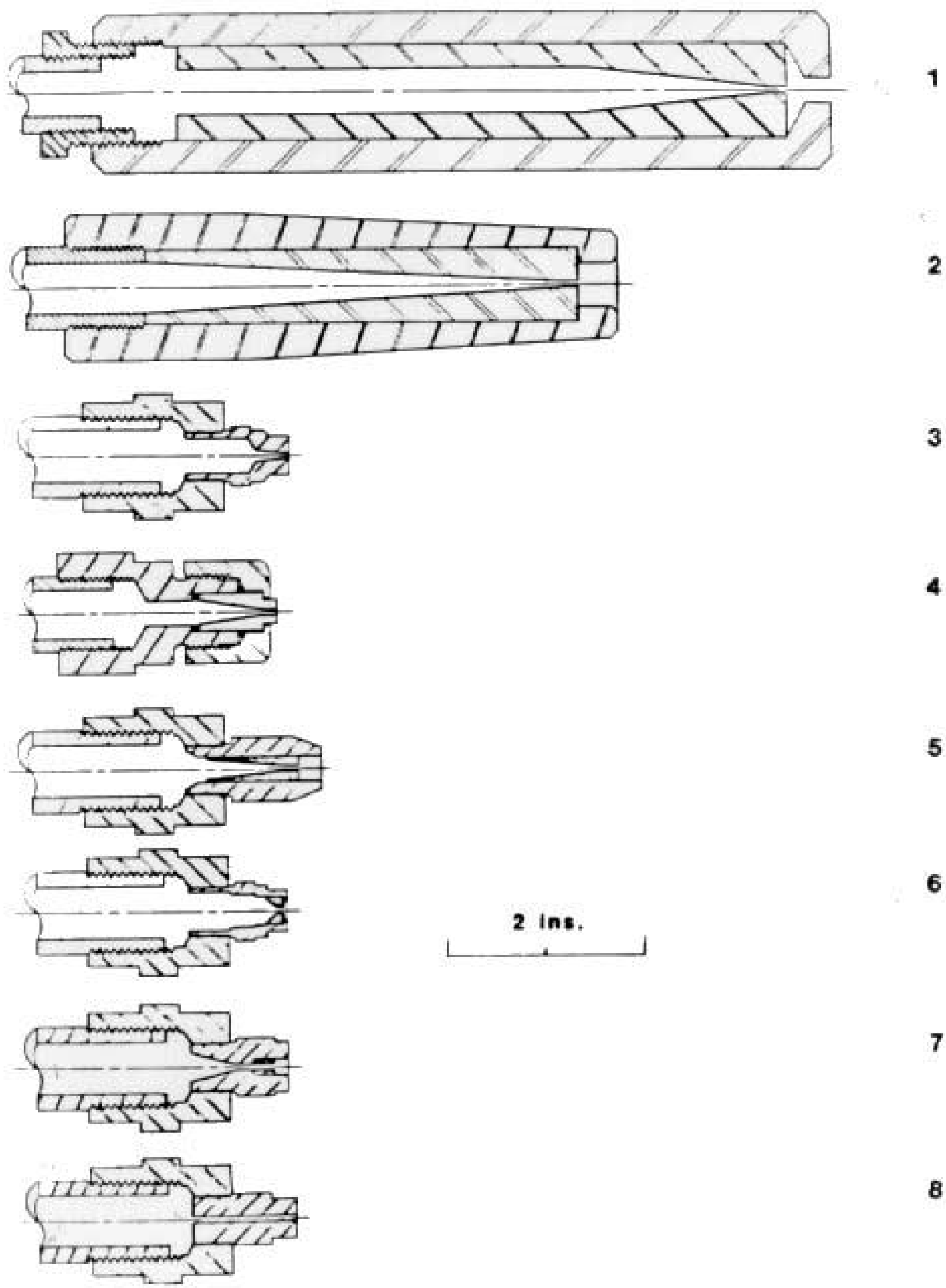

Figure 2. Sectional View of Nozzles Examined

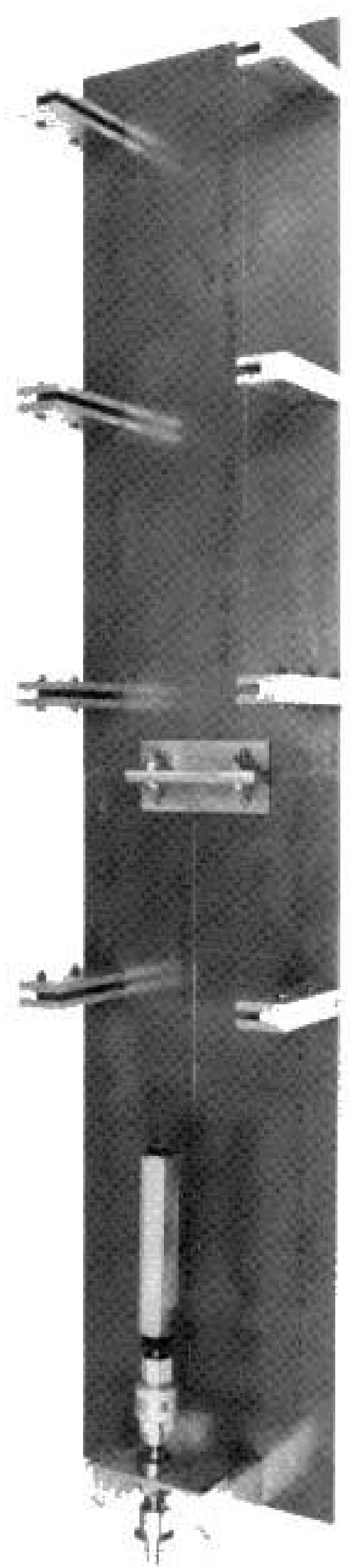

Figure 3. Test Set Up

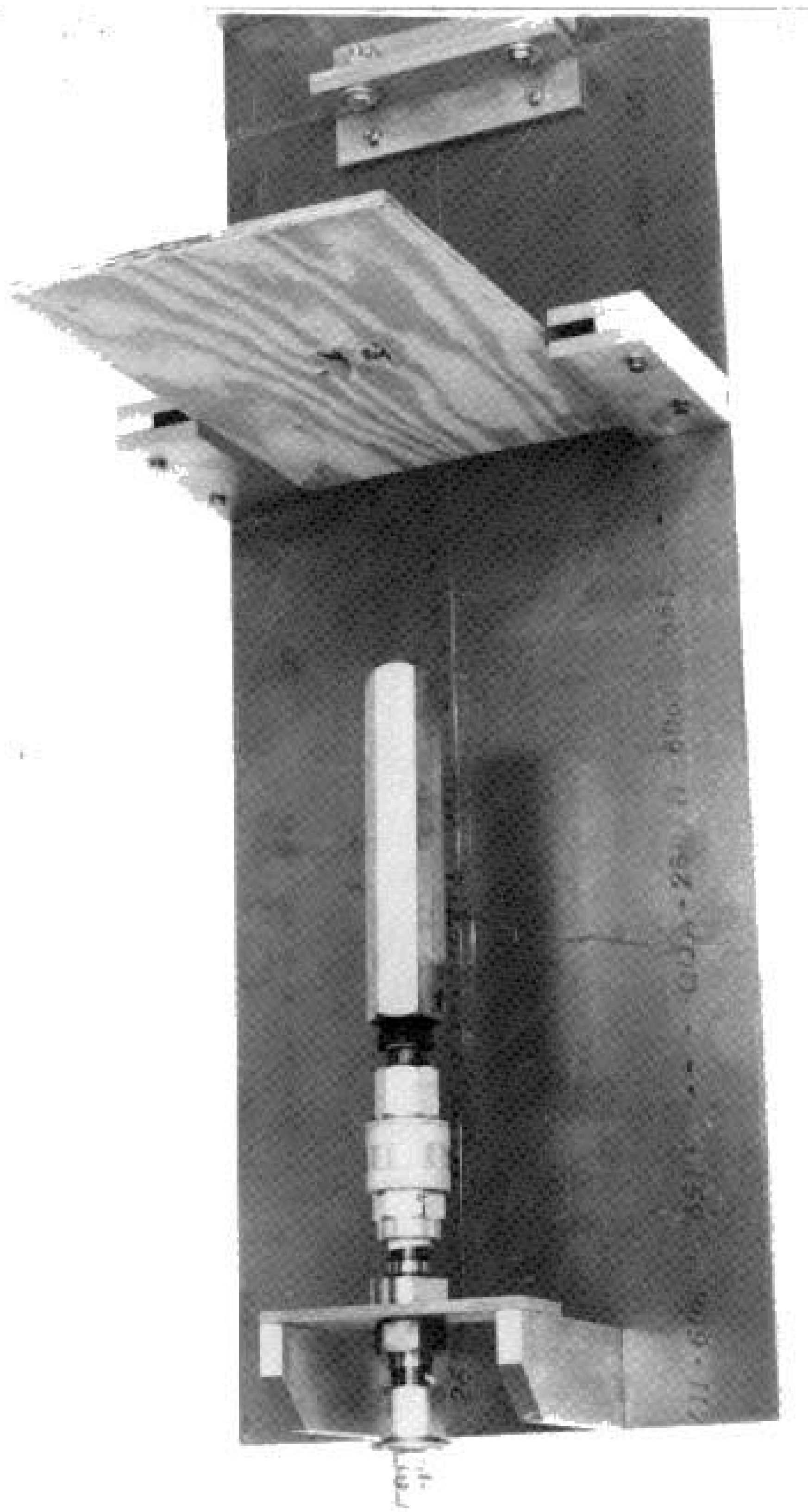

Figure 4. Close-up of Test Set Up

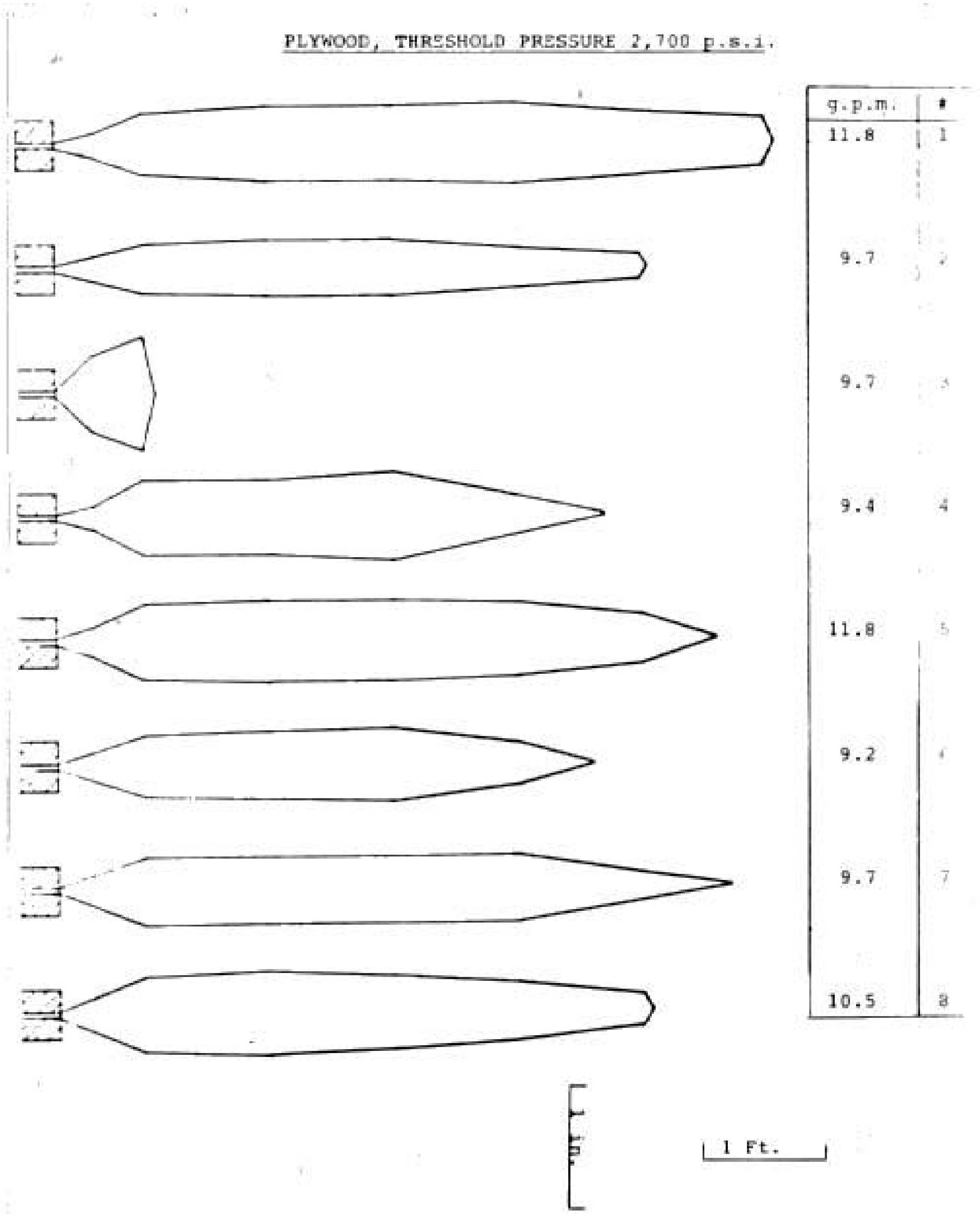

Figure 5. Test Results of Nozzles Examined

SHIP HULL CLEANING WITH SELF-RESONATING PULSED WATER JETS

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ABSTRACT

Cleaning trials on ship hulls in dry-dock have shown that white metal finishes can be achieved with pulsed, self-resonating water jets, using no grit or other additives. Under an ongoing program, these preliminary field trials were conducted in the shipyard at Tracor Marine, Fort Lauderdale, Florida. These tests have demonstrated potential of achieving the desired surface finishes at economically viable rates of cleaning. The success of these trials has motivated the continuation of this program to develop a prototype device for commercial use in shipyards.

INTRODUCTION

Grit blasting of steel ship hulls in dry-dock is an established and successful method. An operator can easily produce a wide range of results -- varying from a gentle and rapid removal of marine growth and loose paint patches to an excellent "white metal" finish suitable for immediate application of the primer coating -- by changing where he stands relative to the hull, and how he sweeps the air-borne particles across the surface. With such a procedure already in place in shipyards all over the world, why then did our company decide to embark on this examination of whether or not water jets -- containing neither additives nor abrasive particles -- could be a competitive replacement for the existing technology? The primary reason is pollution, the major drawback to grit blasting.

The most commonly used material for shipyard blasting of steel is known as "Black Beauty," a slag by-product from the smelting of copper. Although ideal for this use -- in terms of the hardness and sharpness of the particles -- the finer grit particles can be carried for large distances if even a gentle breeze is blowing. Nearby waterways are inevitably polluted by this grit; and cleaning costs in and around the drydock must be added to the significant contribution that the grit adds to the cost per unit area of cleaned hull. Accumulations of grit under floating dry docks may eventually lead to a major dredging effort. And the protective suits required for the grit blasting operator, to protect his eyes, skin, and lungs, can become infernos on even a moderately warm day.

A number of shipyards have begun to acquire some type of water blasting equipment because of the problems listed above. None of the existing water jet systems, however, are capable of economically creating the kinds of white metal finishes (SA-2.5 or SA-3; see the definitions of these surface finishes in Table 1) that many ship owners and the U.S. Navy require. Although the existing water jet systems can rapidly create an SA-1 surface, the white metal requirement cannot be effectively achieved with conventional water jet nozzles when operated within the pressure limitations of even nonstandard, positive-displacement pump units, namely, up to 138 MPa (20,000 psi). Since the costs associated with hoses and other components rise rapidly with pressure;

and there is a rapid decrease in the lifetime of all components as pressure increases, it is very desirable to keep the system pressure within the rating of what we would term a standard pumping unit, i.e., 68.9 MPa (10,000 psi). Pumps and associated hardware which operate at or below 68.9 MPa have been available for many years from numerous suppliers and are now developed to a level of high reliability and safety. This is particularly important in what can only be described as the extremely rugged environment of most shipyards.

The conflicting objectives of minimizing system pressure and yet achieving a rapid and thorough removal of fouling, paint, and rust from a steel hull were partially resolved in an earlier study our company conducted for the U.S. Maritime Administration (1,2). Using the CAVIJET cavitating fluid jet technology, it was demonstrated that pressures of only 13.8 MPa (2000 psi) were sufficient to economically create an SA-1 surface. As discussed in References 2 and 3, a six-nozzle experimental system requiring only 66.3 kW (89 hp.), cleaned at a rate of over 500 m^2/hr (5,400 ft^2/hr). At the time of this study (1975), the CAVIJET system power costs per unit area cleaned were only one-fifteenth of the grit costs for a comparable grit-blasted surface. Experiments, however, for achieving a white metal surface showed that although CAVIJET nozzles operating below 68.9 MPa (10,000 psi) could indeed remove all of the required surface adherents, the cleaning rates per unit of power were far too low to be practical with that type of cavitating water jet.

A new invention -- which, although using entirely different principles, also creates an effective "amplification" of the erosive capabilities of a water jet -- has recently shown itself capable of providing white metal surfaces within the 68.9 MPa (10,000 psi) constraint and at rates that are potentially competitive with grit blasting. This invention, the "SERVOJET™" self-resonating pulsed water jet, and the results of some preliminary cleaning trials on ship hulls in dry dock, are discussed in the following sections of this paper.

THE SERVOJET SELF-RESONATING PULSED WATER JET TECHNOLOGY

The advantages of pulsing to improve the erosive action of a jet -- by either modulating or interrupting the flow, and hence using the waterhammer stress created by a series of individual slug impacts -have long been appreciated by the users of water jets for cutting and cleaning applications. Such interrupted or pulsed jets provide an enhanced erosivity, compared with steady-flowing jets, for either cleaning a substance from a substrate or cutting into a bulk material, because of several interacting physical phenomena. These include: a larger initial impact stress, due to the waterhammer effect; larger outflow velocities, which aid the material removal processes; an increased area of impact, thus allowing delivery of the impact to a greater surface; and short duration, cyclic loadings which serve to more efficiently interact with naturally occurring material flaws and enhance debonding of surface adherents.

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 (4). It was then realized that the same principles of self-resonance could be used to passively interrupt a water jet in air (5, 6, 7). This SERVOJET concept is now being developed for a variety of cleaning applications, ranging from removing soil from aircraft exteriors to stripping worn nonskid coatings from aircraft carrier decks. To date, enhanced erosivity has been obtained over a pressure range of from 4.1 to 68.9 MPa (600 to 10,000 psi), with systems of between 3.7 to 150 kW (5 to 200 hp).

A variety of self-resonating nozzle system designs have been developed (see Figure 1), for either enhancing the creation of cavitation in a submerged jet, or to interrupt a water jet in air. The "PULSER-FED" self-resonating jet (SERVOJET), shown in Figure 1c, was the design found to be most readily adaptable to creating an in-air pulsed jet, and it can be seen from the several configurations in Figure 1 that the PULSER-FED concept is a combination of the "PULSER" (Figure 1a) and "ORGAN PIPE" (Figure lb) configurations. A tandem-orifice Helmholtz resonating chamber (diameter, d_T, in Figure 1c) is tuned so as to excite a standing wave within the organpipe section (length, L, Figure 1c). Peak resonance in this system occurs when the frequencies of the Helmholtz chamber and the organpipe wave are matched to a preferred jet structuring frequency for the exit orifice (d_e 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 the references cited above for further details on the performance of these self-resonating systems.

It is emphasized that no moving parts are required to create the necessary pressure and velocity fluctuations in these nozzle systems. Therefore, the required high-frequency pulsations for optimum erosivity enhancement can readily be created, without the excessive internal component wear and pressure losses associated with mechanically pulsed water jets.

THE EQUIPMENT

Since the objective of these field trials was merely to determine the feasibility of obtaining white or near-white metal finishes on actual ship hulls, the equipment used was selected on the basis of availability and minimal cost. Although this decision resulted in some interesting challenges during the testing, it proved correct since we did achieve our goals for this part of the project. The natural site for these trials was at TMI (Tracor Marine, Inc.), Fort Lauderdale, Florida, since TMI and Tracor Hydronautics are both subsidiaries of Tracor, Inc. A TMI-owned WOMA pump, with 68.9 MPa (10,000 psi) capability at flow rates up to 100 liters/m (26.5 gpm) was used, along with another portable diesel-powered unit, rented for the occasion. The rental pump was also rated for up to 68.9 MPa with flow rates up to 114 liters/m (30 gpm).

The multi-nozzle SERVOJET array is seen schematically in Figure 2. A total of 15 nozzles were available, eight in one manifold and seven in the other. High pressure, 12.7 mm (1/2 in.) ID hoses were run independently from the rental pump and the WOMA to the eight nozzle and seven-nozzle manifolds, respectively. The support plate for the two manifolds was slotted to allow for various orientations and relative positioning of the two sets of nozzles, as seen in Figure 2b. The support plate was mounted to a 51 mm (2 in.) diameter steel pipe, which in turn was clamped (see Figure 3) to the basket of the machine chosen to provide motion for the SERVOJET nozzle array.

This machine, as seen in Figure 4, was a "High Reach," (Model 50F Aerial Work Platform manufactured by JLG Industries) which was available in the TMI shipyard, and hence selected for these feasibility trials. Although its capabilities were less than ideal, a testing technique (see below) eventually evolved which allowed us to achieve our objectives. Use of such a machine for serious commercial cleaning of ship hulls, however, is certainly not recommended.

TEST PROCEDURES AND CONDITIONS

A variety of approaches for moving the nozzle manifolds were tried with the High Reach machine before a method giving reasonably reproducible results was found. We tried swinging the boom in either horizontal or vertical planes; and various "booming-in" and out moves were discarded. The procedure which finally worked may be seen in Figure 5; but only the skill of the operator, Mr. Gary Cunningham, seen here braving the spray and keeping the High Reach moving at a reasonably straight and constant velocity, allowed the effort to succeed. As stated, we do not recommend this -- a strictly experimental method -- for a system designed to clean an entire ship hull. By aligning the wheels of the High Reach to run parallel to the section of hull being cleaned, it was possible to complete runs of about 0.9 to 1.2 m (3 to 4 ft) in length at desired preselected speeds of translation -- after a brief learning period by our operator.

Prior to a run, the desired operating pressure was set by adjusting the RPM on each diesel until the required pressure was indicated on the respective manifold gauge. A length to be cleaned was marked on the hull, and the time to traverse between the start and stop lines was timed by stopwatch. In addition to translation speed, the other variables were system pressure, standoff distance, and orifice nozzle type. When all 15 SERVOJET nozzles were used, the maximum flow capacities available allowed nozzle pressure drops of up to 48.3 MPa (7,000 psi). By successive removal of nozzles (and plugging the openings in the manifolds), a range of pressures up to a maximum of 62.0 MPa (9,000 psi) could be attained (with nine nozzles in operation).

The main difficulties with this experimental procedure were: (a) maintaining a desired standoff distance, and (b) adjusting the angle of tilt and relative positioning of the two manifolds so as to avoid "gaps" (uncleaned strips between the paths cleaned by the individual nozzles), and (c) achieving a sufficiently rapid (and constant) speed of translation to allow adequate evaluation of the effects of this parameter. Variations in the hull plates and the difficulty of steering the High Reach in a perfectly straight path parallel to the hull both contributed to the standoff control problem. However, as seen in Table 2, practice served to improve this control. During these preliminary tests a gap-free total path was achieved by multiple runs across a portion of the hull (see Figure 6).

The following method for analyzing the results was used in order to infer the rates of cleaning achieved in these tests:

(a) The individual-nozzle clean path widths, gaps between clean paths, and total path widths were all measured.

(b) A factor, p, was derived as the estimated fraction of the total path width, w, that was cleaned to each specified degree of surface finish (SA-1, SA-2, or SA-2.5).

(c) The rate of cleaning, A, for a given test run, was then calculated from:

$$A = p\ w\ v \qquad [1]$$

where v is the speed of translation of the nozzles.

Two ship hulls were used for these trials. The first was the Exxtor I, a cargo ship with hull curvatures that made control of the test conditions particularly difficult. The second craft, the Revere Sun barge, was ideal for our purposes. Its straight flat hull surfaces provided large areas of fairly uniform fouling and paint conditions which facilitated the testing and interpretation of results. The Exxtor I hull had a vinyl coal tar primer, with a resin-based antifoulant coating. The Revere Sun barge had a mastic epoxy primer and a vinyl-based antifoulant paint.

Two types of exit orifice nozzle configurations (labeled d_e in Figure 1c) were used. For the majority of the tests (Runs 1 through 21), a circular orifice of diameter 1.1 mm (0.042 in.) was used. This orifice configuration provided a relatively narrower but more intense path of cleaning intensity. The remainder of the tests (Runs 22 to 30) utilized a 15° fan type nozzle, with an equivalent circular orifice of 0.9 mm (0.036 in.). This configuration delivered a wider but less intense cleaning path in comparison to the circular jet.

DISCUSSION OF RESULTS

The conditions and results for these preliminary hull cleaning trials are summarized in Tables 2 and 3. The cleaning rate effectiveness, e_a. is defined as:

$$e_a = A/P\ . \qquad [2]$$

The cleaning rates, A, were calculated using Equation [1], and P, is the hydraulic power delivered by the full set of nozzles used for a given run (values are listed in Table 2). The cleaning rates, for a given surface finish and nozzle pressure drop, are seen to vary widely. These variations were due to changes in standoff distance and in the speed of translation. Over the limited range of speeds available (up to only about 6.5 cm/s (2.5 in./s) which was the limit of the operator's ability to control the High Reach machine), there was essentially no measurable effect from the translation speed parameter. Therefore, this parameter was not listed in Table 3.

Based on laboratory testing, the SA-1 surface cleaning rates seen in Table 3 are as much as an order of magnitude lower than what might have been achieved if we could have translated the nozzles in a more rapid and controlled manner. A desirable standoff distance for SA-1 cleaning was in the range of about 7 to 12 nozzle diameters or about 8 to 13 cm (3 to 5 in.) for the round 1.1 mm nozzles, and about 5 to 10 cm (2 to 4 in.) for the fan nozzles. In summary, these results indicated that pulsed fan jets at a pressure of 48.2 MPa (7,000 psi) are able to achieve an SA-1, "brush-off cleaning," at rates of well over 32 m^2/hr (350 ft^2/hr), with a 150 kW (200 hp) pump unit.

The SA-2, "commercial blast" finish required slightly smaller standoff distances than the SA-1, about 5 to 10 cm (2 to 4 in.) for the round nozzles, and the preferred pressure was about 55.1 MPa (8,000 psi). A conservative estimate of the cleaning rate for the SA-2 finish is well over 14 m^2/hr (150 ft^2/hr) for the same 150 kW (200 hp) pump power. The fan nozzles delivered an erosive intensity which was only marginally capable of producing an SA-2 finish on the Revere Sun barge hull.

A pressure of 62.0 MPa (9,000 psi) was required, with the more intensive round nozzles, to achieve an SA-2.5, "near-white blast." The difficulty of maintaining nozzle alignment made it impossible to achieve a full (untapped) SA-2.5 path, hence, as noted in Table 2, multiple passes were made in order to assess the feasibility of achieving this type of surface. For this reason, we could only infer the SA-2.5 cleaning rates to be at least 7 m^2/hr (75 ft^2/hr), i.e., about one-half the SA-2 rates for the same pump power. Standoff distances for the SA-2.5 finish were comparable to those for the SA-2 surface.

RUST INHIBITING

A water-blasted bare steel surface will rapidly be covered by so-called "flash rust." Hull areas cleaned to an SA-2 or SA-2.5 finish during the SERVOJET trials at Tracor Marine showed this red layer of oxidation within a few hours of exposure to the humid Florida environment. Although it is common practice in Europe to paint over this tightly adhering oxide film (8), there is a resistance in the U.S., from shipyards, ship owners, and paint manufacturers, to placing the AC (anticorrodant) coating over a flash-rusted hull. To examine this facet of possible reluctance to using water blasting for ship hull cleaning, a paintability study was conducted.

A total of twelve test panels were used in this study. Four were cleaned to an SA-2.5 finish by conventional grit blasting; the other eight by water jetting. Of the latter eight, four panels were allowed to thoroughly flash rust before being painted, while the last four were protected with a rust inhibitor having the following components:

Percentage _ by Weight	Component
0.32	Sodium nitrite (NaNO2)
1.28	Diammonium phosphate (NH4) 2HPO4 (also known as: secondary ammonium phosphate (dibasic))
98.4	Water

This inhibitor has been recommended by the U.S. Navy (9) and the Steel Structures Painting Council (10), as being a safe and very inexpensive way to prevent flash rusting of bare steel surfaces. The cost for inhibitor chemicals to protect the complete hull of a typical ship serviced at Tracor Marine would only be about $2.61. Depending on the thoroughness of the application and the environmental conditions, this inhibitor can prevent flash rusting for as long as seven days (10).

All of the test panels were then protected with AC, top coat, and AF (antifouling) coatings at Tracor Marine, using the same procedures and materials as employed on ship hulls. The panels were then subjected to three months of dynamic seawater immersion testing on the Dynamic Drum Apparatus at the Miami Marine Research and Test Station, Miami Beach, Florida. Virtually no difference was seen in the adhesion of the AC layer to the steel surfaces prepared by the three methods. Our consultants (Mr. Carlos Perez, President, MMR&TS; and Mr. Leon S. Birnbaum, formerly head of the Coatings Branch, Naval Sea Systems Command, U.S. Navy) advised that only one to two months of such testing is sufficient to cause adherence failures (blistering, debonding) if an improperly prepared surface is present. This result of no adherence failures merely served to reconfirm the British and European experiences with what has become routine practice in the water jet preparation of ship hulls in drydock.

COMPARISON WITH GRIT BLASTING

The results from the preliminary SERVOJET hull cleaning trials were compared, on a cost per unit area basis, with typical shipyard costs (1983-84 $) for each specified type of surface cleaning by their existing grit blasting method. The assumptions and conclusions from this preliminary cost analysis are summarized in Table 4. It is seen, for each category of surface, that the SERVOJET costs per unit area are competitive with typical grit-blasting costs. And, it should be emphasized that these prices do not include the periodic cleanup efforts necessitated by accumulations of the grit both in and adjacent to the drydocks and synchrolift at this shipyard. These encouraging cost comparisons, based as they were on the admittedly less than ideal test configuration of the preliminary field trials, and the potential for greatly reducing the pollution problems, have motivated our company to decide to continue with the development of SERVOJET-based equipment for cleaning ship hulls in drydock.

CONCLUDING REMARKS

Preliminary drydock hull cleaning trials, using a multi-nozzle SERVOJET self-resonating water jet device, have demonstrated the feasibility of this method as a replacement for existing grit blasting procedures. Although the equipment used for these field trials was rather makeshift, the results showed that costs per unit area of cleaned hull, with an operational system, should be competitive with present grit-blasting costs, and to these costs should be added the grit removal expenses, and the pollution problems associated with existing blasting methods. Acceptable hull surface finishes, up to SA-2.5, "near-white blast," were achieved by the erosive action of impulsive water jets without any additives in the water.

This in-house project is continuing, based on the results reported here, with an on-going effort to design and build equipment which can rapidly and accurately deploy a multi-nozzle SERVOJET cleaning head. This cleaning head will be mounted in a framework which contains hydraulically actuated devices to provide variable and efficient translation speeds for the pulsed nozzle array for each required degree of surface cleaning. The framework will be affixed to a self-propelled, wheeled vehicle, such as one of TMI's scissor lifts, for the next round of field trials. Anticipating successful completion of these evaluations, design and construction of production equipment will follow.

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Table 1
Surface Cleaning Definitions

Type	Definition
Brush-Off Blast Cleaning SA-1 (also: SSPC-7, or NACE-4)	Brush-off blast cleaning is a method of preparing metal surfaces for painting or coating by rapidly removing loose mil scale, loose rust, and loose paint. A brush-off blast cleaned surface is defined as one from which all oil, grease, dirt, rust-scale, loose mil scale, loose rust, and loose paint or coatings are removed completely, but tight mil scale and tightly adhered rust, paint, and coatings are permitted to remain provided that all mil scale and rust have been exposed to the abrasive blast pattern sufficiently to expose numerous flecks of the underlying metal fairly uniformly distributed over the entire surface.
Commercial Blast Cleaning SA-2 (also: SSPC-6, or NACE-3)	Commercial blast cleaned surface finish is defined as one from which all oil, grease, dirt, rust-scale, and foreign matter have been completely removed from the surface, and streaks or discolorations, caused by rust stain, mil scale oxides or slight, tight residues or paint may be found in the bottom of pits; at least two-thirds of each square inch of surface area shall be free of all visible residues, and the remainder shall be limited to the light discoloration, slight staining, or light residues mentioned above.
Near-White Blast Cleaning SA-2.5 (also: SSPC-10, or NACE-2)	A near-white blast cleaned surface finish is defined as one from which all oil, grease, dirt, mil scale, rust, corrosion are completely removed from the surface except for very light shadows, very slight streaks, or slight discolorations caused by rust stain, mil scale oxides, or slight, tight residues of paint or coating that may remain. At least 94 percent of each square inch of surface area shall be free of all visible residues, and the remainder shall be limited to the light discoloration mentioned above.
White Metal Blast SA-3 (also: SSPC-5 or NACE-1)	A white metal blast cleaned surface finish is defined as a surface with a gray-white, uniform metallic color, slightly roughened to form a suitable anchor pattern for coatings. The surface, when viewed without magnifications, shall be free of all oil, grease, dirt, visible mil scale, rust, corrosion products, oxides, paint, or any other foreign matter. The color of the clean surface may be affected by the particular abrasive medium used.

Table 1 Surface Cleaning Definitions

Table 2
Summary of Test Conditions -- SERVOJET Hull Cleaning Trials

Run No.	Ship	Exit Nozzle Type	Nozzle Pressure Drop, MPa (ksi)	Translation Speed, v cm/s	Translation Speed, v in./s	Standoff Distance, X cm	Standoff Distance, X in.	No. of Nozzles	Total Flow Rate, Q ℓ/m (gpm)	Total Hydraulic Power, P kW (hhp)
1	EXXTOR I	Circular Orifice, diameter = 1.1 mm (0.042 in.)	48.2 (7.0)	1.7	0.67	>10	> 4.0	15	193 (51.0)	155 (208)
2				3.5	1.39	9-11	3.6-4.4			
3				4.0	1.57	8-15	3.0-6.0			
4				6.5	2.54	8-15	3.0-6.0			
5				3.6	1.40	8-20	3.0-8.0			
6				1.3	0.52	12-21	4.7-8.2			
7			45.5 (6.6)	2.8	1.11	10-20	4.0-8.0		187 (49.5)	142 (191)
8	SUN BARGE		48.2 (7.0)	3.4	1.33	10-13	4.0-5.0	13	167 (44.2)	134 (180)
9				2.4	0.96	10-13	4.0-5.0			
10				3.9	1.52	8-13	3.2-5.2			
11				4.6	1.83	5-16	2.1-6.4			
12			55.1 (8.0)	4.0	1.57	8-9	3.1-3.6	11	152 (40.2)	139 (187)
13				3.9	1.52	8-11	3.1-4.2			
14				4.9	1.91	>25	> 10.0			
15				3.2	1.26	8-11	3.1-4.2			
16				3.3	1.28	9-10	3.5-4.0			
17				2.3	0.92	9-10	3.5-4.0			
18				3.5	1.39	9-10	3.5-4.0			
19			62.0 (9.0)	3.1	1.24	8-9	3.0-3.4	9	133 (35.1)	137 (184)
20				3.0	1.18	8-9	3.0-3.4			
21				4.3	1.70	8-9	3.0-3.4			
22	REVERE	15° Fan equivalent dia. 0.9mm(0.036in.)	48.2 (7.0)	4.3	1.71	6-9	2.5-3.6	15	153 (40.5)	123 (165)
23				0.9	0.34	5-6	2.0-2.5			
24				3.4	1.33	6-7	2.2-2.7			
25				2.5	0.97	4-5	1.5-1.9			
26				0.8	0.32	3.5-4	1.4-1.5			
27				1.5	0.61	4-6	1.5-2.4			
28			62.0 (9.0)	4.3	1.71	6-7	2.2-2.6	9	102 (27.0)	106 (142)
29				3.8	1.50	6-7	2.2-2.6			
30				2.0	0.78	6-7	2.2-2.6			

Note: The following sets of runs were repeated over areas of the hull (two or three runs per a given area): (10, 11), (12, 14, 15), (16, 17, 18), (19, 20, 21), (28, 29, 30).

Table 3
Summary of Test Results -- SERVOJET Hull Cleaning Trials

Exit Nozzle Type	Surface Finish	Nozzle Pressure Drop MPa(ksi)	Cleaning Rates, $\dot{A}$				Cleaning Rate Effectiveness, e_a			
			Range of Results		Typical Values[a]		Range of Results		Typical Values[a]	
			m^2/hr	ft^2/hr	m^2/hr	ft^2/hr	$m^2/kW\text{-}hr$	$ft^2/hp\text{-}hr$	$m^2/kW\text{-}hr$	$ft^2/hp\text{-}hr$
Circular	SA-1	48.2(7.0)	9.7-54	104-584	16.7-29.7	180-320	0.06-0.35	0.5-2.8	0.12-0.19	1.0-1.5
orifice,		55.1(8.0)	20.4-22.6	220-243	20.8-21.9	224-236	0.15-0.16	1.2-1.3	0.15-0.16	1.2-1.3
diameter		62.0(9.0)	14.8[b]	159[b]	14.8	159	0.11	0.86	0.11	0.86
1.1 mm	SA-2	48.2(7.0)	4.0-33.9	43-365	8.5-18.6	92-200	0.02-0.22	0.2-1.8	0.06-0.12	0.5-1.0
(0.042in)		55.1(8.0)	5.5-17.7	59-191	6.6-13.3	71-143	0.03-0.12	0.3-1.0	0.05-0.10	0.4-0.8
		62.0(9.0)	5.3[b,c]	57[b,c]	5.3	57	0.04	0.3	0.04	0.3
15° Fan, equiva-lent dia	SA-1	48.2(7.0)	5.3-32.8	57-353	10.3-21.5	111-231	0.04-0.26	0.3-2.1	0.09-0.17	0.7-1.4
		62.0(9.0)	21.5[b]	231[b]	21.5	231	0.20	1.6	0.20	1.6
0.9mm	SA-2	48.2(7.0)	2.1-10.0	23-108	4.2-7.5	45-81	0.01-0.09	0.1-0.7	0.04-0.06	0.3-0.5
(0.036in.)		62.0(9.0)	1.0[b,c]	11[b,c]	1.0	11	0.01	0.1	0.01	0.1

a Typical values are defined as within one standard deviation around mean value of full range of results ($\pm 0.5\sigma$).
b These single values are because, although multiple runs were made at each pressure, in several cases (as noted in Table 2), repeated runs were made over the same hull area.
c Cleaning rates based on time required to make three passes over same hull area.

Assumptions (All costs are 1983-84 dollars)

1. Yearly Hull Cleaning: 88,250 m^2 (950,000 ft^2)
2. Equipment Cost Range: \$150,000 to \$300,000 (including maintenance)
3. Depreciation: Five years, linear
4. Labor: Two men at \$23.75 per hour; total: \$47.50 per hour
5. Power: \$0.07 per kW-hr; 150 kW (200 hp) system; therefore: \$10.50 per hour
6. Water: \$0.50 per 3785 ℓ (1000g); 114 ℓ/m (30 gpm) flow rate; therefore: \$0.90 per hour
7. Cleaning Rates: SA-1: 32 m^2/hr (350 ft^2/hr)
   SA-2: 14 m^2/hr (150 ft^2/hr)
   SA-2.5: 7 m^2/hr (75 ft^2/hr)

COST FACTORS: \$/m^2 (\$/ft^2)

Surface Finish	SA-1	SA-2	SA-2.5
Equipment	0.34-0.68 (0.032-0.064)	0.34-0.68 (0.032-0.064)	0.34-0.68 (0.032-0.064)
Labor	1.46 (0.136)	3.41 (0.317)	6.81 (0.633)
Power	0.32 (0.030)	0.75 (0.070)	1.51 (0.140)
Water	0.03 (0.003)	0.06 (0.006)	0.13 (0.012)
Total Cost Range for SERVOJET System	2.15-2.58 (0.20-0.24)	4.63-4.95 (0.43-0.46)	8.82-9.15 (0.82-0.85)
Typical Grit Blasing Costs	3.12 (0.29)	5.06 (0.47)	7.31 (0.68)

Table 4 Cost Analysis of SERVOJET Hull Cleaning System

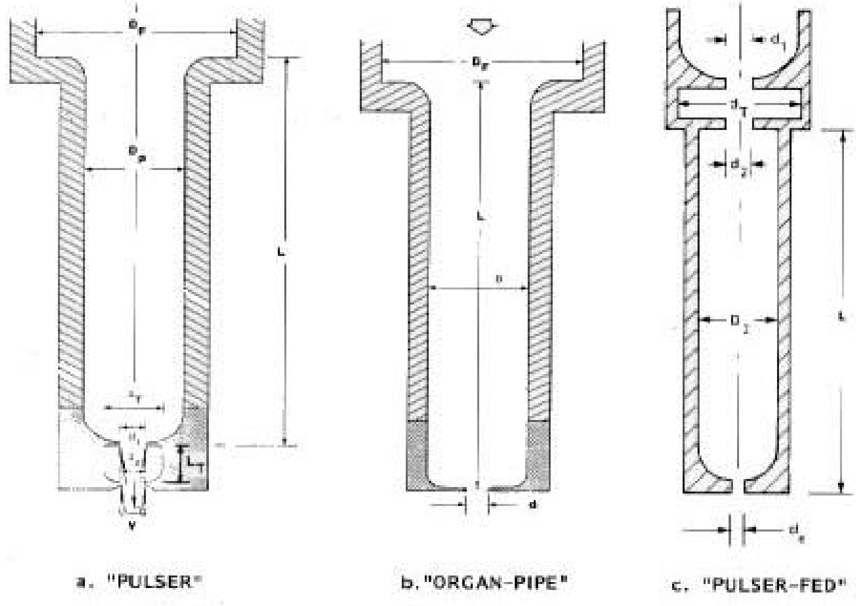

Figure 1 - Self-Resonating Nozzle System Designs

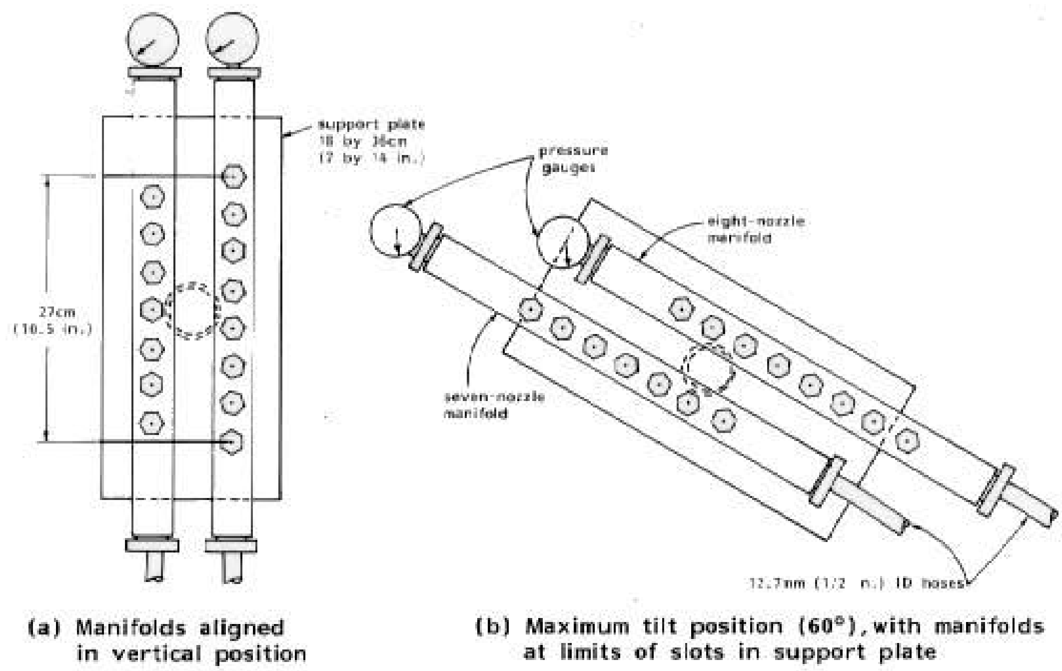

Figure 2 - Multi-Nozzle SERVOJET Array for Drydock Hull Cleaning Trials

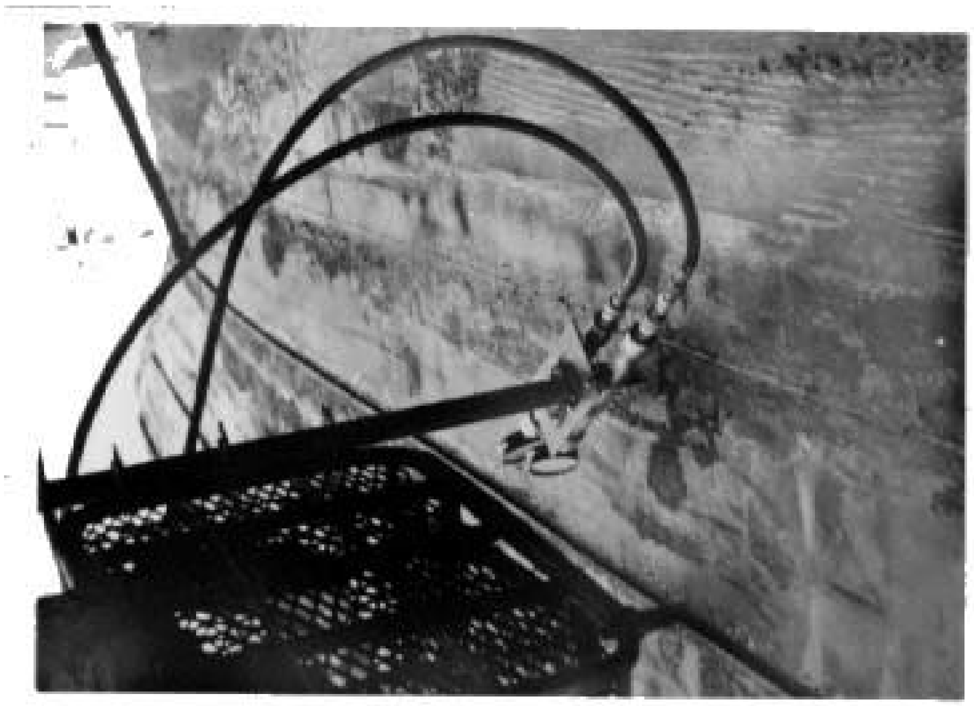

Figure 3 - SERVOJET Array Shown Mounted on Basket of "High Reach" Machine

Figure 4 - The "High Reach" Machine (adjacent to hull of the Revere Sun Barge)

Figure 5 - "High Reach" Machine Being Driven Parallel To Ship Hull. This method of testing was used for majority of hull cleaning trials.

Figure 6 - Results from Runs 10 and 11. Two runs on same hull area, to eliminate gaps (as seen in top of photo from prior run). A "commercial blast", SA-2 finish was achieved here.

RECOMMENDED PRACTICES FOR THE USE OF MANUALLY OPERATED HIGH PRESSURE WATER JETTING EQUIPMENT

U.S. Water Jet Technology Association
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1. INTRODUCTION

These recommended practices cover the personnel requirements, operator training, operating procedures and recommended equipment for the proper operation of all types of high pressure water jetting equipment as normally used by industries concerned with construction, maintenance, repair, cleaning and demolition work. Attention is drawn to the relevant or proposed, OSHQ ASTM and ANSI Standards. It is intended that extensions to this code will be produced, in due course, to cover specialist applications, e.g. multiple-gun operation, pulsed jets, cutting with the use of abrasives and high pressure intensifiers, but in the meantime these practices should be used as far as practicable.

The use of high pressure water jets for cutting and cleaning is a rapidly evolving technology, with current developments occurring. For this reason these practices are dated and the Association shall, biannually, review these practices for any required changes.

2. SCOPE

2.1 These recommended practices are intended to provide guidance on the proper operation of high pressure water jet cleaning and cutting equipment.

2.2 In this document the word "SHALL" indicates a requirement that is to be adopted in order to comply with these recommended practices.

2.3 The term high pressure water jetting covers all water jetting including the use of additives or abrasives at pressures above 70 bar (approximately 1,000 psi).

2.4 These recommended practices are also applicable at lower pressures where there is foreseeable risk of injury. As a guideline these Recommended Practices are applicable where the product of pressure times flow exceeds 560 bar liters per minute (pressure being measured in bar and flow in liters per minute) or 2,000 psi gpm (pressure in psi, flow rate in gpm).

2.5 Any person required to operate or maintain High Pressure Water Jetting Equipment shall have been trained and have demonstrated the ability and knowledge to do so.

3. DEFINITIONS OF TERMS

3.1 **High Pressure Water Jet Systems** - high pressure water jet systems are water delivery systems which have nozzles or other openings whose function is to increase the speed of liquids. Solid particles or additional chemicals may also be introduced, but the exit in all cases will be in a free stream. In terms of these recommended practices, the system shall include the pumps (pressure producing devices), and the hoses, lances, nozzles, valves and safety devices, as well as any heating elements or injection systems attached thereto.

3.2 **High Pressure Water Cleaning** - The use of high pressure water, with or without the addition of other liquids or solid particles, to remove unwanted matter from various surfaces, where the pressure of the liquid jet exceeds 1,000 psig (6.9 MPa) at the orifice.

<u>Caution</u> -- The lower limit of 1,000 psig (6.9 MPa) does not mean that pressures below 1,000 psig (6.9 HPa) cannot cause injury or require any less attention to the principles of these recommended practices. Adequate precautions, similar to those of these recommended practices, are required at all pressures.

3.3 **High Pressure Water Cutting** - The use of high pressure water, with or without the addition of other liquids or solid particles, to penetrate into the surface of a material for the purpose of cutting that material, and where the pressure of the liquid jet exceeds 1,000 psi (6.9 MPa) at the orifice.

3.4 **Lancing** - An application whereby a lance and nozzle combination is inserted into, and retracted from, the interior of a pipe or tubular product.

3.5 **Dump System** - An operator controlled manually-operated device or system that rapidly reduces the pressure to a level that yields a pressure flow at the nozzle that is considerably below the risk threshold.

3.6 **Moleing** - Moleing is an application whereby a hose fitted either with a nozzle or with a nozzle attached to a lance is inserted into, and retracted from, the interior of a tubular product. It is a system commonly intended for cleaning the internal surfaces of pipes or drains.

It can be self-propelled by its backward directed jets, and is manufactured in various shapes, sizes and combinations of forward and backward directed jets.

3.7 **Nozzle -** A device with one or more openings where the fluid discharges from the system. The nozzle restricts the area of flow of the fluid, accelerating the water to the required velocity and shaping it to the required flow pattern and distribution for a particular application. Combinations of forward and backward nozzles are often used to balance the thrust. Such nozzles are commonly referred to as tips, jets, orifices, etc.

3.8. **Operator** - A person who has been trained and has demonstrated the knowledge and experience to perform the assigned task.

3.9 **Operator Trainee** - A person not qualified due to the lack of knowledge and/or experience to perform the assigned task without supervision.

3.10 **Shotgunning** - An application whereby a lance and nozzle combination can be manipulated in virtually all planes of operation.

3.11 **Hose Assembly** - A hose with coupling attached in accordance with manufacturer's specifications.

3.12 **Lance** - A rigid metal tube used to extend the nozzle from the end of the hose.

4.0 EQUIPMENT DEFINITION AND STANDARDS

4.1 **Pressurizing Pump** - A unit designed to deliver high pressure water or other fluid. This is usually based on positive displacement pistons or rubber diaphragm/hydraulic systems, and discharges water into a common manifold to which either flexible hoses, or rigid tubing connecting to lances and nozzles are attached. These pumps can be either mobile or permanently mounted.

The pump should have a permanently mounted tag which provides the following information:

1) Product and supplier.
2) Production model and serial number, or year of production.
3) Maximum performance in terms of gpm and pressure in psi.

4.2 **Relief System** - The system shall be equipped with an automatic relief device on the discharge side of the pump.

4.3 **Automatic Pressure Relief Devices** - These may take the form of:
(a) Pressure Relief valve or Bursting Disc in Holder.
(b) Automatic Pressure Regulating Valve (Unloading Valve).

4.3.1 **Pressure Relief Valve or Bursting Disc in Holder** - Usually mounted on the pump discharge chamber to prevent the pressure exceeding the rated maximum pressure of the whole system.

4.3.2 **Automatic Pressure Regulating Valve (Unloading Valve)** - Limits the pressure at which the pump operates by releasing a preset proportion of the generated flow back to the pump suction chamber or to waste. It may be used to regulate the water pressure from the pump and is individually set for each operation. This device may be integral with the pump hydraulic assembly.

4.4 **Pressure Gauge** - The system should be equipped with a gauge indicating the pressure being developed. Gauges shall have a scale range of at least fifty percent (50%) above the maximum working pressure of the system.

4.5 **Filter or Strainer** - The water system should be equipped with a filter or strainer to prevent particles from restricting the orifices in the nozzle. The strainer or filter should be capable of removing particles smaller in size than the smallest orifice in the nozzle, and usually smaller to protect pumps, etc.

4.6 **Dry Shut-Off Control Valve** - This operator-controlled valve, normally hand-controlled, automatically shuts off flow to the lance and/or nozzle assembly when released by the operator, but retains the operating pressure within the supply line when so shut off. This valve shall be used in systems with an Automatic Pressure Regulating Valve (see Section 4.3.2 of this recommended practice).

Care shall be taken to release the pressure in the dry shut off valve and line when the pump is shut down otherwise the valve operating lever may remain alive. This valve may alternatively be actuated by solenoid or pilot pressure mechanism.

4.7 **Dump System** - The system should be equipped with a device which will either shut down the unit, idle it to a low rpm, bypass the flow or reduce the discharge pressure to a low level. The dump system shall be manually controlled only by the nozzle operator. The dump system actuator device should be shielded to preclude inadvertent operation. This device should immediately shut off the high pressure water stream if the operator loses control.

4.7.1 **Dump Control Valve** - This operator-controlled valve, normally hand-controlled, automatically terminates significant flow to the lance and/or nozzle assembly when released by the operator, thus relieving the operating pressure within the whole system by diverting the flow produced by the pump to atmosphere. A valve size should be selected that will not cause generation of significant back pressure at the maximum possible pumping rate of the pump. This valve may alternatively be actuated by solenoid or pilot pressure mechanism.

4.7.2 **Solenoid & Electrically Operated Control Dump Systems** - All electrically controlled dump systems should be of fail safe design. Voltage of an alternating current (AC) or direct current (DC) dump system handled by personnel should not exceed 24 volts.

4.8 **High Pressure Hose -** This is a flexible hose which connects two components and which delivers the high pressure fluid to the gun or nozzle components. The hose should have a burst rating of a minimum of 2.5 times the intended working pressure. Operating levels below this ratio should require a protective shielding around that hose. The hose should be marked with the manufacturer's symbol, serial number, the maximum permissible operating pressure and the test pressure.

4.9 **End Fittings and Couplings** - High pressure hose end fittings and couplings shall be manufactured to be compatible with the hose and tested as a unit.

4.10 **Jetting Gun Extension** - This is a length or lengths of tube carrying high pressure fluid to the nozzle. Each shall be manufactured from suitable material to suit the application. End connections shall be suitable for the application. The extension is used in conjunction with a control valve (see Section 4.6 & 4.7 of these Recommended Practices). The extension shall have a minimum burst strength of at least 2.5 times the highest actual operating pressure used.

4.11 **Nozzle -** the nozzle creates the water jet or jets at the required velocity, flow rate, pressure, shape and distribution for a particular application. Combinations of forward and backward direct water jets are often used to balance the thrust. Such nozzles may be referred to as tips, jets, or orifices.

4.12 **Water Jet -** A jet stream of water produced from the individual outlet orifice of a nozzle. The shape of the jet is determined by the form of the orifice while the speed at which it travels is determined by the orifice design, orifice area, and flow. The pressure drop at the orifice is a result of an increase in velocity. The two most commonly used jet shapes are the straight-jet and fan-shaped jet.

4.12.1 **Straight Jet** - Concentrates the stream of water over a small area of the work-piece by minimizing the spread. A typical application is for cutting, or for general cleaning of matter with higher shear and/or bond strength.

4.12.2 **Fan Jet** - Spreads the stream of water in one plane, so giving a wide band coverage of the workpiece. A typical application is for cleaning larger areas requiring less energy to remove unwanted matter.

4.13 **Jetting Head Manifold and Spray Bars -** These are pieces of equipment into which individual nozzles are fitted.

4.14 **Foot Control valve** - The operator's control valve (see Sections 4.6 & 4.7 of these recommended practices) may be arranged for actuation by the operator's foot if desired, either in place of, or in addition to, hand-control.

An adequate guard should be fitted to prevent accidental operation and the base plate should be sufficient to ensure stability in use. If on the dump type (Section 4.7) the layout should ensure that the dump line, if used, is restrained from whipping when the valve is released.

An adequate guard shall be fitted to prevent accidental operation and the base plate area shall be sufficient to ensure stability in use. If on the dump type (Section 3.4) the layout should ensure that the dump line, if used, is restrained from whipping when the valve is released.

4.15 **Jetting Gun** - A portable combination of operator's control valve (Sections 4.6 and 4.7), lance (Section 4.10) and nozzle (Section 4.11) resembling a gun in layout and outline.

The control valve is hand-operated, generally by a squeeze action of the hand of the operator, who should always have control of this device and may be of the dry shut off (Section 4.6) or dump (Section 4.7) type, the gun being named accordingly. The hand-control normally takes the form of a trigger or lever which is provided with a guard adequate to prevent accidental operation and which shall have the means of being immobilized in the "off" position by means of a safety catch. The gun may be fitted with a shoulder pad or hand grips to facilitate back-thrust control.

4.16 **Retro Gun** - A retro safety gun is fitted with forward and backward facing jets. This reduces the thrust experienced by the operator. This type of gun is used mainly for underwater jetting operations. The retro balance jet protection tube should be sufficiently long or constructed so as to prevent the operator directing a retro balance jet at himself.

4.17 **Changeover Valve** - An operator-controlled valve designed to properly direct high pressure water flow from the pump (see Section 4.1 of this Code) to one or other items of equipment at the operator's choice. It shall be designed to withstand the maximum system pressure, and can be power operated.

5.0 CARE AND MAINTENANCE OF EQUIPMENT

5.1 **Pump Unit** - The pump unit shall be maintained in accordance with the manufacturer's instructions. Where applicable this should include daily checks on the following items:

(a) Drive unit - lubricating oil, water, hydraulic fluid and fuel levels.
(b) Pump unit - lubricating oil and gear box oil levels.
(c) Hydraulic hose reel - lubricating oil and fluid levels.
(d) Condition of guards and shields.

5.2 **High Pressure Hose** - the high pressure hose should be tested at 1.5 times operating pressure at least once a quarter.

5.3 **Filters and Strainers** - All water filters should be checked at regular intervals, dependent upon the supply water conditions, and in accordance with pump manufacturer's recommendations.
Extreme care should be taken to filter the water source through proper micron filtration, to prevent foreign particles from cutting changeover valves and seating surfaces, and to prevent clogging the changeover valve operating mechanism. Such clogging can cause a loss of control, which can be dangerous to the operator.

5.4. **Hose Assemblies -** All hose assemblies shall be inspected prior to use with respect to the following:

(a) Correct pressure rating and size.
(b) Free from external damage, i.e., broken wires.
(c) All end fittings and couplings are in good order and of the correct pressure rating for the unit operating pressure.

5.5. **Nozzles -** All jetting nozzles shall be kept clean and the orifice shall be checked to ensure that it is not obstructed or damaged before installation. Defective nozzles shall not be used but should be replaced or repaired before installation. During the startup prior to operation, the nozzle should be removed from the lance and the system flushed thoroughly, to remove air and foreign particles.

5.6. **Jetting Guns & Lances -** Jetting guns and lances shall be checked daily and the trigger mechanism and guard given a thorough examination to ensure correct operation. All high pressure connections should be observed during operation of the equipment at pressure. If a leak is observed, the pump shall be shut down and the connection repaired or replaced before further operation.

5.7. **Foot Control Valves** - All foot control valves shall be checked and cleaned daily and the foot mechanism and guard given a thorough visual examination to ensure correct operation.

5.8 **Electrical Equipment** - All electrically-operated high pressure water jetting units shall be checked daily for external damage, with special emphasis placed on connections, junction boxes, switches and supply cables. Care should be taken to ensure that the electrical system is protected from the ingress of water. Correct direction of rotation of the electric motor should be checked on initial installation and after every re-connection.

5.9 **Trailers** - Trailer-mounted units shall be checked daily examining tires, braking systems, jacking points, towing hitch, lights, safety chains, structural damage and general cleanliness. The units should only be towed by vehicles fit for the purpose.

5.10 **Engine Controls** - All throttle cables and engine stop devices shall be checked daily to ensure that they are functioning properly.

5.11 **Maintenance Servicing and Repair** - The following operations should only be carried out by competent personnel:
(a) Manufacturers' servicing requirements.
(b) The following items should be overhauled and checked for correct functioning at manufacturer's recommended intervals:
(i) Pressure relief valve.
(ii) Bursting discs if used.
(iii) Pressure control valves.
(iv) Hand or foot operated dump control valve or dry shut off control valve.

(v) Dry shut off or dump gun.
(vi) Changeover valve.

5.13 **Tools** - When maintaining or assembling jetting systems the correct size tools must be used. The use of adjustable tools having serrated gripping jaws, for example pipe wrenches, which can damage equipment is not recommended, particularly on the crimped portion of a hose fitting.

5.14 **Compatibility** - All component parts and fittings should be checked to ensure they are of the correct size and rating for the unit.

6.0 PROTECTIVE CLOTHING AND PERSONNEL PROTECTION

6.1 **OSHA Compliance** - All applicable OSHA regulations covering personal protective equipment shall be followed.

6.2 **Head Protection** - All operators shall be issued with suitable head protection which shall be worn. Where possible, this should include a full face shield.

6.3. **Eye Protection** - "Suitable" eye protection (adequate for the purpose and of adequate fit on the person) shall be provided to all operators of high pressure water jetting equipment, and must be worn within the working area. Additionally, several states have regulations governing eye protection, which must be conformed with.

Where liquids liable to cause eye damage are encountered, it may be necessary to use either a combination of visor and goggles or a full hood with shield.

6.4. **Body Protection** - All operators should be supplied with suitable waterproof clothing having regard to the type of work being undertaken. Garments should provide full cover to the operator, including his arms. Liquid or chemical resistant suits shall be worn where there is a reasonable probability of injury that can be prevented by such equipment.

6.5 **Hand Protection** - Adequate hand protection should be supplied to all operators and shall be worn when there is a reasonable probability of injury that can be prevented by such equipment.

6.6 **Foot Protection** - All operators should be supplied with waterproof boots with steel toecaps. A metatarsal guard should be used by jetting gun operators.

6.7. **Hearing Protection** - Most high pressure water jetting operations produce noise levels in excess of 90 dB(A) and so suitable ear protection issued in accordance with OSHA Standards must be worn and provision should be made for its regular inspection and maintenance. All personnel and operators should receive instruction in the correct use of ear protectors, so that noise exposure lies within the limits as specified by OSHA.

6.8 **Respiratory Protection** - A respiratory protection program shall be implemented where there is a reasonable probability of injury that can be prevented by such a program.

6.9 **Equipment Limitations -** It should be recognized that protective equipment may not necessarily protect the operator from injury by direct high pressure water jet impact.

7.0 PRE-OPERATING PROCEDURES

7.1 **Planning** - Each job shall be pre-planned. Personnel familiar with the equipment to be cleaned or the material to be cut and the work environment shall meet with the personnel that will be doing the work, and outline potential hazards of the work area, environmental problems, safety standards and emergency aid procedures.

7.2. **Check List** - A check list shall be used to assure that the proper procedures and proper equipment selection are followed (see Appendix 1).

7.3. **Dump Valve** - All systems shall incorporate at least one fluid shut off or dump device. The gun operator must always be able to shut down the water jet by releasing pressure on the trigger, switch, or foot valve pedal.

7.4 **Warning Barriers** - Suitable barriers shall be erected to encompass the hazard area
and signs posted to warn personnel they are entering a hazardous area. The perimeter should be outside the effective range of the jet wherever possible. Barriers may be or rope, safety tape, barrels, etc. as long as they give an effective warning, and are highly visible.

7.5 Hook-up

7.5.1 **Hose** - Hose shall be arranged so a tripping hazard does not occur. Hoses, pipes and fittings shall be supported to prevent excessive sway, and/or wear created by vibration or stress on the end connections, when laid on the ground, over sharp objects or on vertical runs.

7.5.2 **Fittings** - All fittings shall be cleaned and lubricated before installing in the system. Be sure all fittings, hoses and nozzles are fit for the purpose.

7.5.3 **Hose -** All hoses shall be checked for evidence of damage, wear or imperfections. The check shall be made periodically during the operation.

7.5.4 **Preflushing -** The system shall be completely flushed with sufficient water to remove any contaminants before installing the nozzle.

7.5.5 **Nozzles** - All orifices shall be checked in all nozzles for any stoppage, and/or damage or imperfections.

7.5.6 **Electrical Equipment** - Any electrical equipment in the immediate area of the operation that presents a hazard to the operator shall be de-energized, shielded, or otherwise made safe.

8.0 OPERATIONAL PROCEDURES

8.1.1 **Work Area** - Where practical, work pieces to be jetted should be removed from plant areas to a high pressure water jetting area. Where this is impractical, cutting or cleaning in place or adjacent to the installed position can be done with the necessary clearance and permission of the occupier.

8.1.2 **Area Limits** - Area limits applicable to the cutting or cleaning operations shall be defined, and the team shall mark these limits by barriers and notices to warn against access to other personnel. Suitable barriers shall be an approved form of hazard warning, rope or tape, as a minimum. Alternatively, a suitable barrier shield is acceptable at any reasonable distance. Notices should state "KEEP CLEAR, HIGH PRESSURE WATERJETTING IN OPERATION", or other suitable wording.

8.1.3 **Corrosive Materials** - Where there is a possibility of encountering corrosive or toxic materials, the occupier shall be requested to inform the person in charge of high pressure water jetting of any precautions that may be necessary, including the collection and disposal of waste materials.

8.1.4 **Work Surface** - Operators should have good access to the workpiece a safe working platform and secure footing. The area in which work is to proceed shall be kept clear of loose items and debris to prevent tripping and slipping hazards.

8.1.5 **Access** - Access by unauthorized persons into the area where high pressure water jetting is taking place shall be prevented. The area shall be cordoned off and warning notices displayed in prominent positions. The perimeter should be outside the effective range of the jet wherever possible.

8.1.6 **Approaching the Operator** - The occupier shall be requested to inform all personnel likely to require access to the area that high pressure water jetting is in progress. Personnel having reason to enter the water jetting area should wait until the jet is stopped and their presence known. Personnel wishing to have the jet stopped shall approach a team member other than the jet operator. The jet operator shall not be distracted until the jet has been stopped.

8.1.7 **Side Protection** - Target and side shields shall, where feasible, be suitably placed to safeguard personnel and equipment against contact with grit, or solids removed by the jets.

8.1.8 **Protective Equipment** - All personnel working or entering the barricaded area while cleaning or cutting is in progress, shall wear the required protective equipment.

8.2 **Pressurizing the System** - Pressure shall be increased slowly on the system while it is being inspected for leaks and/or faulty components. All leaks or faulty components shall be repaired or replaced. The system shall be de-pressurized for repairs.

8.3 **Team Operations** - In most jetting operations it is accepted practice to employ a minimum of two persons.

8.3.1 **Supervision** - All high pressure water jetting operations shall be controlled by a Supervisor who is trained in all aspects of the jetting operation.

8.3.2 **Number of Operators** - The operation of the high pressure water jetting equipment should be by two or more operators according to the equipment being used and the nature of the job. These operators shall work as a team, with one member in charge. The operator of the gun or lance as described in Section 8.3.3 (below) shall take the lead role while jetting is in progress.

8.3.3 **Gun Operator** - One operator from the team shall hold the lance, gun or delivery hose, with the nozzle mounted on it. His primary duty is to direct the jet.

8.3.4 **Second Operator** - The second operator of the team shall attend the pump unit, keep close watch on the first operator for signs of difficulty or fatigue, and watch the surrounding area for intrusion by other persons or unsafe situations. If required, the operator will shut off the pressure until it is safe to continue. Caution should be exercised in shutting off the pressure rapidly as this can cause the loss of footing by the gun operator.

8.3.5 **Additional Operators -** Further operators are required in the following circumstances:

(a) To assist the first operator with the handling of the lance if it is too long or heavy for one man.

(b) To provide communication if the lance operator is out of sight of the pump unit operator.

8.3.6 **Job Rotation** - The team members should rotate their duties during any job to minimize fatigue to the operator holding the lance or gun.

8.3.7 **Team Leader** - The team leader is responsible for basic equipment checks as detailed in Section 8.9 (below), the preparation of the working area for safe operation, and for obtaining a permit to work where and when required.

8.3.8 **Code of Signals** - Before starting a jetting operation the team members, one of whom must be in charge, shall agree on a code of signals to be used during the operation of the equipment.

8.3.9 **Fitness -** The operator and other team members shall be physically and mentally capable of performing the required operations.

8.4 **Single Person Operation** - Single person operation is allowed where the pressure does not exceed 2,000 psi, and the flow is less than 20 gpm.

8.4.1 **Single Operator Guidelines** - All other recommendations, pertaining to team operations, shall hold.

8.5 **Shotgunning**

8.5.1 **Controls** - The person operating the nozzle shall have direct control of the dump system.

8.5.2 **Attendance** - The system shall never be left unattended when pressurized.

8.5.3 **Multiple Operation** - When more than one shotgunning operation is being performed within the same area, a physical barrier shall be installed or adequate spacing between operators shall be maintained to prevent the possibility of injury from the high pressure water.

8.5.4 **Target Holding** - Objects to be cleaned shall never be held manually.

8.5.5 **Connection Protection** - The point where the hose connects to the gun shall be shrouded by a protective device such as a heavy duty hose, shoulder guard, etc. such as to prevent injury to operator should hose, pipe, or fitting rupture.

8.5.6 **Minimum Length** - Where practicable, the minimum length of the shotgun lance extension should be 4 ft. from the triggering device to the nozzle.

8.5.7 **Hose Protection** - Steel braided hoses should be used on air operated fail safe systems to keep the system from being activated by someone stepping on the hose or running over it.

8.6 **Moleing or Flex Lancing**

8.6.1 **Control** - The operator inserting the nozzle shall have direct control of the dump system.

8.6.2 **Reversing** - A positive method shall be used to prevent the nozzle from reversing direction inside the item being cleaned.

8.6.3 **Retrojets** - During manual operations, the entrance to a line or pipe shall not be cleaned with a nozzle containing back jets without adequate shielding.

8.6.4 **Clearance** - The clearance between the outside diameter of the hose, lance and nozzle assembly and the inside wall of the item being cleaned shall be sufficient to allow adequate washout of water and debris.

8.6.5 **Pressurization** - During manual operation, the nozzle shall be inserted into the tube prior to pressurizing. Conversely, the system shall be de-pressurized before removal of the nozzle from the tube.

8.6.6 **End Identification** - Hoses shall be conspicuously marked no closer than 24" (0.6 m) from the nozzle to warn the operator of the nozzle location.

8.6.7 **Nozzle Support** - Where the length of the nozzle and rigid coupling is less than the inside diameter of the pipe, a length of rigid pipe of not less than the diameter of the pipe being cleaned should be fitted directly behind the nozzle, or a suitable safety shield should be provided to protect the operator. This is to prevent the nozzle turning around 180° and doubling back towards the operator.

8.7 Rigid Lancing

8.7.1 **Control** - The operator inserting the nozzle shall have direct control of the dump system.

8.7.2 **Clearance** - The clearance between the outside diameter of the lance and nozzle and the inside wall of the item being cleaned shall be sufficient to allow adequate washout of water and debris.

8.7.3 **Pressurization** - When under manual operation, the nozzle shall be inserted into the tube prior to pressurizing. Conversely, the system shall be de-pressurized before removal of the nozzle from the tube, unless proper shielding is provided.

8.7.4 **Shields** - When lancing tubes with a rigid lance, a guard should be installed, where practicable, around the lance, to prevent a lance nozzle from being inadvertently withdrawn and causing injury.

8.8 Additives

8.8.1 **Additives** - Any water additive (chemical, detergent, or solid particle) shall be used in accordance with the manufacturer's recommendations.

8.9 Proper Operation

8.9.1 **Startup** - The pump unit shall not be started and brought up to pressure unless each team member is in his designated position, the nozzle is held in or directed at the workpiece, and the lance or gun securely held.

8.9.2 **Adjustments** - Apart from operational procedures no attempt shall be made to adjust any nut, hose connection, fitting etc., while the system is under pressure. The pumps shall be stopped and any pressure in the line discharged prior to making any such adjustment. Care should be taken to release the pressure in the dry shut off gun and the line when the unit is switched off.

8.9.3 **Equipment Malfunction** - If for any reason the water flow does not shut off when the trigger or foot pedal is released, work shall cease until the item has been serviced, repaired or changed, by properly trained personnel.

8.9.4 **Reaction Force** - The operator should be allowed to experience the reaction force of the jet progressively until the required operating pressure is reached. The lowest pressure should be used compatible with the work to be done. The pressure shall not be adjusted without the operator's awareness.

8.9.5 **Effect of Line Pulses** - Operators should be made aware of the reactive effect of pressure in the line which can transmit a severe jolt to the operator when the dump valve or dry shut-off valve is operated. To minimize this effect total hose lengths should be kept as short as possible. Damping devices can be introduced into the system.

8.9.6 **Thermo-Plastic Hoses** - Thermo-plastic hose should not be used for water jetting unless specifically designed and fit for this purpose.

8.9.7 **Operator Positioning** - While operating the team members shall be safely positioned and if any person should encroach into the working area jetting shall be stopped.

8.9.8 **Work Stoppage** - Work shall stop:

(a) In the event that leaks or damage become apparent.

(b) If any person becomes aware of any change in conditions or any hazards being introduced or existing.

(c) If plant or work alarms are sounded.

(d) Any of the recommended practices in this document are not being followed.

8.9.9 **Hose Protection** - All hoses should be protected from being run over and crushed by vehicles, fork lift trucks, etc.

9.0 THE USE OF LANCES AND NOZZLES

9.1 **Lances** - Lances which are rigid or semi-rigid having nozzles fitted to them with any combination of forward backward or 90° angle jets shall be used with either a dump system or dry shut off control valve. When a flexible lance or nozzle mounted on a hose is in use, the jet should not be operated at pressure unless the nozzle is properly positioned inside the workpiece or the operator is protected by screens or proper shielding from the rear-facing jets. If necessary, the lead-in to the workpiece should be cleaned by other methods.

9.2 **Flexible Lances -** Flexible lances used to clean pipes where the inside diameter of the pipe is not small enough to prevent the lance from turning back on itself, shall have a piece of rigid straight tube, slightly longer than the diameter of the pipe fitted immediately behind the nozzle to prevent this happening.

9.3 **Distance Indicator -** When an assembly is used which allows the nozzle to enter the work piece with restricted visibility, the lance, hose or floor should be clearly marked in a manner which enables the operator to judge how far the nozzle is in the workpiece before pressure is applied and, conversely, so that pressure is released before the apparatus is completely withdrawn from the workpiece.

9.4 **Lance Length** - The length of a rigid lance or combination of lances shall be such that the operator can maintain control at all times.

9.5 **Jet Pressure** - The nozzle and minimum operating pressure shall be selected by the operators to allow effective and efficient jetting.

9.6 **Improper Use -** Should an operator enter a manhole or access port for any purpose (preferably with the jetting machine turned off) the hose shall not be used to support his weight when climbing up or down.

9.7 **"T" Pieces** - When using a "T" piece or nozzle carrier "T" (which are devices for producing two equal and opposite jets at the end of the lance and at right angles to the normal flow) it should be inserted into a tube, or vessel, or between two surfaces before the system is pressurized. This is necessary to ensure that should one jet be larger than the other, or one jet become blocked or partially blocked, the operator of the lance will not be spun out of control. When a NT" piece is used to provide a balancing jet on a long lance to clean a single surface it is not always possible to check for equal thrust from both jets in the above described manner, therefore, these lances should be checked by progressive pressure increases. This restraint shall also apply to any form of multi-jet nozzle, the jets issuing from which have a radial component.

9.8 **Confined Working -** Before entry into a confined space for jetting, a certificate of clearance shall be obtained to ensure that access is safe.

10. OPERATIONAL AND TRAINING REQUIREMENTS

10.1 **Qualified Operators** - Only trained personnel shall operate high pressure water jetting equipment, and supervise the training of new operators.

10.2 **Training** - A personnel training program shall be developed by each employer and be presented to each employee before assignment to the employee's first high pressure cleaning job. Such training shall include, as a minimum, coverage of all items listed inthese recommended practices.

10.3 **Cutting Action** - The cutting action of a high pressure water jet and the potential hazard it poses to the human body shall be demonstrated through the use of audio/visual aids or actual use of equipment (by cutting through a piece of lumber, a concrete block, etc.).

10.4 **Personal Protective Equipment** - The minimum personal protective equipment shall be explained. Instructions shall be given as to when and how specific clothing and other types of protective devices shall be worn according to the type of work performed, locations, etc.

10.5 **System Operation** - The operation of the system shall be explained pointing out potential problems and proper corrective action.

10.6 **Control Devices** - The operation of all control devices shall be explained. The importance of not tampering with any control devices as well as the importance of keeping them in proper working order shall be stressed.

10.6.1 **Equipment Maintenance -** It should be pointed out that values and seating surfaces in pressure regulating devices encounter high wear during high pressure water jetting. These items require frequent inspections, maintenance, and/or replacements in order to provide proper operation.

10.7 **Hose** - The proper method of connecting hoses including laying out without kinks, protection from excessive wear, and proper tools to use on couplings and fittings shall be explained.

10.8 **Stance -** The proper stance for sound footing and how to use the various devices for lancing, shotgunning, and molting shall be demonstrated. The trainee, under close supervision, shall use the various devices while the unit is slowly pressurized.

10.9 **Proficiency** - Personnel shall demonstrate knowledge and skill in the proper operation of equipment through practical application.

10.10 **General**

10.10.1 System shall be de-pressurized when

(a) Not in use.

(b) An unauthorized or inadequately protected person enters the barricaded area.

(c) Replacement or repairs are made to the system.

(d) Any recommended practices are violated.

10.11 **Refresher Training -** Operator retraining shall be on an annual basis or more frequent, if needed.

11. PERMANENT CLEANING AREAS

11.1 **Enclosure** - The areas shall be suitably enclosed and warning notices prominently displayed at the access points and perimeters.

11.2 **Access -** Access by persons other than the jetting team shall be strictly prohibited whilst work is in progress. If any unauthorized entry is made, all work shall cease immediately.

11.3 **Hazards** - The working area shall be free from hazards likely to trip personnel and be provided with adequate drainage and lighting facilities.

12. FREEZE PRECAUTIONS

12.1 **Freeze Precautions** - During periods where there is a risk of freezing follow manufacturers' recommendations or take the following precautions, on shutting down.

12.1.1 Remove gun or nozzle from delivery hose.

12.1.2 Pump water from supply tank until level of water is just above the filter.

12.1.3 Add recommended quantity of anti-freeze into water tank.

12.1.4 Place delivery hose into water tank and secure.

12.1.5 Run the pump until the anti-freeze works through the system.

12.1.6 Move selector level to dump or recycle position until the anti-freeze shows in the water tank.

12.1.7 If no supply tank is fitted, follow manufacturer's recommendations.

WARNING: IF A PUMP OR HOSE APPEARS FROZEN, ON NO ACCOUNT MUST THE PUMP BE ENGAGED OR THE ENGINE STARTED IF THERE IS A DIRECT DRIVE TO THE PUMP UNTIL THE SYSTEM HAS BEEN THAWED OUT AND LOW PRESSURE WATER HAS BEEN ALLOWED TO FLOW THROUGH THE SYSTEX TO THE NOZZLE END OF THE LANCE, THE LANCE HAVING BEEN REMOVED.

13. ACCIDENTS

13.1 **Personal Injuries -** In the event that a person is injured by the impact of a water jet, the injury caused may appear insignificant and give little indication of the extent of the injury beneath the skin and the damage to deeper tissues. Large quantities of water may have punctured the skin, flesh, and organs through a very small hole that may not even bleed.

13.2 **Operator Identification -** Immediate hospital attention is required and medical staff must be informed of the cause of the injury. To ensure that this is not overlooked, all operators engaged on jetting should carry am immediately accessible waterproof card which outlines the possible nature of the injury and bears the following text "This man has been involved with high pressure water jetting at pressures up to 14,500 lb./in^2 (100 MPa, 1000 bar, 1019 kg/cm^2) with a jet velocity of 900 miles (1440 km) per hour. Please take this into account when making your diagnosis. Unusual infections with micro-aerophilic organisms occurring at lower temperatures have been reported. These may be gram negative pathogens such as are found in sewage. Bacterial swabs and blood cultures may therefore be helpful."

13.2.1 **Medical Recommendations -** If an accident should occur and high pressure water penetrates the skin, the National Poison Center Office, telephone 81/412/681-6669 may be contacted for best medical measures.

13.3 **Immediate First Aid -** Where medical examination is not immediately possible in remote situations, first aid measures should be confined to dressing the wound and observing the patient closely until medical examination has been arranged.

13.4 **Reporting -** If any person or equipment is accidentally struck by the jet, this fact must be reported to a responsible party.

14. RESPONSIBILITY

14.1 **Purpose -**These recommended practices are provided to assist persons unfamiliar with the operation of water jetting equipment in learning to correctly use the equipment.

The responsibility of correct operation and use of the equipment is the sole responsibility of the operator. The operator should familiarize himself with the identification of high pressure metal fittings, hoses, guns, and accessories. The modification of water jetting equipment or accessories is not recommended without prior written approval by the manufacturer of the equipment.

Serious harm or injury may result from the misuse of water jetting equipment, the use of improper fittings, hoses, or improper attachments. The Water Jetting Association does not accept liability for the use of water jetting equipment by the provision of these recommended practices or warrant that the techniques expressed or implied herein are correct or will prevent harm or injury.

SUCCESSFUL APPLICATION OF AN ABRASIVE WATER JET T0 CUT CONCRETE PAVEMENT UNDER ACTUAL FIELD CONDITIONS

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ABSTRACT

Flow Industries, Inc. with funding from the Gas Research Institute has designed a mobile, high-pressure water jet system for cutting pavement. The system is designed to cut concrete faster, cleaner and be less expensive to operate than conventional cutting systems. The concept is to mix a high velocity stream of water with abrasive particles. Cutting is achieved by a combination of micro-machining mechanism and particle erosion. A cutting rate of 7 inches/minute for a surface consisting of 6 inches of concrete and 2 inches of asphalt is the design objective. The prototype unit includes a 120 hp diesel engine, 3 ultra-high pressure intensifiers plumbed in parallel, water and abrasive storage tanks, automatic controls, and a circular or linear traversing system that includes the abrasive water jet cutting nozzle. In 1984, limited field tests were conducted in cooperation with three gas utilities on the west coast of the United States. The prototype unit successfully met the design objective in many of the tests. The total length of the holes cut at the utilities was 590 feet and the area of the holes was 351 ft. The results of the field tests indicate that the abrasive waterjet was able to consistently cut pavement with repeatable results.

INTRODUCTION

A major time, cost and safety factor in the repair of gas distribution piping beneath roadways is the surface cutting method applied. Conventional equipment used for cutting or breaking pavements include jack hammers, diamond saws, impact breakers, and large cutting wheels. However, these techniques can be labor-intense, expensive, slow and may be unable to cut through all types of thickness’ of pavement surfaces. In addition, these methods often produce a poor quality cut that may complicate and increase the cost of the surface restoration process.

In 1981, Gas Research Institute (GRI) initiated a program at Flow Industries, Inc. (Flow) to develop and demonstrate a water jet pavement cutter for gas utility applications. The result of this program was the development of a mobile pavement cutting system that uses a high-velocity water jet with entrained abrasive particles. The system is designed to cut concrete faster, cleaner and be less expensive to operate than conventional systems. The performance objectives for the abrasive water jet cutting system are:

- Cutting Rate: 7 inches/minute for a pavement consisting of 6 inches concrete and 2 inches of asphalt.

- Hole Size: 1) 3-ft. diameter holes with circular cutting system.
  2) rectangular or irregularly-shaped holes with linear cutter.
-
- Waterjet Parameters: 35,000 psi water at a flow rate of 3 gal1ons/minute.

- Operating Temperatures: 20° to 100° F.

- System Design: Self-contained, highly maneuverable in streets, easy to operate and maintain.

These performance objectives were established by GRI and its gas industry advisors with input from contractors and Flow.

In 1984, limited field experiments were conducted in cooperation with three gas utilities on the west coast of the United States. The results of the field experiments indicate that the abrasive water jet was able to cut pavement consistently with repeatable performance. The performance objectives were met in many of the tests.

RESEARCH PROGRAM

A. TECHNICAL APPROACH

Abrasive-Water Jet Cutting

Flow proposed to use ultra-high pressure abrasive-water jet technology to meet the design goals of the program. Abrasive-waterjet cutting is a micro-machining cutting process where a small diameter ultra-high velocity water jet is used to accelerate abrasive particles which perform cutting. The high velocity water jet is formed by pumping water to high pressure (30,000 psi to 35,000 psi) through a small diameter (0.018 to 0.032 inches in diameter) nozzle. The high velocity jet is discharged through a chamber where abrasive particles are added. In the chamber and the subsequent discharge tube, the abrasive particles are accelerated by the high velocity jet through impact and momentum transfer. The jet and the abrasive particles exit the tube in a high velocity composite jet to perform cutting. A representation of this process is shown in Figure 1.

Abrasive-Water Jet Cutting Tests

Flow performed a series of cutting tests to optimize the abrasive-water jet cutting performance for the asphalt and concrete composite material. When acceptable cutting performance was reached, the system operating parameters and ranges were identified. These parameters were used to establish the system components. Parameters tested included:

- Water jet pressure.
- Water jet diameter.
- Abrasive particle size
- Abrasive type.
- Abrasive flow rate.
- Nozzle traverse rate.
- Number of cutting passes.

Individual parameters were varied to determine the effect on cutting performance. The results were then plotted and analyzed to optimize cutting performance for the 8-inch composite roadway surface. The major emphasis was placed on minimizing the horsepower required to meet the cutting performance goals, to reduce system size, and increase maneuverability.

Cutting depths in concrete vary greatly due to the non-homogeneous nature of the material. Concrete is a composite material consisting of varying size and shape aggregates, grout binder and sometimes steel reinforcing bar. Each of these materials will have different optimum cutting rates and parameters which makes selection of a single set of optimum cutting parameters difficult. Flow performed all cutting tests on a special highway grade (5000 psi) concrete slab. The slab had closely controlled aggregate size and was aged for six (6) months prior to the beginning of the tests. The slab was poured in two lifts of 6 inches and 2 inches to approximate the composite roadway surface. In some cases this material was more difficult to cut than composite roadway surfaces encountered in the field tests.

System Design Parameters

When the parametric tests were completed and the data was analyzed, the system design and operational parameters were selected. The system design parameters which proved sufficient to meet the program performance goals are listed in Table 1.

Table 1. System Design Parameters

Parameter	Value
Pressure	35.000 psi
Flow Rate	3.3 gpm
Waterjet Diameter	0.030 inches
Standoff Distance	0.5 inches
Abrasive Material	Garnet
Abrasive Size	36 grit
Abrasive Flow Rate	5.8 lb/min.

B. CURRENT CUTTING SYSTEM

Cutting System Overview

Flow designed and fabricated a self-contained trailer mounted cutting system, as shown in Figure 2. The system consists of three (3) major components:

- High pressure water power unit.
- Abrasive water jet cutter.
- Abrasive metering and storage tank.

The high pressure water power unit contains a 120 hp water-cooled diesel engine which serves as the sole power source for the cutting system. The diesel engine drives a hydraulic pump to power the three (3) Flow water intensifier pumps. The diesel engine also supplies electrical power for the cutting system auxiliary hydraulic power for the abrasive feed valve. The power unit had capabilities for control by a remote pendant to allow on/off and pressure control.

A portable cutter (traversing mechanism) was designed to perform linear cuts of 60 inch length as shown in Figure 3. Longer cuts were achieved by indexing the cutter to begin a new cut at the end of the previous cut. This design gave maximum flexibility for a wide range of field operations with the different utilities. The cutter weighs about 140 lb and can be set at speeds from 1 ipm to 18 ipm. Speeds of 100 ipm can be achieved with a simple gear change. A remote control is used to turn the portable cutter on-off and for directional changes.

The abrasive metering and storage tank is connected to the power unit for automatic on/off control. The tank has capacity for 200 lbs of abrasive material and was stored in the tow vehicle during the tests. The tank can be operated up to 300 feet from the cutter. A small diameter plastic hose transports the abrasive material to the cutter.

Cutting System Specifications

The system operating specifications are listed in Table 2.

Table 2 System Operating Specifications

Power Unit	
Horsepower operating	125 hp
Water pressure	35,000 psi
Water flow rate	3.5 gpm
Length	13’-8”
Width	71 incs.
Height	96 ins.
Weight dry	5,500 lb.
Water capacity	120 gal.
Fuel Capacity	40 gal.
Auxiliary power	
Electrical power	.5 kw.
Abrasive Cutter	
Length	60 ins.
Width	18 in
Height	18 in.
Traverse speed	1-100 ipm
Weight	140 lb.
Abrasive Storage and Metering System	
Capacity	200 lb.
Flow rate	0-10 lb/min.

System Operation

The trailer mounted cutting system was operated as a production cutting unit during the field tests. The utilities had the desired access holes clearly identified in advance. The Flow cutting team, consisting of a technician and an engineer, arrived at the test site, cut the access hole, c1eaned-up and moved to the next hole.

The cutting unit was designed for single lane operation; therefore, the drive could position the vehicle directly in front of the cut. The tow vehicle contained the abrasive feed and storage tank and the abrasive cutter. Other items stored in the tow vehicle included spare parts, traffic cones and a work bench.

When the vehicle was stopped the cutter was unloaded and positioned on the cut. Three connections were made to the abrasive cutter: 1) high pressure water for the cutter nozzle, 2) electrical power for the traverse mechanism, and 3) the abrasive hose for the cutting nozzle. One additional connection was made from the power unit to the abrasive feed hopper for actuating the abrasive on/off valve. The set-up time varied from 10-30 minutes depending on the crew and conditions. The diesel engine was started and cutting commenced when setup was completed.

Determination of depth of cut was made by one of two methods: 1) direct measurement or 2) visual assessment of the cutting water runoff. Direct measurement was done with a stiff wire inserted into the cut. When the abrasive water jet cuts through pavement into the subbase, the cut will be quite deep and have a soft base. "Through-cutting" the pavement also results in a dirty brown water runoff, whereas non-through-cutting results in a white runoff. Cutter speed was adjusted to a maximum value while maintaining through-cutting. The cutter was repositioned after each cut.

When the cut was completed the vehicle was moved to a new hole. Water replenishment was achieved by fire hydrant when several cuts were made in the same location. A11 sites had hydrants within 250 ft. A hose reel in the truck with pre-fit adapters was used for the hydrant to power unit replenishment.

C. FIELD TEST RESULTS

The abrasive water jet pavement cutter was field tested for one week at each of three west-coast gas utilities--Washington Natural Gas Company (WNG) in Seattle, Washington; Northwest Natural Gas Company (NWNG) in Portland, Oregon; and Pacific Gas and Electric Company (PG&E) in San Francisco, California. During the three weeks of testing, a total of 44 pavement openings were cut that ranged in size from 2 feet by 3 feet up to 7 feet by 8 feet. A wide variety of pavement types were encountered with depths ranging up to 15 inches. Over 23 hours of operation were logged with an approximate down time of 11%.

Cutting Performance

Cutting performance by pavement type is given separately for field tests with WNG, NWNG, and PG&E in Tables 3, 4 and 5, respectively. As shown in Tables 3 and 5, cutting performance in a composite pavement of 2-inch asphalt over 6-inch concrete ranged from 6 to 8 inches per minute (ipm). This met the performance goal of 7 ipm for this type of pavement.

Table 3. Cutting Performance by Pavement Type – WNG.

Pavement Description	Depth of Cut (in)	Number of Passes	Equivalent Cut Rate (ipm)
2 in. asphalt over 4 in. concrete.	thru	1 @ 8 ipm	8
2 in. asphalt over 6-7 in. concrete	7-thru	3 @ 18 ipm	6
6 in. concrete	thru	2 @ 12 ipm	6
6 in. concrete	1.25 – 1.75	1 @ 48 ipm	48

Table 4. Cutting Performance by Pavement Type – NWNG.

Pavement Description	Depth of Cut (in)	Number of Passes	Equivalent Cut Rate (ipm)
7.5 in. asphalt over 4.5 in. red brick over 8 in. concrete	14-15	3 @ 8 ipm	2.7
4 in. asphalt over 8 in. ATB (Asphalt treated base).	10.25-thru	2 @ 8 ipm	4
4 in. asphalt over 4 in. concrete	6.25 - thru	1 @ 6 ipm	6
4 in. ashpalt over 6 in. concrete	8.75 - thru	2 @ 8 ipm	4
6 in. concrete	5.25 – thru	2 @ 12 ipm	6

Table 5. Cutting Performance by Pavement Type – PG&E.

Pavement Description	Depth of Cut (in)	Number of Passes	Equivalent Cut Rate (ipm)
6 – 7 in. concrete	thru	1 @ 8 ipm	8
2 in. asphalt over 6 in. concrete	6.5 -thru	1 @ 8 ipm	8
2 in. asphalt over 8 in. concrete	thru	2 @ 8 ipm	4
6 in. asphalt	thru	1 @ 18 ipm	18
6 – 8 in. asphalt	7.5 – thru	1 @ 12 ipm	12

Plug Removal and Replacement

In ongoing field tests with WNG, the concept of one-piece pavement 'plug' removal and replacement was successfully demonstrated. Usually pavement is broken out with an impact breaker (after the perimeter of the hole is cut with the abrasive-waterjet pavement cutter). While the perimeter cut makes this breakout procedure easier, substantial time savings can be realized by removing the pavement 'plug' intact. The savings are even greater if the same plug can be replaced in the pavement opening. The plug removal is accomplished by installing anchors in the plug and lifting it out with a backhoe, see Figure 4. After performing the underground utility repair, the plus is reinstalled and grouted in place. This procedure has been demonstrated for both rectangular and circular holes. In many applications, plug removal and replacement has the potential for significant cost savings for the utility industry.

Utility Comments

In general, the pavement cutting system was well received by the gas utilities participating in the field tests. The following are some of their comments and recommendations.

The utilities were concerned about the disposition of the water and spoils after the cut. During the field tests, the spoils were washed down the nearest storm drain. Although the material is inert, this practice may contribute to clogging of the drains. To remedy this situation, an integral vacuum system was developed that collects the bulk of the water and spoils during the cutting operation. This system is currently being tested.

Utilities found the noise level of the system acceptable. Measurements indicated noise levels of approximately 100 dBa at a distance of 10 feet from both the diesel engine and abrasive-waterjet nozzle.

The utilities also recommended that the equipment be downsized for greater maneuverability. This will be accomplished when the commercial prototype is developed.

BENEFITS T0 USER

Commercialization of the abrasive water jet pavement cutting system may have a significant impact on the construction industry in general, and could benefit the construction and maintenance operations of utilities. The following are potential benefits when using the system:

1) Reduced cost of operations attributable to increased speed of pavement cutting and removal; simplified tool inventory; and lower costs for pavement restoration because of the clean edge of the pavement cut.

2) Improved public relations through reduced traffic interruption time; lower levels of noise shock and vibration from the abrasive water jet when compared to pneumatically-operated jackhammers and other pavement cutting equipment.

3) Reduced operational hazards to equipment operators as result of the lower level of noise and shock associated with using the abrasive water jet cutter.

A cost-benefit analysis was prepared by Flow to make a realistic assessment of the economic benefit of the abrasive water jet pavement cutter to gas utilities. A comparison was made between the current cost of pavement cutting for the three gas utilities who participated in the field experiments and the estimated cost of a production abrasive water Jet cutter. The average hourly cost and average cost per hole to cut a composite pavement of asphalt over concrete for the three gas utilities on the west coast are $36.11 per hour and $42.47 per hole (2 ft. wide by 4 ft. long). In comparison, the operating costs for the abrasive water jet pavement cutter are $78 per hour or $30.20 per hole (2 ft. wide by 4 ft. long). Average savings were calculated to be $12.27 per hole, representing a 29% cost savings. Figure 5 shows graphically the pavement cutting costs for the three gas utilities and the abrasive water jet cutter. This analysis indicates an advantage, on a per hole basis, for using an abrasive water jet over conventional techniques. In addition, the projected benefits or savings resulting from improved operational safety, and the decrease in restoration costs were not included in the analysis.

FUTURE PLANS AND PROSPECTUS

Currently, GRI is sponsoring additional research at Flow to field test two pre-production abrasive water jet systems in cooperation with ten gas utilities. In addition to the trailer mounted power unit used during the field experiments, a skid-mounted power unit on a flatbed truck was for use in the field tests. The purpose of the field tests is to: 1) establish system performance and operating costs for a wide variety of work environments; 2) assess system reliability and repeatability of performance; and 3) determine system design improvements and operational procedures for optimal performance. Each pre-production unit wi11 be tested at five utilities. Upon completing the tests, production rates and costs for the abrasive water jet system will be determined and compared to conventional pavement cutting techniques used by the utilities.

If successful, the end result of the field tests will be an abrasive water jet system ready for commercial introduction. Flow plans to introduce the pavement cutter beginning in early 1986.

ACKNOWLEDGEMENTS

The authors wish to thank Northwest Natural Gas Company, Pacific Gas and Electric Company, and Washington Natural Gas Company for their time and effort in making the field tests a success.

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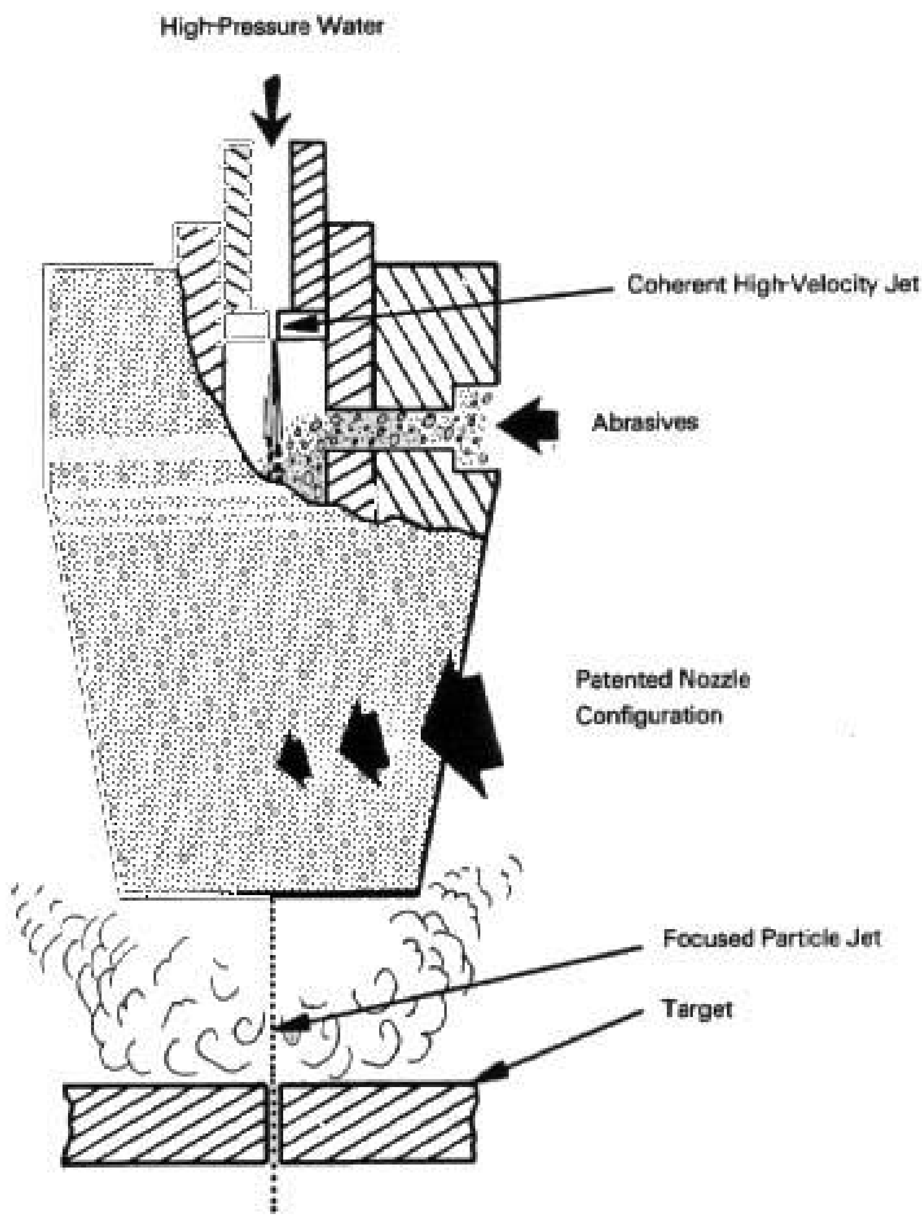

Figure 1. Secondary Abrasive Injection into High Velocity Waterjet

Figure 2. Abrasive Waterjet Cutting System

Figure 3. Portable Abrasive Waterjet Cutter

Figure 4. Pavement Plug Removal

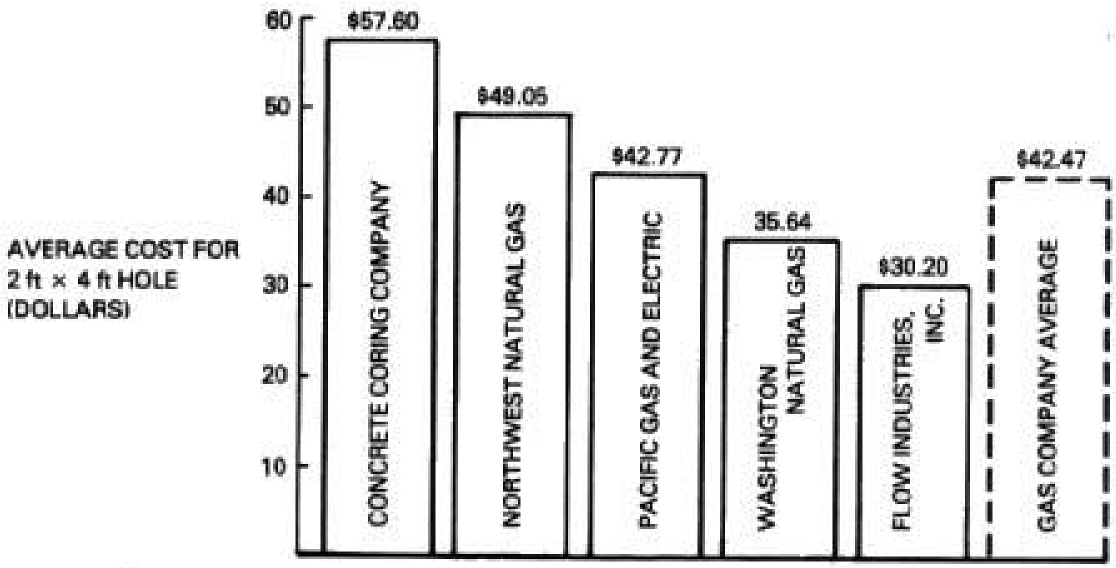

Figure 5. Average Cost Comparison for Cutting 2 ft x 4 ft Access Hole Through 8 in. Roadway Surface

AN ABRASIVE JET DEVICE FOR CUTTING DEEP KERFS IN HARD ROCK

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ABSTRACT

This paper proposes a method for cutting deep slots in hard rock with an abrasive water jet. The method uses a collimator that enables a high velocity abrasive jet to retain its cutting ability for distances up to 4 ft from the nozzle. This collimation is achieved by affixing a pipe downstream of the abrasive mixing chamber. The cutting jet retains its cutting ability while inside the pipe so that a kerf is cut in hard rock by the jet as it emerges from the pipe. The ability to cut 4 ft downstream from the nozzle allows the cutter head to follow into the kerf and thereby to extend the depth of cut. This paper describes the design of a collimated abrasive jet kerf cutter and presents rock cutting data.

INTRODUCTION

The Bureau of Mines is conducting research in the area of waterjet technology for selective mining of ore from surrounding waste rock in underground mines. Many underground metal mines unnecessarily remove large volumes of waste rock along with the metal values when the ore occurs in thin beds or veins, the thickness of which is smaller than the smallest practical size of headings. The resulting dilution of the ore values is undesirable because of increased costs and waste disposal requirements.

An effective means of selectively mining or separating ore from waste during the mining cycle would be beneficial. Attempts have been made to achieve selective mining using mechanical cutters. These approaches have met with little success because of problems with wear while cutting deep kerfs in hard rock.

The Bureau has developed a kerf cutter for hard rock that uses a high-velocity abrasive water jet. The device differs from other abrasive jet cutters in its ability to follow into a cut and thereby cut deep (4-ft) kerfs in hard rock. Other abrasive jet rock cutters are limited to cutting in a single plane.

Some high-velocity nonabrasive water jets are capable of following into a cut because they are generated in a rotating dual orifice cutting head. However, rotary couplers (swivels) are not generally appropriate for high velocity abrasive slurries because of wear. Kerf cutting with a rotary water jet is severely limited regarding the hardness of the rock cut and requires super-high pressure fittings, hoses, and pumps.

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Abrasive jet kerfing permits kerf cutting in the hardest rocks at moderate pressures (10,000 psi), for which hoses, fittings, and pumps are readily available. This paper outlines a method for cutting deep kerfs in hard rock without rotating the cutting nozzle. Using a collimated abrasive water jet, kerfs are produced by multiple passes with the cutting head following into the kerf. The phenomenon of mechanical collimation and the description of the collimators represent new technology and are the most important parts of the research described.

EQUIPMENT

The abrasive jet kerf cutter consists of a pump, an abrasive entrainment assembly, a collimator, and a traversing system mounted on a drill carrier. Pressurized water from the pump is conducted through flexible high pressure hose to a water nozzle in the abrasive entrainment device. The nozzle, when incorporated in the internal geometry of the entrainment assembly, creates a high velocity jet that causes a suction by the venturi effect. This suction causes an air-and-sand mixture to flow from a sand pick-up probe through a hose into the abrasive entrainment chamber. Here the sand and air are entrained into the flow, and the resultant slurry travels at speeds of 600 to 1,000 ft/sec through a pipe collimator. The length of this pipe varies depending on the depth of the kerf being cut.

The abrasive jet cutter requires a pump capable of generating a flow of 21 gpm at 10,000 psi. The pump used was driven by an electric motor and thus is inconvenient for field operations where access to electric power is limited. Such inconvenience can be avoided by the use of a diesel driven pump such as that used in the quarry cutting test described in the field test section.

Twenty feet of 10,000 psi working pressure hose connects the pump to the water nozzle. Hose flexibility permits the abrasive jet cutter to be traversed across the rock face without restriction.

The abrasive entrainment assembly (fig. 1) is a modified tee-section. Water flows linearly from a single 0.081-in nozzle through the mixing chamber and out the collimating pipe. Sand and air, drawn into the mixing chamber through the tee, are entrained and flow out the collimating pipe as a sand, air, and water slurry.

A collimator, consisting of schedule 80 steel pipe (0.55-in ID), is inserted into the outlet of the abrasive entrainment assembly. The length of this pipe collimator was varied from 1 to 4 ft depending on the depth of the kerf being cut.

The abrasive jet slurry wears the interior walls of the collimating pipe. These pipes commonly wear through after approximately 30 min of use. However, pipe life is extended if the pipe is rotated at intervals during use to radially distribute the wear. These pipes are considered expendable because they cost about $0.50/ft.

The abrasive entrainment assembly and collimator are traversed across the rock face on a parallel slide mechanism driven by a chain and sprocket arrangement and confined by microswitches at the ends of the traverse.

The abrasive entrainment device, collimator, and traversing system ensemble is mounted on the arm of a drill carrier.

ENTRAINMENT

Dry sand from a 55-gal drum is drawn with air through a pick-up probe inserted into the sand. The pick-up probe is an assembly of two concentric pipes open at the bottom and connected by a hose at the top to a slurry entrainment assembly. The outer pipewall is perforated near the top of the assembly and the inner pipe is perforated near the bottom. The venturi effect caused by the flow of a 10,000-psi, 21 gpm jet downstream of a 0.081-in diameter nozzle in the slurry entrainment assembly causes air to enter the pick--up probe through the top perforations.

The air flows down through the annular space between the pipes and enters the base of the inner pipe through the perforations. The air mixes with sand, and the resultant mixture is drawn up the pipe, through 20 ft of 0.75-in ID hose and into the slurry entrainment assembly.

It is necessary to mix dry sand with air to avoid clogging in the transfer hose.

Sand is entrained into the water jet at the rate of 18 lb/min. Figure 2 is a high-speed photograph through a transparent tee connector showing the entrainment of sand into a 10,000 psi, 21 gpm water jet. Note that some of the sand is drawn upstream toward the nozzle. This backward flow of sand is probably the result of the occurrence of a low-pressure region adjacent to the nozzle through which no water flows. The figure illustrates that a transfer of momentum occurs between the water jet and the sand particles. Sand-particle velocities of 1,000 ft/sec -have been measured downstream of the entrainment assembly.

The mixing chamber has the configuration shown in figure 1. Note that it is a simple arrangement providing a port for the nozzle, an abrasive inlet port, and a slurry exit port into which a collimator is fitted. The mixing chamber is made of stainless steel and measures 3 x 2 x 1.75 inches.

COLLIMATION

A jet constrained to flow in a pipe will not dissipate and will retain its cutting ability significantly beyond the distance at which- jets passing through air are effective. The jet is constrained to flow linearly, i.e. it is collimated. This linear flow enables the loss of momentum of the jet to be minimized, enabling the abrasive jet to cut deep kerfs as far as 4 ft from the nozzle.

The water from the pump (21 gpm at 10,000 psi) enters the pipe collimator after having been accelerated to 1,250 ft/sec through a 0.081-in diameter nozzle and having entrained sand and air. Sand particles are accelerated by the jet in the abrasive entrainment chamber and retain most of their kinetic energy as they pass through the collimator. Sand particle velocities as high as 1,000 ft/sec have been measured at the outlet of the collimator pipe by double exposure high-speed photographs, such as figure 3. Two exposures were taken at 10μ sec intervals, resulting in a double exposure picture from which the distance traveled by a particle of abrasive in 10μ sec can be determined.

The abrasive particles are randomly distributed in the jet cross section. This distribution was ascertained by evaluating the pattern of impact craters left on polished steel plates (fig. 4) by abrasive jets containing a known number of sand grains. These plates were placed at similar standoff distances from collimating pipes of various lengths in order to determine the effect of collimator length and abrasive distribution. The study showed that the abrasive is well mixed in collimating pipes of 1 ft to 6 ft long.

Mining applications of abrasive jet cutting requires a large amount of abrasive, thus it is imperative that a low-cost abrasive be used. The commercial sand-blasting sand selected was low cost ($70 per ton), dry packaged, convenient in size, and effective. The size distribution and chemical composition is listed in Tables 1 and 2. This dry abrasive flowed readily in a suction hose connected from the sand pick-up probe to the abrasive injection port in the mixing chamber.

Tabl e 1. - Size distribution of abrasive.
(Source: Bureau of Mines, Twin Cities Research Center, Research Services Group, Analysis of Grab Sample, Dec., 1983)

Screen (mesh)	Wt. pct	Size (mm)
+ 20	32.2	+.83
-20/+28	46.3	+.59
-28/+35	17.8	+.42
-35/+65	3.7	+.21
-65	0	-.21

PARAMETRIC STUDY

The cutting ability of a collimated abrasive jet is dependent upon certain operating parameters. In order to elucidate this dependency for the 10,000 psi, 21-gpm abrasive jet, a parametric study was performed using cutting depth as the dependent variable. Table 3 shows the operating conditions and data for the effect of abrasive feed rate on the depth of cut for a single pass in Salem limestone (unconfined compressive strength of 6,367 psi, as given in reference 1). These data show that the cut depth levels off at feed rates in the vicinity of 18 lb/min, the maximum. At this feed, rate the entrained sand grains presumably interfere with each other's ability to be accelerated by the water jet to effective velocities for cutting.

Table 2. - Chemical composition of abrasive. (Source: Twin City Testing and Engineering Laboratory, Inc., Feb. 1978)

	Pct
Loss on ignition	0.97
Al_2O_3	4.48
Fe_20_3	2.45
CaO	1.91
MgO	1.29
$S0_3$	.09
Na_20	.04
K_20	.04
SiO_2	88.61
Other inert materials	.12
Sum:	100.00

TABLE 3. - Effect of abrasive feed rate on the depth of cut in Salem limestone

Feed rate (lb/min)	Depth of cut (in)
18.5	2.13
15.6	2.16
1l 7	1.95
6.3	1.63
0	.65

Table 4 and 5 contain the data and operating conditions showing the effect of varying the rate at which a jet is traversed across a block of Salem limestone (unconfined compressive strength 6,367 psi) on the depth of penetration of the jet in a single pass across the rock block. A graph of these data (fig. 5) shows that the penetration depth decreases as the traverse rate increases. This does not indicate, however, that low traverse rates are the most efficient mode for rock kerfing. When the volume removed per pass is divided by the energy of the jet, a relationship emerges (fig. 6) that indicates that the faster traverse rates are more efficient.

The effect of the length of the collimating pipe on depth of cut was investigated by fitting the abrasive jet cutting head with various lengths of collimating pipe. The depth into samples of Oneota dolomite (compressive strength 15,000 psi) cut by 10,000 psi, 21 gpm abrasive jet issuing from these collimators was measured. Similar tests were performed with a 4 ft collimator, a 3-ft collimator, a 2-ft collimator, and a l-ft collimator.

TABLE 4. - Effect of traverse rate on the efficiency of cutting Salem limestone

Traverse rate (in/sec)	Cut volume (in^3)	Cutting efficiency (in^3/Btu)
6.0	4.5	.0215
3.0	6.8	.0162
2.0	9.8	.0156
1.0	18.5	.0147
.5	32.8	.0130
25	55.0	.0109

TABLE 5. Operating conditions for tables 3 and 4.

Water jet nozzle:	
Rate (Q) gpm	21
Diameter (d) in	0.08
Power Btu/sec	104.8
Abrasive: Feed rate lb/min	18.4
Collimating pipe, in:	
Length	12
ID	0.5
Standoff	1.0

The effect of the length of the collimating pipe on the depth of jet penetration is shown in table 6. Note that the cutting depth is essentially invariable as a function of collimator pipe length for lengths up to 3 ft.

TABLE 6. - The depth of cut of an abrasive jet into the Oneota dolomite as a function of collimator pipe length.

Pipe length (ft)	Depth of cut (in)
4	2.5
3	3.25
2	3.25
1	3.25

DEEP KERF CUTTING

The ability to keep an abrasive jet collimated for several feet without significant loss of cutting capability can be used to cut deep kerfs in hard rock because the collimating pipe can be advanced into the cut to increase the depth of cut. Just as the cutting ability decreases very quickly with increasing nozzle-target standoff distance in uncollimated water jets, so the collimator pipe-target standoff distance should be kept to within 1 in to insure most effective cutting. Thus the collimating pipe should be advanced to within 1 in of the bottom of the kerf in order to deepen a kerf in hard rock.

It is not practical to rotate the collimating pipe because the high-speed abrasive would quickly destroy any swivel downstream of the nozzle. Standard methods of cutting clearance by rotating two diverging jets cannot be used for abrasive jet kerfing because the cutting head cannot be rotated. Thus deep kerfing with an abrasive jet must be accomplished using translating motions only. This is done by cutting two parallel kerfs and wedging out the rocks between (fig. 7). This technique was used to cut a rectangular hole through a 4 ft cube of Salem limestone (fig. 8).

FIELD TEST

A field test was conducted in cooperation with the Vetter Stone Co. at its Kasota, MN, quarry using an abrasive mixer-collimator mounted on a drill carrier (fig. 9). Cutting was performed on a bench of Oneota dolomite. The Oneota dolomite is a buff-colored, fine-to-medium grained dolomite of Lower Ordovician age. Cores taken from this ledge indicate that the unconfined compressive strength of the rock cut is 15,000 psi.

The main task of the field test was to remove a 12 x 30 x 33 in. block from the bench. This was done by cutting three vertical slots and wedging the block loose from the horizontal bedding plane (fig. 10). Each slot was cut the entire 2-1/2-ft depth of the bench by cutting two parallel kerfs, wedging out the rib between, and advancing the collimator into the slot (fig. 11). This process was repeated three to five times to obtain a 30-in cut.

CONCLUSION

The main point of this paper is to emphasize that an abrasive jet retains its rock cutting ability while confined in a pipe for distances up to 4 ft. It is this retention of cutting ability that permits kerfs as deep as 4 ft to be cut in dolomite, limestones, granites, and other hard rocks.

The concept of a collimated abrasive jet, i.e. a jet rendered parallel by being confined in a pipe, is introduced in this paper. Collimated jets can cut deep kerfs in hard rock because the collimating pipe can follow into a kerf for distances up to 4 ft. At present, the clearance in the kerf necessary for the pipe to follow into the kerf is maintained by cutting two parallel kerfs and removing the intervening web of rock. Research is continuing on the development of a method for the collimated jet to cut its own clearance in a single pass.

REFERENCE

Krech, W. W., F. A. Henderson, and K. E. Hjelmstad. A Standard Rock Suite for Rapid Excavation Research. BuMines RI 7865, 1974.

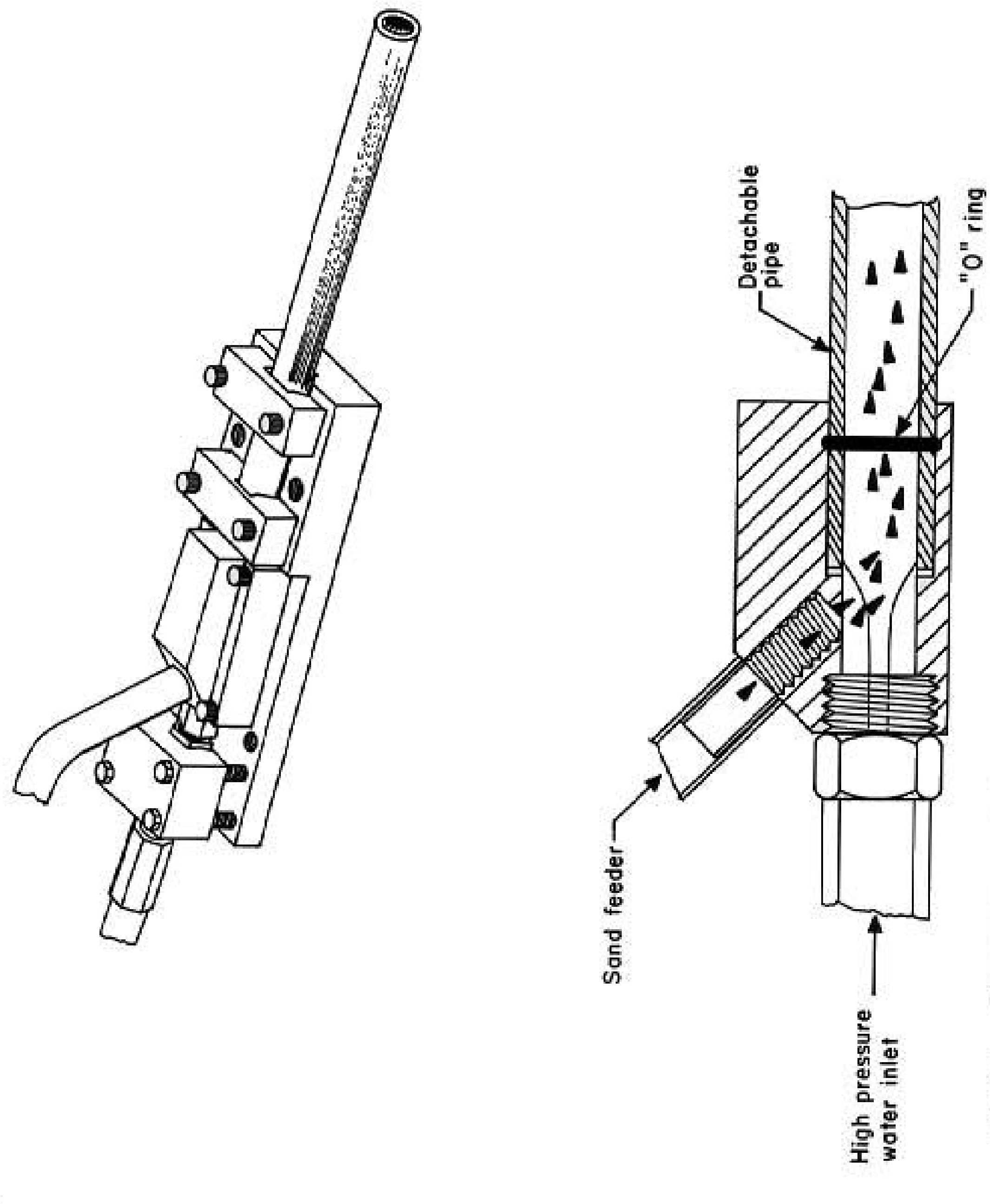

Figure 1. The abrasive entrainment assembly.

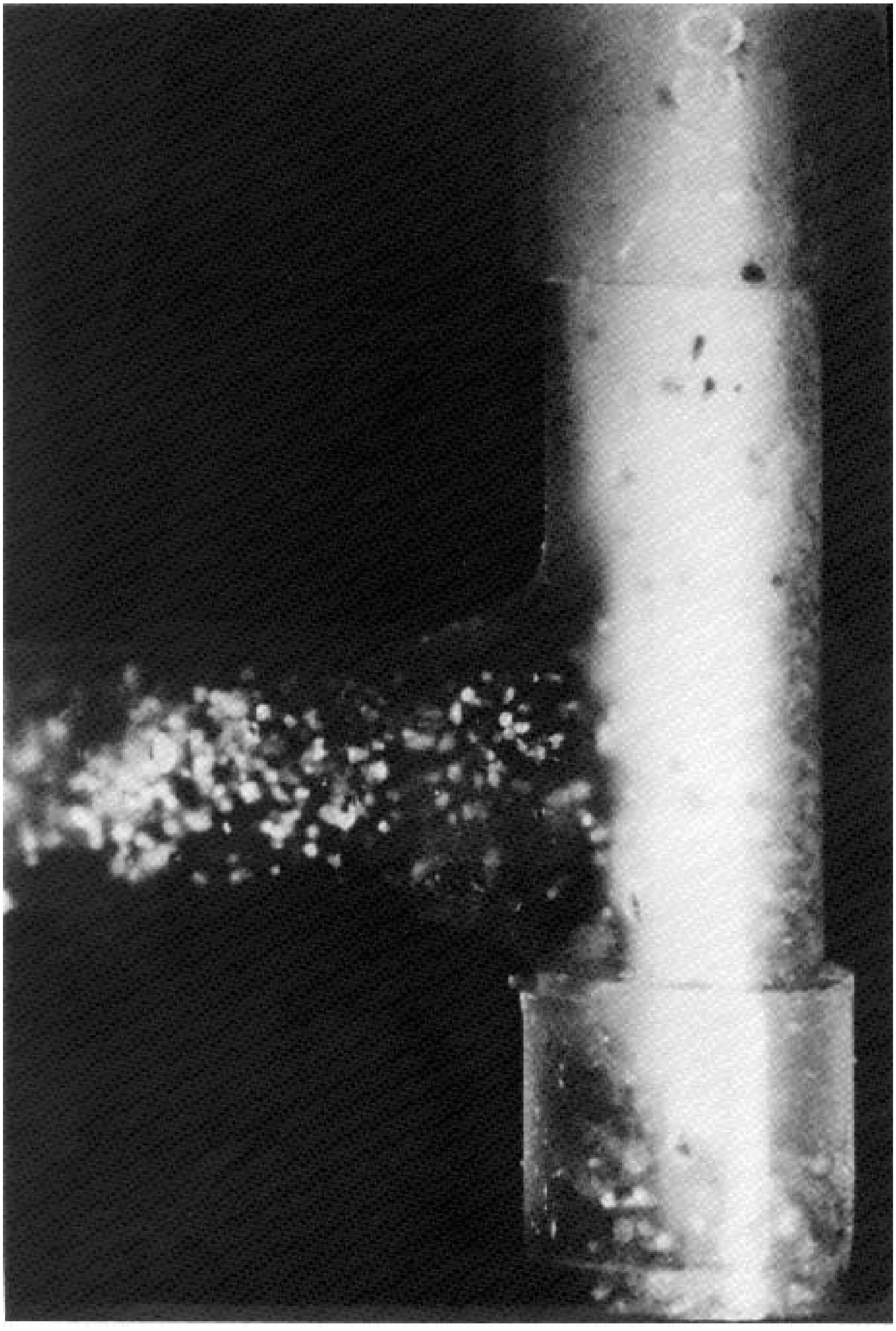

Figure 2. High speed photograph of sand being entrained into a 10,000 psi, 20 gpm jet.

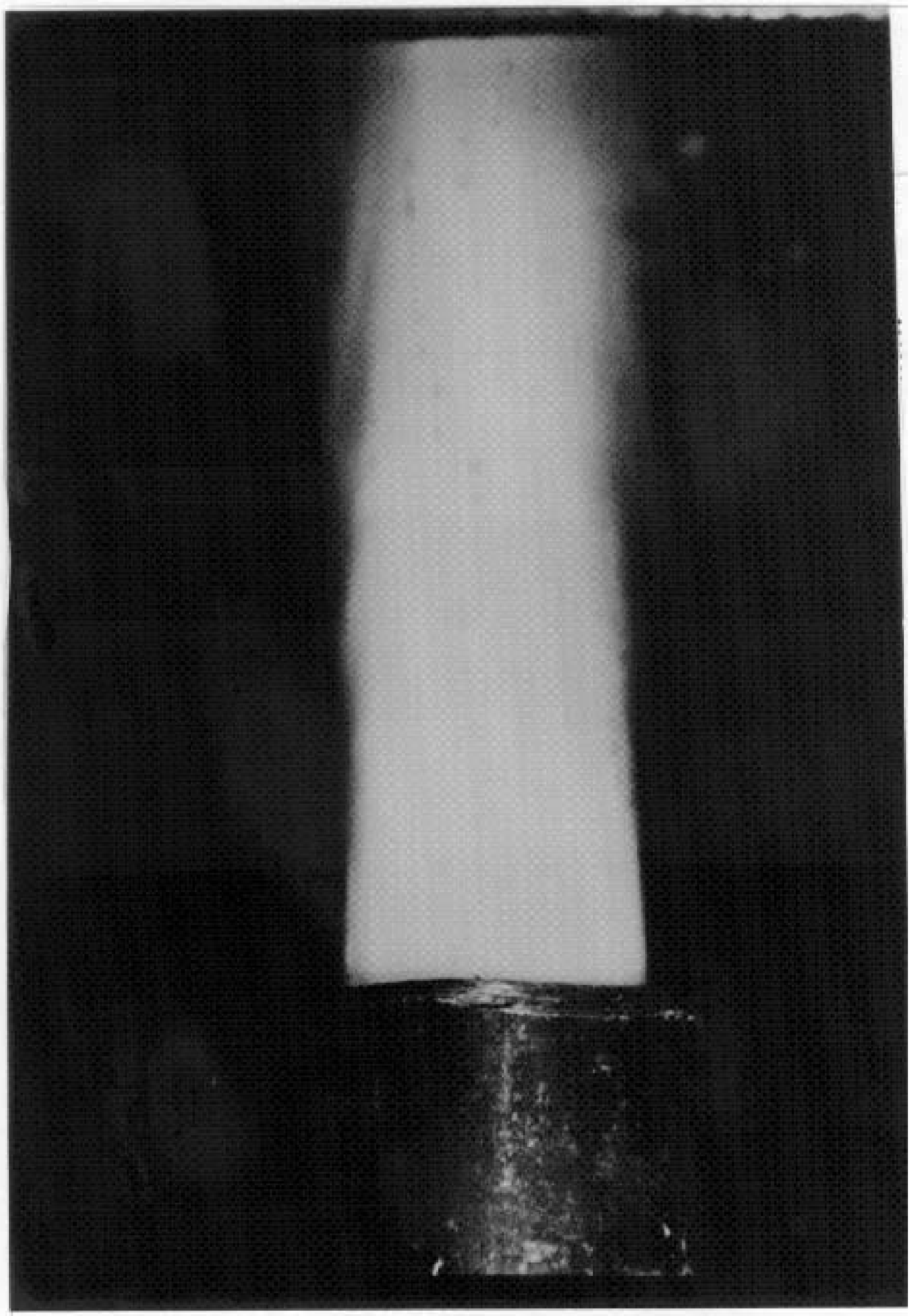

Figure 3. Double exposure of a high speed photograph of abrasive in a 10,000 psi, 20 gpm abrasive jet

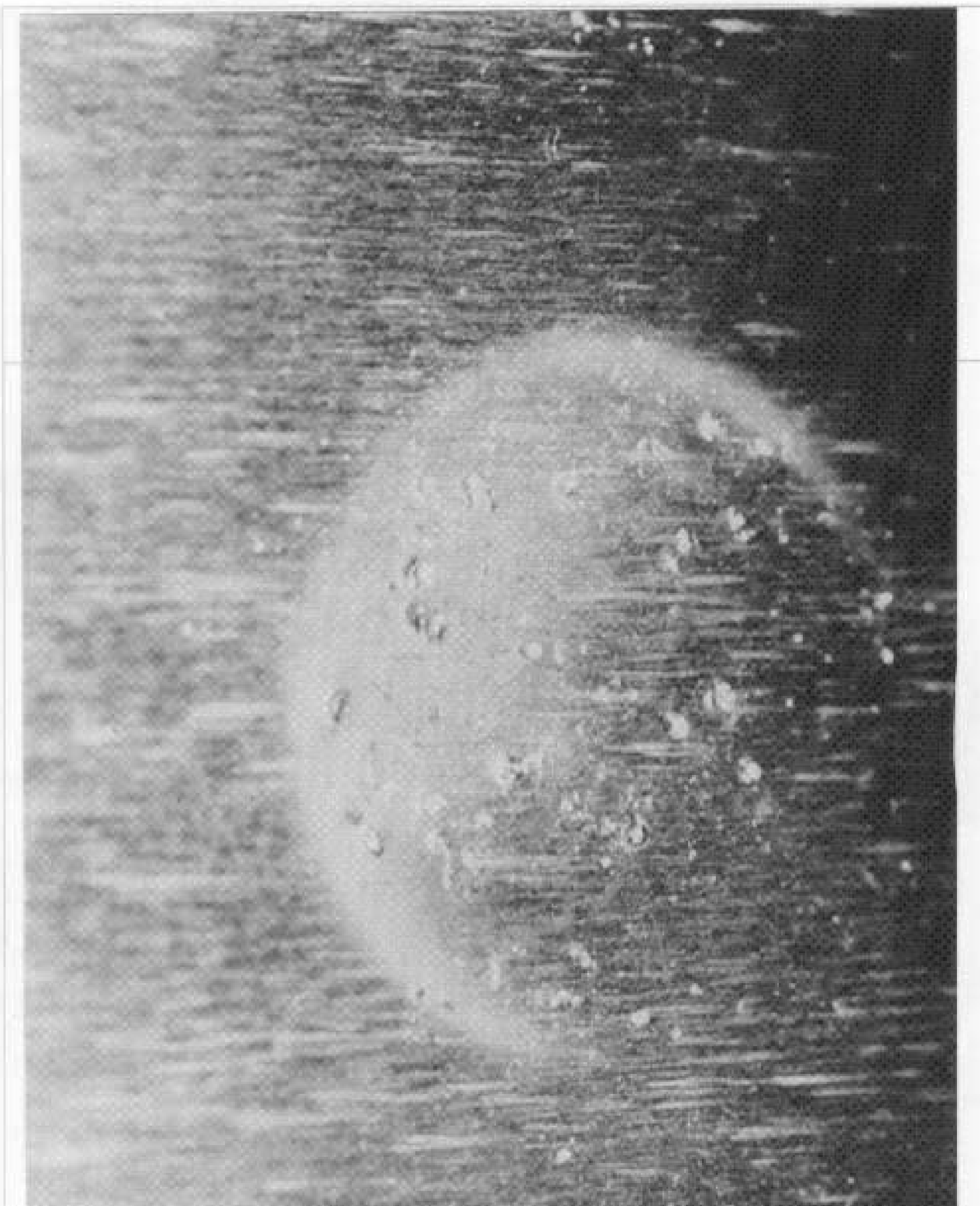

Figure 4. Pattern of impact on a polished steel plate showing distribution of abrasive jet.

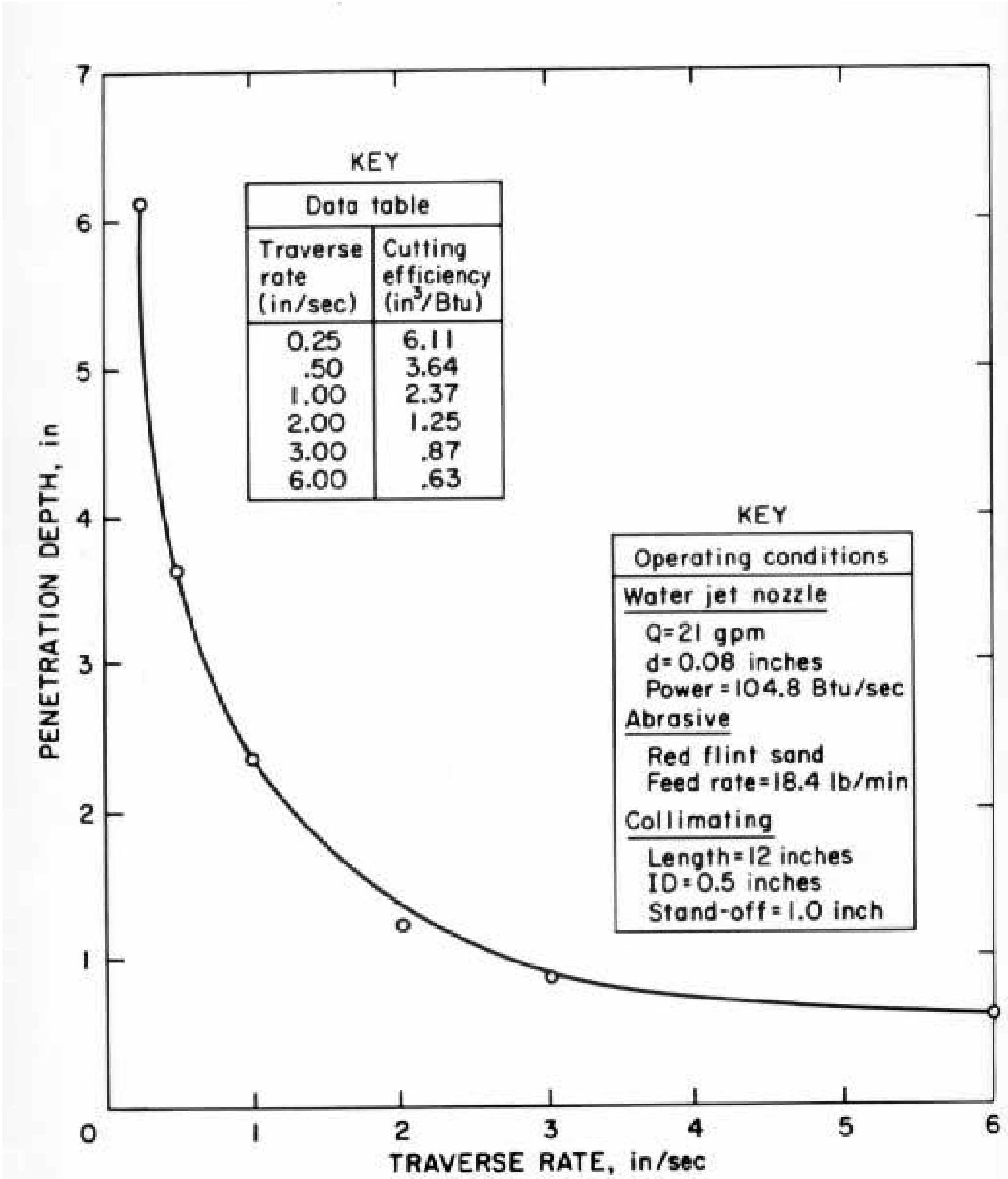

Figure 5. The variation of the depth of penetration of a 10,000 psi, 20 gpm abrasive jet into Salem limestone in a single pass as a function of traverse rate.

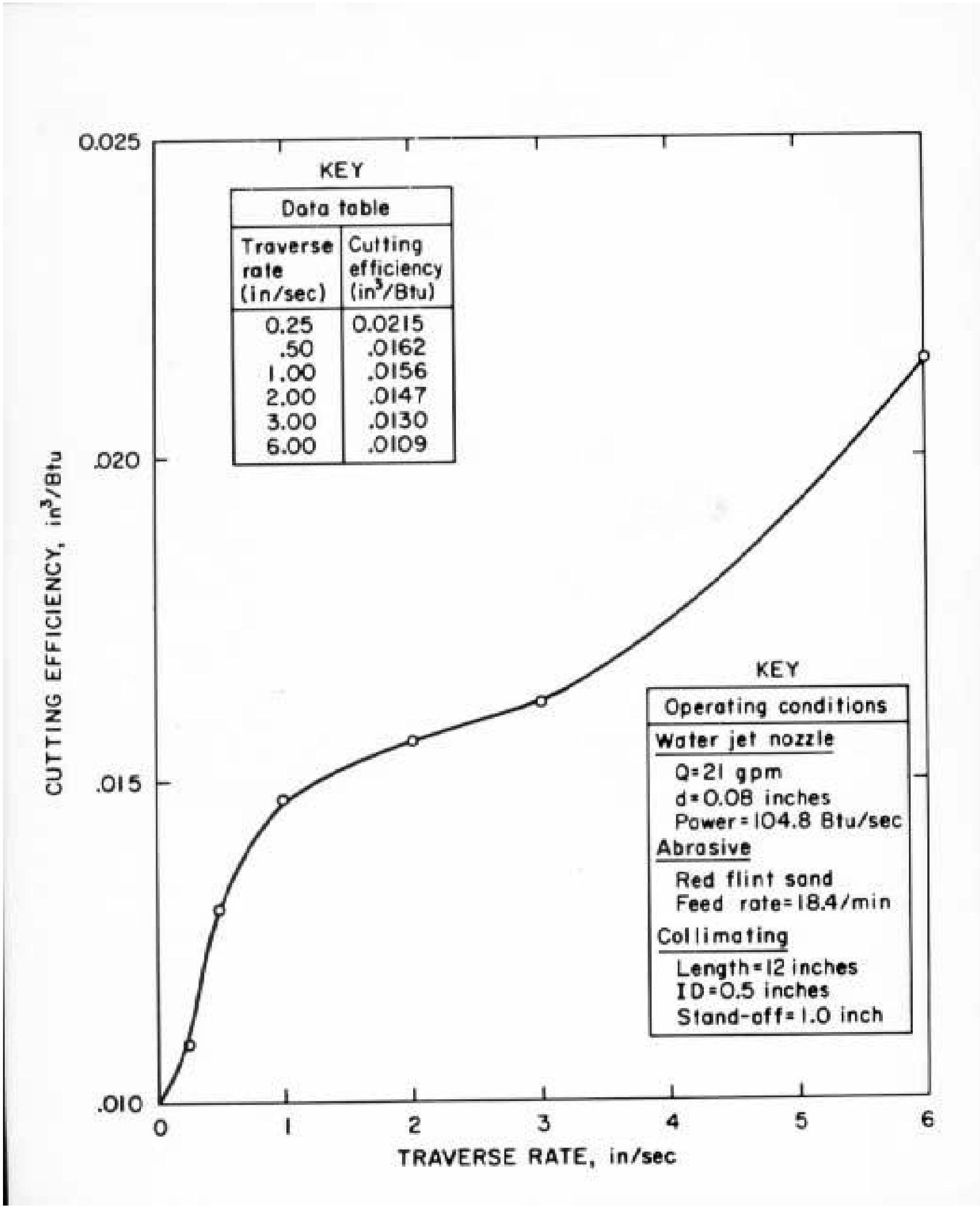

Figure 6. Cutting efficiency as a function of traverse rate in Salem limestone.

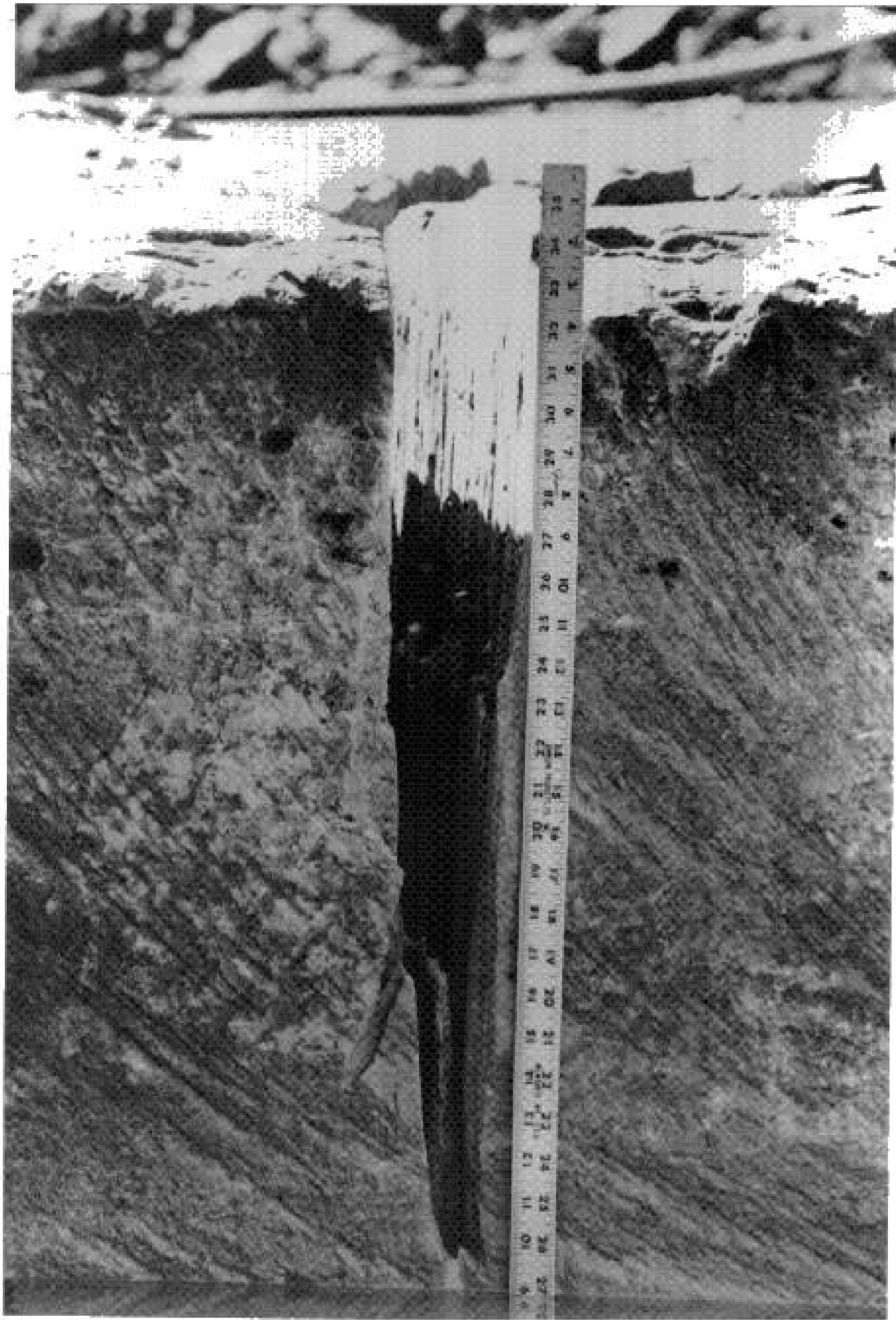

Figure 7. Deep kerf showing intervening rib in the Oneota dolomite.

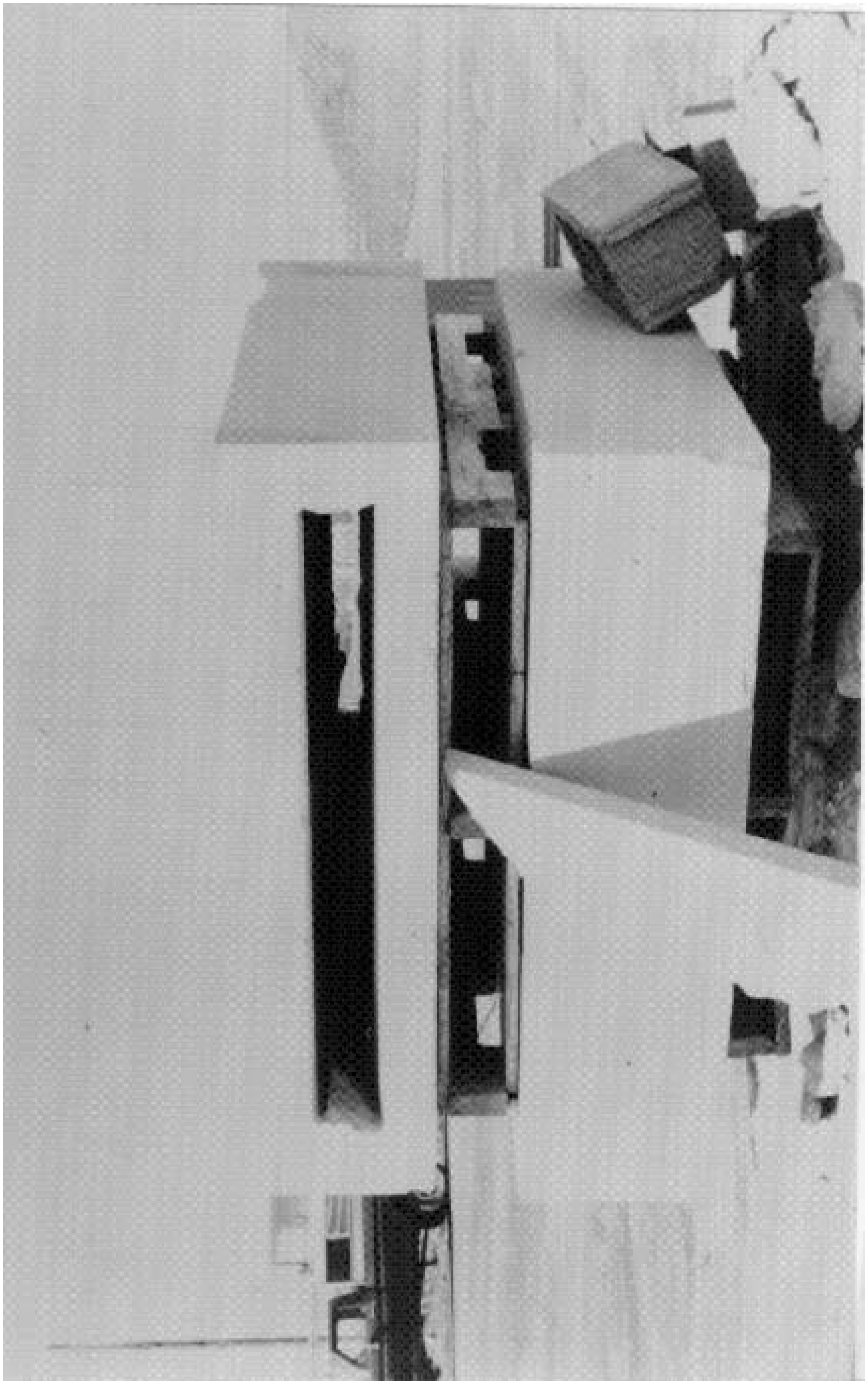

Figure 8. Kerf cut through a 4 ft. cube of Salem limestone.

Figure 9. Abrasive jet mixer-collimator ensemble mounted on a drill carrier.

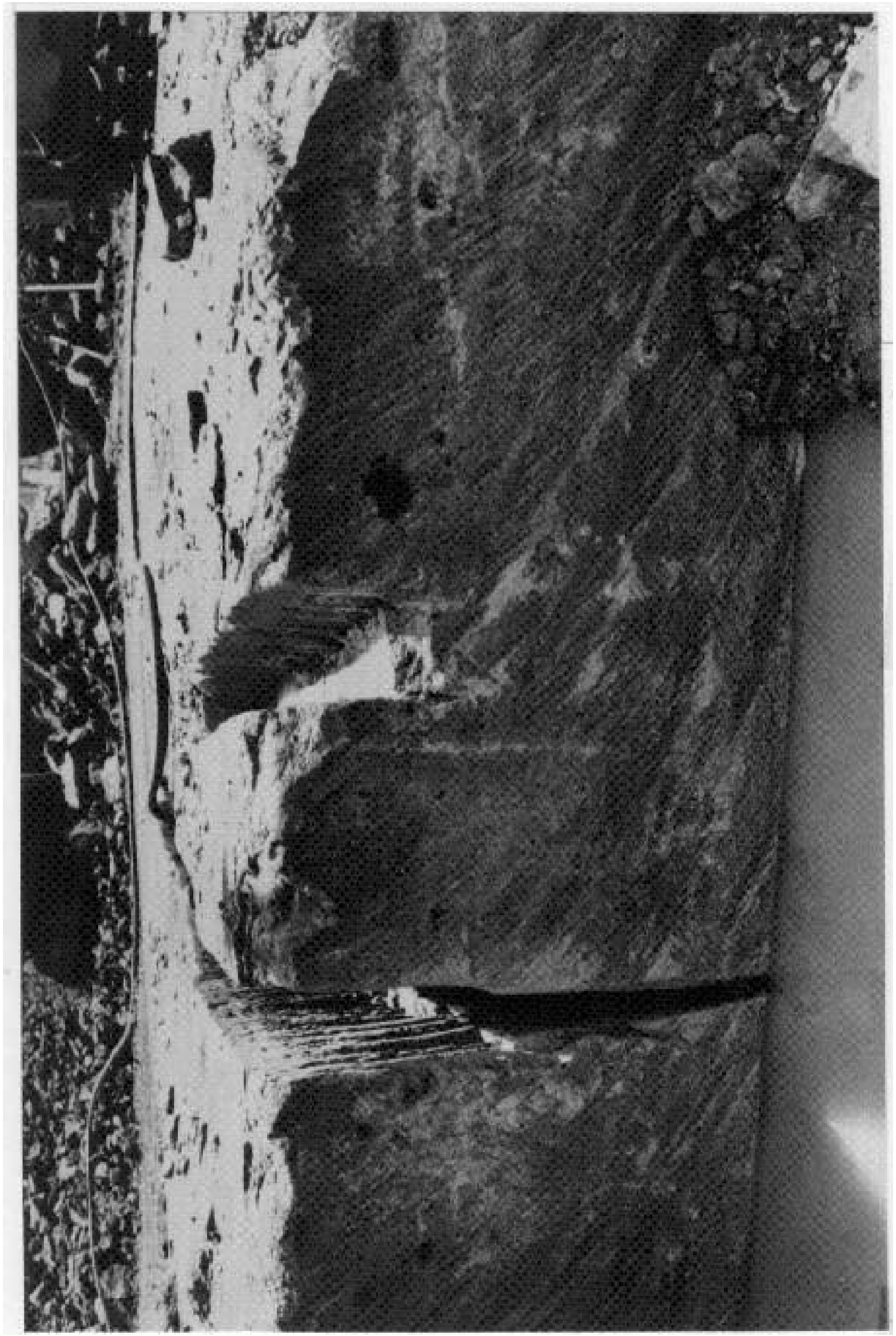
Figure 10. Block with kerf cut on periphery.

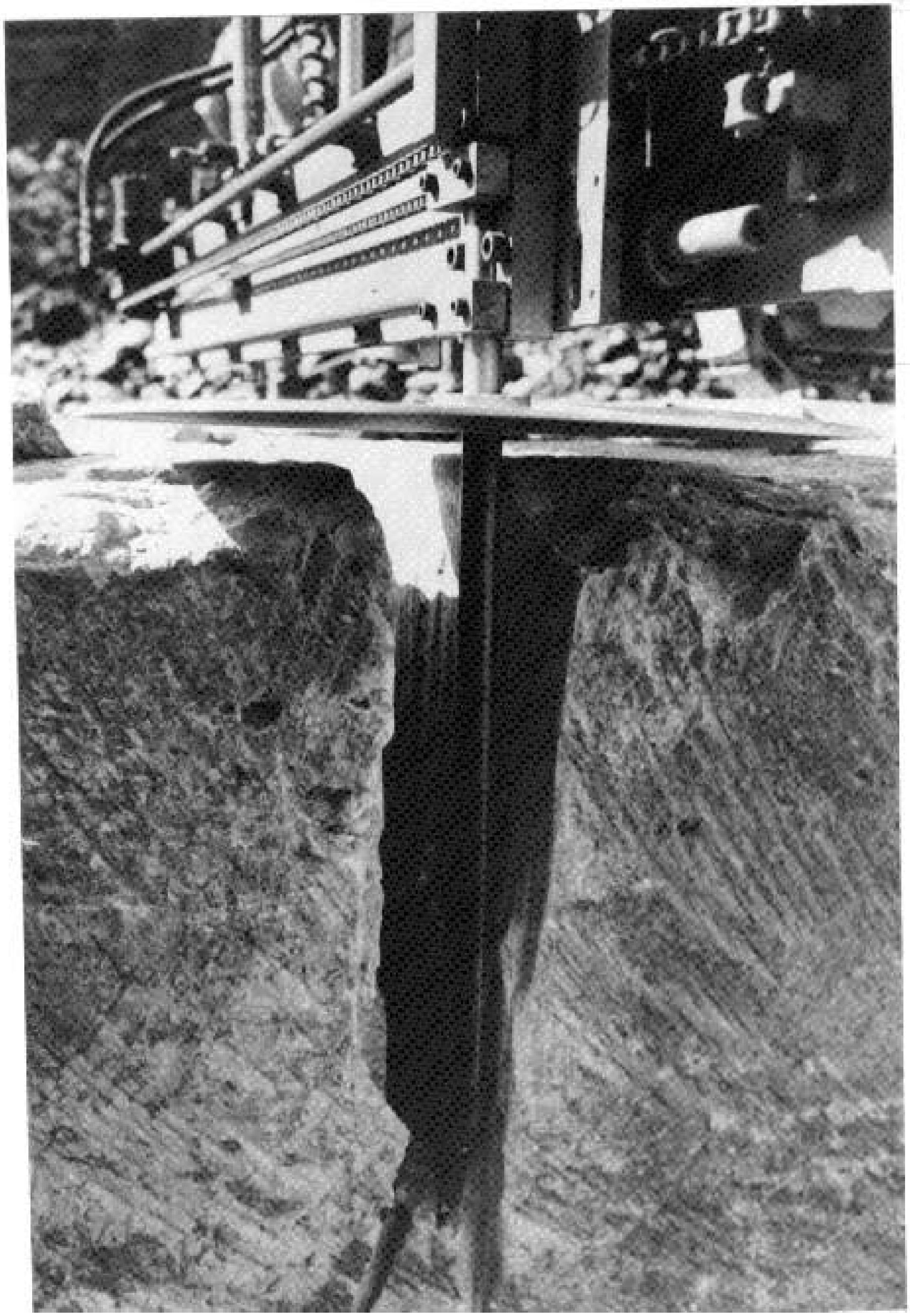

Figure 11. Deep kerf with collimator pipe.

ABRASIVE-WATERJET DEEP KERFING IN CONCRETE FOR NUCLEAR FACILITY DECOMMISSIONING

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ABSTRACT

The feasibility of using abrasive-waterjets for decommissioning nuclear facilities was demonstrated using nozzle assemblies specially designed to enter the kerf and continue cutting. Linear cutting tests on concrete indicated higher cutting efficiency at reduced abrasive flow rates. The highest performance index, (cm^2/kW•kg), was obtained using a 0.38 mm jet at 206 MPa with an abrasive flow rate of only 0.54 kg/min. In deep kerfing experiments on reinforced nuclear grade concrete, the cutting rate for a relatively high-power jet (60 kW, d_n = 0.8 mm, P = 241 MPa) was 19.5 cm^2 /hr kW (kg/min). Cuttings and water were 80% contained using a unique vacuum system. Optimization of deep kerfing parameters and pick-up efficiency is expected to increase cutting rates to 2 - 3 m^2/hr, with nearly 100% containment. Cost analysis indicated that this technique would be competitive with conventional bulk removal methods.

INTRODUCTION

Approximately 500 contaminated nuclear facilities in the United States have been identified for decommissioning (Ref. 1). A major part of most decommissioning jobs is the decontamination and removal of concrete. This can range from minimum removal of radioactive material to the complete disassembly of a facility (Ref. 2).

Concrete decontamination and demolition require special care during decommissioning activities. Precautions must be taken to prevent the release of radioactive particles when activated concrete sections are removed. Also, difficulties arise because concrete is porous and often contains numerous cracks which hold the contaminants (Refs 3, 4, 5). High radiation levels may prevent personnel from entering the area to be decontaminated. In such cases the equipment must be remotely operated.

In addition to the problem of radiation, nuclear structures typically contain a large volume of concrete with reinforcing bars. Base mats may be 8 m thick, and biological shields up to 3 m thick. Conventional concrete removal methods cannot efficiently remove these structures while containing contaminants from release into the environment (Ref. 6).

Commercial nuclear power plant decommissioning will require a far greater effort than any job attempted to date. For example, a typical pressurized water reactor plant contains approximately 127,400 cubic meters compared to about 3,600 cubic meters in the largest power plant dismantled to date (Ref. 7). Because of the enormous volumes involved, contaminated concrete will have to be segregated to reduce the waste volume for controlled storage. Existing demolition equipment is too slow and cumbersome to perform such work efficiently; new techniques are needed (Ref. 4).

The purpose of this work was to investigate abrasive-waterjet techniques for cutting deep kerfs in reinforced concrete. Prior to this effort, the abrasive waterjet nozzle, positioned above the concrete surface, could make cuts to a maximum depth of about 61 cm (Refs 8, 9). A new nozzle concept had to be developed to overcome this depth limitation. This nozzle would be narrow enough to work in the kerf created by previous cuts, and thus would not be depth limited. Results of research on such new concepts are discussed in this paper.

BACKGROUND

There are a wide variety of bulk concrete and surface removal techniques, many of which have been used to remove concrete in nuclear decommissioning projects. Table 1 lists the techniques for which information is available, together with approximate removal rates, costs and operational limitations. The data were obtained from technical papers and reports (Refs. 7, 8 & 9). Approximate costs were provided by the U.S. Department of Energy (Ref. 8). The selection of a technique depends upon certain unique requirements. The ideal equipment characteristics for performing this work include:

1. Contain all cuttings, i.e.,
    - no release of airborne contamination, and
    - no recontamination due to the cutting operation.
2. Generate no additional waste in the removal process.
3. Remove only the contaminated surface, leaving radiation-free concrete.
4. Cut concrete and reinforcing bars simultaneously.
5. Perform all cuts on all surfaces (ceilings, walls, floors, contoured surfaces, etc.) without tool changes.
6. Adapt easily to remote operation, automation, and use in confined areas (Ref. 10)
7. Generate no shock, vibration or excessive noise while operating.
8. Easy to operate by personnel in protective clothing.
9. Easy to repair.
10. Economically feasible.

Conventional demolition equipment meets few of the ideal features listed. However, such equipment is not designed for use in a contaminated environment. Therefore, it is not useful in applications where the greatest concerns are contamination control and safety (Ref. 11). Abrasive-waterjet equipment, on the other hand, is well suited to this task and can be custom-designed for decontamination applications.

Abrasive-waterjets are formed by mixing small diameter (0.01- to 0.03-inch), high velocity (up to 750 m/sec) waterjets with abrasive particles. The mixing process occurs in a specially designed mixing and accelerating chamber, as shown in Figure 1.

The abrasive particles exit the acceleration section at high velocities and become capable of cutting even the hardest material (Refs. 12, 13).

EXPERIMENTAL INVESTIGATION

The experimental investigations conducted in this research include the following efforts:

· Linear cutting tests with abrasive-waterjet nozzles not entering the kerf.
· Cutting tests with abrasive-waterjet nozzles entering the kerf.
· Tests with catching devices to pick up abrasives, water and cutting.
· Analysis of used slurry to determine the feasibility of recycling.

The following sections describe the results obtained during this preliminary research effort.

Linear Cutting Tests

Linear cutting tests were conducted on concrete with compressive strength of about 34.5 MPa and density of 2483 kg/m^3. A sample of this concrete, with abrasive-waterjet cuts revealing the size and distribution of aggregates, is shown in Figure 2.

Figure 3 shows the depth of cut obtained at different passes with a 32.8 kW jet working at 380 MPa and consuming 3.62 lpm of water and 0.9 kg/min of abrasives. It is evident that deep cuts can be made with a low power jet when high pressure is used. This relationship is clarified in Figure 4, which shows cutting results obtained with 15.4 and 28.5 kW jets operating at 207 MPa and 310 MPa, respectively. From this graph it can be calculated that the lower power jet is more efficient in terms of rate of kerf area generation per unit power and unit abrasive flow rate which will be termed here as "performance index". This performance index will be 14.82 cm^2/kW-kg for the lower power jet and 8.74 cm^2/kW-kg for the higher power jet. These numbers use the depth of cut obtained with the first pass. If the effect of number of passes is introduced, then the lower power jet will consume the same energy in 7 passes as the higher power jet does in 4 passes. However, the cumulative depth of cut produced will be slightly larger for the higher power jet.

Figures 5 and 6 show other examples of the effect of pressure, and consequently power, on the depths of cut obtained. In Figure 5, jet powers are 26 kW and 20.7 kW at the corresponding pressures of 234 MPa and 200 MPa. The performance index with single pass is 10.07 cm^2/kW-kg at 200 MPa and 6.84 cm^2/kW-kg at 234 MPa. This difference is reduced when both jets consume equal energy after 4 passes (5.13 cm^2/kW kg) for the lower power jet and after 5 passes (3.8 cm^2/kW-kg) for the higher power jet.

Table 2 summarizes results on Figures 3-6. From this table it is evident that, at a kerf of about 28.5 cm, a 0.38 mm jet at 206 MPa and 0.54 kg/min of abrasives will be most efficient (6.84 cm^2/kW-kg). This combination of parameters is also most efficient for single pass cutting (14.8 cm^2/kW-kg). This high efficiency is attributed to the reduced abrasive flow rate used.

Deep Kerfing Nozzle Designs

To achieve deep cutting in concrete, the abrasive-waterjet nozzle shown in Figure 1 was to be redesigned so that it could enter the cut. The process consisted of cutting a kerf (slot) wide enough to introduce a nozzle head, thus extending the slot depth. This required that the nozzle head be as small as possible to reduce the energy requirement for cutting. It was clear that some energy would be consumed in widening the kerf to achieve a deep cut. On the other hand, reducing nozzle head size might limit its capacity to transmit enough hydraulic power and abrasives to the cutting area to achieve faster cutting rates.

The nozzle design concepts tested in Phase I included:

- A rotating and slightly angled abrasive-waterjet nozzle (Figure 7a).
- A non-rotating head with side jets to produce a controlled slot width with a center jet to remove material in between slots (Figure 7b).
- A non-rotating nozzle head that utilizes the spreading of the abrasive-waterjet stream to produce a wide enough slot (Figure 7c).

Figure 8 shows a sketch of the first concept and Figure 9 shows that the nozzle design for the second concept. Only one angled jet, rather than the three shown in Figure 9, was used to determine concept feasibility. The multiple nozzle concept was simulated by traversing the single nozzle three times with proper jet orientation. The principal differences between the nozzles in Figures 8 and 9 are in size and means of abrasive feed. In some tests the rotating deep kerfing nozzle was used without rotation to represent the second nozzle concept. Figure 10 shows photographs of second and third deep kerfing nozzle concepts.

Deep Kerfing Experimental Results

The nozzle shown in Figure 8 utilized waterjet orifice sizes of 0.305 mm and 0.51 mm at a pressure of 241 MPa, which corresponds to hydraulic power levels of 8.9 kW and 23.9 kW, respectively. Maximum abrasive flow rates for 400 µm (60 mesh) and 700 µm (36 mesh) garnet sand were 0.9 kg/min and 0.77 kg/min, respectively. These maximum abrasive flow rates represent the capability of this small nozzle assembly to handle abrasives. Rotations were 80 to 300 rpm and linear traverse speeds 10.2 cm to 30.5 cm per minute, respectively.

The optimum rotational speed was found to be 100 rpm, and optimum traverse speed was 20.3 cm/min for these jets. The incremental depth of cut was 3.8 cm per traverse. However, uncut aggregate in the slot did not allow the nozzle head to enter the kerf. Initially, it was necessary to cut two slots and then remove the material between the slots by rotating the nozzle head.

In the next series of experiments, nozzle rotation was eliminated. In these tests two closely spaced cuts were made and the material between them was removed by subsequent linear traversing. This method was more effective in removing material, and it produced a slot with well defined edges and bottom. Traverse rates were varied from 10 to 91 cm/min. A rate of 30.5 cm/min was found to be most effective.

The typical depth of cut increment using this method was about 5 cm to 6.4 cm. The average observed kerfing rate is about 0.23 m^2/hr for the 24 kW jet with 0.77 kg/min of 700 μm garnet sand. Previous research efforts (Ref. 9) indicated that cutting 30.5 cm thick concrete at a rate of 1.86 m^2/hr requires 74.6 kW jet power with 1.8 kg/min of abrasives. The cutting efficiency in this case is 2.28 cm , while the efficiency for the deep kerfing example mentioned above is about 2.07 cm /kW-kg. This suggests that, even with the burden of kerf widening for deep slots, cutting efficiency is reasonably within the range of the best current slotting methods.

To improve cutting rates, more power and abrasive flow rates will be needed. The nozzle design shown in Figure 11 was used to deliver more power to the cutting surface. Tests using this nozzle were conducted on a nuclear grade concrete obtained from UNC Nuclear Industries in Richland, Washington. This concrete contained various size of rebar and steel plate reinforcements. Due to the steel reinforcement, the cutting rate was 0.185 m^2/hr with the 24 kW nozzle head. Nuclear grade concrete was cut faster than the concrete with large aggregates for which Figures 3 - 6 were generated. When the relatively high-power jet (60 kW, d_n = 0.81 mm, P = 241 MPa) was used, the cutting rate for a 0.91 m-thick reinforced nuclear grade concrete was 0.95 cm^2/kW-kg. Because a limited number of tests were conducted, this cutting rate can not be considered optimized performance. Observation of the cutting process led to the following conclusions for improved efficiency:

- Nozzle diameter should be reduced.
- Angles of jets should be optimized.
- Optimum energy distribution between jets should be determined.
- Removal of cut material from the kerf will expose more uncut surface to the jet, improving cutting efficiency.

A third series of tests employed the nozzle head shown in Figure 10. Although there might be great advantages to this nozzle design, no successful in-kerf advance was achieved. The cutting of aggregates was insufficient to allow the nozzle to enter the kerf. Further research on improved distribution of hydraulic energy over the nozzle cross section may prove the feasibility of this technique.

Catching Water, Abrasives and Cuttings

Figure 12 shows a sketch of the catcher concept used. Two shop vacs were used to suck water, abrasives and concrete cuttings from the slot. Leading and trailing suction heads were attached to shop vacs and to nozzle heads. Deep kerfing tests conducted with this suction device indicated the following from few experiments:

- 80% to 90% of spent material was removed from the slot.
- The trailing suction head recovered up to 95% of the material removed.
- Unrecovered material was mostly water that flowed out of the slot or sprayed upward.

This effort indicates that a more powerful suction device may be needed. Also, a more closely attached suction head will be more effective. Using a shroud to seal the slot may also prevent sprayback losses.

Recycling of Water and Abrasives

After cutting with 400 µm abrasives spent material was collected and sieved for approximate abrasive particle size analysis. Also, a collected sample was used for cutting. The results (Table 3) showed that more than 96% of the spent abrasives were fractured. When these spent abrasives were dried and recycled, there was a decrease of about 44% in cutting rates. When fractured abrasives were recycled in a slurry, there was a 67% decrease in cutting rate. Considering these results, the use of garnet sand may not be feasible for recycling.

More accurate particle analysis of the fines showed that 30% of the solids collected are below 1 micron in size. The water flow rate used in this test was about 9 kg/min and the flow rate of solids (abrasives and cut material) was about 0.9 kg/min. Thus, the concentration of fines below 1 µm in recycled water may be calculated at 3%. This concentration may be further reduced using finer filters and mixed bed deionizers, which remove soluble particles (Ref. 14). Such particles will not affect the intensifier pump, but may increase wear in very small waterjet orifices.

COST COMPARISONS

Deep and controlled kerfing with abrasive-waterjets was demonstrated to be technically feasible. Performance was unmatched by any other technique, as discussed in Section 2. However, a cost comparison can be made for relatively shallow cuts by other methods for concrete pavement cutting. A cost comparison of abrasive-waterjet cutting versus diamond saw cutting and drill/break methods used by a commercial concrete cutting contractor and three utility companies revealed that the abrasive-waterjet method was about 25% more economical (Ref. 15). This comparison was extended to deeper cutting using the preliminary results obtained on cutting nuclear grade concrete. The approximate breakdown of costs, shown in Table 4, indicate a total hourly cost of $53 for abrasive-waterjet cutting. This estimate assumed that abrasives would be used only once. The factors and expectations that offset this cost include:

- Additional cost of a recycling system
- Reduced cost of abrasive consumption
- Reduced cost due to improved cutting rates with an optimized tool
- Reduced cost of wear due to possible reduction in abrasive flow rate with optimized nozzle.

CONCLUSIONS

Linear cutting tests indicate that high cutting efficiency can be obtained in concrete when the depth of cut is relatively shallow. For deep kerfing, this performance can be repeated by advancing the tool into the slot after each pass. It is technically feasible to cut deep kerfs in concrete and reinforced concrete with abrasive-waterjet nozzles entering the cut kerf. This was demonstrated by cutting nuclear grade concrete with steel reinforcing bars.

Cutting efficiency of deep kerfing nozzles matches or exceeds that for conventional slotting methods. It is estimated that a 75 kW abrasive-waterjet cutting system can cut deep kerfs in reinforced concrete at a rate of at least 0.93 m^2/hr, regardless of depth. Optimization of deep kerfing parameters may improve cutting rates to 1.86 - 2.78 m^2/hr.

Partial success was achieved in catching and pick-up of water, abrasives and cuttings. Raising pick-up efficiency from 80% (obtained in this study) to near 100% is achievable with a properly designed suction system. This requires further investigation. Recycling of garnet abrasives has limited feasibility due to particle breakage. Steel grit abrasives may be needed for recycling. Water recycling is technically possible and can be achieved with appropriate filtration systems.

A comparison with other techniques indicates that abrasive-waterjet cutting is economically feasible. Further optimization and cost reductions could increase the practicality of abrasive-waterjets for decommissioning applications, making this the preferred technique for most operations.

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TECHNIQUE	DESCRIPTION	RATE M^3/DAY	COST $\$/M^3$	COMMENTS
CONTROLLED BLASTING: Massive reinforced standard concrete (Non-radioactive)	Holes are drilled in concrete, loaded with explosives, and explosives are detonated using a delayed firing technique. The rebar is cut after the detonation.	77 - 306	144	Rebar must be cut after blast and fog spray is used to control dust.
CONTROLLED BLASTING: Massive non-reinforced standard concrete (non-radioactive)		190	20	
CONTROLLED BLASTING: Lightly reinforced standard concrete (non-radioactive)		153	52	
CONTROLLED BLASTING: Non-reinforced high density concrete (Radioactive)		4.5 - 6	52	Removal rate includes inefficiency due to personnel and area contamination control and radiation work area control.
CONTROLLED BLASTING: Lightly reinforced standard concrete (Radioactive)		4.5 - 6	280	
CONTROLLED BLASTING: Massive reinforced standard concrete (Radioactive)		3 - 4.5	560	76 cubic meters per day removal rate is possible if adequate space is available to use large capacity loading and hauling equipment.
WRECKING BALL: Concrete block structure	A 2- to 5-ton ball is dropped or swung into the structure to be demolished. The technique is recommended for non-radioactive structures less than 1 meter thick.	46	13	A wrecking ball is used after all detectable radiation has been removed from the structure.

Table 1. Bulk Concrete Removal Techniques

TECHNIQUE	DESCRIPTION	RATE M^3/DAY	COST $\$/m^3$	COMMENTS
CORE STITCH DRILLING	Close-pitched holes are drilled around the desired area to be removed.	Slow	High	The technique is difficult to implement where accessibility is limited since the slab must be sheared free and removed.
MANUAL BREAKING	Small chipping hammers and large jackhammers are used to break the concrete surface.	0.7 - 15	42	Rebar must be cut separately. Water spray is used to control dust.
EXPANDING COMPOUND	A chemical compound is poured into predrilled holes. After 10 to 20 hours the compound expands causing the concrete to fracture.	Slow	--	It is not recommended for concrete less than 0.3 m thick. If reinforcing rods are present, it must be used to expose the rods for conventional cutting.
ABRASIVE WATERJET - DEEP KERF CUTTING	Abrasive particles are mixed with high velocity water jets to cut concrete and reinforcing rods. The cutting nozzle enters the kerf to make deep cuts.			The depth of cut is limited only by the length of tubing attached to the nozzle.
ABRASIVE WATERJET - SINGLE PASS CUTTING	Abrasive particles are mixed in a high velocity waterjet to cut concrete and rebar. One pass of the jet is used to cut up to 0.6 m deep.			Best results are obtained cutting concrete with rebar up to 0.3 m thick. Deeper cuts require dispropor-tionality more time to perform.

Data gathered from Refs. 7, 8 & 9.

Table 1. Bulk Concrete Removal Techniques (Cont.)

TECHNIQUE	DESCRIPTION	RATE M^3/DAY	COST $/$M^3$	COMMENTS
WRECKING BALL: Non-reinforced standard concrete		38	17 - 35	
WRECKING BALL: Lightly reinforced standard concrete		30	26 - 52	
WRECKING BALL: Heavily reinforced standard concrete	Not recommended for heavily reinforced concrete.	--	144	
BACKHOE MOUNTED RAM	A hydraulic or air power ram with a point mounted on the arm of a backhoe. An operator positions the ram to remove walls and floors	6 - 23	56	The ram is primarily used on lightly reinforced concrete less than 0.6 meter thick. Rebar must be cut separately and fog spray used to control dust.
FLAME CUTTING	A high-temperature (8,871°C) flame jet is used to decompose concrete. Used for thick concrete less than 2 m thick, with or without reinforcing rod.	0.09m^2/hr	2044/m^2	Not recommended for radioactive concrete.
ROCK SPLITTER	A hydraulically powered expanding wedge is inserted into a predrilled hole to split layers of concrete.	190	--	The reinforcing rods must be cut. Impractical for cut over 0.3 m deep.
BLADE SAWING	A diamond or carbide tipped wheel is used to abrasively cut a kerf.	0.1 m^2/min	10 - 39	Maximum cut is 1 meter deep. Water is used to control dust. The water must be contained in a contaminated environment.

Table 1. Bulk Concrete Removal Techniques (Cont.)

d_n (mm)	P (MPa)	Power kW	lpm	$\dot{m}a$	N	h (cm)	Performance Index $cm^2/kW \cdot kg$
0.36	379	32.81	3.63	0.9	1	13.97	6.84
					3	28.83	4.75
0.38	207	15.16	3.07	0.55	1	8.43	14.82
					7	27.61	6.84
	310	27.85	3.79	0.73	1	12.24	8.74
					4	29.13	5.13
0.45	200	20.75	4.35	0.73	1	10.92	10.07
					5	27.94	5.13
	234	26.38	4.73	1.02	1	12.70	6.84
					4	28.70	3.8
0.51	164	19.28	4.88	0.76	1	8.56	8.55
					5	28.07	5.51
	200	25.61	5.37	1.02	1	11.61	6.46
					4	28.70	3.99

d_n = waterjet orifice diameter (mm)
P = waterjet pressure MPa
lpm = water flow rate in liters per minute
$\dot{m}_a$ = abrasive flow rate in kg per minute
N = number of passes
h = depth of cut in cm
u = traverse rate in cm/min
$h \cdot u$ = rate of kerf area generation in cm^2/min
W = brake (motor) horsepower (input)
Performance Index = $h \cdot u / N \cdot kw \cdot \dot{m}_a$

Table 2. Summary of Cutting Results and Efficiency Data
(from Figures 3 through 6)

Table 3. Abrasive Distribution After Cutting

Mesh Size (Inch)	Collected Weight	% Weight Collected
0.375 and over	15 grams	0.7%
0.066 - 0.375	21 grams	0.9%
0.031 - 0.066	2 grams	0.09%
0.024 - 0.031	0.4 grams	0.01%
0.014 - 0.024	38 grams	1.7%
under 0.014	2163.6 grams	96.6%
TOTAL	2240.0 grams	100%

Table 3. Abrasive Distribution After Cutting

ITEM	HOURLY COST
\$150,000, 150 HP pump, 7000 hr lifetime, 5 years at 13% interest	14.00
Abrasives, 3 lb/min., 10 C/lb	18.00
Cost of Nozzle Wear	5.00
Power 10 C/kwh	11.00
Maintenance	5.00
TOTAL	\$53.00

Table 4. Operating Cost of Abrasive Waterjet System

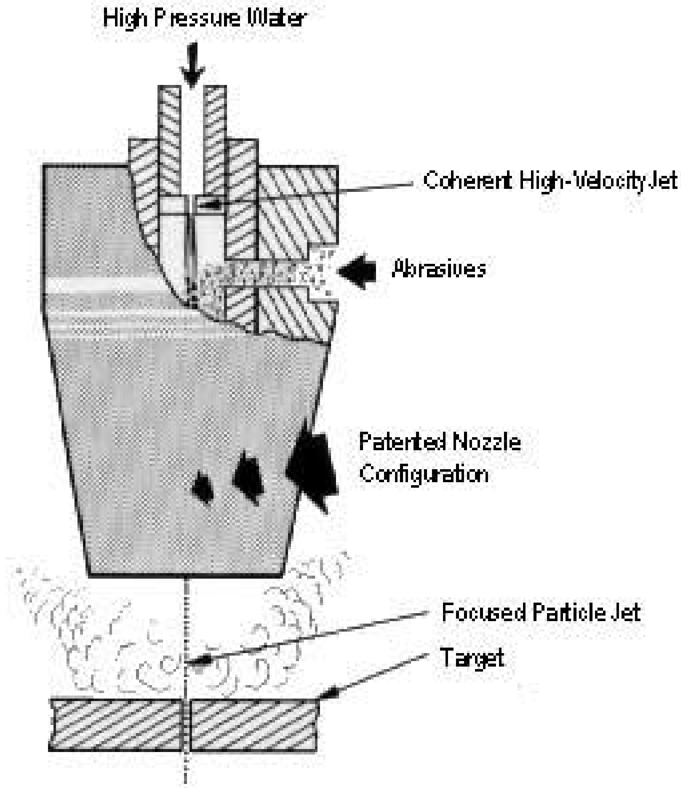
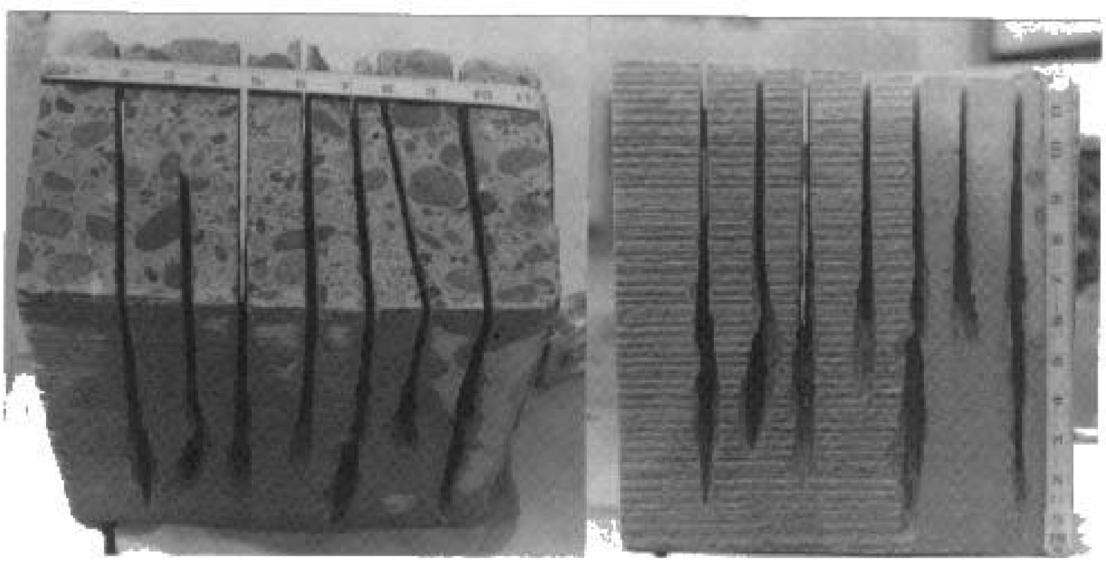


Figure 1. Abrasive Waterjet Nozzle



Top View Side View

Figure 2. Typical Abrasive Waterjet Cuts Made in Concrete

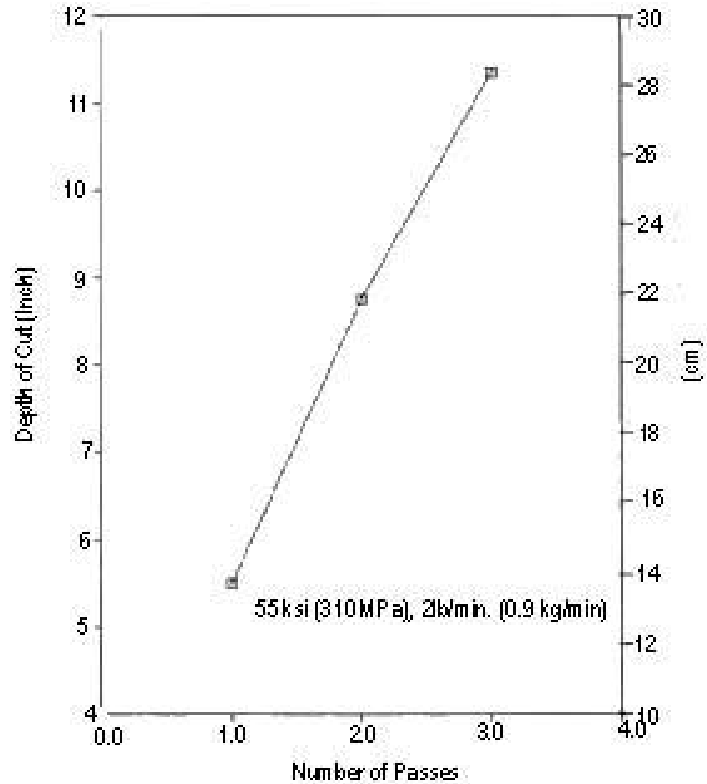


Figure 3. Effect of Number of Passes with d_n =0.014 inch (0.356 mm)

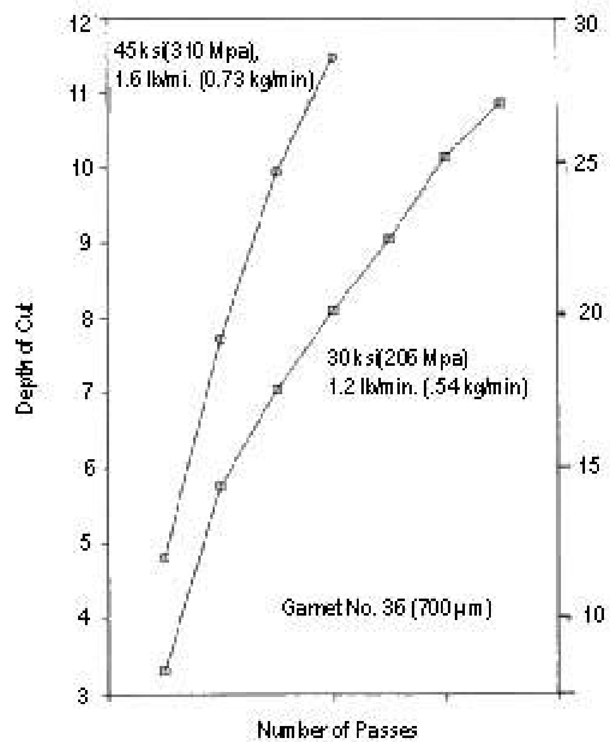


Figure 4. Effect of Number of Passes with d_n =0.0125 Inch Diameter (0.432mm)

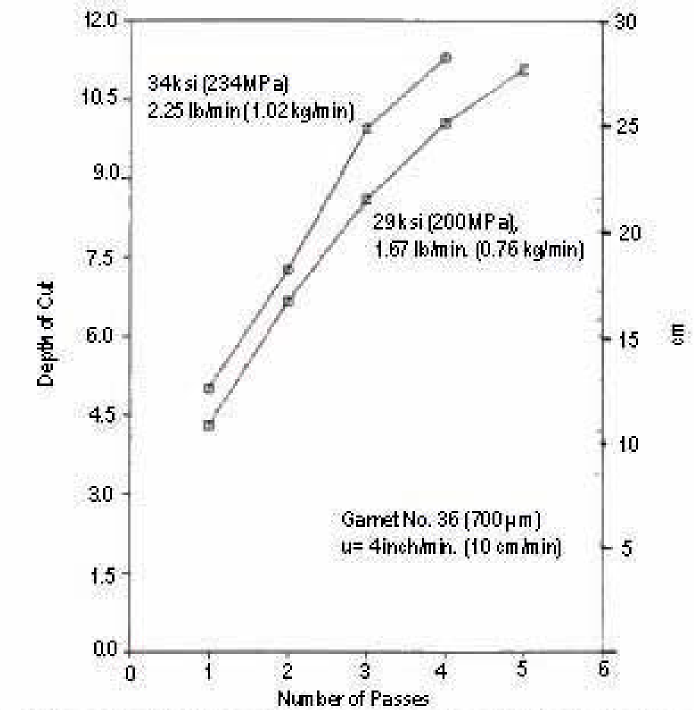


Figure 5. Effect of Number of Passes with d_n = 0.018 Inch Diameter (0.457 mm)

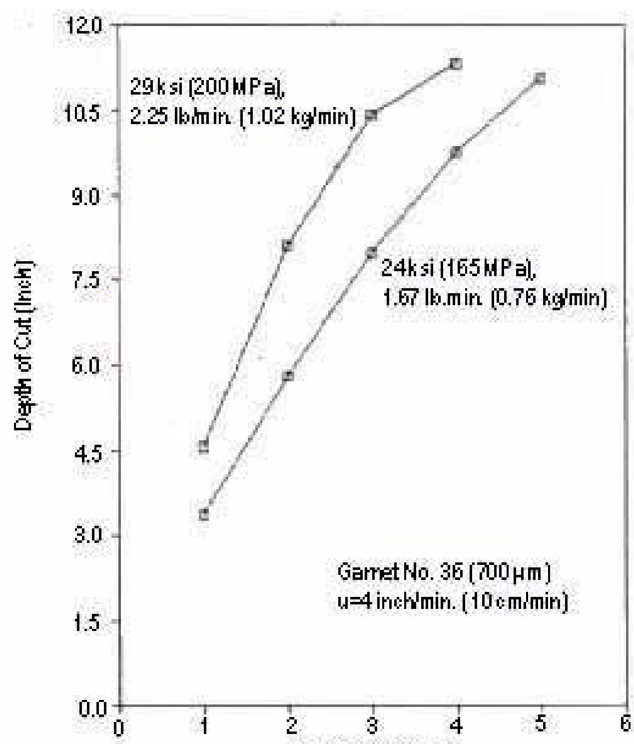


Figure 6. Effect of Number of Passes with d_n = 0.020 Inch Diameter (0.508 mm)

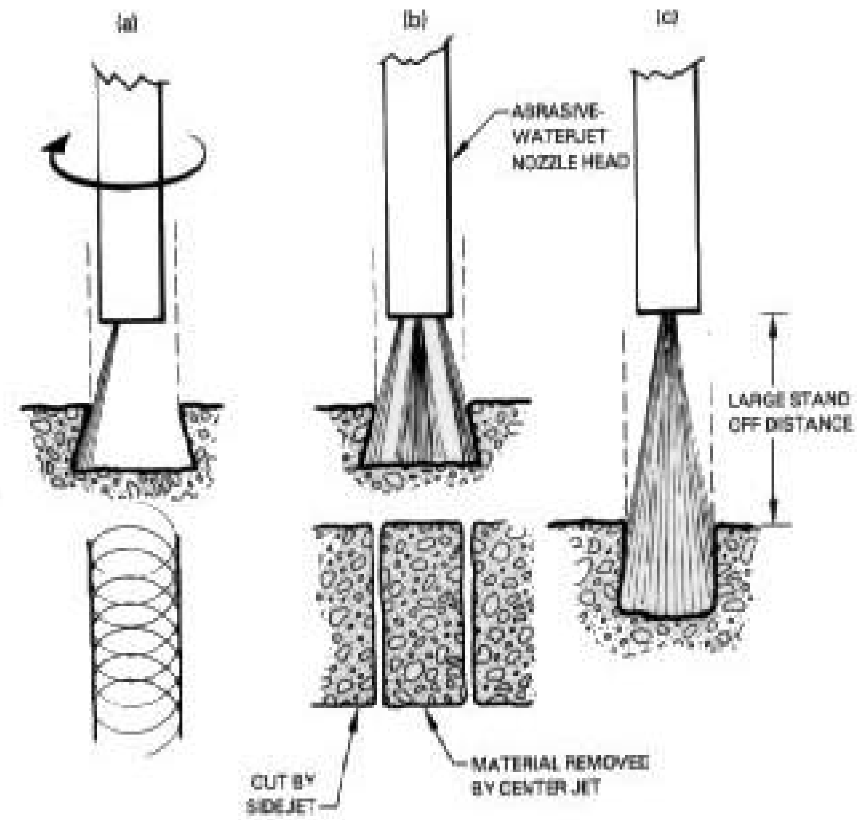


Figure 7. Concepts of Deep Kerfing Nozzles

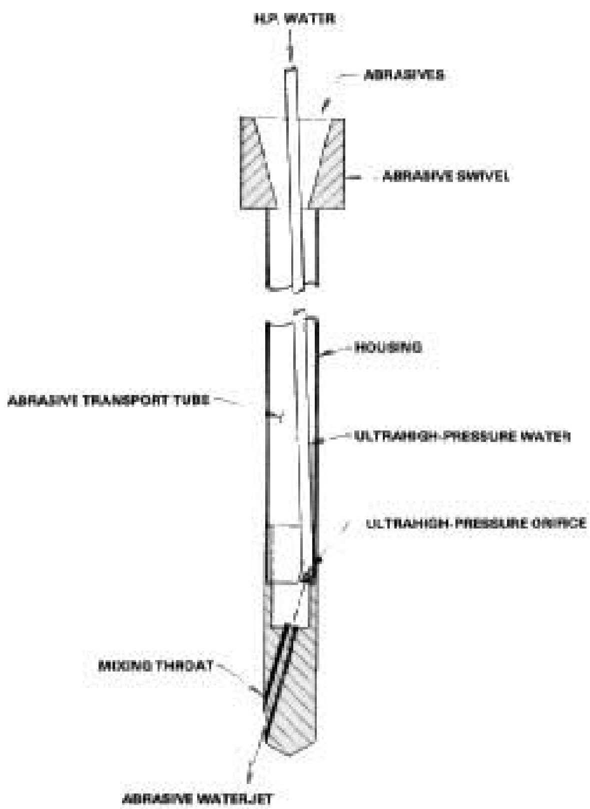


Figure 8. Rotating Nozzle

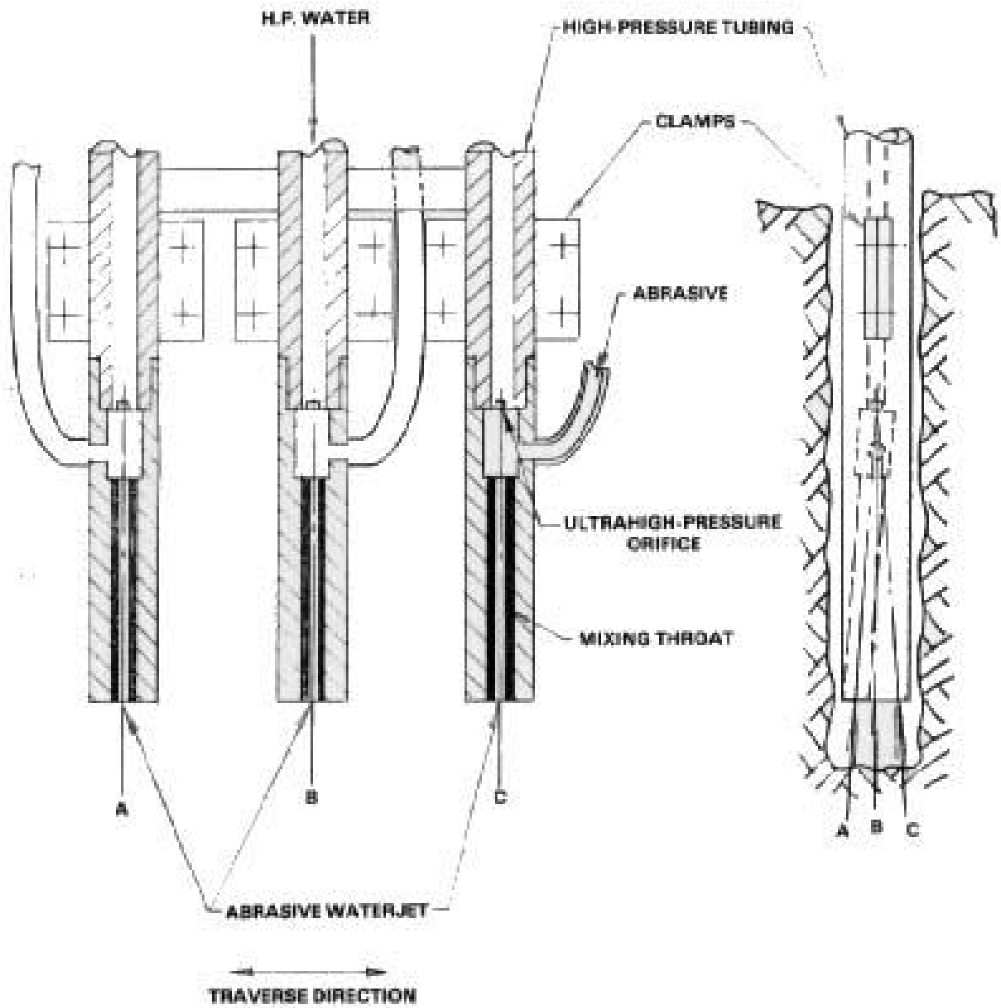
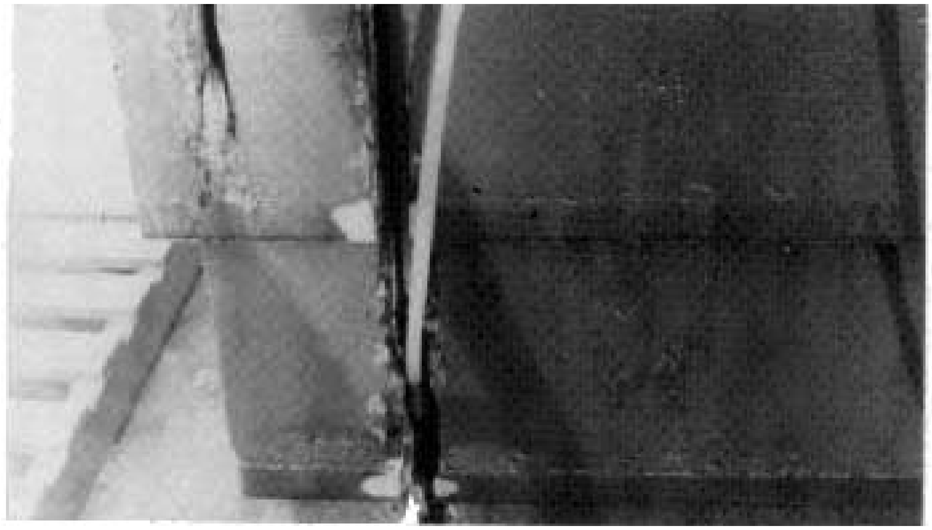


Figure 9. Non-Rating Offset Nozzle

Figure 10. Deep Kerfing Nozzles (0.56 inch diameter) (14.22 mm)



a. Nozzle shown in deep kerf

b. Cutting through

c. Nozzles next to the cut face

Figure 11. Cutting of 3ft (0.924m) Thick Concrete Plug

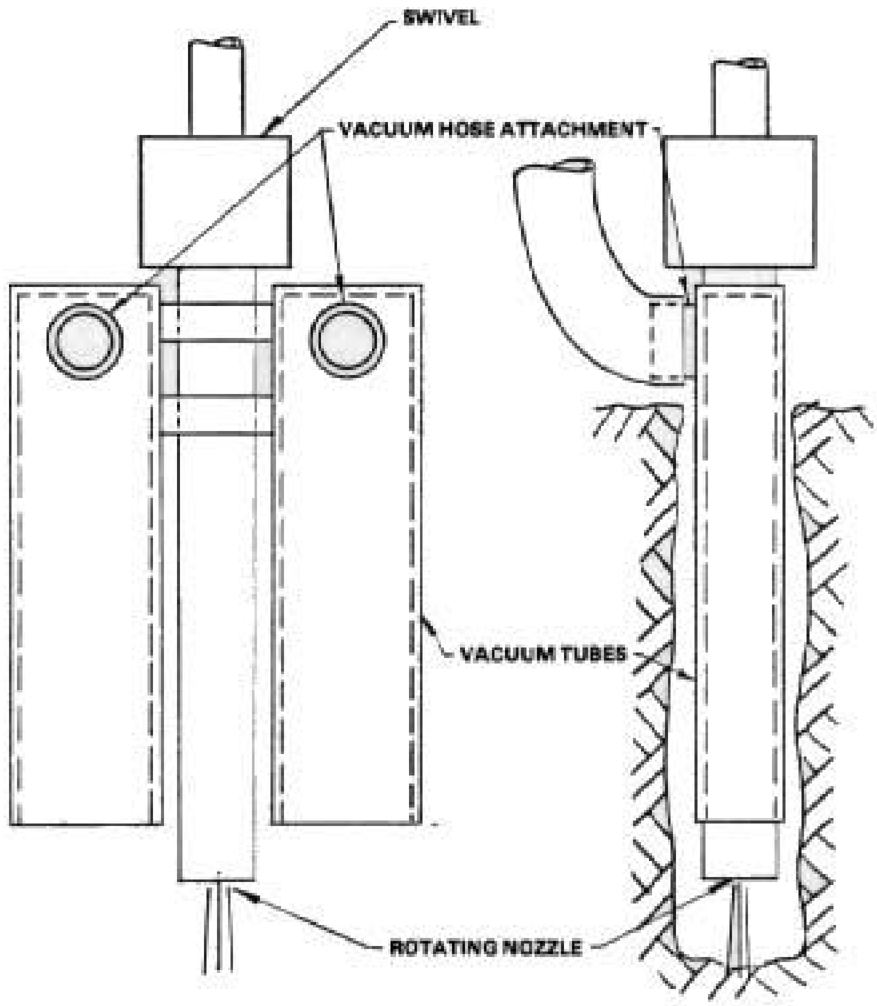


Figure 12. Rotating Nozzle with Catcher

ULTRA HIGH VELOCITY WATER JETS

by

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ABSTRACT

The paper describes the anatomy and impact phenomena of gelled water jets propelled at velocities up to 9,500 m/s. These high jet velocities are obtained explosively using shaped charges. The velocity of the jet is calculated using a framing camera which provides 30 photographs at 5 µsecond intervals. The photographs are also used to study the jet anatomy both before and during impact.

At a charge standoff of 300 mm the jet impacts a 25 mm diameter maraging steel anvil which is acoustically coupled to a 1 metre long maraging steel Hopkinson bar of the same diameter. The impact pressure of the jet is measured using two electrical resistance strain gauge stations positioned at 10 and 30 diameters along the Hopkinson bar. When the stress pulse reaches the strain gauges it is assumed to be one-dimensional leaving a simple calculation to obtain the jet impact pressure.

The results show little correlation between the maximum impact pressure and impact velocity. A variation in jet coherence exists which is associated with the shaped charge propulsion system. The range of measured impact pressures is 250 N/mm^2 to 850 N/mm^2 and of impact velocities 4900 m/s to 8000 m/s. An interesting phenomena arising from the study of photographs of the jet during impact is the presence of a seemingly coherent core within the jet.

INTRODUCTION

This paper presents an experimental study of the flight and impact characteristics of gelled water jets. Maximum jet velocities approaching 9500 m/s are reached using shaped explosive charges as a means of propulsion.

The velocity and anatomy of the water jet are studied using a high speed rotating mirror camera. The impact pressures are measured by firing the waterjet at the end of a long steel bar (a Hopkinson bar) which is instrumented with electrical resistance strain gauges.

The results from a series of fifteen tests are presented. in each test a standoff of 300 mm is used, the only variable being the mass of water propelled by the shaped charge.

SHAPED CHARGE THEORY

In order to propel the water jet at very high velocities, shaped charges can be used instead of the more conventional water cannon arrangement of Watson et al (ref. 1). This section presents a simple theoretical analysis of shaped charge propulsion based on the classic paper of Birkhoff et al (ref. 2) and the work of Johnson (ref. 3).

A given mass of explosive detonated in contact with a rigid material succeeds in causing a crater in the material if the charge is shaped. To increase the depth of the crater and the effectiveness of the explosive the shaped charge can be lined with a suitable material, e.g. copper, gelled water. A cone of liner material backed onto the shaped explosive is shown in Fig. 1. On detonating the explosive, the walls of the cone collapse forcing the gelled water to concentrate on the axis and to give rise to a 'jet' and a 'slug' (See Fig. 2). if the angle of the cone and the explosive are correctly chosen, the jet although of small mass can move at velocities in the order of 10,000 m/s. This effect became known as the Neumann effect in Germany and as the Munroe effect in the U.K. and the U.S.A.

From Figs. 1 and 2 the initial apex angle of the wedge is α but that part of it which has been traversed by the detonation wave takes up an angle β where $\beta>\alpha$. The collapse of the liner is analyzed hydrodynamically assuming the liner material to flow as a non-viscous fluid. Fig. 3 shows the collapse of the liner. In this system the slug and the jet will both leave the collision point (M) with identical velocities, V_2. However, point M itself is moving with a velocity, V_1, hence the absolute speeds are:

$$V_{slug} = V_1 - V_2 \qquad \text{Eq. 1}$$

$$V_{jet} = V_1 - V_2 \qquad \text{Eq. 2}$$

By applying the principle of momentum, Johnson (3) further deduces the following equations:

$$\frac{M_j}{M_s} = \frac{1-\cos\beta}{1+\cos\beta}(\tan\frac{\beta}{2})^2 \qquad \text{Eq. 3}$$

where M_j and M_s are the masses of the jet and slug respectively.

The length of the jet,,l, can be predicted using Eq. 4, where 'h' is the perpendicular height from base to apex of the shaped charge cone or wedge.

$$\ell = (1 + \tan\alpha.\tan(\frac{\beta+\alpha}{2})) \qquad \text{Eq. 4}$$

To solve Equations 3 and 4 the value of β must be known. The angle β can be calculated using Eq. 5 (Johnson) with measured experimental data. In this equation, V_j is the water jet velocity and V_d is the velocity of detonation of the explosive.

$$V_j = 2.U_d.\sin\frac{\beta-\alpha}{2}.\cos\frac{\alpha}{2}/(\sin\frac{\beta}{2}.\cos\alpha) \qquad \text{Eq. 5}$$

Solution of equation 5 by the substitution of experimental results predicts β to be approximately 54°.

EXPERIMENTAL METHOD AND EQUIPMENT

The shaped charge

In each test the shaped charge consisted of a 125 mm length of 38 mm diameter plastic tube, an L2A1 detonator, a booster pellet, a tufnol housing and 172 g of plastic explosive (type PE4 with a density of 1600 kg/m^3 and a detonation velocity of 8200 m/s) as shown in Fig. 1. The cone angle (2α) for each test is 60° and liner thicknesses of 1.5 mm, 3 mm and 6 mm are investigated. The thickness of the liner is measured perpendicular to the sloping surface of the explosive.

The composition of the gelled water used to line the shaped charge is 3.2 g of carageenan mixed with 200 ml of water. The gelled water is cast into the shaped charge using a former set at a preconditioned height above the explosive.

High speed photography

The flight and impact of the water jet are recorded photographically. The camera used is a Barr and Stroud, type CP5, with 35 mm Ilford HP5 black and white film and takes thirty pictures of 18.4 mm diameter at interframe fines of between 0.5 and 15 µseconds. For this series of tests,. an interframe time of 5 µseconds is used giving approximately 8 in-flight photographs of the jet before impact occurs.

The Hopkinson bar arrangement

To measure the impact pressure of the water jet, a 25 mm diameter Hopkinson bar is used. The bar is manufactured from G125 maraging steel, with a 0.2% proof stress of 1960 N/mm^2. The bar is 1000 mm long with two electrical resistance strain gauge stations positioned at 250 mm and 750 mm along the bar (fig. 4). The voltage output from the electrical resistance strain gauges is recorded on a digital storage oscilloscope.

To minimize impact damage, the water jet is fired at an anvil which is acoustically coupled to the Hopkinson bar. The anvil is 25 mm long and is manufactured from a 25 mm diameter bar of G110 maraging steel with a 0.2% proof stress of 1807 N/mm^2. A mild steel witness plate is placed around the anvil to detect "off-target" impacts (Plates 5 and 6). In each test the distance from the base of the shaped charge liner to the anvil is 300 mm.

RESULTS

This section analyses the results presented in Table 2.

Velocity measurements

The velocity profile of the water jet tip with distance is similar for each gel liner thickness and the following general observations have been obtained:

(i) the jet reaches it's maximum velocity almost instantaneously.
(ii) a period of slow but steady deceleration (consistent with air drag) occurs up to a flight distance of approximately 180 mm.
(iii) a sharp drop in velocity occurs in the 190 - 250 mm range which is followed by a slow deceleration.

The two main mechanisms of deceleration are likely to be air drag and Taylor instability (ref. 4). The emergence of "fingers" from the jet tip (Plate 1) has been attributed to Taylor Instability by Field and Lesser (ref. 5) who also observed a rapid drop in jet velocity after the development of the instability. The length of jet travel (L) before the onset of Taylor instability and before significant jet breakup, was calculated by Field and Lesser using Equation 6. In this equation, a is the surface tension of water, (ℓ is the jet radius, $\rho\ell$ and $\rho\alpha$ are the densities of water and air respectively, u is the jet velocity and K is a numerical constant with value of ~26 for the onset of instability and ~ 77 for significant jet breakup.

$$L = K\ell^{3/4}\sigma^{1/4}\rho\ell^{1/2}\rho a^{-3/4}u^{-1/2} \qquad \text{Eq. 6}$$

If Taylor instability is the cause of the sharp drop in velocity of the water jet then the 6 mm gel thickness jet would experience this velocity drop at a later stage than the 1.5 mm gel thickness jet because of it's increased radius. However, this is not the case as shown in Table 2. The rapid deceleration point occurs at a similar distance for each gel thickness suggesting perhaps that the apparently coherent jet core seen photographically, (Plates 2, 3, 4), is constructed of smaller water drops whose size do not change very much with gel thickness. Taylor instability may occur locally in these drops. Application of Equation 6 to a water jet with a velocity of 8500 m/s, where significant jet break-up and deceleration occur around 215 mm, suggests a radius of approximately 4.5 mm.

Both the maximum and impact velocities of the water jet decrease with increasing gel thickness. This is to be expected as an increase in gel thickness also increases the mass of water to be propelled by the explosive as shown in Table 1.

The jet velocities quoted in Table 2 are minimum values calculated from the photographs. The actual jet velocities after correction for possible errors in the measured camera framing rate can be a maximum of 14% higher than the listed values.

Physical jet characteristics

As soon as the water jet emerges from the shaped charge the tip or front of the jet is ablated causing a shroud of small water particles to form around the main body of the jet. These particles are particularly prone to air drag and as a consequence they appear to move backwards and radially outwards relative to the main jet. This droplet shroud prevents conventional photography of the jet core (Plate 1). The products of ablation are present throughout the entire jet flight time but shortly after impact, the shroud disappears and an "apparently" coherent jet is displayed. The diameter of the coherent jet is seen to increase with gel thickness as would be expected (Plates 2, 3 and 4).

The theoretical length of the jet as predicted by Equation 4 is 50 mm. Photographic evidence suggests that this is an underestimate as the jet appears to have a length of at least 150 mm.

Impact pressure

The pressure-time traces obtained for jets from each gel thickness are shown in figs. 5, 6 and 7. The pressures are calculated assuming that the ablated jet front impacts over the entire 25 mm diameter anvil area. The sharp initial rise in the pressure traces is caused by the continual compression of the water jet until a state of shock wave release is achieved (ref. 6). In the jets produced by the 6 mm gel liners a secondary peak occurs approximately 100 µsecs after the first peak. This is thought to be caused by the impact of the slug, (the theoretical mass of the slug is presented in Table 1).

The trend displayed in the impact pressure results cannot be satisfactorily explained by either the water hammer (Eq. 7) or Bernoulli (Eq. 8) equations.

$$p = \rho_j c v_j \qquad \text{Eq. 7}$$

$$p = \frac{1}{2}\rho_j v_j^2 \qquad \text{Eq. 8}$$

The magnitude of the impact pressure appears to be primarily a function of how well the jet is formed. Variations in jet formation could produce non-uniform mass and velocity distributions both along and perpendicular to the axis of the jet. The results show much higher impact pressures and greater anvil pitting for the 1.5 mm and 3 mm liner jets than for the 6 mm liner jet.

The length of the pressure pulse and the area under the pulse appear to be controlled by the degree to which the slug complements the pressure peak obtained from the initial impact of the jet, i.e. they are dependent on the relative mass of the gel liner.

Impact damage

The type of damage caused to the anvil by the 1.5 mm and 3 mm liners is similar and can best be described as “multiple pitting” (Plate 5). This multiple pitting points to the fact that the jets are not continuous but are composed of small packets of water grouped together.

The jets formed by the 6 mm liner inflict negligible damage to the anvil (Plate 6). The ring impact around the anvil would seem to be the result of a jet with a hollowed out cone at the front. Chou et al (ref. 7) state that this cone is the result of a radial velocity component imparted on the water during the initial stages of formation of the shaped charge jet. When the liner material flows towards the collision point at a supersonic speed (relative to the liner material properties) a detached shock wave is formed at the collision point which imparts an outward radial velocity component to the water jet. In the 6 mm liner the increased mass of fluid at the collision point must exaggerate the radial velocity component to the extent that a hollow cone is formed.

CONCLUSIONS

Water jet velocities of over 9000 m/s have been produced using shaped charges lined with gelled water. Impact pressures of 850 N/mm^2 have been produced by these jets and there is an optimum liner thickness for high impact pressures.

ACKNOWLEDGEMENTS

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Cone angle (degrees)	Gel liner thickness (mm)	Gel liner volume (mm^3)	Gel liner mass (g)	Cone half angle a (degrees)	Cone collapse angle β (degrees)	Jet mass (g)	Slug mass (g)
60	1.5	3101	3.10	30	54	0.64	2.46
60	3.0	5639	5.64	30	54	1.16	4.48
60	6.0	9249	9.25	30	54	1.91	7.34

TABLE 1 PHYSCAL PROPERTIES OF SHAPED CHARGE CONFIGURATIONS

Test number	Gel thickness (mm)	Maximum velocity (m/s)	Impact velocity (m/s)	Deceleration point (mm)	Coherent jet (μsec)	Maximum pressure (N/mm^2)	Pressure pulse (μsec)	Area under pressure pulse ($N.\mu sec/mm^2$)	Maximum pit depth (mm)	Comments
1	1.5	8655	6755	205–245	–	522	64	12,885	2.3	
2	1.5	8480	6705	195–235	19	723	122	45,292	6.3	
3	1.5	9335	7745	170–210	23	598	–	–	2.3	Strain gauge broke after impact
4	1.5	9068	9060	Not present	17	806	90	24,423	12.3	One predominant pit
5	1.5	8920	7260	190–230	19	510	111	23,402	1.5	
Average	1.5	8890	7305	190–230	19.5	630	97	26,500	4.9	
6	3.0	7945	5760	210–240	20	675	129	26,709	7.4	One predominant pit
7	3.0	8540	6550	245–275	20	645	132	41,254	5.0	
8	3.0	8310	6330	200–240	28	853	98	26,416	6.3	One predominant pit
9	3.0	–	–	–	–	623	124	26,139	2.7	Jet not photographed
10	3.0	8225	6660	185–220	20	734	106	26,292	4.4	
Average	3.0	8255	6325	210–240	22	705	117	29,362	5.2	
11	6.0	8480	5915	180–220	32	383,303	172	25,823	1.1	Ring impact around anvil
12	6.0	8420	5290	180–215	29	287,526	146	28,519	0.8	" " " "
13	6.0	7660	4910	160–195	29	256,471	183	31,299	0.9	" " " "
14	6.0	8110	5600	185–215	33	271,463	158	28,618	0.3	" " " "
15	6.0	8140	5810	185–220	33	239,406	149	28,630	0.1	" " " "
Average	6.0	8160	5505	180–215	31.2	285,435	161	28,577	0.6	

Table 2. A Summary of the Test Results

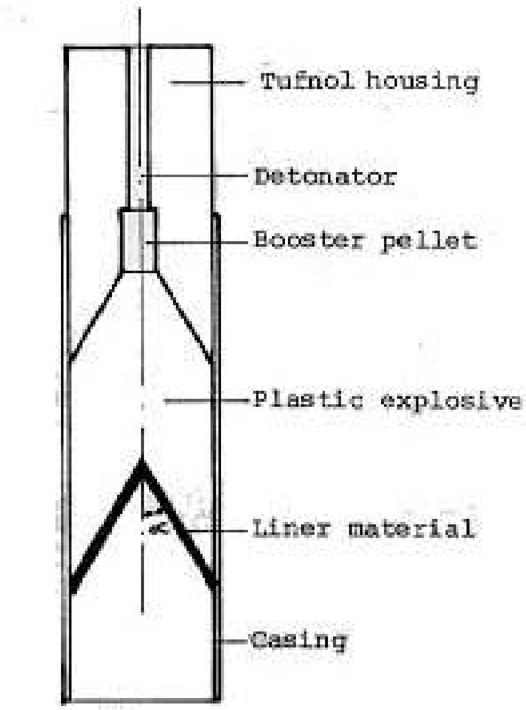

Fig. 1 A Typical Shaped Charge

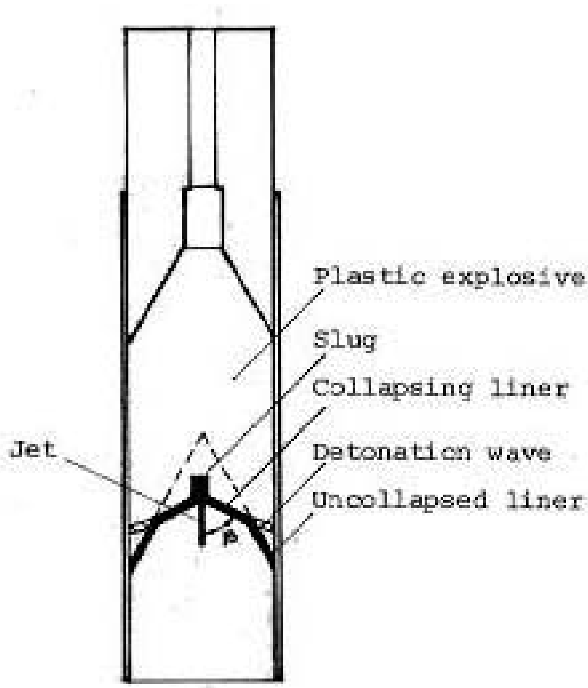

Fig. 2 A Collapsing Shaped Charge

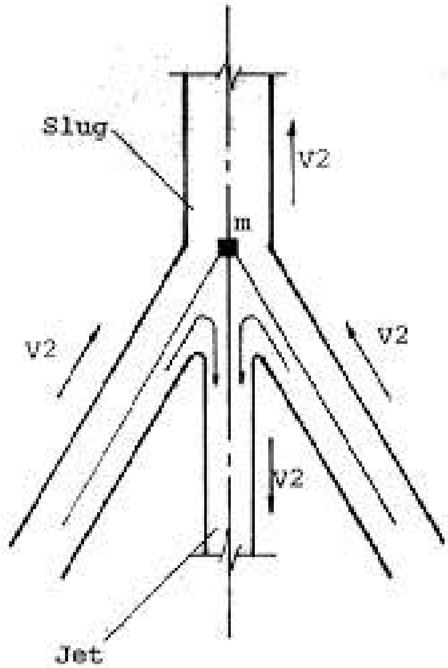

Fig. 3 Fluid Flow at Liner Collapse

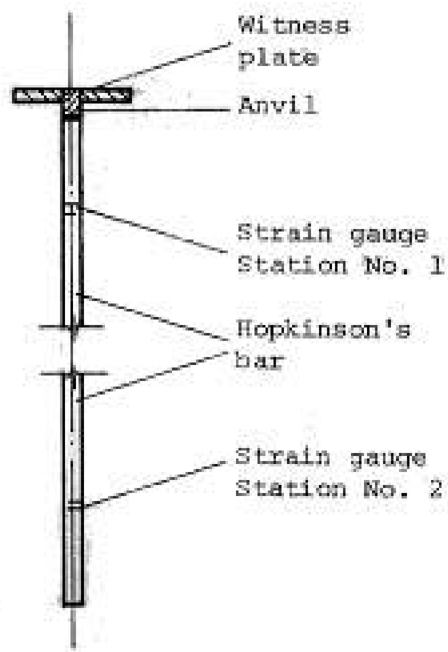

Fig. 4 Hopkinson Bar Arrangement

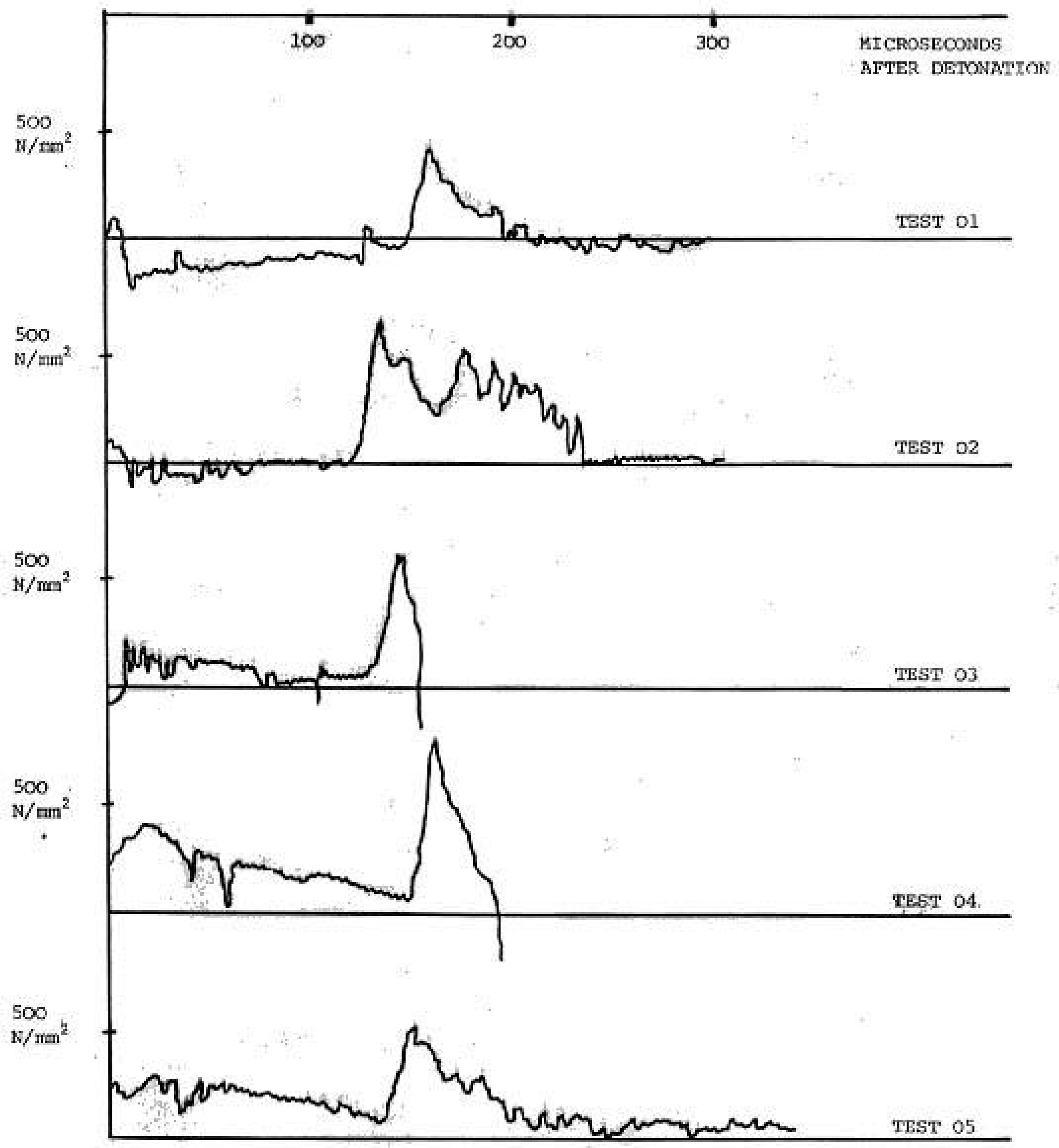

Fig. 5 Pressure Time Traces for 1.5 mm Liner Thickness

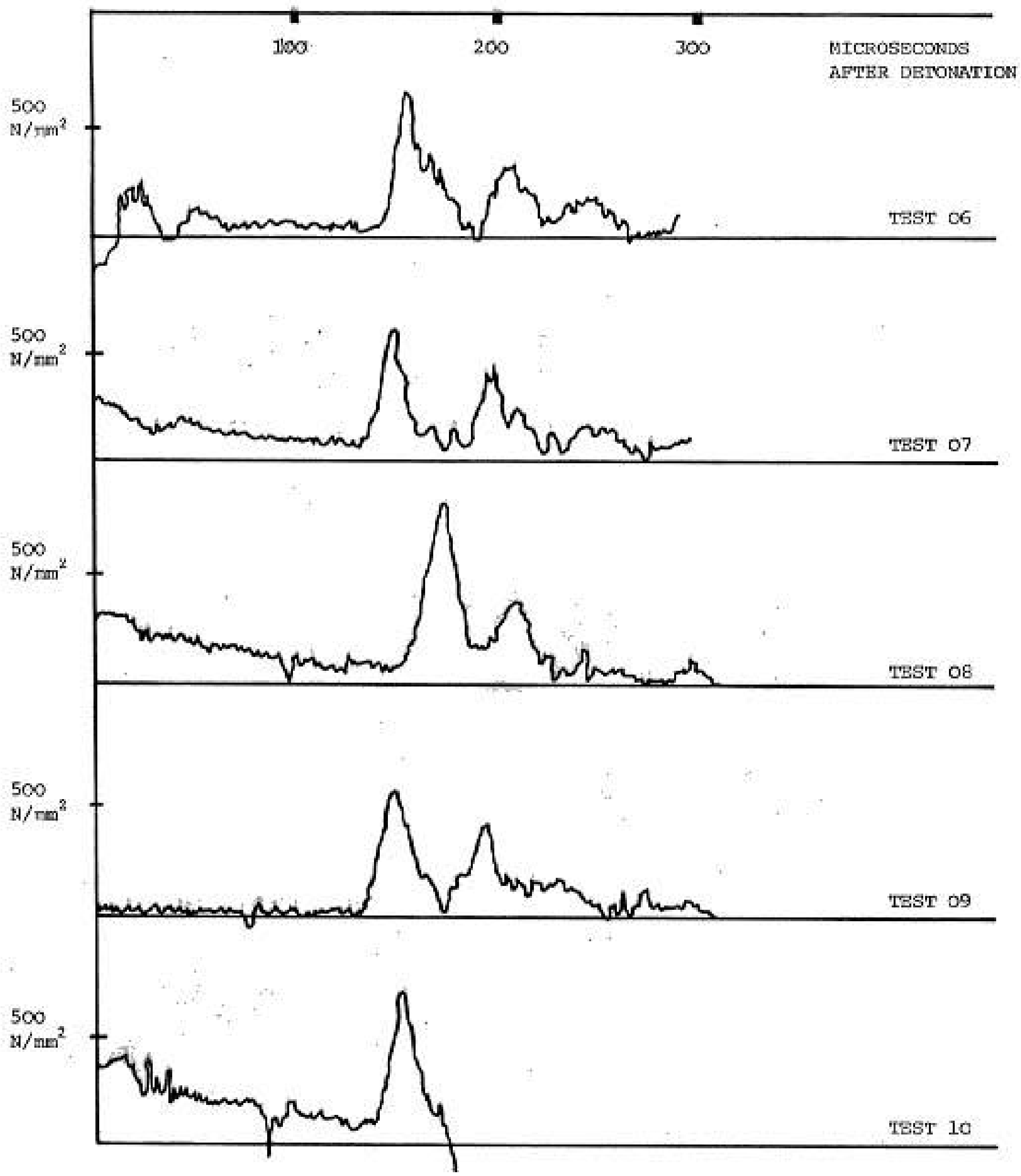

Fig. 6 Pressure Time Traces for 3mm Liner Thickness

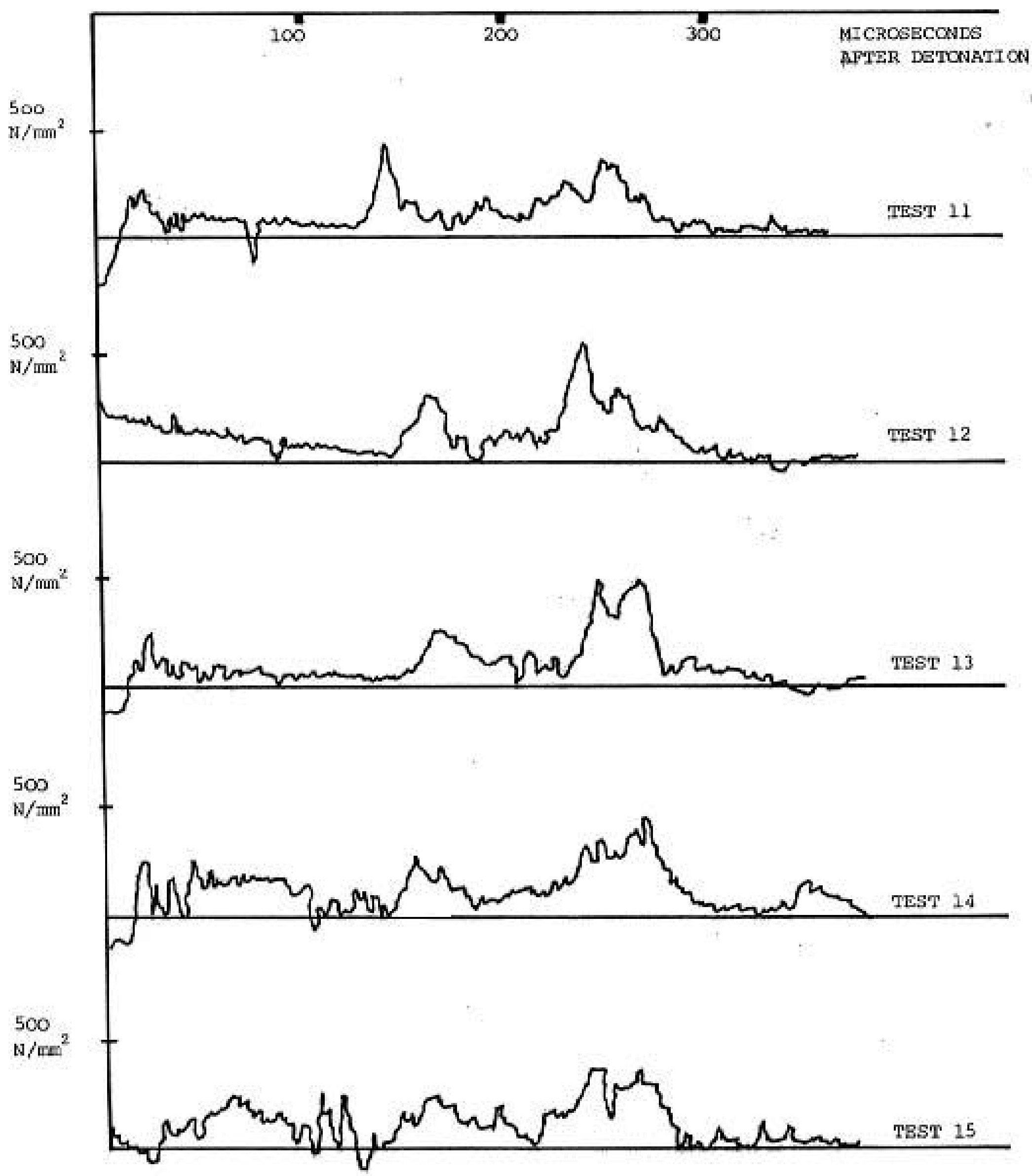

Fig. 7 Pressure Time Traces for 6mm Liner Thickness

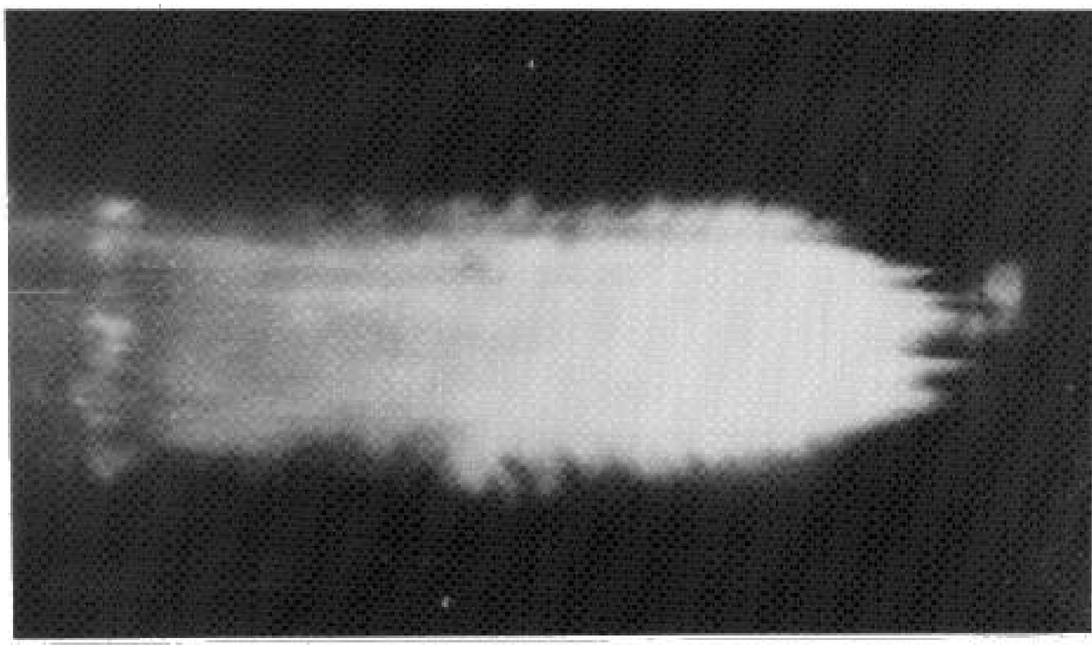

Plate 1 Waterjet in flight (TEST 7) showing "fingers" and droplet shroud at front of jet

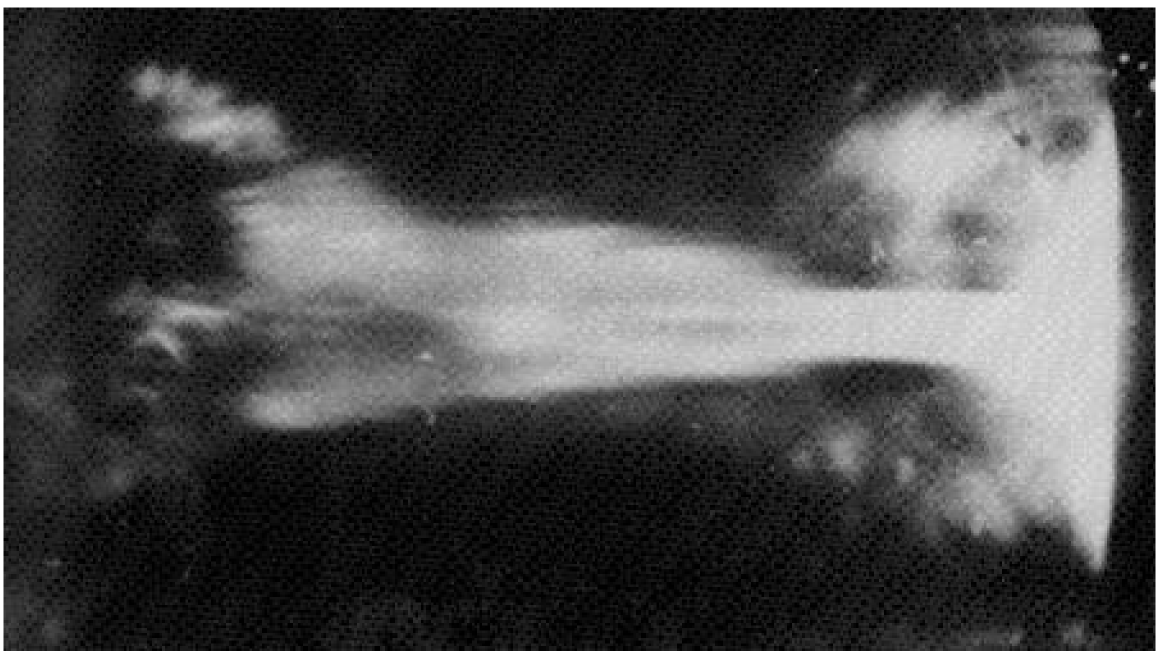

Plate 2 Impact of jet formed by 1.5m liner thickness (TEST 4)

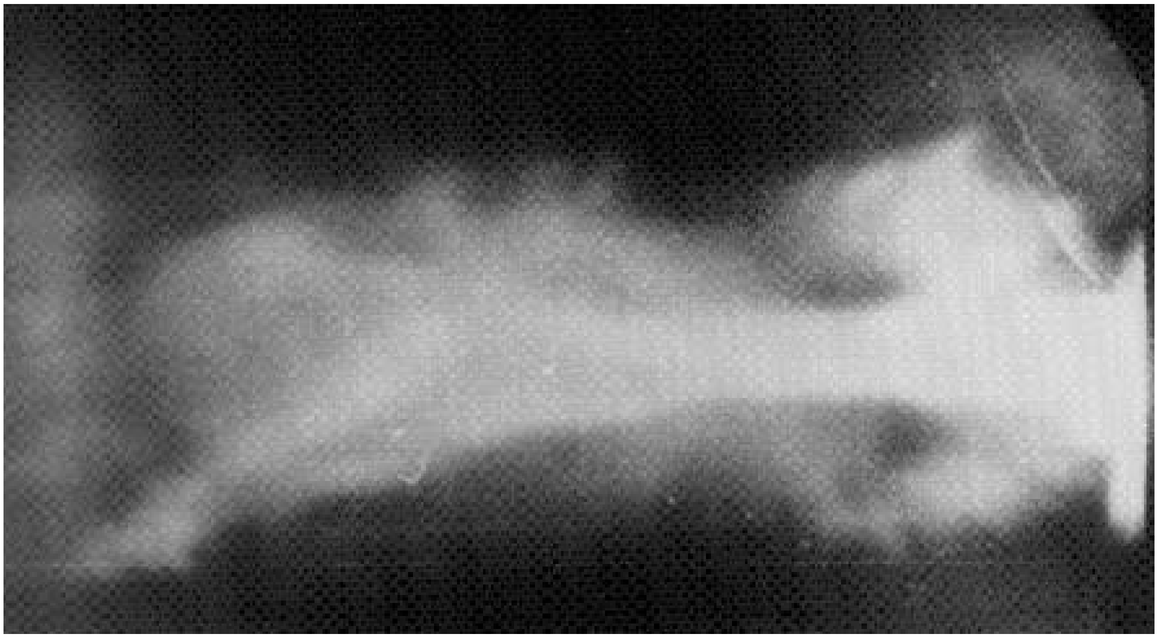

Plate 3 Impact of jet formed by 3mm liner thickness (TEST 10)

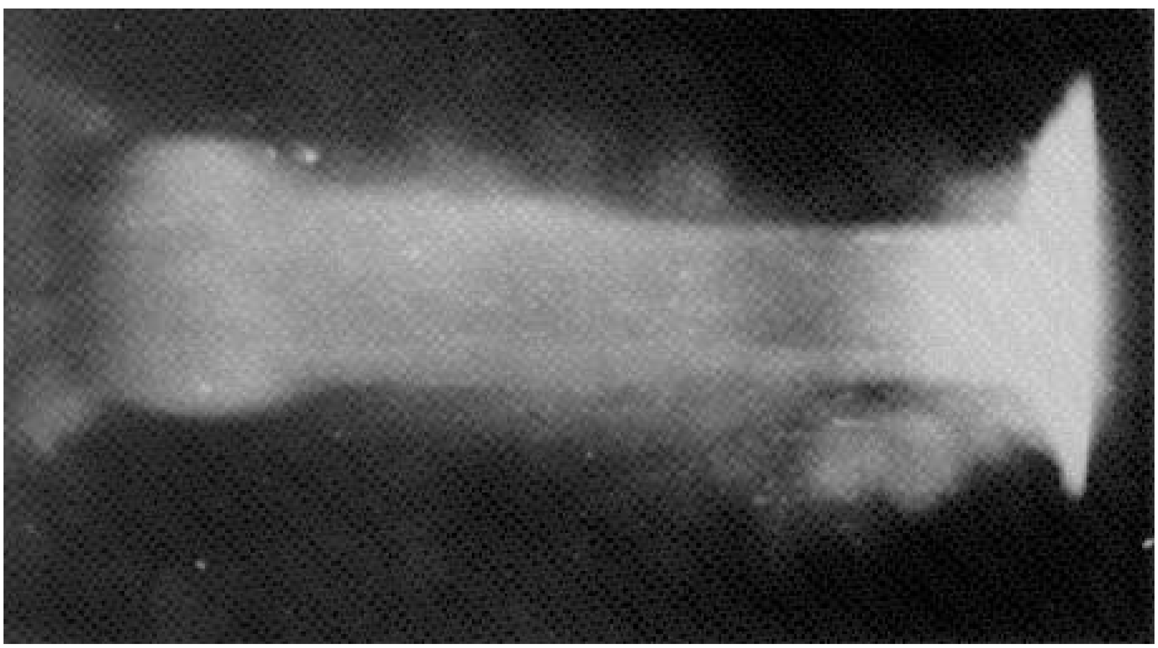

Plate 4 Impact of jet formed by 6 mm liner thickness (TEST 11)

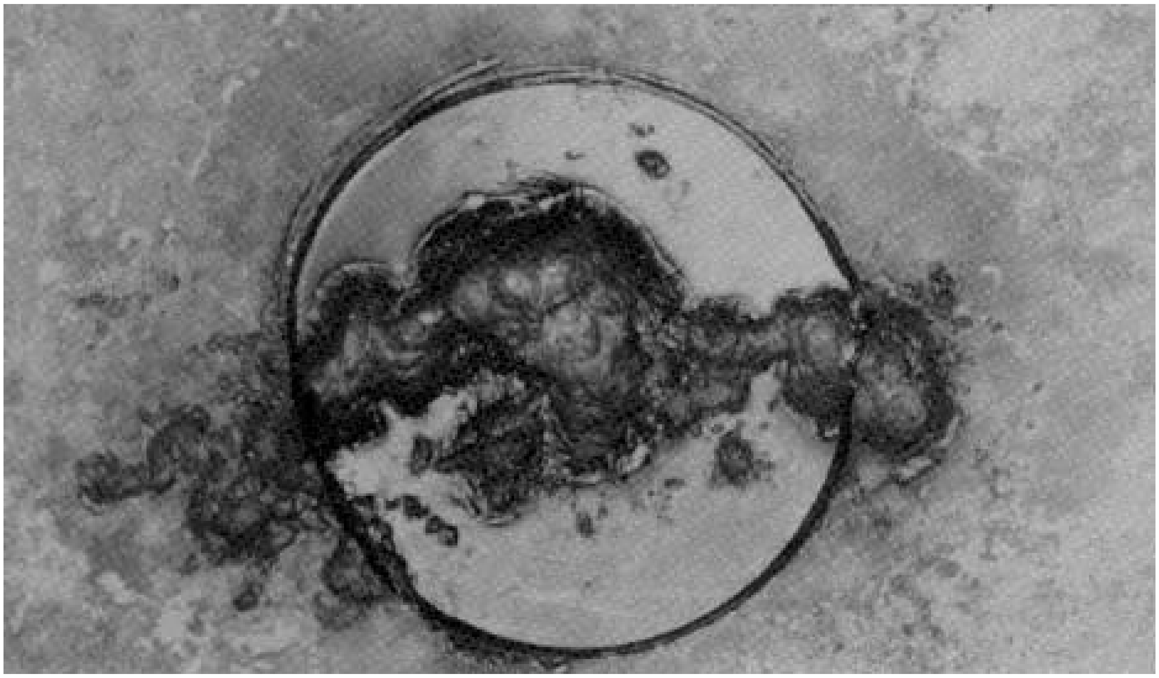

Plate 5 Anvil Damage (TEST 2)

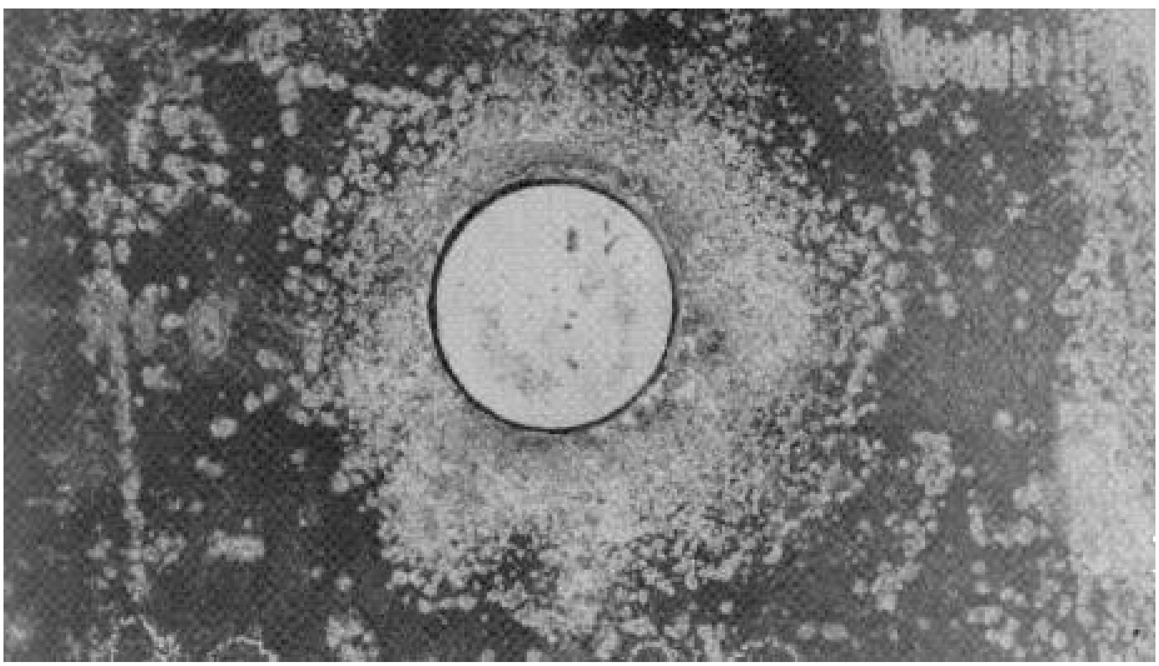

Plate 6 Anvil and witness plate damage (TEST 14)

INVESTIGATION ON ANATOMY OF CONTINUOUS WATERJET FOR UPDATING JET PERFORMANCE

by

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ABSTRACT

In the present paper the anatomy of a continuous waterjet is studied with a model of homogeneous atomization and the integral method of calculation, based on which new formulae for determining the length of jet initial section, and the boundary and distribution of dynamic pressure in the jet core are derived. As a result, an optimal stabilizer is suggested for updating the jet performance.

NOMENCLATURE

- b - thickness of mixing section of jet, $b = y_e - y_i$
- C_o - empiric constant characterizing initial disturbance;
- d_o - nozzle exit diameter;
- L - length of initial section of jet, $\overline{L} = \frac{L}{r_o}$
- P_o - exit dynamic pressure of jet;
- P_m - main section dynamic pressure in the jet core;
- r_o - nozzle exit radius;
- u, v - component of average velocity on axis x & y;
- u_o- outlet (or discharge) velocity of jet;
- u_m- main section velocity in the jet core;
- x, y - abscissa, ordinate of Cartesian coordinates, $\overline{x} = \frac{x}{r_o}$
- y_e - distance between x axis and outer boundary of jet mixing section;
- y_e'- distance between x axis and actual boundary of jet main section;
- y_{eo}- distance between x axis and outer boundary of mixing section of jet transition cross-section area $y^*_e = \frac{y}{r_o}$, $y^*_{eo} = \frac{y_{eo}}{r_o}$, $\overline{y}_e = \frac{y_e}{y_{eo}}$
- y_I - distance between x axis and inner boundary of jet mixing section;
- α - empiric index characterizing boundary growth of actual jet main section;
- ε - viscosity coefficient of vortex;
- η - dimensionless ordinate of jet initial section, $\eta = \frac{Ye - y}{b}$
- K - average mass concentration;
- K_o- mass concentration at nozzle exit, $\mu = \frac{\rho_a}{\rho_w}$

ξ - dimensionless ordinate of jet main section, $\xi = \dfrac{Y}{Ye}$

ρ - average density anywhere in the flow field;

ρ_o - density at nozzle exit, here $\rho_o = \rho_w$;

ρ_a - density of air;

ρ_m - density of water in the core of jet main section;

ρ_w - density of water;

T - temperature;

θ - angle of diffusion of waterjet.

INTRODUCTION

High-pressure water jets, as newly developed techniques, are increasing attractive. These techniques show great potential for cutting and cleaning (so-called 'liquid blasting'), because they have many noticeable advantages, such as working efficiency, costs and safety, over conventional cutting tools.

However, water jet technology, on the whole, is still in its infancy. Development of experimental, and in particular, theoretical investigations are of great importance for updating performance and cutting capabilities of water jets. To approach this aim, the authors have further studied jet anatomies using state-of-the-art instruments to achieve better understanding and in evaluating the effects of many different factors (parameters) on jet performance. This will also provide the basis for predicting and designing waterjet devices.

As a rule of thumb, calculation procedures of two-phase flow (i.e. the high-speed water jet stream and the low-speed, even quiescent, atmospheric air medium) are based on empiric formulae or semi-empiric predictions to date. However, the situation appears to have improved in recent years. The earlier researcher G.N. Abramovich has successfully calculated velocities in the jet core using integral method of calculation (ref. 1). Shavlovsky, HaiChow Li have also suggested several empiric formulae (ref. 2,3). Recently, Yanaida has conducted precise measurement of the two-phase flow (ref. 4). Summarizing test data using the Abramovich method of calculation, Yanaida presumably suggested a novel theoretical model, and then, derived a formula for calculating the distribution of dynamic pressure in the high-speed jet core. But as you know, this procedure seems to be quite complicated for practical use (ref. 4, 5, 6).

In addition, a profusion of experimental studies has been focussed on identifying the effects of parameters (especially, the optimal nozzle shape and size) on jet performance. On the other hand, flow conditions prior to inserting the water into the nozzle have demonstrated non-negligible effects on the jet structure, for example, the effect of initial disturbances on the diffusion process of the jet stream. Unfortunately, any satisfactory solution to these problems has not yet been obtained. Therefore a usual water jet stream would be swiftly diffused through the surrounding air, provided that stand-off distance or jet was enlarged to a certain extent. As a result, it will lead to a decrease in cutting effectiveness, such as the rate of advance, depth of cut or penetration rate.

As is usually the case; swing and oscillation (or rotation) will lead not only to a considerably increase in the productivity of coal by the jet roadheader, but will also reduce the effective stand-off distance of the jet by intensifying the jet stream diffusion.

Recognizing the need and great importance of improving jet performance, the authors initiated major efforts in an extensive investigation to introduce a stabilizer into the inlet tube prior to the nozzle. Theoretical calculation procedures, test set-up, instruments etc. will be described in this research report.

CALCULATION OF WATER-JET VELOCITY FIELD

In general, the waterjet stream will speed up steadily as it goes through the nozzle. The initial disturbance and the concomitant vortexes are enhanced with the increase of water stream velocity during flow. The surrounding air will be entrained into the high-speed jet stream due to intensive tranverse exchanges of mass and momentum between water and air particles. Experimental observations show (Fig. 1-a) that the jet initial section will consist of a potential-flow core of constant dynamic pressure (i.e. constant velocity) and two-phase water-air boundary layers, which will be formed after the jet stream is a certain distance from the nozzle exit. The interaction of the waterjet with air will result in dispersion of the jet stream itself into droplets of quite different sizes within the boundary layers. Furthermore, along the horizontal axis of the velocity field, as shown in Fig. 1: the smaller the horizontal distance from the inner boundary C_A, the bigger the water droplet size. And vice-versa: the greater the distance from C_A, the finer the water droplet. The potential-flow core of constant velocity ends in the transition plane BAB due to the continuous entrainment of air. And then, with the water jet stream mixing fully with the air, the jet main section[1] occurs beyond plane BAB. In this section the width of jet stream is gradually expanded, and consequently the dynamic pressure in the jet core is gradually reduced. As is the case for the mixing area of the initial section, the jet stream is disintegrated by the entrained air into droplets: the greater the distance from the plane BAS, the finer the water droplet. Finally, the waterjet stream is completely atomized.

Disintegration of the waterjet stream into droplets in turbulent flow is an extremely complicated process. It is necessary to employ a model of homogeneous atomization as shown in Fig. 1-b, to determine the length of the jet, representing the jet structure and being an important parameter of the hydraulic monitor. This model postulates that the finest particles of the water mist are uniformly mixed with the air, and the density of this water-air mixture will be described later.

In addition, some presumptions and approximation methods are used to abbreviate the theoretical calculation; such as:

1. The effect of viscous force in neglected, because the flow condition involved is a well-developed turbulent flow.
2. Owing to the lack of a pressure gradient, static pressure throughout the flow field is presumably the same as atmospheric pressure.

[1] Note: Here the transition section between the initial and main sections was neglected in order to simplify the engineering predictions and calculations (ref. 7).

3. The waterjet stream at the nozzle exit being uniform, an empiric constant is used to represent the initial disturbance for its nonuniformity correction (ref. 7).
4. In view of the very high velocity of the water jet, gravitational effects can be neglected in comparison with the inertial force.
5. The jet flow field involved is within the region of so-called 'turbulent automodelling'.

Schlichting's velocity profiles testified by experimental studies are employed and written as follows (ref. 1,8):

$$\frac{U_o - U}{U_o} = \left(1 - {}^{3}\!/_{2}\right)^2 \qquad (1)$$

The linear concentration distribution profiles are written as below :

$$\frac{K_o - K}{K_o} = 1 - \eta \qquad (2)$$

By definition, a correlation of the density distribution with the mass concentration distribution of the water droplets will be found. The ratio of density any where in the flow field to water density at the nozzle exit can be written as:

$$\frac{\rho_o}{\rho_w} = \frac{\mu}{1 + (\mu - 1)K} \qquad (3)$$

The continuity equation of the water-air two-phase flow field can be written as

$$\frac{\partial}{\partial x}(\rho u y) + \frac{\partial}{\partial y(\rho u y)} = 0 \qquad (4)$$

The diffusion equation of momentum and boundary conditions are as below:

$$\rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial y} = \frac{1}{y}\frac{\partial}{\partial y}\left(y \rho \varepsilon \frac{\partial u}{\partial y}\right) \qquad (5)$$

and

when

$$\left.\begin{aligned} y = 0,\ u = u_m,\ \frac{\partial u}{\partial y} = 0, \\ y = y_e,\ u = 0,\ \frac{\partial u}{\partial y} = 0, \end{aligned}\right\} \qquad (6)$$

Using transformation, the conservation equation of momentum of jet stream can be written

$$\int_0^{y_e} \rho u^2 y dy = \frac{\rho_w u_0^2 r_0^2}{2} \qquad (7)$$

Using Eq. (7), the momentum conservation equation between the nozzle outlet plane and any plane inside the initial section can be obtained:

$$\rho_w u_0^2 \pi r_0^2 = \int_0^{y_e} \rho u^2 \left(2\pi y\right) dy = \int_{y_i}^{y_e} \rho u^2 (2\pi y) dy + \rho_w u_0^2 \pi y_i^2 \tag{8}$$

Replacing by y = y_e - b and t_e = y_i + b, and substituting Eqs. (1)-(3) into Eq. (8), a quadric algebraic equation of y_i

$$y_i^2 + A_1 b y_i + (A_1 - A_2) b^2 - Y_0^2 = 0 \tag{9}$$

where

$$A_1 = 2\mu \int_0^1 \frac{\left[1 - \left(1 - \eta^{3/2}\right)^2\right]^2}{1 - (1 - \mu)\eta} d\eta$$

$$= -2\mu\left\{\frac{11}{18(1-\mu)} + \frac{37}{35(1-\mu)^2} + \frac{53}{20(1-\mu)^3} + \frac{12}{3(1-\mu)^4} - \frac{15}{2(1-\mu)^5} + \frac{1}{(1-\mu)^6} + \frac{\ln\mu}{(1-\mu)^7} + \frac{4}{(1-\mu)^{5.5}} \ln\frac{1+\sqrt{1-\mu}}{1-\sqrt{1-\mu}}\right\}$$

$$A_2 = 2\mu \int_0^1 \frac{[1 - (1 - \eta^{3/2})^2]^2 \eta}{1 - (1 - \mu)\eta} d\eta$$

$$= -2\mu\left[\frac{32}{77(1-\mu)} + \frac{11}{18(1-\mu)^2} + \frac{37}{35(1-\mu)^3} + \frac{53}{20(1-\mu)^4} + \frac{12\ln\mu - 7}{3(1-\mu)^5} - \frac{15}{2(1-\mu)^6} + \frac{1}{(1-\mu)^7} + \frac{\ln\mu}{(1-\mu)^8} + \frac{4}{(1-\mu)^{6.5}} \ln\frac{1+\sqrt{1-\mu}}{1-\sqrt{1-\mu}}\right]$$

Employing the characteristic velocity U* and model of 'momentum velocity', the thickness of the mixing section of the jet can be determined (ref. 7).

Assuming

$$u^* = \frac{\int_0^b \rho u^2 dy}{\int_0^b \rho u dy} \tag{10}$$

Therefore

$$b = cF(\mu)x, \tag{11}$$

where e is a constant, and

$$F(\mu) = \frac{u_o}{u^*} = 2\mu \frac{A_3}{A_1}$$

where

$$A_3 = \int_0^1 \frac{1 - (1 - \eta^{3/2})^2}{1 - (1 - \mu)\eta} d\eta = -\frac{1}{1-\mu} - \frac{7}{2(1-\mu)^2} + \frac{1}{(1-\mu)^3} + \frac{1}{(1-\mu)^4} \ln\mu + \frac{2}{(1-\mu)^{2.5}} \ln\frac{1+\sqrt{1-\mu}}{1-\sqrt{1-\mu}}$$

In the case where the medium is of the same phase (i.e. water jets spurting into quiescent water), μ= 1. Test results show that the thickness of the jet mixing section is

$$b = c_o x \qquad (12)$$

where, c_O is an empiric constant representing the initial disturbance of the turbulent flow.

By comparing Eq. (11) with Eq. (12), we obtain

$$C = \frac{Co}{F(\ell)} \qquad (13)$$

where F(1) = 1.32344.

To solve Eq. (9) using Eq. (11), the length of jet initial section can be written

$$L = \frac{A_1}{2C\mu A_3\sqrt{A_1 - A_2}} \qquad (14)$$

and the ordinates of the outer boundary of the jet initial section are

$$y^* = \theta_1\bar{x} + \sqrt{1 - \theta_2\bar{x}^2} \qquad (15)$$

where

$$\theta_1 = 2c\mu\left(1 - \frac{A_1}{2}\right)\frac{A_3}{A_1}$$

$$\theta_2 = 4c^2\mu^2\left(A_1 - A_2 - \frac{A_1^2}{4}\right)\frac{A_3^2}{A_1^2}$$

Test results indicate that the outer boundary of the main section of the atomized jet is a straight line. Thus using a calculation method described in the literature (ref. 9), (i.e. using a tangent at the margin points of transition plane BAB for the approximate substitution of the boundary of the jet main section), the ordinates of the jet main section boundary can be written as:

$$\bar{y}_e = K_o + \delta_o\bar{x} \qquad (16)$$

where

$$Ko = 2\left(1 - \frac{A_2}{A_1}\right)$$

$$\delta_o = 2c\mu\frac{A_3}{A_1}\left(\frac{2A_2}{A_1} - 1\right)\sqrt{A_1 - A_2}$$

For T = 200 C, $\mu = 0.001207173$
$A_1 = 0.01376$
$A_2 = 0.01277$
$A_3 = 6.00129$

Therefore

$$\bar{L} = \frac{40}{Co} \tag{17}$$

and
$$Y_e = 0.144 + 0.0214\, C_o X \tag{18}$$

When $C_o = 0.27$, we get tan = 0.186. However, using test data on spraying water jet given by Yanaida, we get tan = 0.144. This means that comparing our theoretical calculation with test data, the error is about 20%.

Owing to the discrepancy between the theoretical flow-model of homogeneous atomization and an actual water jet stream (as shown in Fig. 1-a), the boundary of the jet main section will no longer be a straight-line. Therefore, it is necessary to introduce an empiric index characterizing this boundary growth of the actual jet main section. This means that the ratio of the ordinate of the boundary of the main section to that of the boundary in proximity to plane BAB varies with an index law, written as

$$\bar{Y}_e = \bar{Y}_e^{\alpha} \tag{19}$$

where, Y'e = Ye/y_{eo}, Y'e is the ordinate of the boundary of the actual water jet.

Employing Eq. (7), the momentum conservation equation between the main section and the transition plane BAB can be derived in order to determine the dynamic pressure in the jet core of main section, thus,

$$2\pi\int_0^{Y_{eo}} \rho u^2 y dy = 2\pi\int_0^{Y'_e} \rho u^2 y dy \tag{20}$$

According to test data described in the literature (ref. 3), within a certain range of random error there exists, on the whole, similar dynamic pressure profiles (i.e. distributions) as follows:

$$\frac{\rho u^2}{\rho_m U_m^2} = \phi(\xi) \tag{21}$$

From Eqs. (20) and (21), we obtain

$$\frac{\rho_m U_m^2}{\rho_o U_0^2}\left(\frac{Y'_e}{Y_{eo}}\right)^2 = 1 \tag{22}$$

In view of Eqs. (18) and (19), distribution of dynamic pressure in the jet core can be written in the following form:

$$\frac{P_m}{P_o} = \frac{\rho_m U_m^2}{\rho_o U_0^2} = \frac{1}{(0.144 + 0.0214 Co\bar{\bar{X}})^{2\alpha}} \qquad (23)$$

TEST FACILITIES AND PROCEDURES

Experimental investigations by the authors consisted of two main components: (a) measurement of dynamic pressure in the jet core and (b) optimization of stabilizer design.

Part (a): Intensity and compactness of waterjet stream are characterized by the dynamic pressure in the jet core, of which the measuring system is shown in Fig. 2. A three-plunger high-pressure pump rated at a pressure of 350 bar and output of 75 kw was used for this test. Pipeline consists of high-pressure hose, quarter bend, inlet tube, lance and nozzle-holder as well. A high-speed waterjet issueing from the nozzle goes through a certain stand-off distance in atmospheric air and enters a center hole of 0.3 mm diameter on the target, where, the water jet strikes the strain gauge transducer. Dynamic pressure in the core of the water jet stream is tranformed into voltage using a transducer. Readings of voltmeters show the variation of dynamic pressure with different stand-off distances including the time-mean maximum and minimum values.

The-holder can be adjusted in three directions. One voltmeter is parallelly connected with the transducer for indicating stand-off distance adjustment. Stand-off distance in this test was adjusted within O to 1,000 mm, The nozzle conic angle was 13^{o} .

Part (b): In order to optimization stabilizer design, including type, size and location of the stabilizer, as well as the shape and size of the inlet tube, *- type and sieve-type stabilizers and inlet tubes of various sizes were tested and measured repeatedly. High-speed camera equipped with common lens and telescopic lens were employed for further observation of jet structures and their variations.

TEST RESULTS AND DISCUSSION

More than ten curves were selected from about 200 experimental curves and classified into four groups.

In order to eliminate the effect of the additional energy dispersion caused by disturbance due to the lack of stabilizer, and evaluate the effect of inlet tube design on the jet anatomies, dimensionless parameters, such as C_O, α , were introduced in calculations.

By comparison of theoretical calculation with test results, as shown in Fig. 3, some theoretical curves were obtained on the basis of different values of C_o and α . Where, C means that (1) the initial disturbance prior to the nozzle exit, (2) discrepancy of model, i.e. degree of atomization.

As shown in Fig. 5, when $C_o = 0.27$ and $\alpha = 0.551$, the curves of the author's theoretical calculation were perfectly matched with test results by Yanaida and Hai Chow Li (ref. 3,4); and when $C_o = 0.13$ to 0.26; and $\alpha = 0.56$ to 0.68, results of theoretical calculation conformed with that of tests by these authors.

It is essential to indicate that: (1) nozzle quality and jet performance are very well; (2) this method is quite suitable for demonstrating and evaluating jet anatomies.

STABILIZER DESIGN AND ITS COLLIMATING EFFECT

Intensive disturbance and concomitantly axial and radial vortexes were formed when the high-speed water went through the complex pipeline and nozzle. High-speed photos have also shown that the total water jet stream consisted of several finer water stream filaments which fluctuate up and down frequently and result in a serious vibration of the pointer of the voltmeter. Digital voltmeter measurements showed that the difference between the maximum and minimum voltage is more than ten times, but the time-mean voltage is considerably decreased.

The axial vortex which accompanied water flow could be step-by-step collimated and water flow could be, of course, restored into natural quiescence, provided that the lance was long enough. In general, the lance length required is 35 to 50 times larger than the lance diameter. The longer the lance length, the better the collimating effect.

However, this is often unacceptable from a mechanical point of view. If the lance length is too long, complications will be raised, such as, intensive vibration of the lance, serious fatigue and even failure (breakage). The way of solution seems to fix a stabilizer in the lance to reduce its length. The better collimating effect, as author's tests verified, would be obtained, provided that the ratio of lance length of its diameter is set within 15 to 20.

Tests have also verified that the better type of stabilizer appears to be the vane-type, such as fish-shape etc. Fig. 5 shows the stabilizer of * shape. It is necessary to call attention to

(1) the arrangement of stabilizer blades (vanes) has to be geometrically symmetric and uniform;
(2) radial arrangement of the longer and the shorter vanes crosswise seems to be promising;
(3) stabilizer length required is 2.5 to 4 times larger than its diameter.

The positioning of the stabilizer in the lance is quite critical for the collimating effect. The stabilizer must be fixed in the lance a certain distance from the nozzle-holder. The distance stated above should be selected to be 1.5 to 2 times greater than the lance diameter, as shown in Fig 5.

Comparing test results with or without a stabilizer, other parameters being identical, the effect of the stabilizer on waterjet performance is quite obvious even surprising, such as:

(1) length of jet initial section is increased by 50% to 60% even more;
(2) attenuation of dynamic pressure in the jet core speed-down, as shown in Fig. 6, the dynamic pressure in the jet core has increased by a factor of up to two at stand-off distance of about 1,000 mm;
(3) angle of diffusion of waterjet seems to be reduced.

Field tests have also demonstrated that jet performance would be improved and that the reasonable working range of the waterjet would be expanded on the basis of good design and correct positioning of the stabilizer. For example, using a stabilizer, the effective stand-off distance of the jet roadheader is increased by 55 to 70% and the capability for cutting coal is increased by a factor of two to three, as shown in Fig. 6.

CONCLUSIONS

(1) A good agreement of theoretical calculation with test data has reached through this extensive investigation, and empiric constant representing jet anatomies has been revealed its acceptability.

(2) By taking advantages of the stabilizer, both engineering and economic profit has been made on some respects, such as, improvement of jet quality, increase in effective stand-off distance of jet and capability of cutting coal.

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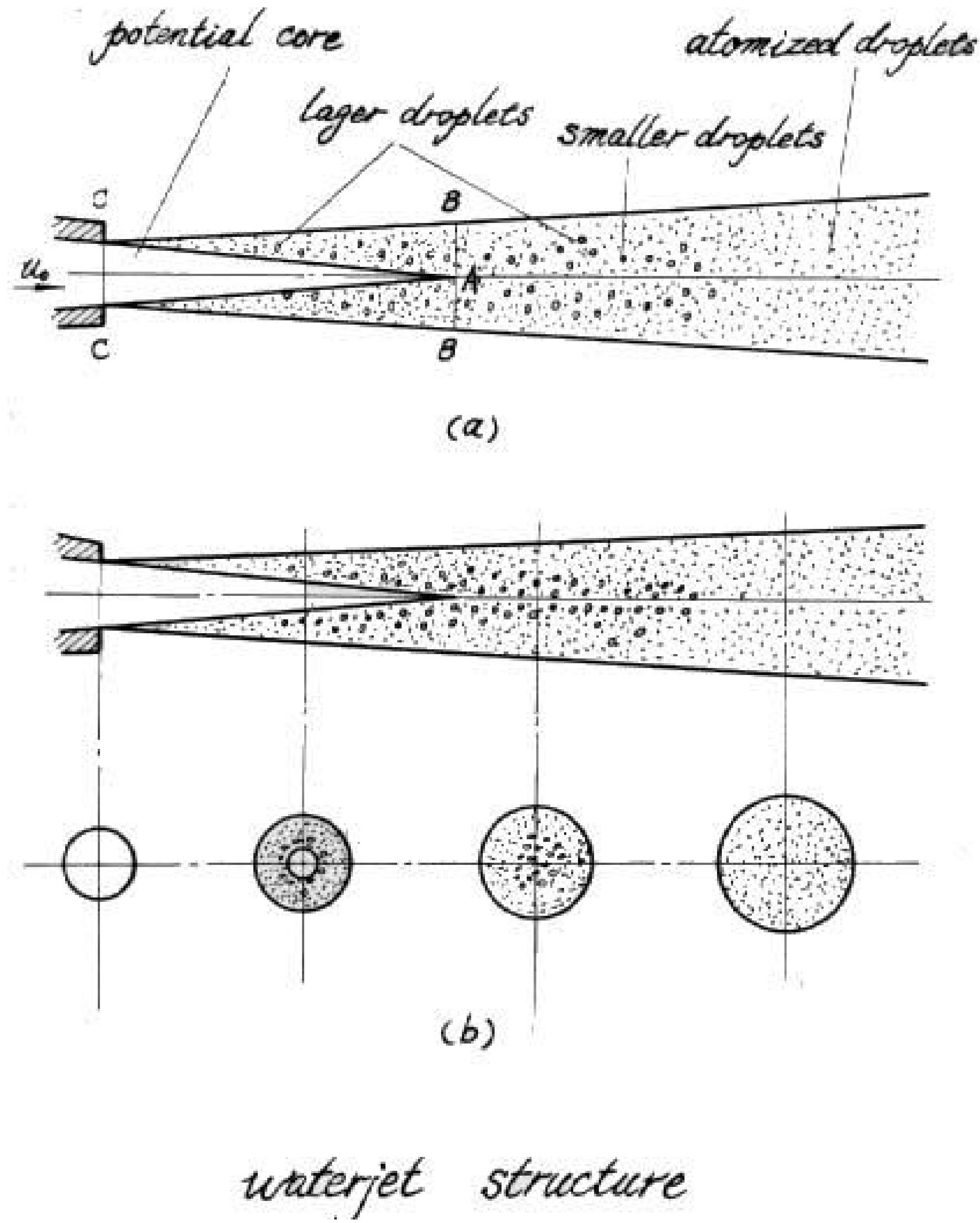


Fig.1

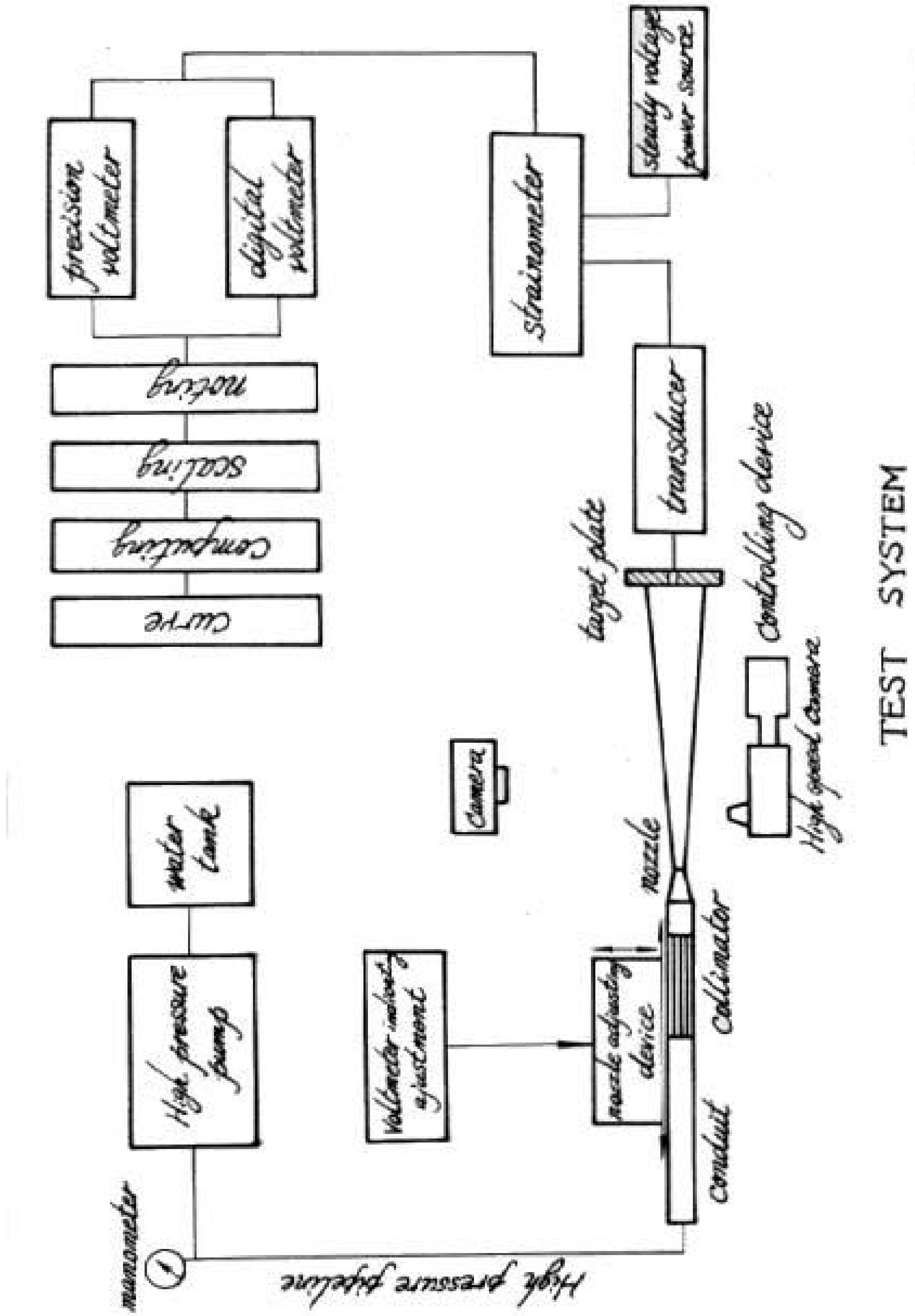
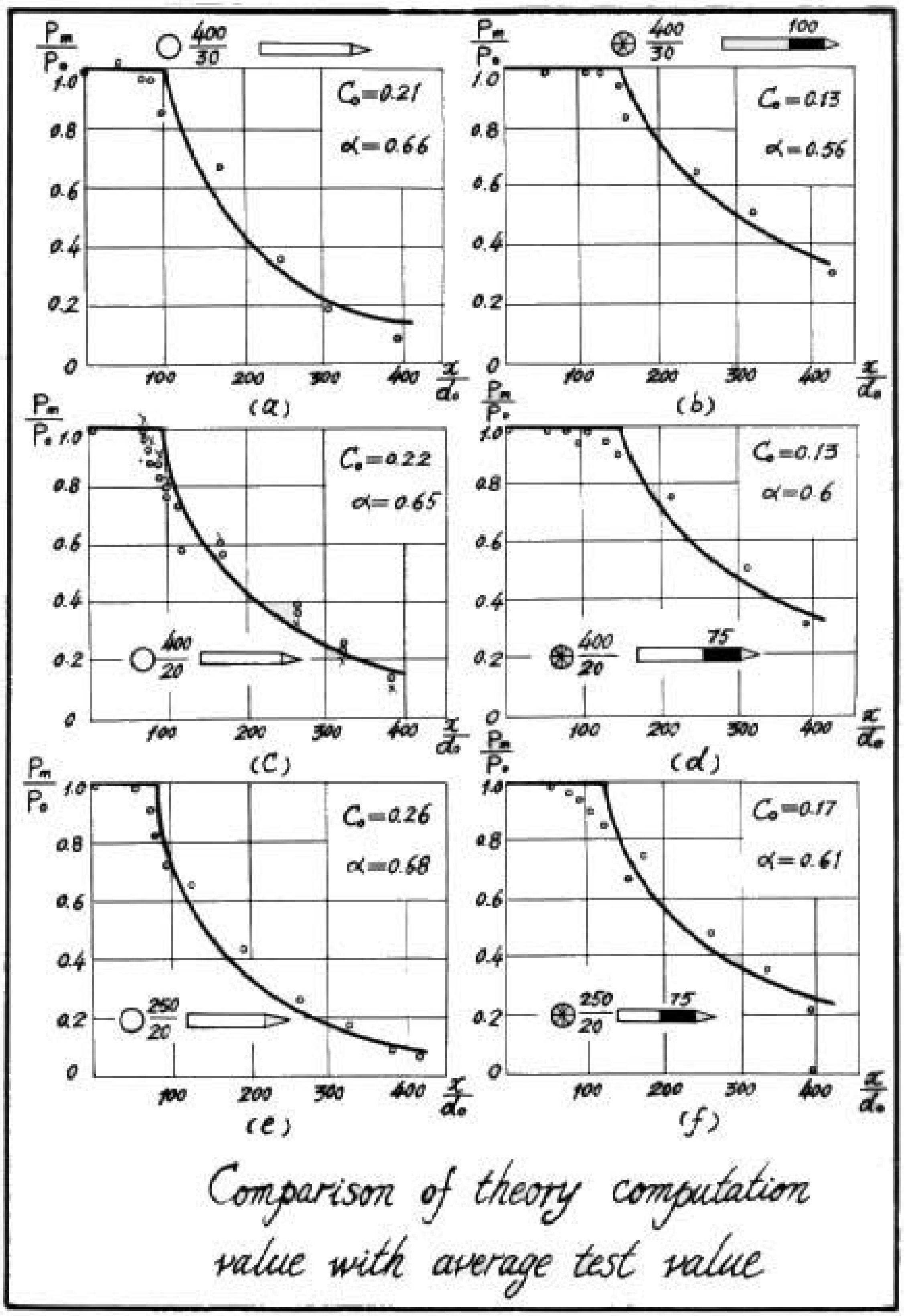
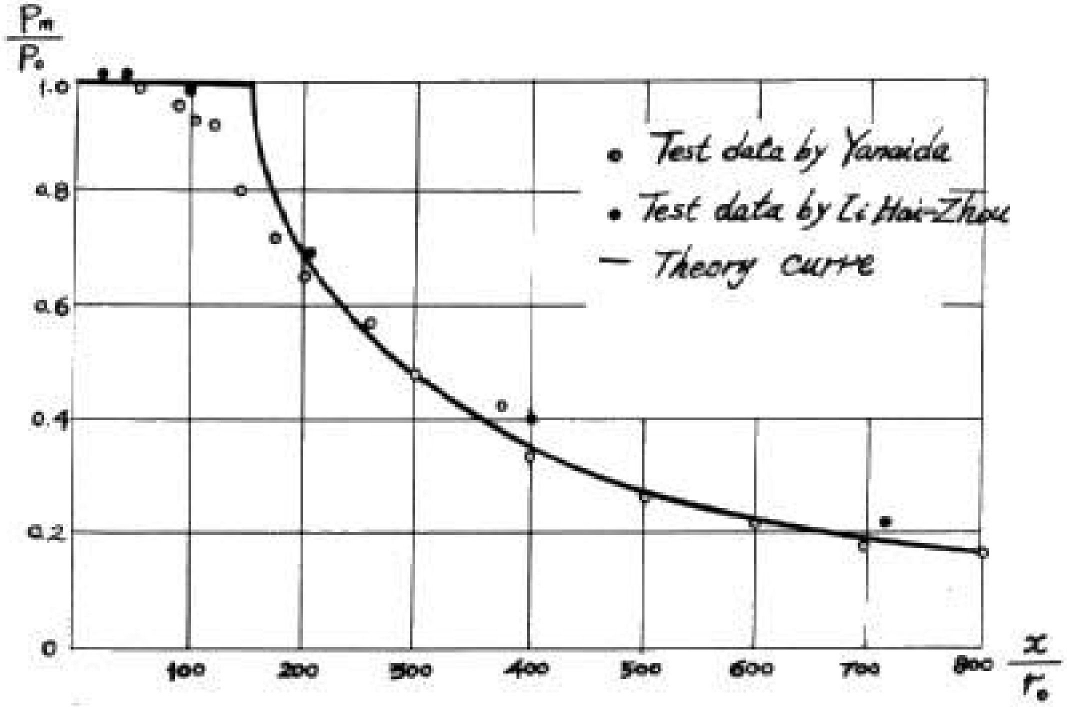

precision voltmeter
digital voltmeter
strainometer
steady voltage power source
noting
scaling
computing
curve
transducer
target plate
controlling device
TEST SYSTEM
High speed camera
camera
nozzle
collimator
water tank
High pressure pump
Voltmeter indirect ajustment
nozzle adjusting device
conduit
manometer
High pressure pipeline

Fig. 2



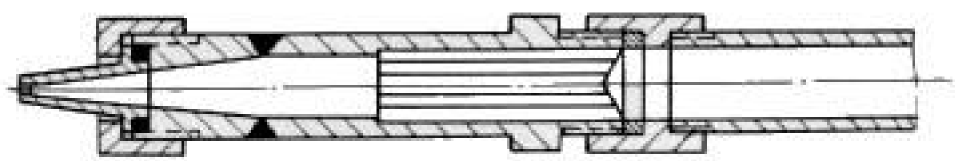
$\frac{P_m}{P_o}$
$\frac{400}{30}$
$C_o = 0.21$
$\alpha = 0.66$
(a)
$\frac{400}{30}$
100
$C_o = 0.13$
$\alpha = 0.56$
(b)
$\frac{400}{20}$
$C_o = 0.22$
$\alpha = 0.65$
(c)
$\frac{400}{20}$
75
$C_o = 0.13$
$\alpha = 0.6$
(d)
$\frac{250}{20}$
$C_o = 0.26$
$\alpha = 0.68$
(e)
$\frac{250}{20}$
75
$C_o = 0.17$
$\alpha = 0.61$
(f)
$\frac{x}{d_o}$
Comparison of theory computation value with average test value

Fig. 3



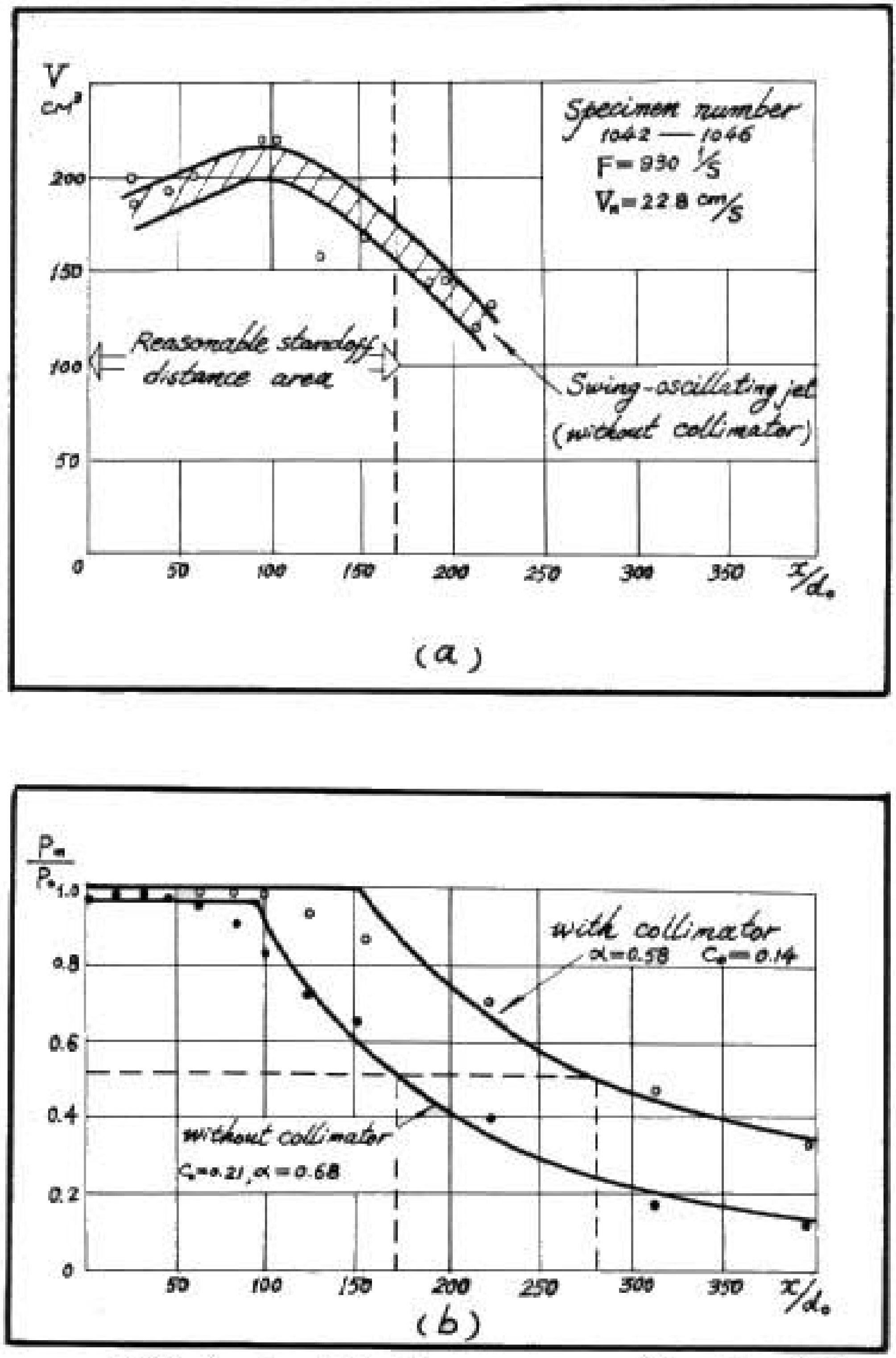
Axial dynamic pressure distribution of high pressure waterjet

Fig. 4



Location of collimator

Fig. 5



Effect of collimator on reasonable standoff distance area and axial dynamic pressure

Fig. 6

DEVELOPMENT AND BASIC REGULARITIES OF WATERJET CUTTING TECHNOLOGY IN CHINA COAL INDUSTRY

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ABSTRACT

The way of developing the waterjet cutting technology may be different in various countries, and especially in developing countries. Based on further studies of our own regularities of development, advances in the waterjet cutting technology will be made by leaps and bounds.

A brief summary of development of the waterjet cutting technology in the People's Republic of China in the area of the coal industry is presented in this paper. Current applications characteristics and regularities of development are also discussed in this report.

INDUSTRIAL USES OF LOW-PRESSURE WATERJET

China has been one of the sources of hydraulic mining technology. In the early part of the 20th century, hydraulic excavation of topsoil and minerals, such as stannum (Sn) had been brought into use in open-pits. In the fifties, hydraulic mining of coal spread by big leaps. A variety of manual-operated or hydraulic-controlled monitors generating the low-pressure coherent waterjet had been proved to be effective for winning coal, of which the Probugiakov hardness coefficient is up to 1.5. The productivity of coal was approximately 50 ton/hr under a water pressure of 30 to 50 bar and nozzle diameter of 18 to 28 mm. Tangshan Hydraulic Coal-Mining Research Institute conducted extensive investigations into nozzle design, monitor performance and hydraulic coal-mining operational procedures as well. Aimed at increasing the productivity of hydraulic monitors, the working pressure of water jet had to be raised up to 200 bar in the sixties. The productivity of the raw coal, as a consequence, was raised to 180 to 200 ton/hr (1), (2), as shown in Fig. 1.

An exploratory survey of the potential application of high-pressure water jet was initiated in the late sixties. A decade later, the high-pressure waterjet, the so-called 'water knife', gave a challenge to conventional mining tools such as the coal-cutter, tunnelling and drilling machines. At present the swing-oscillating jet, and the pulsed jet are being put into field application in coal mine entry driving. The technique of waterjet deep-kerfing in coal seams for releasing underground gas has been verified to be promising for field application. In addition, cleaning of mine vehicles using the plain water jet has become successful for commercial use.

DEVELOPMENT OF JET ROADHEADER IN COAL-SEAM

Experimental studies of waterjet roadheaders for coal seam drivage were conducted by China Institute of Mining with the cooperation of Zaozhang Coal Mine Administration in 1979. The working pressure was up to 500 bar. High-pressure pump of three-piston type and hoses were all made in China. Two different types of jets, i.e. (1) combination of horizontal swing with perpendicular oscillation, and (2) combination of horizontal swing with steady rotation, were recommended in tests (3). Superimposition of a perpendicular oscillating velocity onto the swing velocity resulted in an increase of the relative velocity of the water jet against the coal face being cut, in turn, considerably enhancing the energy efficiency of waterjet. At the same time, the comparatively wide fragmentation zones were cut through the coal face (Fig. 2) (4). Deep-kerfing in coal seams will be induced by the combination of formation of a fragmentation zone with repeated cuts.

Laboratory test results indicated that depth of cut, at a certain extent, is directly proportional to the number of pass of the jet (5). The jet roadheader has already been tested and employed in Beijing, Tanshan, Zaozhang and Xintai etc. Coal Mine Administrations. The total amount of coal seam entry drivage is more than 1,000 meters.

For the purpose of reducing the energy consumed, operation procedure of coal mining (Fig. 3) could be stated as below.

1. At first, a deep kerf will be formed by repeated cuts at the bottom of the coal face;
2. Fragmentation zones with a reasonable spacing will be cut from the bottom to the top. The ridges left between two adjacent cutting zones will collapse under the induced normal thrust so that a profusion of large bulk coal will be produced;
3. Collapse of a large quantity of coal may take place when (a) the bottom kerf is deeper (b) parameters of water jet are tailored to the cutting conditions (c) jointing and bedding planes in the coal seam are more developed;
4. An additional cut along both side-walls of the roadway has to be accomplished by water jet after the cross-section area has been cut through.

Discussion on the operation procedures in all details was given in the paper (6). Major parameters of Swing-oscillating jet are as follows:

Water pressure	350 to 500 bar
Nozzle diameter	1.8 to 2.5 mm
Swing velocity	Up to 1 m/sec
Oscillation frequency	14 to 24 1/sec
Oscillation amplitude	60 to 100 mm
Hardness coefficient of coa1	less than 1.2
Humidity content in coal	less than 10%
Productivity of coal	0.8 to 1.6 ton/min

The jet roadheader has many advantages over conventional methods such as:

1. In comparison with drilling-blasting method, it simplified the working process of entry drivage. This leads to increased advance rates by a factor of 2 to 5;
2. Dense coal dust has effectively been suppressed by droplets of water. The average dust concentration measured in-situ is less than 8 mg/m^3;
3. Waterjet cutting device is light, but robust easily moved, less backward reaction thrust;
4. Humidity content of coal, in general, is less than 10%, and seems to be suitable for transporting by the belt conveyor;
5. No crack and destruction has been found in the surrounding coal or rock bodies. It is thus able to enhance roadway stability and reduce repair costs as well.

In order to expand specifications of this device, researchers of China Institute of Mining are (a) suggesting a new project of design of the waterjet assisted roadheader and (b) developing an investigation to raise the working pressure of the waterjet to a reasonable level(5)

POTENTIAL APPLICATION OF WATER JET DEEP-KERFING FOR GAS PRODUCTION

Utilizing hydraulic monitors, the early tests for releasing mining gas and preventing spontaneous bursting were conducted in the late fifties. In 1977, investigators of Fushun Coal Mine Safety Research Institute conducted an experimental programme of deep-kerfing in coal seams using waterjet for the purposes cited above in Hongwei Coal Mine (Hunan Province, China). Test set-up mainly consisted of a plunger-type pump rated at a pressure of 700 bar and a nozzle-holder with lance. Two nozzles have been fixed radially in the nozzle-holder and aligned in the same diameter of the holder. The power output of water jet is up to 320 kw with a nozzle diameter of 2 to 4 mm and a water pressure of 400 to 700 bar.

The field test procedure was as described below:

(1) Using a drilling machine, three to six holes of diameter 130 mm and depth 7 to 11 m have been drilled and distributed parallel to or sectionally across the face and throughout the coal, as shown in Fig. 4 and 5.

(2) By feeding the lance in to the back end of each hole, and setting nozzles in the horizontal plane, the water jets will be drawn backward at an adjustable rate of translation making it possible to connect two adjacent holes by means of waterjet deep-kerfing. Thus, many horizontal kerfing planes will be formed for purposes of releasing gas pressure and diminution (even elimination) of hazards of gas-and dust bursting. Fushun investigators have also carried out an alternative project for large-scale field application for releasing and extracting underground gas in Hebi Coal Mine Administration (Henan Provinces China). Two different nozzles have been tested, such as (a) nozzle diameter of 2 to 4 mm and water pressure of 300 to 350 bar, output of jet within 50 to 80 kw (b) nozzle diameter of 5 to 8 mm, water pressure of 100 bar and output of jet within 25 to 60 kw. Deep-kerfing in coal seams has been used to cut through more than one hundred holes of 50 to 80 m depth. The amount of gas extracted by this technique was increased by a factor of 2 over that by the conventional method. Annual output of gas was 24% to

40% of the total reserve of gas within the coal-mining area. It is significantly important that this technique contributed to improving coal mine safety, and of course, promptly increasing the productivity of raw coal (8).

APPLICATION OF WATERJET CLEANING TECHNIQUE

Mine vehicles for underground or surface transportation of coal, rock debris or slurry are often very dirty. In particular, a thick deposit (thickness of up to 400 mm) of fine coal or debris is left and consolidated on the inner wall of vehicles. This leads, to a large intent, to reduce the volume of vehicle, and considerably increases the maintenance cost of mining and transportation. Jiangsu Coal Mine Research Institute (Jiangsu Province, China), etc. have successfully developed a cleaning system which consists of a piston pump rated at a pressure of 120 to 180 bar and using a rotating nozzle holder. The time to clean is 50 sec for cleaning a coal-loaded vehicle, 80 sec for cleaning a debris-loaded vehicle.

Furthermore, on a fully-mechanized long-wall coal face more than one hundred batches of hydraulic supports has to be equipped. Waterjet cleaning technique has already been used for repairing a large quantity of cylinders of the hydraulic supports. Jiangsu researchers have tested a set-up for removal of oil and water emulsion deposits stuck inside the cylinder by means of an abrasive water jet or plain water jet. In order to clean heavy-duty vehicles used on the underground transportation lines, the China Institute of Mining has carried out an alternative project that is under development.

DEVELOPMENT OF WATER CANNON

Since 1970, two parallel programs, the extrusive and the percussive water cannon, have been pursued for the purpose of development of an adequate and inexpensive equipment for coal mine entry drivage. Jiangsu researchers tested, at first, the extrusive water cannon, and then, the percussive type with the financial support of the National Science And Technology Commission and the Ministry of the Coal Industry P.R.C.. Field tests and application have been carried out in a coal mine (Fig. 6.7). Test results are briefly described as below:

Impact pressure	6,000 to 7,400 bar
Nozzle diameter	6 to 10 mm
Content of rock and coal	Approx. 50% each
Hardness coefficient	6 to 7 (for rock)
	2 to 3 (for coal)
Rock and coal breaking amount per shot	34.5 Kg

In particular, the cost of electric power consumption per cubic meter of rock and coal mixture was reduced by one-thirds, other conditions being identical (6). Attention has to be paid to suppressing the cloudy dust and terrible noise, although the test results seem to be promising.

In addition, an alternative design of the percussive water cannon has been tested by the China Institute of Mining. This project will be reported in details in this conference by Dr Da-Zhong Cheng, associate professor at the China Institute of Mining, one of the explorers of the waterjet technique in People's Republic of China (9) .

PRELIMINARY CONCLUSIONS

Learning through our own experience, characteristics and preliminary regularities of development could be concluded as follows:

(1) Recognizing the need and importance of coal mine entry development prior to coal production, the lack of adequate and cheap equipment, and the necessity of improving underground working conditions. We initiated major efforts in developing high pressure water jet cutting and blasting techniques based upon previous technical experiments and earlier success in hydraulic coal mining.

(2) Development of water jet has been handicapped by the lack of the advanced high-pressure techniques. Therefore, at the present time the continuous water jet, of which the working pressure has to be limited within the range of 350 to 700 bar, could be promptly spread for industrial applications, as stated above. The point is that equipment and ancillaries required are within the present state of the art in China

(3) Energy efficiency must be updated as far as possible. For this purpose the swing-oscillating jet as well as the rotating jet have been employed. The interrupted waterjet is of great interest for the same purpose, and is under development at the China Institute of Mining.

(4) High pressure pulsed jets show great potential for cutting and breaking hard rock. However, its development appears to be inherently limited by lack of very good quality materials and the advanced manufacturing art. It is expected to spread the pulsed waterjet as increasingly great progress in science and technology is made in China.

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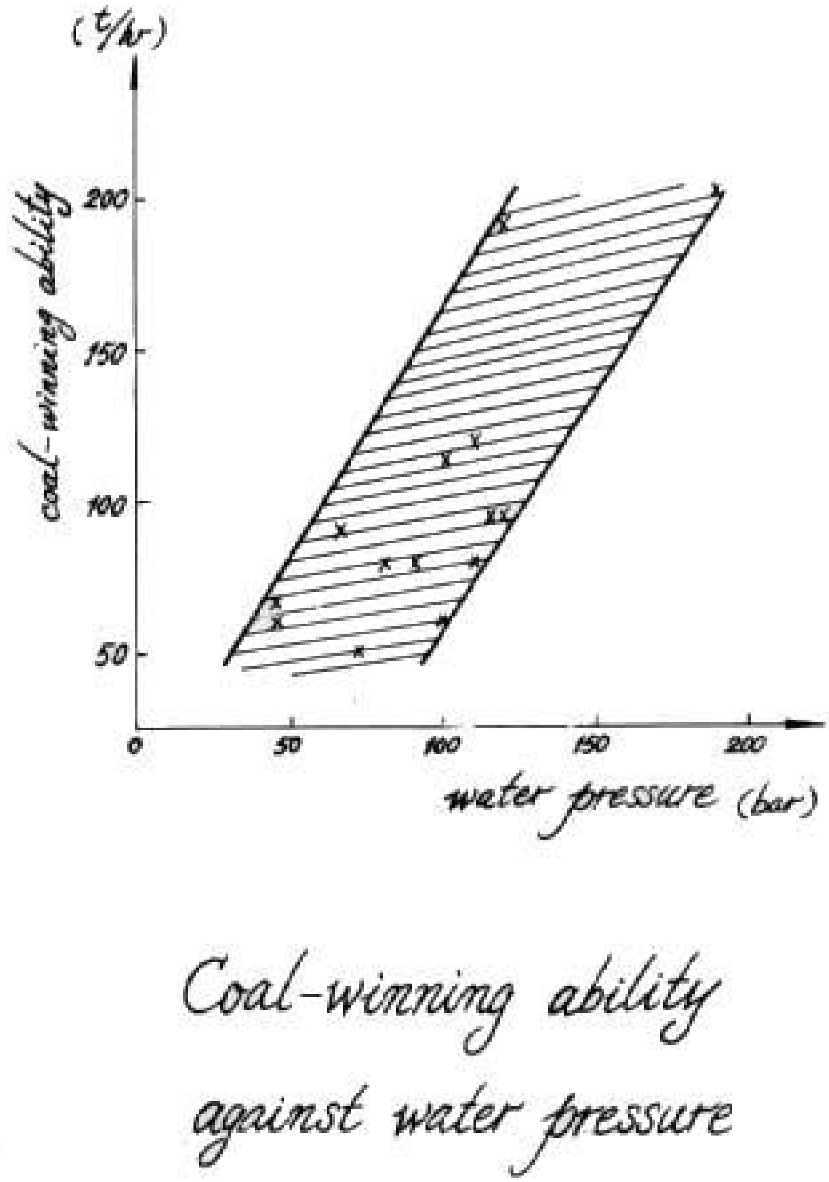
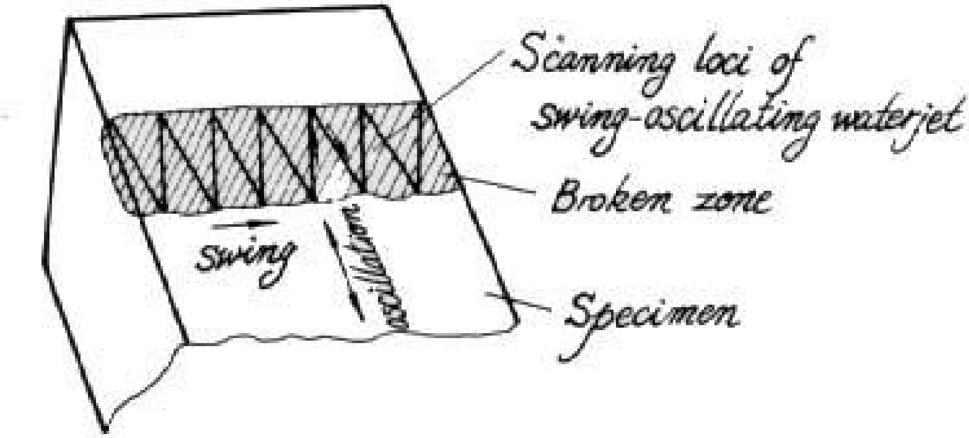
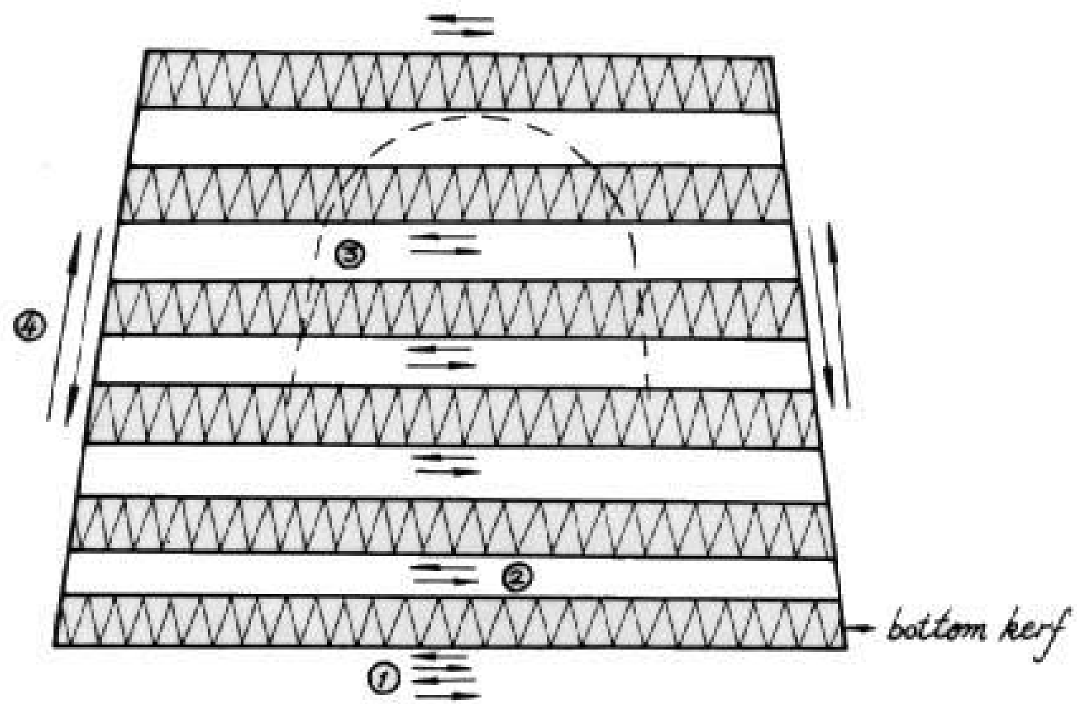


Fig. 1



Scanning loci of swing-oscillating waterjet
and widely broken zone

Fig. 2



Sequence of winning coal
using a swing-oscillating waterjet on roadway

Fig. 3

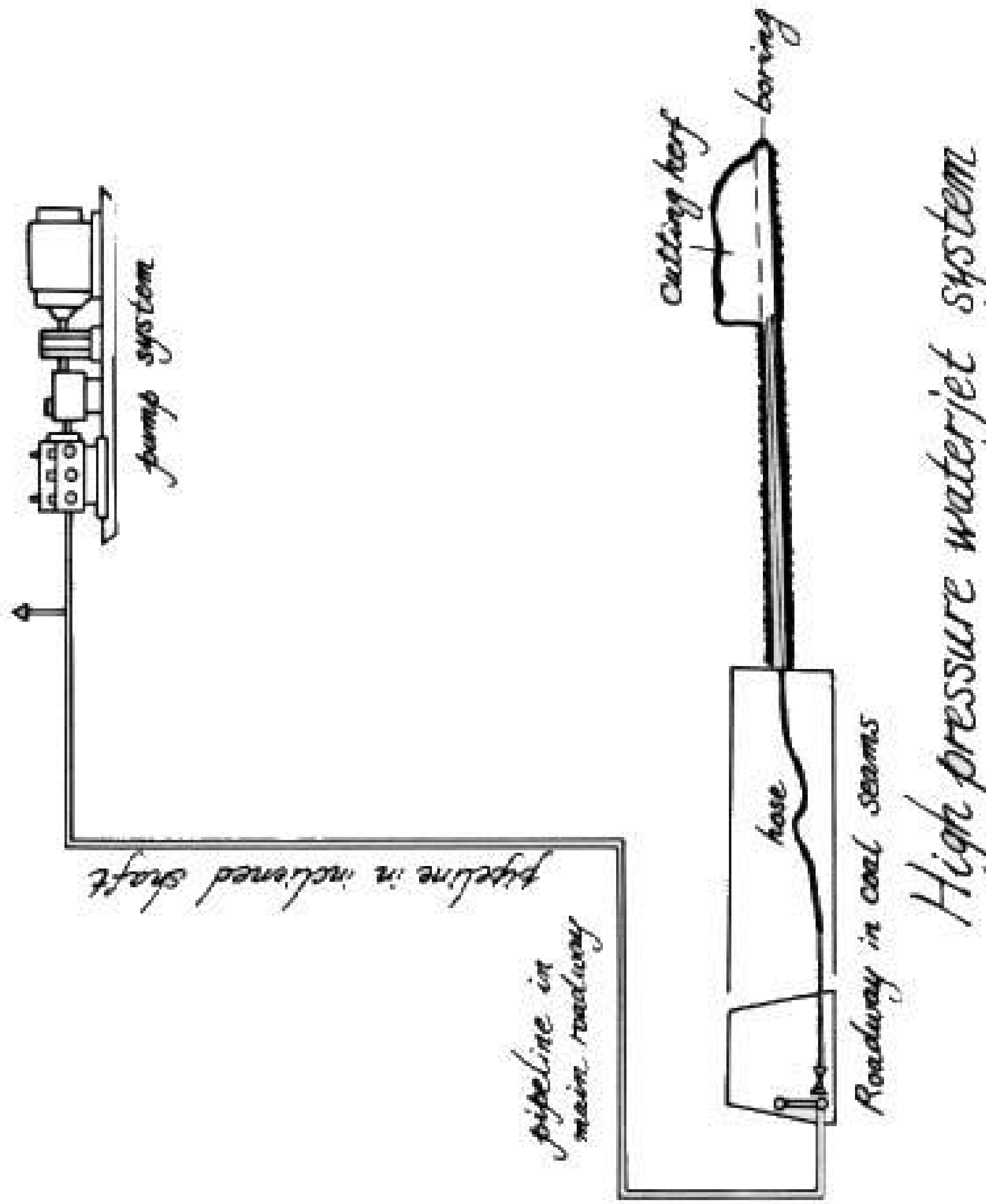
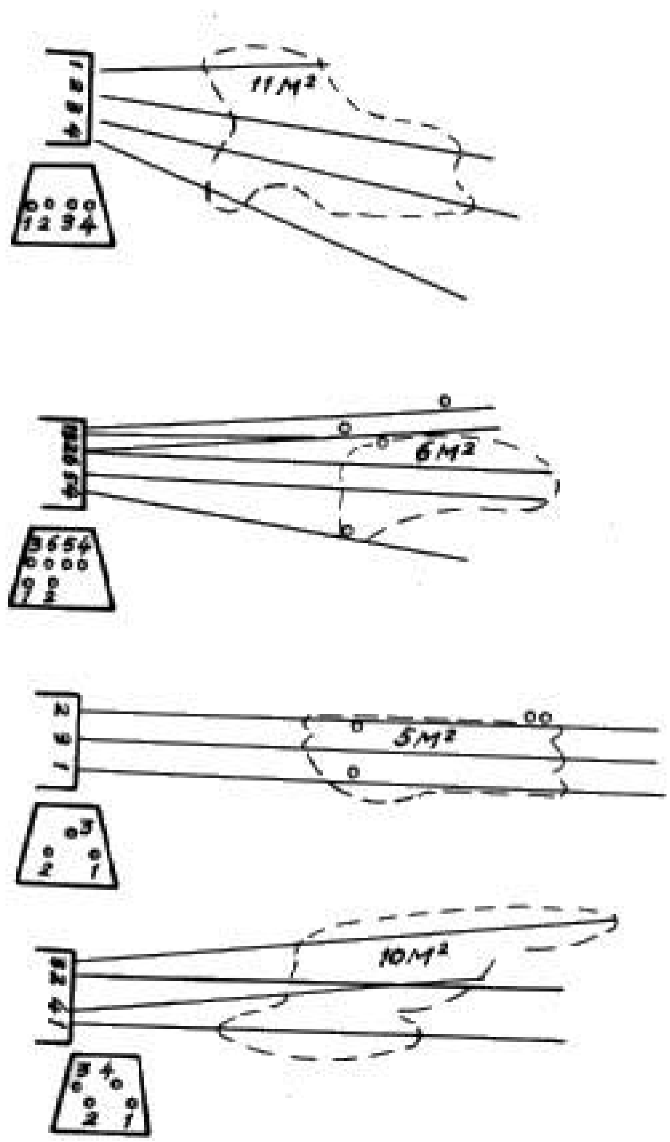

pump system
cutting kerf
boring
pipeline in inclined shaft
pipeline in main roadway
hose
Roadway in coal seams
High pressure waterjet system

Fig. 4



Crack distribution due to waterjet cutting

Fig. 5

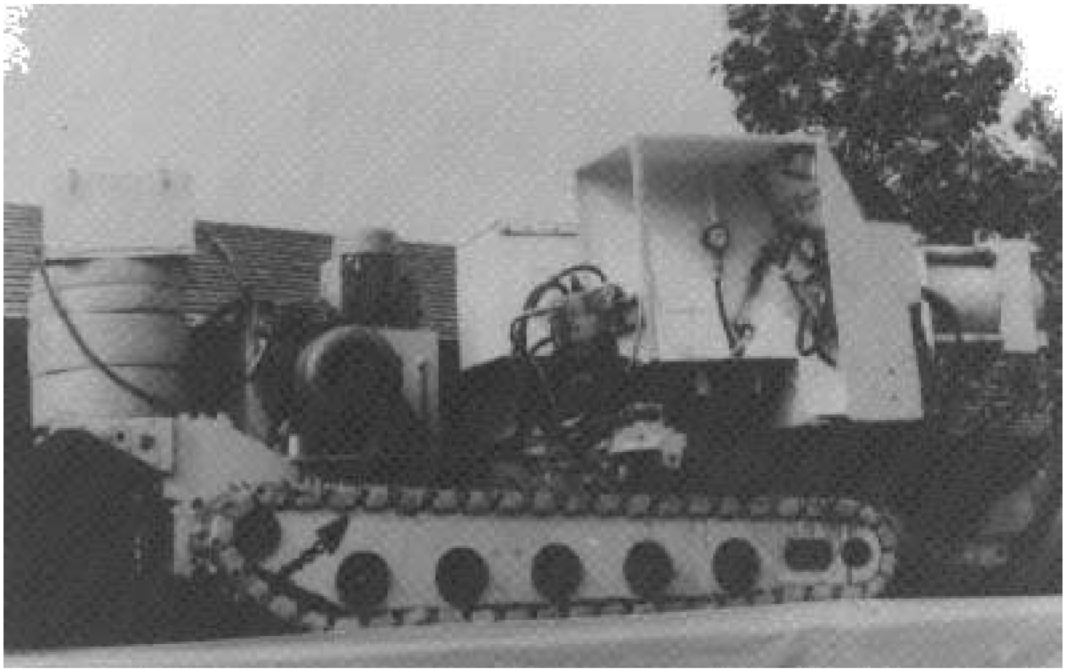
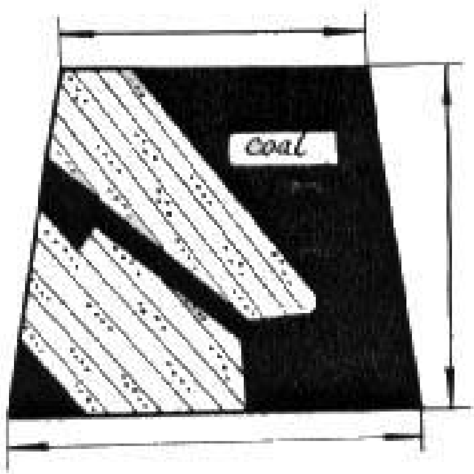


Fig. 6



Cross sectional area: 6.4 m²

Percent of coal: 59%

TYPE OF ENTRY

Fig. 7

HIGH PRESSURE WATER JET APPLICATIONS TO ROADHEADERS

by

J. Straughan C.Eng. F.I.minE.
Director of Exports
Dosco Overseas Engineering Ltd

SYNOPSIS

Following laboratory and field experiments by M.R.D.E. in association with the U.S.B.M. a definite benefit from the use of high pressure water on roadheading machines was identified. This technology was made available to the manufacturers and Dosco were to develop the MRDE/USBM system concurrently with the lance through the boom.

This paper reviews the work carried out by MRDE at the Middleton mine on the prototype roadheader and the subsequent work carried out by Dosco on the MK2A and MK2B roadheaders.

Particular emphasis is placed on the engineering difficulties encountered when transmitting water at a pressure of 69 MPa (10,000 psi) from a moving platform to a rotating head in a mining environment. How these problems were solved is explained as is the current philosophy for Dosco machines equipped with high pressure water assisted cutting.

INTRODUCTION

Boom type roadheaders were introduced into the mining industry in the early 1950's but it was not until the 1960's, with the rapid advances in longwall mechanization that the full potential for roadheaders was realized in the U.K. The machine which was to lead the revolution was the Dosco D.R.C.L. which was the fore runner of the Dosco MK 2A Roadheader. These machines were successfully used in most types of coal measure strata but for the more difficult, abrasive rock, heavier more powerful machines were subsequently developed.

Heavier machines could generate greater arcing and lifting forces as well as accommodating more powerful cutting booms. This enabled harder rocks to be cut by larger roadheading machines. This line of development was not to continue indefinitely however and the limiting factor was found to be the cost of replacement tungsten carbide tools in the cutting head.

The advent of High Pressure Water Assisted Cutting (H.P.W.A.C.) was seen by the mining industry as a means of overcoming the limitation placed on roadheaders by tungsten carbide tools and enabling rotary cutting to be utilized in rock types previously beyond its scope.

Much of the early laboratory work carried out in the U.K. on water assisted cutting was undertaken by the Mines Research and Development Establishment

(M.R.D.E.) at Bretby. It was MRDE in association with the U.S. Bureau of Mines that were to sponsor and build the "prototype" high pressure water assisted roadheader which was tested at the Middleton mine test site.

Following the Middleton mine trial of H.P.W.A.C., manufacturers were invited to inspect the prototype machine and to propose high pressure water systems of their own which would be suitable for use in coal mines. At this time Bretby were developing the offset gearbox which would facilitate the use of a small diameter rotary seal which could be placed at the back of the cutting head shaft. This would simplify access to the rotary seal on the cutting boom.

The use of the offset gearbox corresponded with the Dosco philosophy of providing a system which could be retrofitted to existing machines. Therefore much of the MRDE prototype design was incorporated into the Dosco "preproduction machine". Concurrent with the development of the offset gearbox Dosco, were also designing a lanced boom where water was to pass through the center of the boom from the motor to the cutting head. As a consequence of field trials with both types of boom standardization on the lance design was agreed.

BACKGROUND

Early laboratory work carried out by MRDE was sufficiently encouraging to prompt a full-scale underground test of a H.P.W.A.C. system mounted on a roadheader. The building of this "prototype" machine was sponsored by MRDE and the U.S.B.M. The test vehicle which was chosen as the prototype was a Dosco MK2A roadheader. The MK2A needed to be modified to accept a water system capable of generating and transmitting high pressure water to the pick point. To generate this pressure the following components were needed in addition to the standard machine.

a) A high pressure pump capable of sustaining an output pressure of 69MPa (10,000 psi).
b) A high pressure rotary seal between the cutting head and the drive shaft.
c) A cutting head with a high pressure nozzle (1).

A basic Dosco MK2A machine has no facility for accommodating a high pressure pump and so a free standing pump was selected. This was mounted on a sledge and towed behind the machine as it advanced. Later Dosco designs would incorporate this feature on board the machine.

The pump comprised of two units:- the hydraulic pump which was powered by a 75 kW (100 hp) electric motor and raised the oil pressure to 14 MPa (2,000 psi) and an intensifier ram which was mounted on the boom and raised the water pressure to 69MPa (10,000 psi).

The high pressure water generated by this system was fed into the cutting shaft in the same manner as low pressure water when used for dust suppression (Fig 1). It was clear that the existing shaft seals on the MK2A would be totally inadequate at these higher pressures and so an alternative seal was sought. Unfortunately no proprietary seal was available and so several new designs of seals were developed by M.R.D.E.

The cutting head was to be based on the standard Dosco design with suitable modifications. The cutting head contained twenty two cutting picks and fifteen water jets, the majority of which were in forward part of the head.

This prototype machine was to be tested at the Middleton mine in Derbyshire, where a homogeneous band of limestone suitable for testing purposes was easily accessible. This limestone has a compressive strength of 116MPa (16,800 psi) which under normal operating conditions would be outside the working range of the MK2A. Cutting test were carried out in the limestone with and without high pressure water so that a comparison between the two could be made.

It was soon apparent that H.P.W.A.C. has a dramatic effect on the ability of the MK2A to cut the limestone. In the absence of high pressure water the cutting tools in the head failed almost on contact with the strata and the motor driving the cutting head frequently stalled on overload. With high pressure water the machine was stable when cutting with up to a 50% reduction in cutting forces. Cutting tool life was increased and there was no sign of incendive sparking (2). The results also showed that high pressure water reduced the quantity of air borne dust liberated by the picks by up to 50%.

During the trial period it was found that the life of the high pressure rotary seal was totally unacceptable for use in a production environment. Various designs of seal were tried with only limited success, the working lives varying from one to thirty hours. This fell well short of a working life of 500 hours which was required.

The results obtained from the trial of the prototype machine indicated that considerable benefit could be gained from H.P.W.A.C. At this point Dosco were invited by the N.C.B. to develop a roadheader capable of being used underground in coal mines.

PRE PRODUCTION MACHINE

The Dosco Pre Production machine which was used to test High Pressure Water Assisted Cutting in a full production mode in a coal mine was again the MK2A Roadheader. At this time however the MK2A was undergoing a full modification process and this was extended to incorporate H.P.W.A.C. From the outset it was decided that the Pre Production Machine should be totally self contained and designed with operational safety at the forefront.

The Pump

Two alternative systems for raising a water pressure of 69MPa (10,000 psi) were considered.

a) An Intensifier (as the prototype)

b) A Water Pump.

It was intended that the intensifier system would be evaluated by M.R.D.E. and since the MK2A had an insufficient hydraulic oil capacity to power an intensifier, a free standing power pack would be required. Clearly this was not desirable.

A water pump would need to be driven from an electric motor, so either an existing motor had to be modified or an extra motor would be required. The solution to this problem was replacing the existing power pack motor with 150kW (200hp) motor with a double ended rotor. The power pack was drive off one end and the water pump off the other.

The pump used to generate the high pressure water was of radial piston diaphragm type with all mechanical components operating in an oil bath. This raised a water pressure of 69 MPa (10,000 psi) with a flow of 0.9 l/s (12gal/min).

As the pump could not be pressurized a water tank was mounted above the bridge belt conveyor.

Boom Assembly

The most persistent problem with the prototype machine has been the inability of the water seals to withstand a working pressure of 69 MPa (10,000 psi). For this reason the M.R.D.E. offset boom was to be used on the pre production machine (Fig 2). By using an offset gearbox the back of the cutting head shaft was exposed, enabling a small diameter high pressure rotary seal to be fitted.

For safety reasons special forged cutting heads were designed for use with H.P.W.A.C. These are now standard and are not interchangeable with low pressure heads.

The head was designed with point attack cutting tools with a jet stand off distance of 2mm (.08"), and the discharge from the nozzle to the tip of the tool was approximately 130 mm (5.1"). The sprays were 0.6 mm (.025") diameter sapphire nozzles, and to achieve 69MPa (10,000 psi) only the first three sprays in each start (9 in all) could be used for High Pressure Water Assisted Cutting.

Hydraulic System

Transmission of the high pressure water from the pump to the rotary seal was through rigid steel piping. This system required the design of special oscillating joints for the arcing and lifting pivots.

Water entered the system in the normal way through a standard stop valve, strainer and pressure reducing valve. At this point the water flow was split, one feed going through the standard cooling and dust suppression circuit and the other going to the H.P.W. circuit.

The feed to the H.P.W. circuit first passed through a restrictor to control the flow into the circuit and the through a 25-30 micron strainer before passing into the tank mounted above the belt conveyor. From the tank the water passed through a 6 micron filter to the high pressure pump.

From the high pressure pump the water at a pressure of 69 MPa (10,000 psi) flowed through the stainless steel tubing and fittings through the high pressure rotary seal to the cutting head shaft. In the line was a pressure gauge relief valve and a Tee piece which was to accept low pressure water for dust suppression when high pressure water system was not required.

Safety

The system was so designed that the high pressure water system could only be selected if all the normal safety procedures had been carried out. The start up sequence was as follows:

1. The low pressure water supply to the machine has to be switch on so that water flowed through the cutting head jets.
2. Water must flow for eight seconds before the electric motors energize, first the hydraulic motor then the cutter motor.
3. With the hydraulic circuit energized a B & G valve was used to activate the hydraulic clutch. (This valve is linked mechanically to the cutter motor).

PRE PRODUCTION TRIAL AT DOSCO TEST SITE

Prior to being put to work in an underground environment the pre production machine was first of all tested in the Dosco Works. On the Dosco test site a concrete block was cast into which were placed several large limestone blocks so that the effects of cutting a full rock face would be simulated (Fig 3).

During these cutting trials several problems were identified and remedial action was taken.

Boom Assembly

When cutting the mock rock face the machine was subjected to excessive vibration which was also present when H.P.W.A.C. was selected. To rectify this problem the cutter motor was changed from a speed of 1460 rpm to 960 rpm reducing the head speed from 66 rpm to 44 rpm. This action reduced the vibration considerably.

The rotary seal used during the surface tests was the M.R.D.E. carbon faced seal (3). Reliability of this seal was found to be poor and the main cause of leaks within the seal were found to be:

a) Carbon faced seal tended to rotate thus damaging the 'O' ring.
b) If the steel pipe and joint connection were not correctly lined up the sealing faces tended to spring apart.

The same problems have been experienced by MRDE at Bretby and the possibility of using a modified seal had been examined. However to keep the project running the carbon faced seal was retained but its reliability was still in doubt.

Cutting Head Design

When cutting limestone in the test block the pick tips on the cutting head shattered even when using H.P.W.A.C. The point attack picks were replaced by forward attack picks, and modifications carried out to the head which reduced the distance from the nozzle to the pick tip to 85 mm (3.3"). This gave a far better performance.

After a period of inactivity a problem was experienced with blocked jets. It was found that scale was the biggest problem and this could have only come from within the cutting head. As a consequence of this all internal water galleries were plated.

Water Distribution System

A problem was identified with the rotary joints which were necessary because of the steel pipe distribution system. The seals in the joints were leaking and the pipes themselves could wear through if subject to rubbing. It was thought that the joints could be leaking because the seals were oscillating and not rotating hence preventing the seals from bedding in; a new stepped seal was therefore designed by Dosco.

Difficulties were also experienced with the overload valve and pressure gauge which was subjected to high pressure pulsations. This was overcome by designing a equalizer block.

Power Pack

When conducting the cutting trials pump cavitation was found to be a problem. This was caused by an air pocket in the inlet chamber of the pump which when drawn through the pump caused a high degree of pulsation. This problem was solved by piping the top of the inlet chamber to the water tank thus keeping it full of water. The 6 micron filter was moved to the input side of the water tank.

The trials on the Dosco test site were useful in identifying problem areas before the system went underground. There were however, still problem areas which had not been resolved when a "heads of agreement" trial for the machine was agreed between the N.C.B. and Dosco. As a result of this agreement it was not possible to complete the full trials program planned by Dosco before the machine was dispatched to the N.C.B. for underground trials.

BENTINCK COLLIERY

The test site for the integral high pressure water system was at Bentinck Colliery which is some twelve miles north of the city of Nottingham. At Bentinck Colliery several coal seams are worked and these are separated by over two hundred meters of mixed strata. The drivage selected for the H.P.W.A.C. trial was cross measure between the Blackshale and Tupton Coals. The drivage was to be driven to the rise for about 550 meters at 1 in 7 excavating an arch profile 4.9 m x 3.7 m (16ft x Lift) supported by H section girders. The strata projected for the drivage consisted mainly of mudstones, but sandstone with a strength of 138 MPa (20,000 psi) was expected.

When the machine arrived at the Colliery it was assembled on the surface so that the operators could be trained in the operation of the high pressure system. Only after this training was completed was the machine taken underground.

Once underground the machine was used to enlarge an existing roadway and create a junction before the drift could be started. This process coupled with the holiday period took over three months to complete.

Once the machine was established in the drift high pressure water assisted cutting was tried in the softer mudstones in the junction area. In this area the mudstones were heavily jointed and the effect of the water on the floor was to cause transportation problems. As a consequence the use of the high pressure system was to be suspended until the harder strata was encountered which was after 350 meters of roadway has been driven.

As expected further problems with the system were encountered but due to floor water problems and intermittent use of the system no conclusive results could be drawn from the trial.

Boom Assembly

The offset boom was longer and heavier than a standard boom and as a consequence hydraulic flow through the over center valve caused a shuddering of the boom. This was rectified by a re-designed valve.

A twin rotary seal was fitted to the machine but due to vibration the seal shaft soon broke. This was replaced by a single seal supplied by Bretby. This seal also experienced difficulties, one of which was wear on the running shaft. Various types of heat treatment were tried to solve this but the problem still existed.

The weight of this seal was also a problem as it could not be supported and the threaded portion of the seal was prone to shear.

Cutting Head Design

The 0.6 mm (.025") diameter sapphire nozzles were replaced by 0.4 mm (0.017") diameter tungsten nozzles. These had the benefit of reducing the water flow, which had proved to be such a problem and also producing a more coherent jet. Clogging of the jets was still found to be a problem and may in part have been due to contamination of the pit water supply.

Water Distribution System

The use of all steel pipes for the distribution system proved difficult as the system had no 'give' so high pressure flexible hoses were used in a number of places. By accepting the use of these hoses the oscillated joints were eliminated.

Power Pack

Despite making several modifications to the high pressure pump the reliability of the system was inadequate. After several months work the eight piston pump was replaced by a four piston pump in an effort to reduce the flow but this pump also proved to be unreliable.

The clutch also proved to be troublesome underground. The manufacturer put the problems down to ingress of dirt and moisture.

6 CONCLUSIONS DRAWN FROM BENTINCK

As a consequence of the experience at Bentinck Colliery the lanced boom which had been developed concurrently with the MK2A trial was selected as standard for this and all machines (Fig 4). This would overcome many of the problems experienced with the trial machine.

The lanced boom

a) Uses standard over center valves.
b) Has no great increase in weight.
c) Is only 100 cm longer than a standard boom.
d) Uses seals mounted in the lift arm making them more accessible.
e) Uses cutting head flange mounting holes of standard pitch.
f) Uses a universal boom for high and low pressure.

Seals

Two types of seal are available depending upon the duty requirement of the system. A low pressure rotary seal is available for pressures up to 21 MPa (3000 psi). This seal has undergone over 2000 hours of testing at the M.R.D.E. laboratories. For pressures up to 69 MPa (10,000 psi) a high pressure seal is available which has been tested for 1000 hours.

Power Pack

Due to the problems experienced with the diaphragm type pump it was decided that a suitable replacement should be sought. It was eventually decided to use the F.S.W. ram pump which would be vertically mounted and fitted to the motor through a new Dosco gearbox.

Because of the problems experienced with the hydraulic clutch a replacement solid coupling was devised. By this method the pump would be energized all the time but would only deliver high pressure water to the jets when required.

MK2B MACHINE AT DEVCO

A MK2B Roadheader (Fig 5) equipped with a medium/high pressure water system was supplied to the Devco coal mines in Nova Scotia in 1984. This machine was designed by Dosco concurrently with the "pre production" MK2A and the transmission system chosen included the lanced boom and F.S.W. pump, driven by a double ended motor. The specification of this machine is shown on Table 1.

The machine was supplied to Devco on a trial basis and was to drive an access drift from the surface to the Phalen Seam at the new Lingan Phalen Mine. The Dosco Roadheader was used to drive number 3 slope on a three shift eight hour basis. The gradient of the drift was twenty two percent.

The trial was carried out against an alternative suppliers machine also equipped with high pressure water assisted cutting which was to drive drift No.2.

The water system supplied with the MK2B was limited to 34.5 MPa (5000 psi) although it was able to generate 69 MPa (10,000 psi). The cutting head was .53 m (21 inches) in diameter and contained 24 Kennametal point attack cutting tools and 20 high pressure water jets of 0.4 mm (0.017") diameter.

The machine started production on Wednesday, 21st November 1984 initially working on only one shift per day. The strata which was encountered during the drivage was composed of coal, mudstones and shales and caused no cutting problems. Water in the drift was a problem but this was due mainly to strata water from the roof and floor. This water necessitated the use of submersible pumps at the face of the heading.

After a six month trial period the decision was made by Devco to purchase two more MK2B machines for use at the colliery. This decision was based on the reliability of the total system which was supplied by Dosco.

The final water distribution was altered slightly to obtain the best dust suppression arrangement. Only 12 jets delivering 27.4 l/min (6 gal/min) were used with the remaining 41 l/min (9 gal/min) being returned to the water tank.

FUTURE DEVELOPMENTS

Dosco have designed and developed High Pressure Water Assisted Cutting Systems for use with roadheaders, which can be supplied either as a total package or in kit form to be fitted to an existing machine. This of necessity, has required more than one approach to solving the problem of transmitting high pressure water to the pick point. The design of totally new systems has not been easy and many problems both mechanical and operational have been identified during the field trials. From trials however, successful designs of machine have emerged and the adoption of the lanced boom as standard for machines such as the MK2A and MK2B has many benefits at both low/medium and high pressure.

To date underground experience with high pressure water assisted cutting is limited and of the trials so far completed two were with systems working up to 34.5 MPa (5000 psi). The information obtained from trials at 69 MPa (10,000 psi), have established the following benefits from H.P.W.A.C.

i) Better cutting ability in harder rock
ii) Extended life of cutting tools
iii) Better dust suppression
iv) Significant reduction in incendive sparking

It is not clear at this stage what is the optimum water pressure to be used with each machine and in which rock type. To evaluate this, much more work is needed on high pressure systems with working pressure in excess of 69MPa (10,000 psi).

At medium pressures however dramatic improvements in dust suppression, incendive sparking and tool life may be achieved compared to systems working at normal pressures.

It is important to realize that these two systems are different in terms of machine specification. A medium pressure system up to 21 MPa (3000 psi) may be generated by a hydraulic pump which drives a ram pump and utilizes a low/medium pressure rotary seal. The high pressure system is driven by a ram pump, driven by an electric motor having a special high pressure rotary seal and a high pressure cutting head.

Against the proven benefits of high/medium pressure systems has to be balanced the associated problems of floor water. However work is now in hand to produce a phased high pressure water system which limits the volume of water put to ground. This may be achieved by monitoring the arcing and lifting forces and automatically generating high pressure water when the hydraulic system is under load.

In coal mines high pressure water assisted cutting will only be required for the small percentage of strata which is currently beyond the range of the existing design of roadheader. If these bands are thin or encountered for only a short period conventional methods may be preferred to H.P.W.A.C. However the benefit of H.P.W.A.C. may be realized in more massive mineral deposits which are currently worked by explosives and may in the future be within the working range of roadheaders.

In conclusion, Dosco are confident that H.P.W.A.C. systems will be capable of extending the working range of roadheaders as well as improving the working environment. We are however mindful of the technical, economic and mining considerations which will affect any machine in the field. It is only by a sound development program based on practical experience that the full potential of these systems can be achieved. Dosco believe that their current philosophy for pressure raising and transmission is the best available to date, but development work is still being carried out to refine existing systems and to produce new systems.

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3. Seminar on Water Jet Assisted Roadheaders for Rock Excavation (Pittsburgh May 1982).
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TABLE 1 - MK2B SPECIFICATION

Overall Dimensions and Weight	
Length	9.80m (32' 2")
Height (standard machine)	1.96m (6' 5")
Width (across frame)	2.70m (8' 10")
Width (at apron)	3.00m (9' 10")
Weight approx.	41,000kg (90,364 lbs)
Tracking Arrangements	
Tracks (single chain type)	With double and triple bar grousers
Width of track pads	457mm (18")
Distance between track centres	1.90m (6' 3")
Tracking speed (sumping)	0.05 m/s (9.5 ft/min)
Tracking speed (flitting)	0.17 m/s (33 ft/min2
Ground pressure	117.2kPa (17 lbf/in)
Max incline/decline gradient	25 percent (1:4)
Max cross gradient	14 percent (1:7)
Conveyor Arrangement	
Scraper conveyor	Twin round or strap link chain type
Scraper conveyor width	610mm (24")
Scraper conveyor speed	1.20 m/s (230 ft/min)
Power Pack Arrangement	
Hydraulic tank capacity	approx. 1550 1 (340 gal)
Hydraulic system working	pressure 140 bar (2000 lbf/2n)
Hydraulic system pilot pressure	31 bar (450 lbf/in)
Hydraulic pumps	One four section gear pump and one radial variable piston pump
Water system working pressure	20 bar (300 lbf/in)
Minimum water supply required	0.2 l/s (3 gal/min)
Electrical	
Electrical system voltage	ll00V, 60Hz
Power pack motor	180 kW (240 hp)
Cutting head motor	97 kW (130 hp)

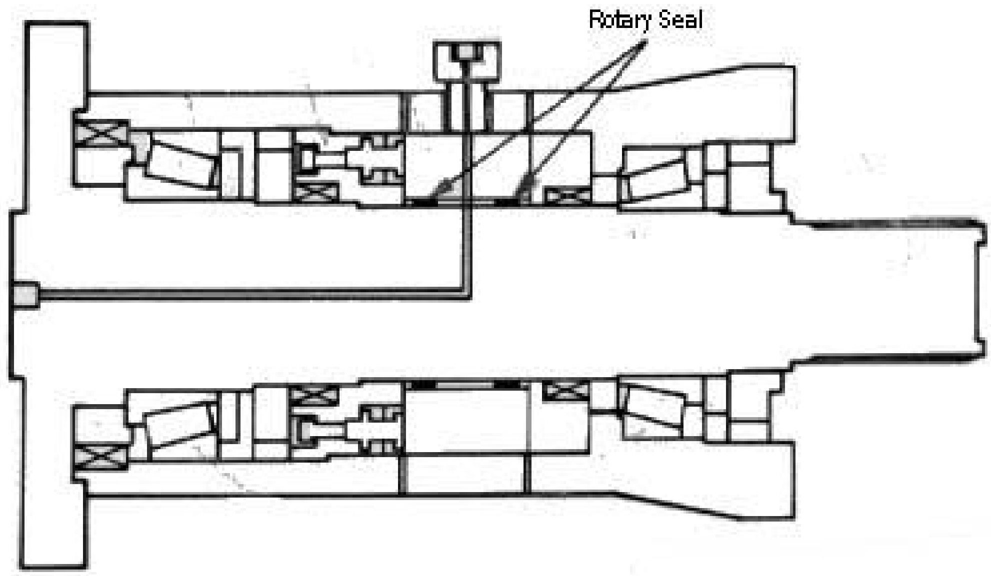

Fig. 1 Section Through MK2A Cutting Head Shaft Assembly

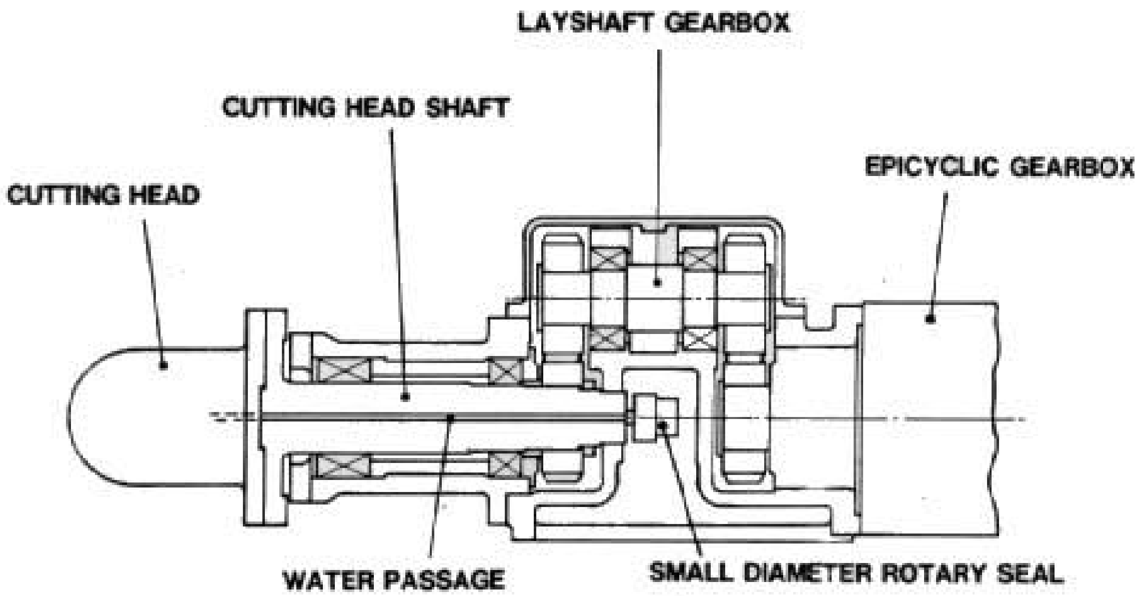

Fig. 2 Section Through Layshaft Gearbox

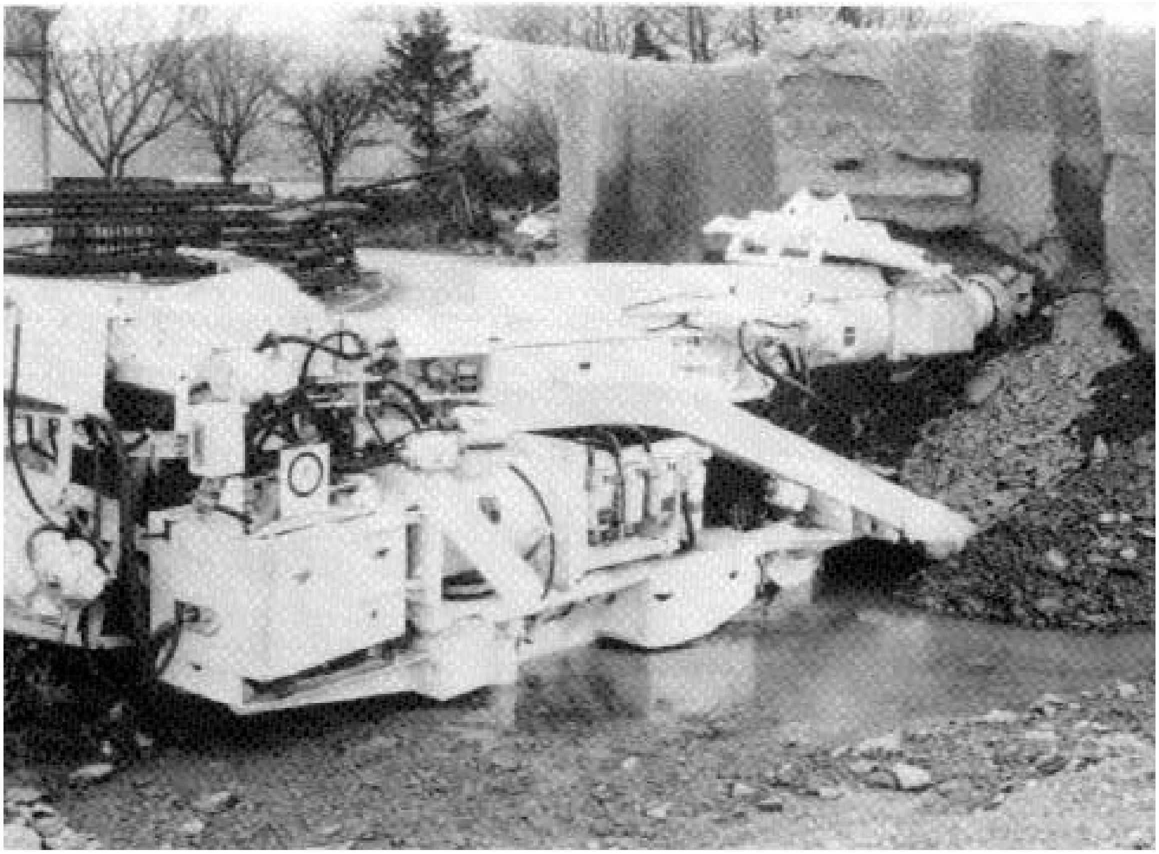

Fig. 3 Pre Production Machine on Dosco Test Site

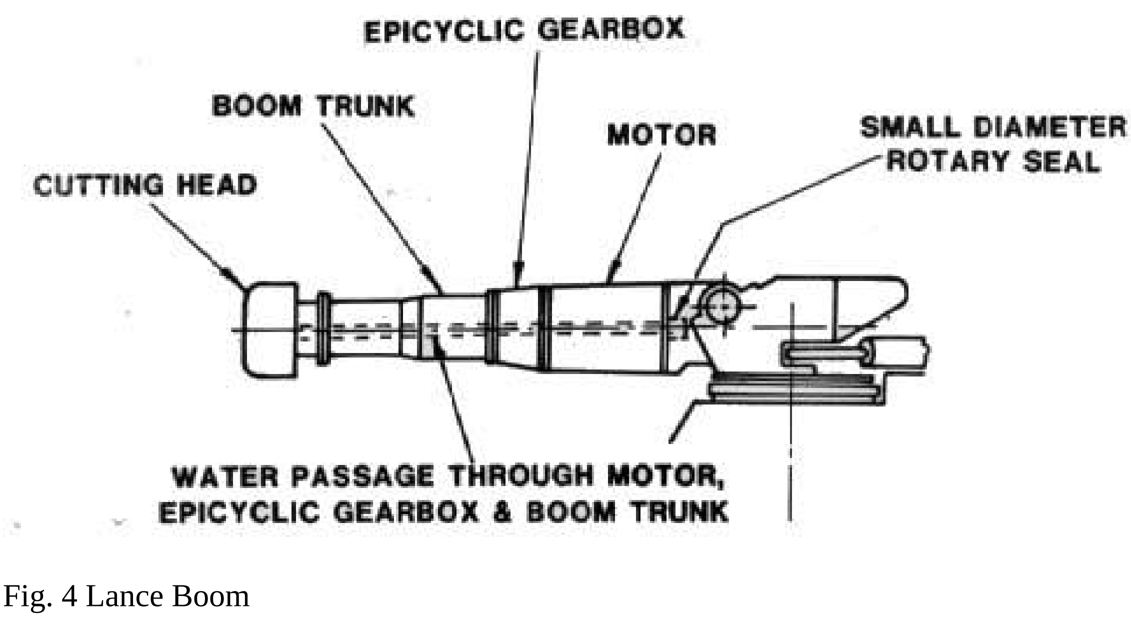

Fig. 4 Lance Boom

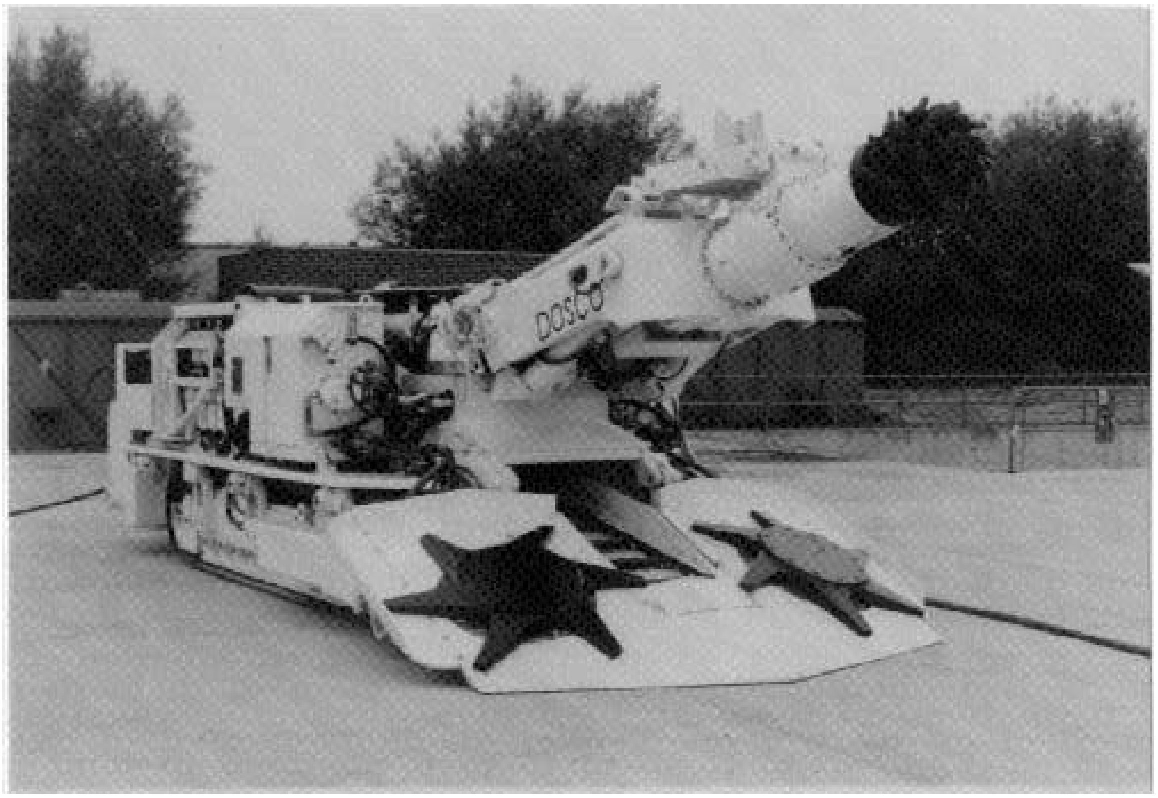

Fig 5. Dosco MK2B Roadheader

TESTS ON THE APPLICATION OF WATERJETS FOR BOOM-TYPE ROADHEADERS IN THE GERMAN HARDCOAL INDUSTRY

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ABSTRACT

At present 115 boom-type roadheaders are in service in the West-German hardcoal industry for the mechanized drivage of about 42 % of development roadways in coal and rock strata. In order to enhance the application of this successful driving method it should be possible to cut also harder rocks efficiently and economically. Bergbau-Forschung of Essen has therefore undertaken comprehensive tests on the assisting effect of high-pressure waterjets arranged on conventional cutting heads of boom-type roadheaders.

The tests are running on a surface test rig on an original scale. The drive of the cutting head has a capacity of 200 kW and a sluing force of 186 kN. The waterjets are produced by two intensifiers of a previous design, 2 x 250 kW, yielding a water pressure of 4000 bar at 60 l/min flow. The first experiments were carried out on a ripping type cutting head fitted with conical picks and jet nozzles of 0.6 mm diameter. The test rock consisted of high-strength concrete and of abrasive natural sandstone.

Early results showed a noticeable reduction of cutting force and tool wear when pressures of up to 1800 bar and chemical additives were adopted. Remote segmental control proved as a useful means of keeping water consumption low. Further trials to be carried out also underground will show where the technical and economic optimum for jet-assisted cutting lies. If the trials are successful, it is expected to widen the application range of boom-type roadheaders to 60 %.

GENERAL

Some 500 km of underground roadways are driven each year in the hardcoal industry of the Federal Republic of Germany. They are required to develop the coal seams and prepare them for winning operations. Most of them are driven in mixed strata. Their proportion of rock varies between 20 and 90 %. The balance is coal. Their average cross-section is 20 m^2.

To drive these development roadways, the West-German hard coal industry has been using boom-type roadheaders for more than two decades now. They are faster, with higher performance than drivages by drilling and blasting and have several advantages as to safety and ergonomics.

The number of these machines has been going up ever since 1972 (column on the fig. 1). 115 of them are in service at present.

Over time more and more development roadways have been driven by the boom-type roadheader. In 1984 their proportion amounted to some 42 % of all the mixed drivages produced (uppermost curve on fig. 1).

The average drivage rate attained in roadways using boom-type roadheaders amounts to 7 m per day (lowermost curve on the fig. 1). This is more than double the rate achieved with conventional drilling and blast-firing. There are, however, many mechanized roadways where much higher advance rates are achieved and where 20 m per day is nothing unusual.

There are at present 15 different designs of boom-type roadheaders in use in the West-German hard coal industry (fig. 2). They are offered by 6 different suppliers. Depending on the application they are meant for they are of different sizes, with weights varying between 24 t and 115 t. The installed capacity - for the cutting head alone - varies between 100 and 350 kW. Continuous improvement of the equipment is sought in order to achieve still better performances.

All of the machines are crawler-mounted and include a loading system (fig. 3). They are equipped with two different types of cutting heads, i.e. the milling type and the ripping type cutting head. The milling type cutting head performs better in soft rock, whereas the ripping type cutting head is better suited for harder rock.

These are the machines ideally suited for soft and medium-hard rock where cutting performances between 30 m^3/h and 80 m^3/h are attained, depending on the type of rock.

As soon as harder rock of higher abrasivity (F in fig. 4) is involved, cutting becomes more difficult and the performance declines. - The abrasivity factor F characterizes the abrasive properties of a rock. The factor is a product of tensile strength, quartz content and grain size of the quartz. - As is shown on the figure, pick consumption and pick costs tend to soar as soon as a certain hardness and abrasiveness of the rock are exceeded. This break-even point oscillates around a compressive strength σ_D between 80 and 100 N/mm^2 and an abrasivity factor F of 0.3 N/mm. If hardness and abrasivity of the rock are beyond these levels, picks with hard metal inserts either are subjected to high wear or they will break away.

Today, application of boom-type roadheaders is dictated by the abrasive behavior of the rock, for it is practically impossible to cut hard and abrasive rock economically. Wherever such rock is encountered unforeseen, blast-firing has to be adopted additionally. And wherever abrasive rock is expected, boom-type roadheaders cannot be used altogether.

Bergbau-Forschung in the Federal Republic of Germany therefore started five years ago seeking new approaches towards successful cutting of hard rock. Among the different physical means available to destroy rock Bergbau-Forschung selected jet cutting as it was not only the most promising method in terms of relatively fast implementation but also in terms of practical application (fig. 5).

By the definition 'jet cutting' we understand the combination of a known mechanical cutting tool with high-pressure waterjet. This means that the question is not to replace the previous, so-called conventional cutting technology by a completely new one, but rather to assist the former by means of additional energy provided by the waterjet.

Our deliberations as to the subsequent transfer of waterjet assistance to boom-type roadheaders underground start from the fact that within the economically viable range of mechanized entry drivages, i.e. when having to cut through soft to medium-hard rock, the known cutting methods will continue to be used (fig. 6). This means that a machine equipped with waterjets will, as soon as working in easily cuttable rock strata will function in the conventional way as it has hitherto.

The waterjets are to be adopted as an additional means only if the encountered proportion of harder rock and the to be expected high wear on the conventional tools would impair the economics of mechanized drivage. In practical operations this means that waterjets are needed exclusively if e.g. a band of sandstone has to be cut as well. It goes without saying that with such combined application there may also be encountered transitional zones or rock types which due to their extremely high abrasivity and hardness cannot be cut economically even when adopting waterjets.

With another concept of waterjets in combination with mechanical tools, cutting is permanently assisted by the jets, i.e. not only in hard, but also in soft and medium-hard rock. This allows the use of machines of a reduced size and reduced power input.

It will have to be proven by subsequent underground application which of the two concepts will make its way.

Preliminary tests, basic research and studies on the application of waterjets undertaken in other countries showed that the exclusive use of waterjets for destruction of hard rock is uneconomic.

The preliminary tests carried out at this juncture have led to the following conclusions: Waterjets alone can do only a limited amount of the cutting job so that mechanical tools to spall off the rock are indispensable. An ordinary, sharp cutting tool will be most advantageous from an energetic point of view. Cutting exclusively by means of waterjets would be inferior to tool-cutting, if one considers energy requirement. Any combination should therefore be selected so as to optimally utilize the specific advantages of waterjets and have at the same time done most of the rock-loosening job by mechanical picks. In this case the picks subjected to the highest load are assisted by waterjets in order to enhance performance at lower pick wear.

The application of such combination between waterjets and mechanical cutting tools according to the above principle would therefore not cause any problems to be integrated in the known designs of boom-type roadheaders.

INVESTIGATIONS WITH BERGBAU-FORSCHUNG GMBH

The investigation program 'Jet-Assisted Rock Cutting' initiated by Bergbau-Forschung was started in 1979 by preliminary tests on separate picks or groups of picks (fig. 7). These tests provided us with the knowledge on the required type of nozzle, nozzle diameter, optimum stand-off distance between the nozzle and rock, required water flow and most favorable orientation of the jet in relationship to the point of the mechanical cutting tool which in our case was a point-attack type pick.

Planning and constructional design of the waterjet components to prepare the first trials on a ripping type cutting head took a long time. At first the valves and swivels did not sustain the high operational loads at water pressures up to 2500 bar and had to be improved. Also construction and installation of the test rig took considerable time as a complete ripping type cutting head had to be equipped with high-pressure lines.

The trial phase then started in 1983. Afterwards, by mid 1984, the test program properly speaking could be commenced.

When applying waterjets to boom-type roadheaders to be used in carbonaceous rock, a total water flow of 60 l/min shall not be exceeded (fig. 8). With the multitude of picks and nozzles arranged on one cutting head and given the water volume per nozzle which is determined by the nozzle diameter and water pressure, the water supply had to be restricted to that part of a rotating ripping type cutting head which was actually cutting into the rock. Waterjets were moreover not allowed to be discharged into the area behind the roadheader since they could endanger both humans and equipment. Prerequisite for using waterjets with ripping type cutting heads therefore was proper dosage of the water which finally was attained by means of segmental control.

The segmental control which suited our purposes and included the remotely controlled preselection of 16 segments, was developed and manufactured by Flow Industries, Kent, Washington (fig. 9). The unit functions on electro-hydraulically controlled high pressure valves. Having 16 segments available for selection it was possible to define the most advantageous combination and number of operating waterjets in the different cutting situations. The most favorable constellation was 3 or 4 segments at a time, equaling 12 to 15 jet nozzles permanently gushing out water.

After these preliminary trials the segmental control was integrated in the test rig (fig. 10). The picture shows on one side the complete cutting head including point-attack type picks and waterjets built into the original roadheader and on the other side a complete control valve unit including the swivel for the high-pressure water.

The tests at the rig of Bergbau-Forschung on a ripping type cutting head driven by a 200 kW motor were commenced in mid 1984. The cutting speed at the largest diameter of the pick periphery amounts to 1.8 m/sec. The cutting head is equipped with a total number of 75 picks with one nozzle each of 0.6 mm diameter. High-pressure water is supplied by two hydraulically powered intensifiers of an older design and 2 x 250 kW drive capacity. They deliver 60 l/min water flow with a maximum pressure of 4000 bar.

Trials started with a test block of 45 m^3 consisting of a hard special concrete (fig. 11). The high-pressure components as well as the complete system as such were tested first. Another objective was to verify, on a complete cutting head system, the stringency of theoretical knowledge on waterjet technology as had been derived both from literature and experiments. This led us to certain corrections e.g. at the nozzle position, segmental control, and cutting sequence.

The following trials were to supply information on the effectiveness of waterjet assisted point-attack type picks in very hard and abrasive natural rock. To the effect were cast into the test block big sandstone cubes of 2.2 N/mm abrasivity factor and high tensile strength, as shown on the picture.

FIRST TEST RESULTS

As far as the tests run with the set-up as outlined and on an original scale are concerned, no more than some early interesting results are available so far. They are based on individual tests and indicate certain trends. Subsequent tests are necessary to back up these first findings.

During the tests were measured both the action of waterjets on the sandstone - the cutting tracks are clearly visible on the fig. 12 - as well as the wear on the hard metal tools. Furthermore were determined any and all machine parameters, e.g. forces, torque, rotation speed, sluing speed, water pressure, water flow and the volume of rock cut.

During the first test series we varied the nozzle position in relationship to the pick (fig. 13). It was found out that the nozzle position directing the waterjet in front of the pick was most appropriate to transfer the jet energy into the rock. In this case the point of impact of the waterjet should be as close as possible to the pick without touching it though, - in order to avoid cavitation at the hard metal pick. In addition, the nozzle should not stand off more than 30 mm from the jet impact point on the rock. This configuration may nevertheless give rise to damages at the nozzle. For this reason we used high-polymeric additives of the Nalco type during all of the tests. This allowed an increase in nozzle stand-off distance to a maximum 50 mm, with the jet concentration being maintained.

The nozzle diameter was varied from 0.3 to 0.6 mm. 0.6 mm nozzles of sapphire gave the best results. Since for practical reasons the water flow had to be limited to 60 l/min, no bigger diameter was allowed although segmental control was available.

The first thing which was derived from the outcome of the above tests was the dependency of sluing force on water pressure at the nozzles (fig. 14). The sluing force represents the force required to move the ripping type head horizontally at constant rotational and sluing speed. The reduction of sluing force in the higher ranges of water pressure is obvious although the individual values show a wide scatter which is probably due to the procedure during individual tests. The tendency of a noticeable reduction of sluing force by 36 % along with, let us say, 1000 bar increase in water pressure becomes apparent though.

Waterjets exert an equally noticeable influence on the wear of point-attack type picks (fig. 15). To this effect is determined the weight loss on the whole batch of tools of the cutting head. Here, again, the tendency goes to a marked reduction in pick wear along with the higher water pressure; such reduction amounts to more than one third if a by 1000 bar higher water pressure is applied.

Another beneficial effect is the decrease in torque at the cutting head with increasing water pressure. This is evidence of lower pick loads at higher water pressure. One may vice versa conclude that with constant torque and sluing force an enhanced cutting performance can be expected from the action of waterjets. This means also that cutting of harder rock becomes possible.

Besides the measured values several other results were yielded which demonstrate the advantages of waterjets. Part of this was evidenced by high-speed photography whereby it became visible that hard metal tools attacked the rock having been loosened by waterjets more aggressively so that deeper cutting depth were obtained.

Another important factor for the application of waterjets is the energy balance. The lower curve in fig. 16 shows the electro-mechanical drive power of the ripping type head alone. Additional energy has to be put in, of course, for generating high-pressure water. To produce e.g. a water flow of 60 l/min at 1000 bar, an additional input power of some 120 kW is required. As is represented, this roughly equals the power required for the ripping type head alone. The total drive power (upper curve in fig. 16) goes up with increasing water pressure. When applying 1500 bar pressure, a total power of about 315 kW is required. The above shows, however, that such input power is well within the range typical for heavy boom-type roadheaders of recent design. A technical and economical optimum for the combination between waterjets and mechanical cutting tools is therefore expected, in this order of magnitude of energy expenditure. It has to be admitted though that the economic aspect has not been the main point of interest so far. What matters now is the verification of the technical applicability of jet-cutting.

Upon completion of more thorough testing on the ripping type cutting heads, Bergbau-Forschung plans to carry out similar tests on cutting heads of the milling type. The tests will be paralleled by underground trials. To the effect design modifications e.g. at the cutting head are already been carried out. The most critical point is to protect the high pressure lines. Furthermore a segmental control reduced in size and limited to 8 segments will be required to be received within the cutting head.

Besides mere cutting assistance by waterjets the relevant tests also had some attractive side affects. Just to mention the effective cooling of the cutting tools. In addition, waterjets avoid the formation of undesirable sparking in abrasive rock and dust development during the cutting action is largely reduced. In view of a subsequent practical use, the waterjet equipment allows at the same time a segment controlled interior nozzle arrangement for water pressures around 200 bar. Such water spray system is nowadays essential for safe rock cutting with any boom-type roadheader.

PROSPECTS

The development activities in the field of rock cutting by boom-type roadheaders, as far as the Bergbau-Forschung is concerned, are focussed on jet-assisted rock cutting. If after the encouraging start of our endeavors some optimum can be found reconciling the technical and economical expenditure with the cutting efficiency of jet-cutting, the proportion of mechanical roadway driving could be greatly enhanced. We expect to broaden the application range from now 42 % to ultimately 60 %, since jet-cutting would allow us to economically handle also abrasive rock of abrasivity factors up to 1.0 N/mm. If the expectances are met, the known designs of boom-type roadheaders can also be used in pure rock roadways.

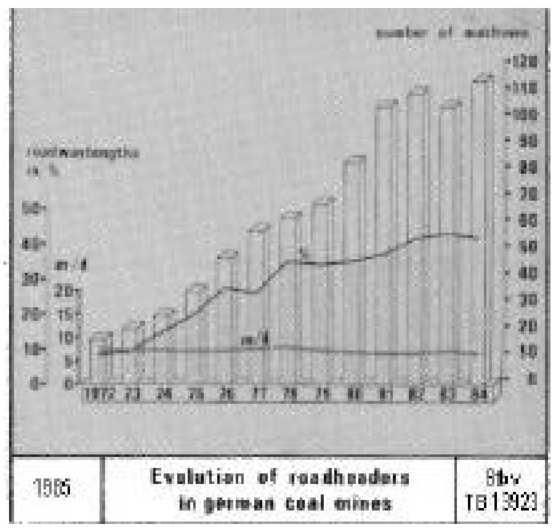


Figure 1

Figure 2

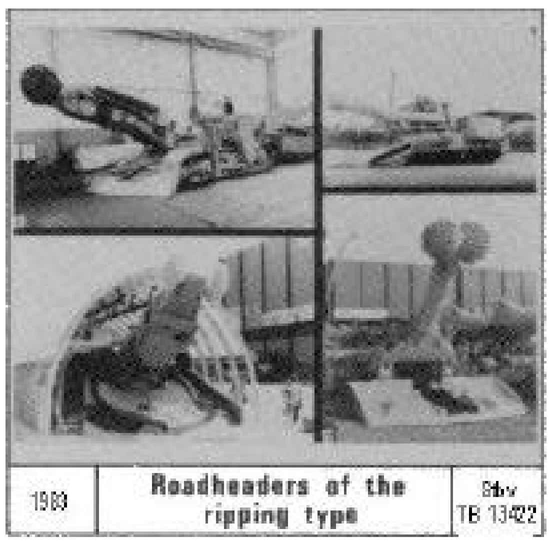


Figure 3

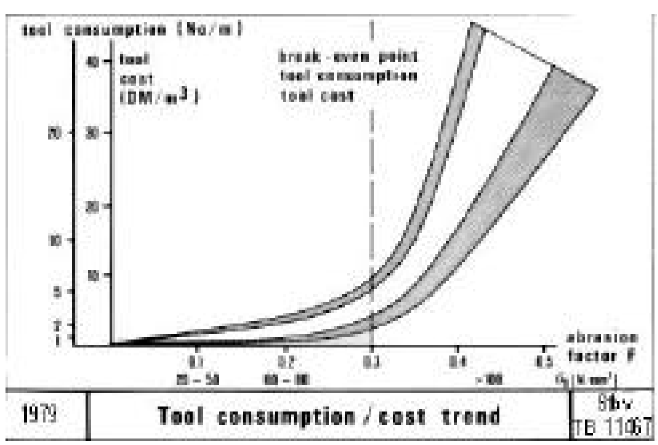


Figure 4

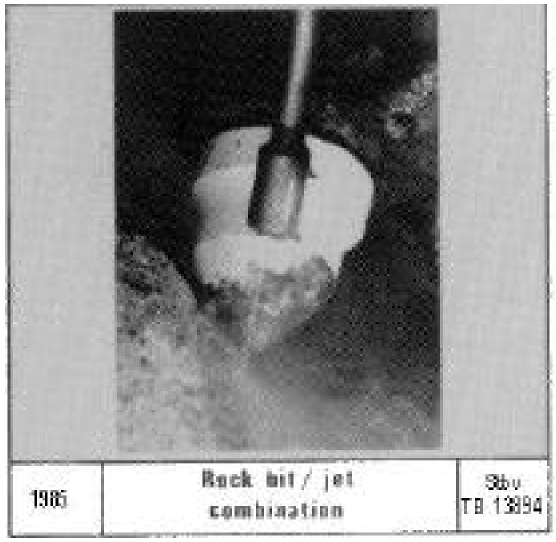


Figure 5

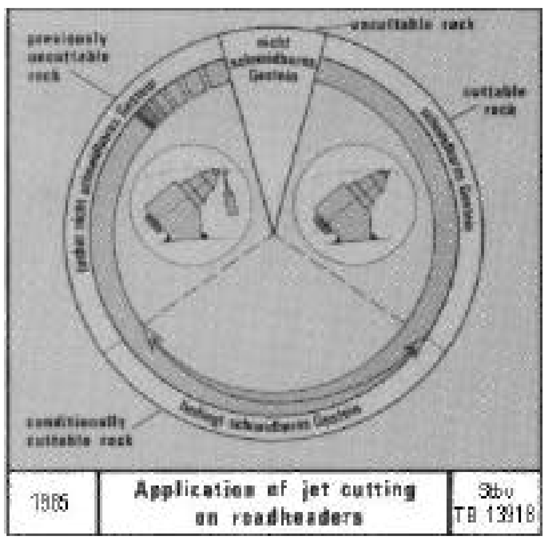


Figure 6

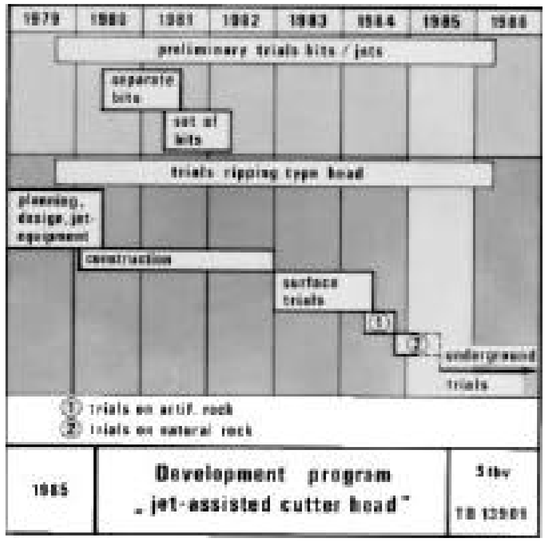


Figure 7

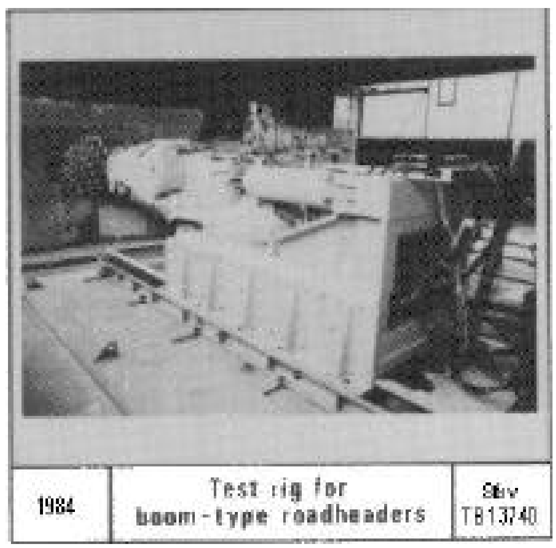


Figure 8

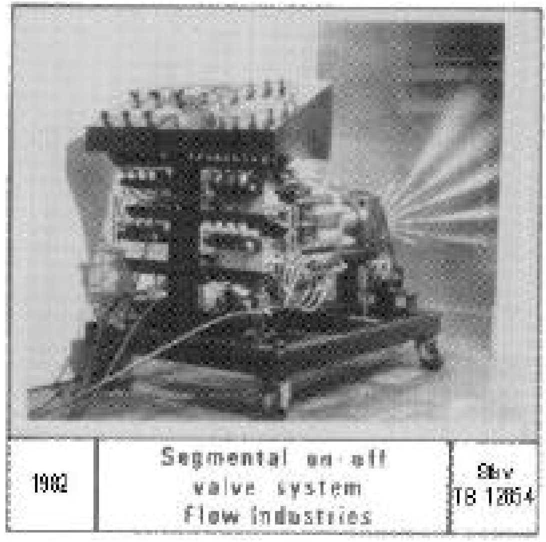


Figure 9

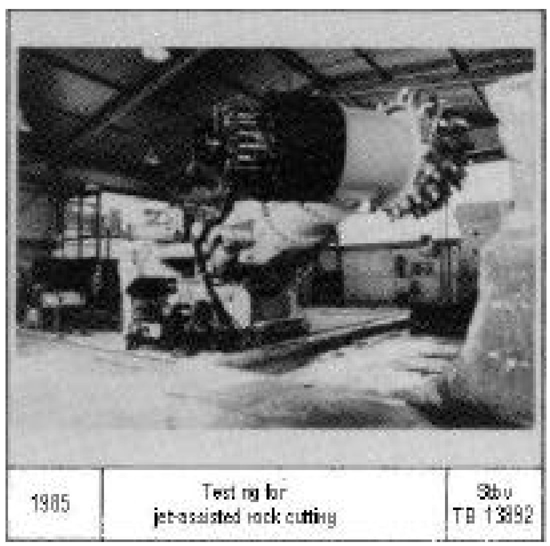
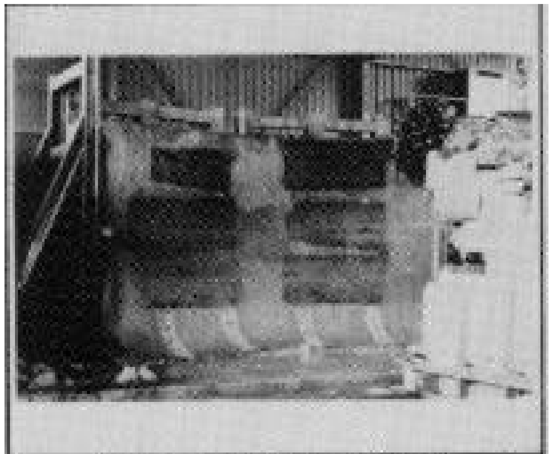
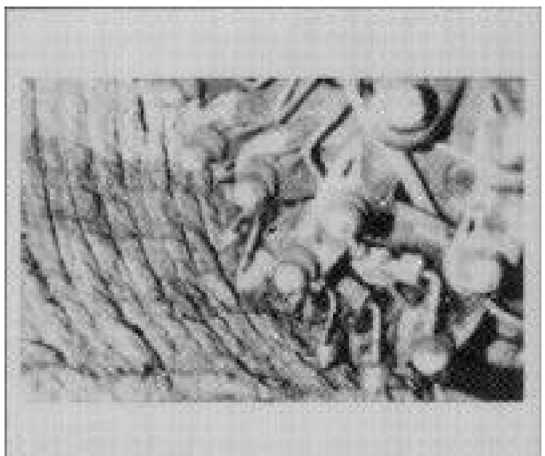


Figure 10



1985 | Test rock for jet assisted cutting | Stbv TB 13899

Figure 11



1985 | Tracks of jet-assisted cutting | Stbv TB 13848

Figure 12

Figure 13

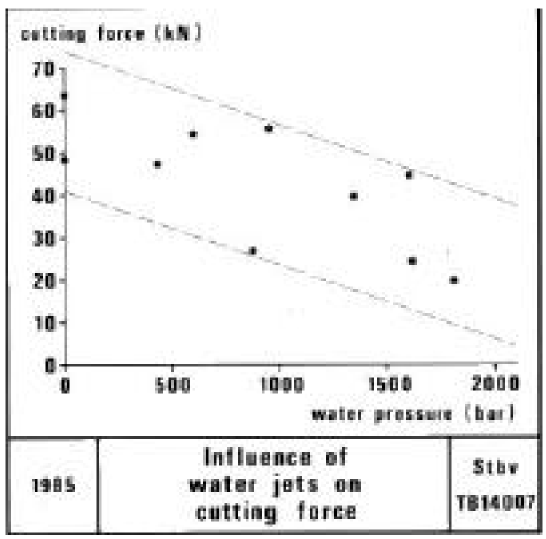


Figure 14

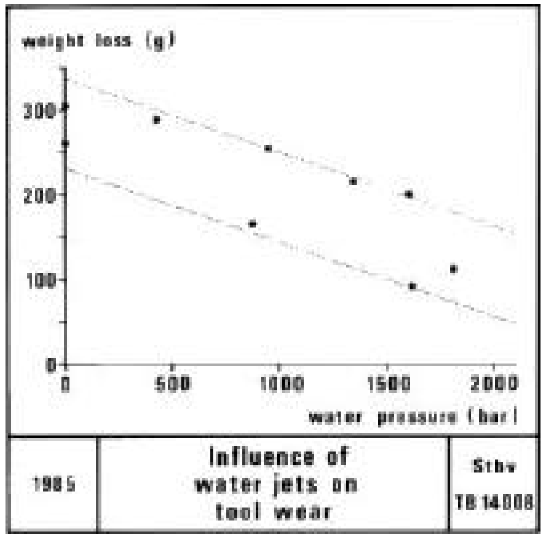


Figure 15

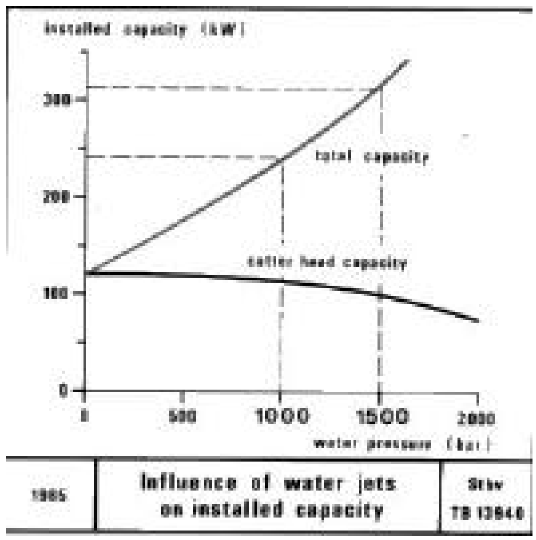


Figure 16

AN ASSESSMENT OF ROTATING HIGH PRESSURE WATER JETS FOR DRILLING AND SLOTTING OF HARD ORE BEARING ROCKS

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ABSTRACT

An extensive program is underway in the laboratory to assess the feasibility of using the rotating high pressure water jets for drilling and slotting of hard ore bearing rocks. The samples of rocks were supplied by a consortium of Canadian mining companies. In this paper preliminary results obtained with both plain and carbide tipped nozzle bodies of various configurations are presented. The maximum pressure and flow were 103.5 MPa and 88 liter/min respectively. The rotational speed was 240 RPM. The main conclusions from these preliminary results were as follows: 1. To penetrate or slot these rocks at the rates specified by the mining companies (a 60 cm/min for drilling) would require pressures beyond 100 MPa. 2. The performance of the carbide tipped nozzle bodies was significantly better than the corresponding plain nozzle bodies. 3. Microstructural properties such as grain size appear to control the penetrability of water jets through rocks.

NOMENCLATURE

A_H	Mean cross-sectional area of the drilled hole, cm^2
A_I	Cross-sectional area of the inner orifice in the nozzle body, cm^2 (Fig. 1)
A_N	Cross-sectional area of nozzle body, cm^2
A_O	Cross-sectional area of outer orifice(s) in the nozzle body, cm^2 (Fig. 1)
A_S	= kerf area, cm^2
A_T	Total cross-sectional area of orifices in the nozzle body, mm^2
C	Compression wave velocity in the rock, m/sec
D_B	Diameter of nozzle body, cm
D_H	Diameter of hole, cm
D_I	Diameter of inner orifice, mm (Fig. 1)
D_O',D_O''	Diameter of outer orifices, mm (Fig. 1B)
E_A	Specific kerfing energy, J/cm^2
E_R	Modulus of deformation of rock, MN/m^2
E_V	Specific energy based on volume of material removed, J/cm^3
f	Porosity of rock (%)
H_P	Total hydraulic power of jets, kW
k	Permeability of rock, μD
L	Length of cylindrical outlet of nozzle, cm (Fig. 1)
ℓ	Mean grain size of rock (mm)
M*	Material factor (uncorrected)
N	Rotational speed of drill, RPM
P	Pressure at nozzle inlet, MPa
r',r''	Radial distances between orifices in the nozzle body, mm

R	Mean drilling rate (feed rate), cm/min
S	Standoff distance, cm
s'	Total surface area of grains per cm^3 of rock, $(\mu m)^2$
V_{TR}	Traverse speed of the rock under the jet, cm/s
θ_I	Angle of inclination of inner orifice, deg. (Fig. 1)
θ_o	Angle of inclination of outer orifice, deg. (Fig. 1)
α	Nozzle entry angle, deg.
ρ_R	Dry bulk density of rock, kg/m^3
σ_C	Uniaxial compressive strength of rock, MN/m^2
σ_t	Uniaxial tensile strength of rock, MN/m^2

INTRODUCTION

The program on cutting (drilling, slotting and fragmentation) of rocks with rotating high pressure water jets has been active in the Gas Dynamics Laboratory for the past several years (Refs. 1 to 4). In the work reported to date, the main objectives were the following:

1. To study the influence of nozzle body size (scaling at constant power input) and nozzle body configurational parameters, viz., the angle of inclination of inner and outer jets (Fig. 1) to the axis of rotation, disposition of total available flow between the inner (D_I) and outer (D_o) orifices and the shape of the nozzle entry, conical or straight on nozzle body performance.
2. To study the effect of rotational speed on specific energy.
3. To clarify the influence of rock properties on the penetrability of water jets through rocks.

In order to achieve these objectives, drilling and slotting tests were conducted in the laboratory on unconfined blocks of Muskoka pink granite (Refs. 1 to 4), Ottawa limestone (Ref. 2), Anabel marble (Ref. 2) and Nepean sandstone (Ref. 3). All the tests were carried out with a series of dual orifice nozzles at three different rotational speeds (120, 240 and 420 RPM) and at pressures up to 69 MPa, the maximum hydraulic power being 58 kW. These earlier results (Refs. 1 to 3) showed that the optimum nozzle configuration depends on the type of rock material and the rock properties which control the penetrability of water jet are micro (grain size, etc.) and not the macro (compressive strength, etc.) properties; low rotational speeds (~ 200 RPM) are close to optimum. In Ref. 3, an attempt was also made to compare the slotting performance of rotating jets with a presently existing technique in a quarry. This brief comparison clearly demonstrated the potential of water jets for quarrying and mining applications. Encouraged by these findings, work was extended into drilling and slotting of hard ore bearing rocks. This was requested by a consortium of Canadian mining companies consisting of Falconbridge, Inco, Kydcreek and Noranda. The purpose was to obtain data to evaluate the feasibility of using the moderately high pressure (~100 MPa) rotating water jets for mining applications.

The work is still in progress and therefore only preliminary results are presented in this paper. Tests were conducted at pressures up to 103.5 MPa on unconfined blocks of samples (Fig. 5) received from Falconbridge, Inco and Noranda. The penetration rates in these samples were found to be considerably lower than those achieved in granite at the same operating conditions (Refs. 1 and 2). An attempt has been made to account for this in terms of the microstructural properties of the rock samples. These results also clearly indicated that to achieve the penetration rates demanded by the mining companies (~ 60 cm/min for drilling) would require pressures beyond 100 MPa.

EQUIPMENT AND PROCEDURE

The equipment consisted of two high pressure pumps, a rotating device, a traverse trolley and a set of nozzle bodies. The first pump was a Wheatley quintuplex pump driven by a 112 kw diesel engine (GM-6V53). The pump was rated to deliver 87.8 liter/min at 83 MPa. The second was custom built to NRC specifications by Indescor Hydrodynamics Inc. This pump was capable of delivering 76 liter/min at the rated pressure of 103.5 MPa. Nozzle body configurations for both large (> 1.905 cm) and small size (0.9525 cm) nozzle bodies are shown in Fig. 1. The design specifications of the nozzles employed in this program are listed in Table 1. It should be noted that some of the configurations in this table are new. They were designed on the basis of the results obtained previously in the laboratory (Refs. 1, 2 and 3). More details on these nozzles are given in a subsequent section. Experimental procedure and method of analysis of data for drilling and slotting were the same as described in Refs. 1 and 3 respectively.

TEST MATERIALS

The test materials were unconfined blocks (up to 50 cm in length in the direction of drilling) of ore bearing rocks received from Falconbridge, Inco and Noranda mines. The rock samples were highly inhomogeneous as will be evident from the results presented later. A number of properties of the rock samples were measured in the laboratory and these are listed in Table 2. Although permeabilities were measured at various effective pressures (confinement pressure-pore pressure), for brevity the value at only one effective pressure is listed in the table. Petrographic analyses of the samples were also performed and these are depicted in Figs. 2, 3 and 4. A brief description of the composition of the rock samples are given below (granite for comparison only).

Noranda (Figs. 2A and B)

Two main rock types occurred in these samples, viz., quartz diorite and biotite uralite gneiss. Ore content varied from 5 to 75% of the rock (high density reflects high ore content in the rock). The ore is composed mainly of pyrrhotite (up to 51%) with small amounts of chalcopyrite (4 to 8%) and pentlandite. The rock is very fine-grained (Table 2) and the volume of cracks measured amounted to about 1.5%. An examination of the drill chippings revealed that fractures occurred in chalcopyrite and not through the harder pyrrhotite.

Falconbridge (Figs. 3A and 3B)

This rock is an altered norite of somewhat variable composition. The least altered norite (Fig. 3A) contained about 35% plagioclase feldspar while the most altered contained about 14%. The quartz content varies from about 5% in the least altered rock to 20% in the most altered. The amount of ore minerals (pyrrhotite, pentlandite, chalcopyrite and cubanite) varied from 10 to 25% of the rock. The grain sizes varied from 0.24 to 0.434 mm (very fine-grained) and the volume of cracks was about 0.5%.

Inco (Fig. 4B)

The rock is a very fine-grained chloritized amphibolite containing ore minerals, the other major component being quartz (14%). The ore content varied from 5 to 90% of the rock. The samples also contained 5% plagioclase feldspar, 5% pyroxene and in some areas up to 25% hornblende. The volume of cracks amounted to about 0.21%. The ore consisted mainly of pentlandite, pyrrhotite, pyrite and chalcopyrite.

Muskoka Pink Granite (Fig. 4B)

The rock consisted essentially of quartz, feldspar and amphibole. The size of the individual crystals varied, the maximum being just over 1 cm. Well developed cleavages occurred in the amphibole. Microcracks are mostly confined to the grain boundaries (Fig. 4B). The volume of cracks was about 18% which is considerably higher than the ore bearing rocks described above.

EXPERIMENTAL RESULTS

The rock samples received from the mines were highly inhomogeneous and as such many difficulties were encountered in drilling or slotting them. Some of these are shown qualitatively in Fig. 5 where holes drilled in Falconbridge and Inco and slots cut in Noranda are depicted. In Falconbridge samples (Fig. 5A) due to the presence of planes of weakness, the holes drilled with plain nozzles were oval in shape and the rock often tended to split along these planes. In Inco samples (Fig. 5B), on the other hand, surface spatting (which prevented the free movement of the nozzle) was a major problem. The carbide tipped nozzles appeared to offer some advantages in this respect, for the frequency of such occurrences was reduced considerably with their use (Fig. 5C).

The experimental data acquired in the investigation are plotted in Figs. 6 to 13. For the sake of clarity only a few plots are presented here. More details are given in Ref. 5. The data pertaining to granite were taken from Refs. 1 and 2. Essentially the plots show the influence of nozzle body parameters (θ_I θ_o, scaling, etc.) and the pressure (rotational speed was maintained constant at 240 RPM) on specific energy (E_V or E_A) for drilling and slotting. The effect of rock property (M* = material factor) on the rate of penetration is summarized in Fig. 13. Data points are connected by lines only for the sake of clarity.

DISCUSSION

It should be stated at the outset that there is a great deal of scatter in the data due to the high inhomogeneity of the rock samples. This will become obvious later when the effect of pressure on specific energy (Fig. 10) is discussed. Consequently, some of the statements made here might be dubious and need to be verified by further work in the laboratory. These preliminary results have been quite useful in planning this further work in the laboratory.

Figure 6 shows the influence of the outer angle (θ_0) of the jets on the specific energy (E_V) for drilling. It is quite obvious that the ore bearing rocks were much more difficult to drill than granite. An examination of Table 2 (properties of rocks) reveals that neither the compressive nor the tensile strength appear to control the penetrability of waterjets through the rocks (more about this later). The outer angle has, as observed in the previous investigations (Refs. 1 to 3), a profound influence on the nozzle body performance. The optimum angle, once again, appears to be a function of the rock type. While it is in the neighborhood of 20° for granite, for ore bearing rocks it seems to lie beyond 30°.

Figure 6 brings out two more important points. The first one is concerned with the increase in power loading by increasing the flow at a constant pressure. Nozzles A10 and All were designed to allow for the increased flow of 87.8 liter/min at 69 MPa (see Table 1). Increasing the flow was found to have an adverse effect on specific energy (compare All with C2). Reducing the outer angle (it was felt, as discussed in Ref. 3, that scaling, that is, altering A_T/A_N, should be accompanied by changes in the basic nozzle body configurational parameters) had even more adverse effect (compare A10 with All for Noranda and Falconbridge). This is either due probably to the inhomogeneity at the locations drilled or, since the rock samples were fairly impermeable (Table 2), the large quantity of water could not simply move fast enough through the rock to enhance the penetration rates. The second one is to do with the use of carbide tips on the plain nozzles. Comparison of 5TP with C2 or 6BTP and 6TP with A6 (these three nozzles were three orifice nozzles designed to balance the jet reactive forces) clearly shows that the performance of the tipped nozzles was significantly superior to that of the plain nozzles. It should be noted that balancing the nozzles seems to improve their performance even better (compare 5TP with 6BTP). This is probably due to the fact that an unbalanced nozzle would not run true causing jamming and other problems, yielding apparently high values of specific energy. These problems are reduced to a large extent by the balanced nozzles.

In Fig. 7 the effect of changing the inner angle for a constant outer angle of 20° on specific energy is indicated. In all the nozzle bodies used in the previous investigations (Refs. 1 to 3), inner angle was maintained constant at 10°. This was based on purely geometrical considerations. That is, disregarding the influence of the rock properties, for the nozzle body sizes involved, the inner jet should impinge on the rock surface as close to the axis of rotation as possible. This is borne out by the results plotted in the figure. An increase in the inner angle from 10 to 20° had a negative effect on specific energy. Nozzle A8 was balanced (the jets emerged in the opposite directions with respect to the axis of rotation) whereas A7 was not. Based on the arguments given above, the performance of A8 should have been just about the same or better than A7. The discrepancy is probably due to the scatter in the data as stated elsewhere. The ease with which granite can be drilled compared to the ore bearing rocks should be noted.

The next configurational parameter of interest is A_O/A_T (flow through the outer orifice/total available flow). Its influence on specific energy is shown in Fig. 8. Although the trend is not clear, the optimum value of A_O/A_T appears to be in the neighborhood of

0.8 for the ore bearing rocks. As discussed in Ref. 2, the grain size and the manner in which they are plucked by the high speed water should influence this parameter.

The effect of scaling, that is, varying the size of the nozzle body for a given power input, on specific energy for drilling is depicted in Fig. 9. Several observations can be made from this figure, but owing to scatter in the data, their validity becomes doubtful. For Noranda samples, for the nozzle bodies used in the earlier investigations (C2, D2, D3 and SC2-3), E_V remains virtually constant with scaling. On the other hand, for the newer nozzle bodies (series A in Table 1) which were designed on the basis of the findings in the earlier investigations (Ref. 3), the trend is not clear. The figure does indicate however that simple scaling (increasing D_B at constant $A_T/A_N = 0.008$) has adverse effect on specific energy (compare A12 with C2 for Noranda). It is interesting to note that for the nozzles with $A_T/A_N = 0.032$, the straight entry nozzles did better than their counterparts, the conical entry nozzles (compare C2 with SC2-3 and Al with A2). This is encouraging because as the nozzle body size is reduced, it becomes difficult to incorporate conical entry orifices in the nozzle body.

In Fig. 10, specific energy is plotted against nozzle pressure for drilling. Here the scatter (indicated by arrows) in the data is quite obvious, for as the pressure is increased one would expect the specific energy to decrease (Ref. 2). The observed fluctuations in the magnitudes of specific energy for Noranda (nozzle C2) and Inco (nozzle D2) can only be attributed to local inhomogeneities in the rock samples. Once again the superior performance of carbide tipped nozzles should be noted.

The very limited results obtained on slotting of Noranda samples with rotating water jets are plotted in Figs. 11 and 12. As discussed in great detail in Ref. 3, for slotting, the specific kerfing energy (E_A) is a measure of performance. The data indicate that for slotting with plain nozzles, small nozzles ($D_B = 9.525$ mm, $A_T/A_N = 0.032$) with smaller outer angles ($\theta_O \approx 15°$) are better than the large nozzles ($D_B = 19.05$ mm). However, as pointed out earlier, due to the complicated structure of the rock samples, the slot width was often non-uniform and this posed considerable difficulties for traversing the plain nozzle body over the length of the slot during the second and subsequent passes. In this respect, the carbide tipped nozzles had a definite advantage over the plain nozzles (see nozzle 6BTP in Fig. 12. See also Fig. 5C.) How far the size of the carbide tipped nozzles could be reduced remains to be investigated.

Now coming to the question of rock properties, it has already been shown (Refs. 2 and 3) that microstructural properties control the penetration of water jets through the rocks. In Ref. 3, a correlation was proposed for the rate of penetration in terms of a parameter, termed the material factor, M* (total surface area of the grains per unit volume of rock). It was shown that as the material factor increased, the rate of penetration decreased. Since the ore bearing rocks were fine-grained compared to granite (Table 2 and Figs. 2, 3 and 4), their material factors should be relatively large, and therefore they should be much more difficult to penetrate. Figure 13 where the rate of penetration is plotted against the material factor strongly confirms this hypothesis. The correlation could be made more realistic by taking into account the presence of cracks, micro fractures and the permeability which augment the rate of penetration. It is believed that this can be accomplished by correcting the material factor to account for these properties. A method is presently under investigation.

CONCLUSIONS

The conclusions from the preliminary results obtained in this investigation were as follows:

1. The drilling and slotting of ore bearing rocks with rotating high pressure water jets are technically feasible. However, to achieve the penetration rates specified by the mining companies (a 60 cm/min for drilling), pressures beyond 100 MPa would be required.
2. Carbide tipped nozzles have definite advantages over plain nozzles and deserve further study.
3. Microstructural properties of rocks control the rate of penetration of water jets. Further work to confirm these observations is in progress in the laboratory.

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Table 1. Design Specification of Nozzle Bodies (Fig. 1)

D_B = 19.05 mm, A_T/A_N = 0.008, θ_I = 10°, ~ = 20°

r/D_B = 0.304 [For all nozzles, except where noted]

NOZZLE TYPE	D_I (mm)	D_O' (mm)	D_O'' (mm)	θ_O (Deg)	REMARKS
A1	1.092	1.321	-	15	D_B = 9.525, A_T/A_N = 0.032
A2	1.092	1.321	-	15	D_B = 9.525, A_T/A_N = 0.032, α = 180°
A3	1.067	0.660	1.100	15	D_B = 9.525, A_T/A_N = 0.032, α = 180°
A4	1.092	1.321	-	15	D_B = 12.70, A_T/A_N = 0.018, α = 180°
A6	1.067	1.067	0.742	20	r'/D_B = r''/D_B = 0.300, α = 180°
A7	1.181	1.181	-	20	θ_I = 20
A8	1.181	1.181	-	20	θ_I = 20 and Balanced
A9	0.742	1.511	-	20	A_O/A_T = 0.8
A10	1.400	1.750	-	15	A_T/A_N = 0.0138
A11	1.400	1.750	-	20	" "
A12	1.400	1.750	-	20	D_B = 24.9 mm
C1	1.181	1.181	-	20	A_O/A_T = 0.50
C2	1.092	1.321	-	20	A_O/A_T = 0.60
C3	0.914	1.400	-	20	A_O/A_T = 0.70
C2	1.092	1.321	-	20	D_B = 9.525, A_T/A_N = 0.032
SC2-3	1.092	1.321	-	20	D_B = 9.525, L/D = 7.5, α = 180°
D2	1.092	1.321	-	25	
D3	0.914	1.400	-	25	
D2	1.092	1.321	-	25	D_B = 9.525, A_T/A_N = 0.032
D3	0.914	1.400	-	25	D_B = 9.525, A_T/A_N = 0.032
E2	1.092	1.321	-	30	
E3	0.914	1.400	-	30	
5TP	1.067	1.300	-	20	CARBIDE TIPPED NOZZLE (Fig. 1C) D_B = 22.2 mm (WITH TIPS)
6BTP	1.067	1.067	0.742	20	TIPPED AND BALANCED, r'/D_B = r''/D_B = 0.300
6TP	1.067	1.067	0.742	20	TIPPED AND BALANCED, r'/D_B = r''/D_B = 0.300, L/D = 6, α = 180°

Table 1. Design Specification of Nozzle Bodies (Fig. 1) D_B=19.05mm, A_T/A_N=0.008, Θ_I = 10°, Θ_D = 20°, r/D_B=0.304 [For all nozzles, except where noted]

Property	Mean Value	Range	Coefficient of Variation (%)*
ρ_R	2627	2603-2649	0.5
σ_t	8.0	5.3-10.0	16.6
σ_c	165.5	139-175	7.2
C	4115	3767-4456	4.8
E_R	44540	37230-52400	9.9
Grain size, mm	1.35	0.2-4.4	14.9
Porosity, %	1.235	0.61-2.65	56.9
M*	4.0		

1. Mushkoka Pink Granite (Ref. 1)

ρ_R	3166	2710-4260	17.7
σ_t	12.9	5.2-27.0	60.6
σ_c	154.2	92-227	30.2
C	5200	3490-6577	17.1
E_R	90500	34500-183400	45.9
ℓ, mm	0.222	0.103-0.341	53.6
f (%)	1.47	0.74-2.20	-
κ	0.072	(at an effective pressure of 1.84 MPa)	
M*	24.40		

2. Noranda Samples
* Standard deviation expressed as a percentage of the mean value.

Table 2. Measured Properties of Rocks

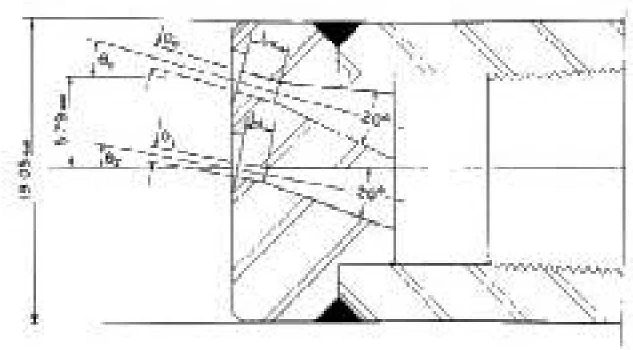
Property	Mean Value	Range	Coefficient of Variation (%)
ρ_R	3290	2910-4420	19.4
σ_t	16.0	8.4-19.7	26.1
σ_c	159.8	102-206	25.1
C	6307	4640-7110	15.4
E_R	128240	95120-157300	18.5
ℓ, mm	0.156	0-0.312	100.0
f (%)	1.30	-	-
κ	0.075	(at an effective pressure of 1.37 MPa)	
M*	34.6		

3. INCO Samples

ρ_R	2850	2820-2880	1.5
σ_t	13.3	9.3-15.5	26.4
σ_c	255.0	237-273	10.0
C	6110	5780-6450	8.0
E_R	106890	94110-119670	16.9
ℓ, mm	0.229	0.24-0.434	89.5
f (%)	1.12	1.08-1.19	5.2
κ	0.582	(at an effective pressure of 1.84 MPa)	
M*	26.6	-	-

4. Falconbridge Samples

Table 2. Measured Properties of Rocks (cont'd)

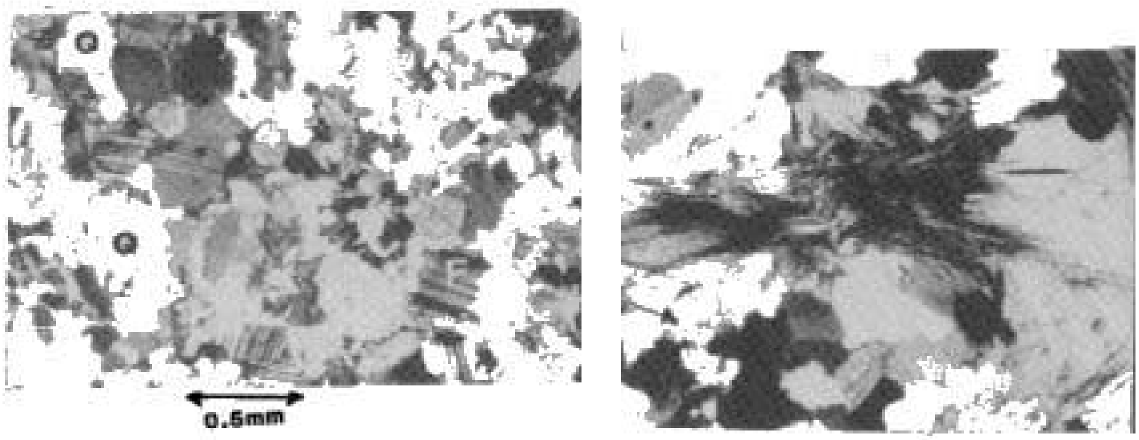


(A) NOZZLE CONFIGURATIONS (TABLE 1)

(B) PLAIN BALANCED NOZZLE

(C) CARBIDE TIPPED NOZZLE

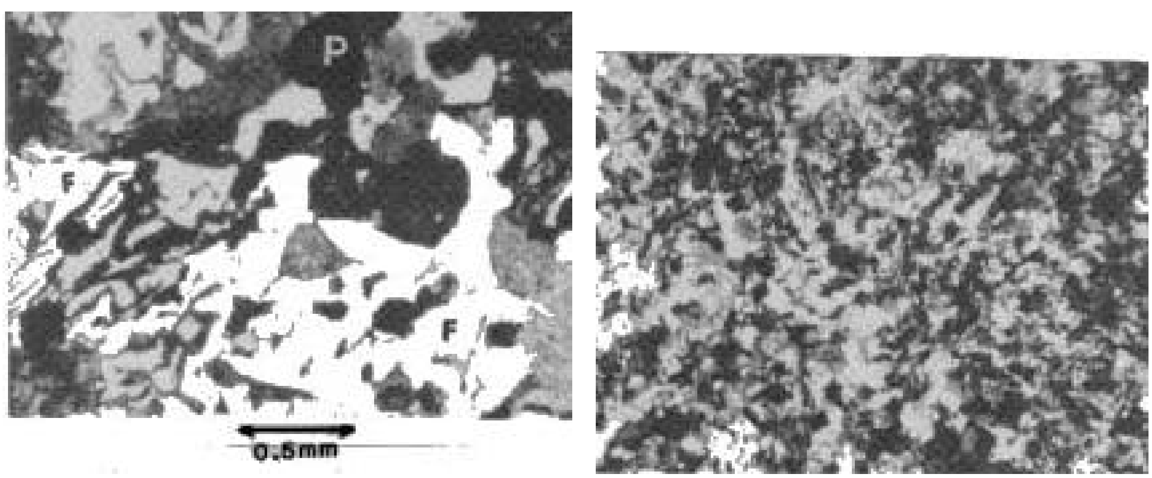
Fig. 1 Nozzle Bodies Used for Drilling and Slotting



a) Quartz Diorite
Quartz 27% Feldspar 54%

b) Biotite Uralite Gneiss
Quartz 60% Biotite 25%   Uralite 13%

Fig. 2 Photomicrographs of Noranda Samples



a) Coarse fraction (less altered norite)

b) Fine fraction (more altered norite)

Ore minerals 19%; Quartz 17%; Feldspar 28%; Chlorite 36%

Fig. 3 Photomicrographs of Falconbridge Samples

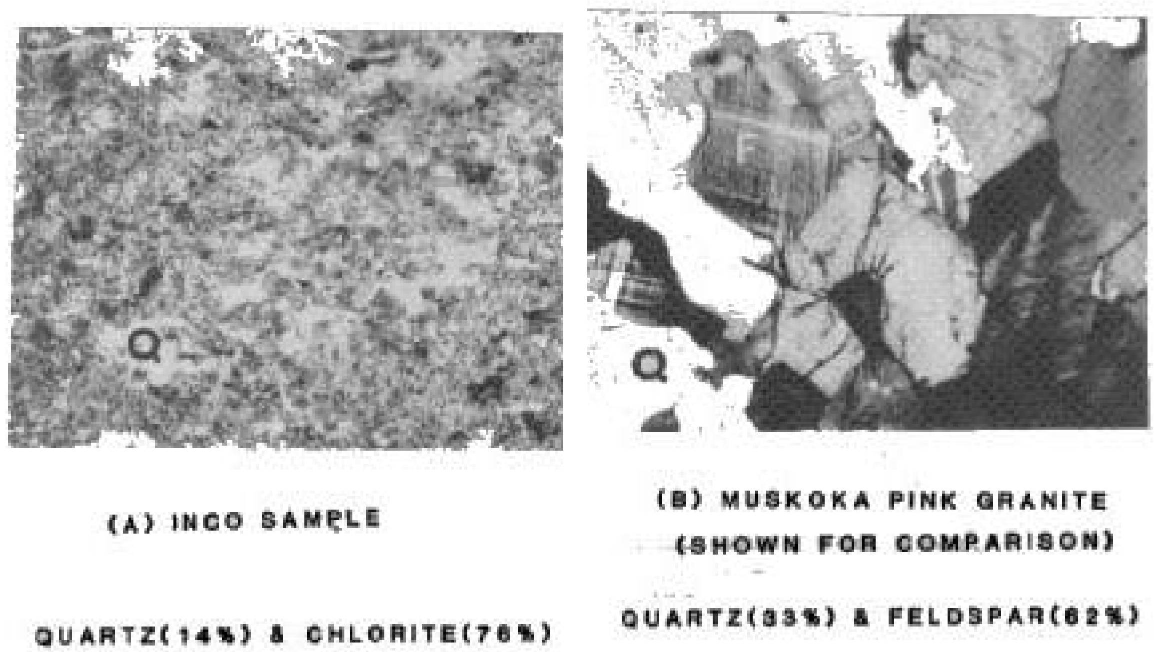


Fig. 4 Photomicrographs of INCO &Pink Granite

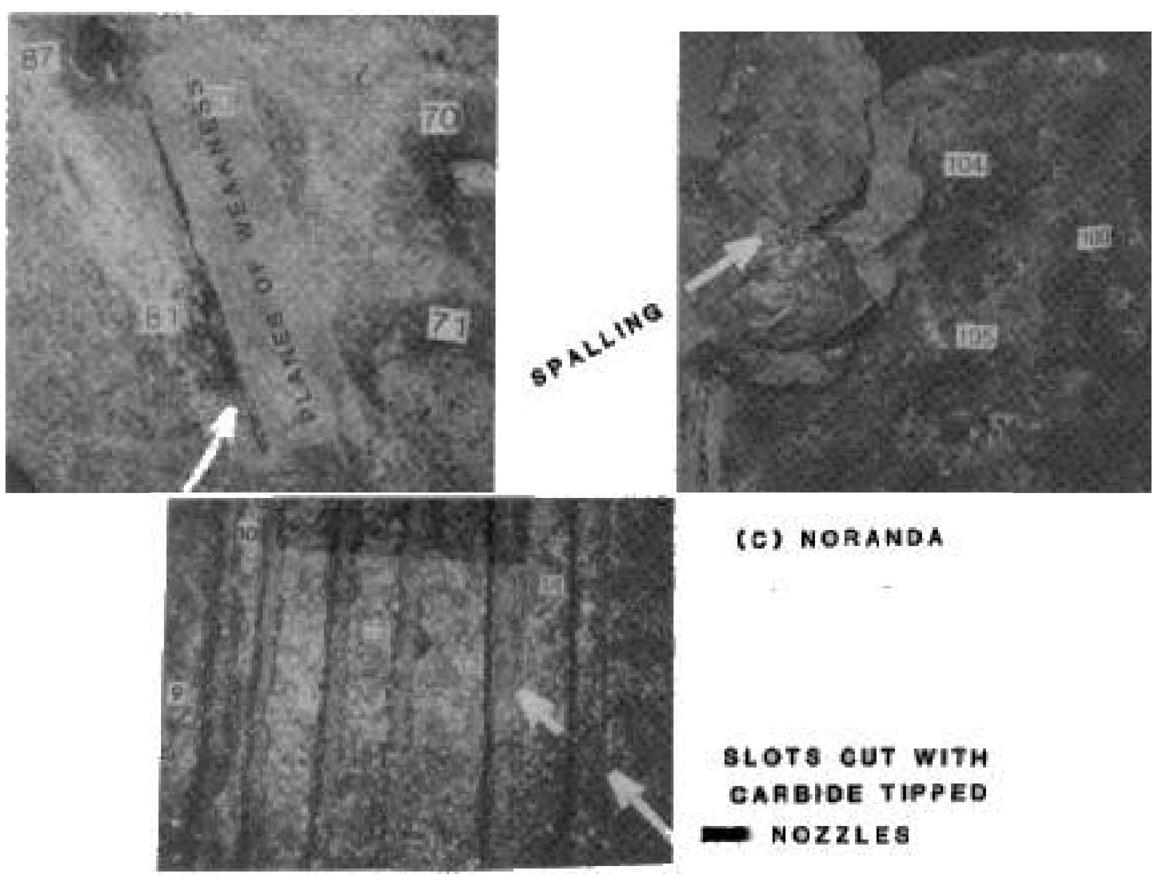


Fig. 5 Typical Holes and Slots Cut in the Samples a) Falconbridge b) Inco c) Noranda

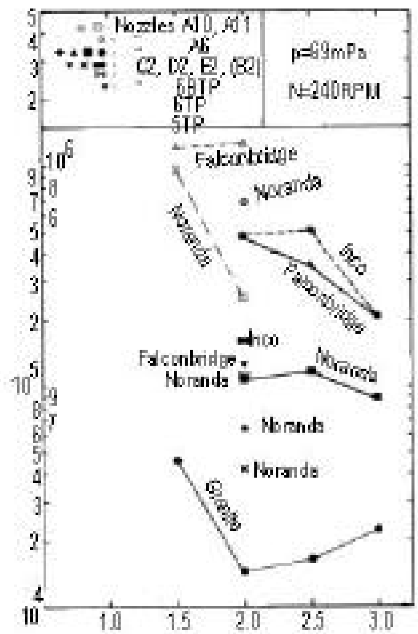


Fig. 6 Plot of E_V Against θ_O

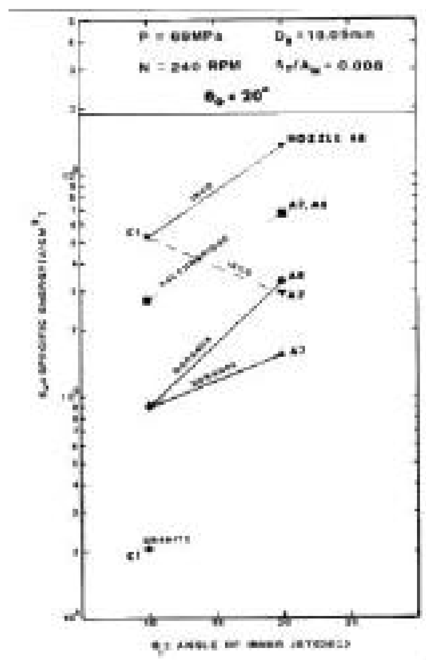


Fig. 7 Plot of E_V Against θ_I

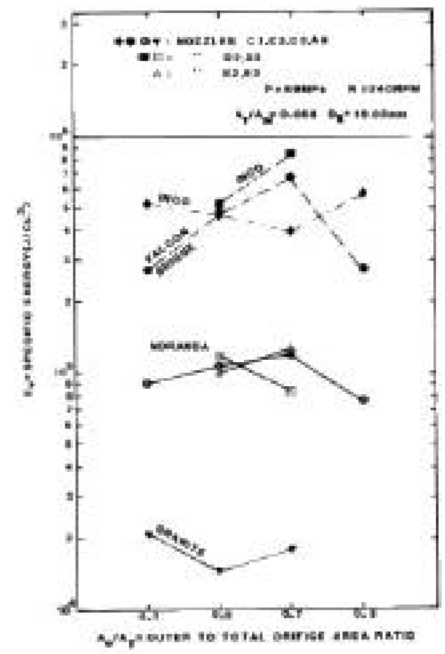


Fig. 8 Plot of E_V Against A_O/A_T

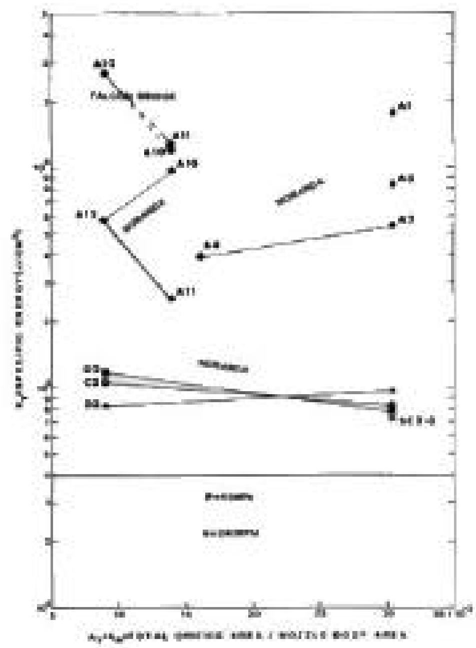


Fig. 9 Plot of E_V Against A_T/A_N

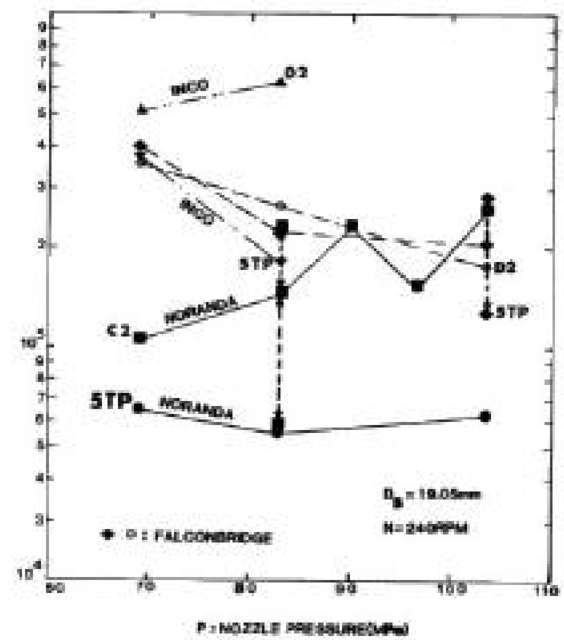


Fig. 10 Plot of E_V Against Pressure

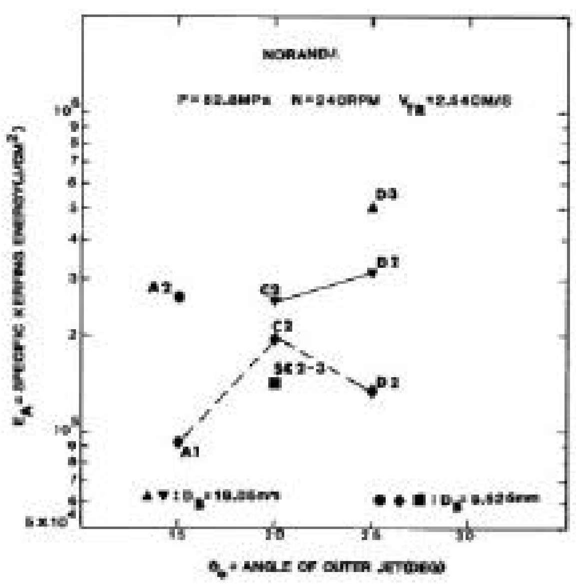


Fig. 11 Plot of E_A Against θ_o for Slotting

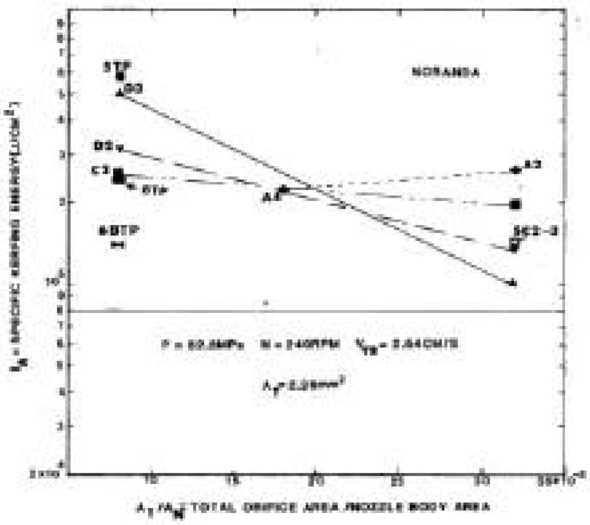


Fig. 12 Plot of E_A Against A_T/A_N for Slotting

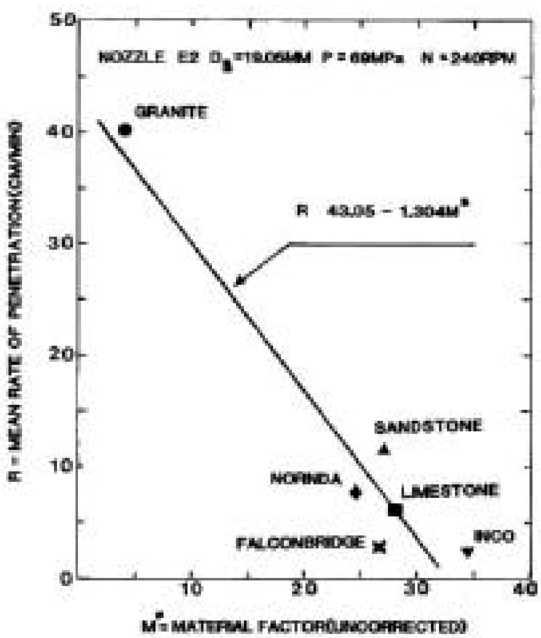


Fig. 13 Plot of R Against M* for Drilling

CUTTING TRIALS WITH A WATER-JET-ASSISTED
LONGWALL COAL SHEARER

By P. D. Kovscek[1], R. J. Evans,[2] and C. D. Taylor[3]

ABSTRACT

A longwall shearer at the Pittsburgh Research Center Mining Equipment Test Facility was retrofitted with a water-jet-assisted cutting system. Testing was conducted on a simulated coal block with jet pressures ranging from 1,000 to 6,000 psi. A triplex pump supplied this high-pressure water to one shearer drum with 32 bits. High-pressure nozzles adjacent to each bit block directed the jet to impinge 1 mm in front of the bit. A small-diameter high-pressure swivel was used to transport water to the rotating drum. Test results indicate dust reductions up to 90 pct when using 6,000-psi jets. Compared with dry cutting and low-pressure sprays, reductions in drum torque and thrust ranged from 10 to 30 pct with the 6,000-psi jet-assisted cutting system.

INTRODUCTION

Domestic production of coal by longwall methods has increased since the early 1960s to account for about 12 pct of the total underground production. Average productivity in the United States is 700 to 1,200 tons per shift compared with 300 to 400 tons per shift for room-and-pillar mining. Coal resource recovery of 85 pct, along with increased productivity, is making longwall mining the preferred coal extraction method.

Despite its advantages, the longwall has several constraints that pose problems affecting its operation. A study of industrial engineering data from U. S. longwall faces (ref. 1) has shown that the greatest single contributor to total reported downtime is delay at the shearer. These delays account for 28 pct of the total downtime, of which 75 pct is due to mechanical operations, the most significant being bit change. This one activity accounts for 36 pct of all mechanical delays (8 pct of total downtime). Cutting system improvements that require fewer bit changes would provide an immediate benefit in terms of increased production.

Laboratory research conducted at the Pittsburgh Research Center (PRC) with water-jet-assisted cutting (using pressures ranging from 1,000 to 6,000 psi) has indicated that there are many potential benefits: increased bit life, significantly reduced dust, reduced methane ignition, reduced fines, and reduced drum thrust, torque, and vibrations.

Water-jet-assisted cutting is rapidly finding its way from development through laboratory research to underground operations. In Europe, at least five jet-assisted roadheaders are already in service along with another 15 machines sold throughout the world. In the United States a manufacturer is currently testing a water-jet roof bolter underground.

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METHODOLOGY

Previous laboratory testing at PRC with a single-pick in-seam tester fitted with a water-jet-assisted cutting system showed that, when cutting on a simulated coal block, the optimum cutting parameters are 6,000-psi jet pressure, 0.6-mm nozzle with 13° tapered Leach and Walker design, and jet location of 1 mm in front of the mechanical bit. Further testing with a multiple-bit cutterhead to determine the effect of bit velocity on jet-assisted cutting efficiency showed that there was an insignificant drop in jet-assisted cutting efficiency at high bit speeds on that particular rock sample.

The objective was to apply the design parameters obtained from a single-pick in-seam tester to a commercial mining machine. The in-seam tester, as shown in figure 1, is a hydraulically powered single pick that cuts upward in a linear plane at 26 ft/min and is instrumented to measure and record the orthogonal cutting forces as illustrated in figure 2.

The rationale for using an in-seam tester is that one of the major difficulties facing investigators conducting laboratory coal cutting tests is to ensure that the relatively small samples being cut are representative with respect to the confined ground-induced stresses. This is more of a problem with coal than with other rock types because of cleat and joint presence in the coal mass and dehydration and temperature changes that occur when blocks of coal are removed from the mine. The in-seam tester will allow use of full-scale tools to avoid potential errors in force scalar relationships.

The research was conducted on a Joy 1LS[4] longwall shearer retrofitted with a new drum designed around a 6,000-psi water-jet-assisted cutting system. The cutting drum was mounted on the left-hand boom of the shearer. A test program was devised to determine the net improvement in shearer cutting performance when using jet-assisted cutting, along with evaluation of the power and energy relationships. High-pressure water was transported to the rotating drum through a small-diameter, high-pressure rotary seal located in the drum. The high-pressure water was supplied by a 150-hp triplex pump with a capacity of 50 gal/min at 6,000 psi.

The cutting drum (fig. 3) is designed to have six equal water-conducting segments. Each segment receives high-pressure water at the hub and radiates it to all the bits contained in that segment. The design specifications of the cutter drum are as follows:

Number of starts:	2
Total wrap angle:	437°
Vane web width:	28 in
Drum bit tip diameter:	54 in
Drum speed:	47 r/min
Lateral bit spacing:	progressive starting at 1-1/2 to 2 in
Cutting bits:	radial, K-107 or equal

[4] Reference to specific products does not imply endorsement by the Bureau of Mines.

Machine Description:

Shearer:	Joy 1-LS1 Double drum-extended boom
Haulage:	chainless (rack and sprocket)
Dimensions:	36 in high; 30-in web; 54-in-diam drum
Haulage motor:	3 5-hp DC
Cutterhead motor:	100-hp AC
Cutter drum speed:	46 r/min
Bit tip velocity:	650 ft/min

TEST PROCEDURE

Cutting tests (fig. 4) were conducted on a 60-ft long by 6-ft high simulated coal block composed of coal, fly ash, and concrete. Although the simulated coal block is more abrasive than coal (because of high silica content), its cutting properties are similar. Table 1 compares the cutting properties of the simulated coal block to the Winchester No. 5 block coal and Pittsburgh Seam coal. The simulated coal block has a reasonable Hardgrove grindability index when compared with Pittsburgh Seam coal and overall, the simulated coal block is homogeneous and isotropic.

	Hardgrove grindability	Density, lb/ft^3	Compressive strength, psi	Shear strength, psi	Normal force, lbf		Cutting force, lbf		F_n/F_t	
					Mean	Peak	Mean	Peak	Mean	Peak
Simulated coal block.	62	106	885 - 950	140	454	1,675	656	2,625	0.69	0.64
Pittsburgh Seam coal.	63	85	3,200	120	446	1,200	802	2,100	.56	.57
Winchester No. 5 coal block.	NA	NA	NA	NA	376	1,635	596	3,245	.54	.50

NA Not available.

TABLE 1. - Comparison of physical and cutting parameters between the PRC's simulated coal block and underground coal seams

The simulated coal block consists of bituminous coal with a nominal size of 1.5 to 2.0 in. A mixture is made up of 10 PPV (parts per volume) bituminous coal, 8 PPV flyash, 1 PPV Portland cement, 1.5 PPV water, and is cast in place. The average bulk density when cured is 106 lb/ft^3. The curing time of the simulated coal block prior to the water-jet-assisted cutting trials was 15 months.

Three planned passes were made on the coal block, as shown in figure 5. The first pass consisted of seven equal segments with varying water-jet pressures using 0.6-mm nozzles as follows:

Segment	Water pressure, psi
1	6,000
2	5,000
3	4,000
4	3,000
5	2,000
6	1,000
7	varied 6,000 down to 1,000

Depth-of-web measurements were taken at 1-ft intervals of each cut to accurately-identify the true volume of coal removed.

The second pass of the coalcrete block was performed at optimum jet pressure obtained from data generated during pass 1. The second pass was divided into two segments. The first data cut was 35 ft long at optimum jet pressure; the second cut was 10 ft long at low jet pressure. Cutter bits were weighed after each pass, and depth-of-web measurements were taken at 1-ft intervals. A third pass, consisting of two data cuts, simulated conventional dust-suppression water sprays during the first 35-ft cut; the second cut (approximately 15 ft) was performed without water to obtain baseline data of longwall dry cutting.

The coal shearer had a number of variables that were kept constant during the cutting operation. The shearer tram rate was maintained at 5 ft/min or approximately 1.3-in depth of cut, and ranging arm and shearer tilt jacks were set to maintain a horizontal cut.

Dust levels were measured during the cutting operation by drawing a portion of the dust generated through a 2-1/2-in plastic pipe mounted adjacent to the shearer cutting head. This pipe was attached to a suction fan that drew the dust-laden air into a dust box and then through the fan mounted on the rear of the dust box. Two nylon cyclones (10-mm), in the dust box, separated the respirable dust. Two RAM-1 real-time dust sampling instruments were used to draw respirable dust from the cyclone sampling heads and to continuously measure the dust concentrations. The concentration data from each instrument were recorded by a dual-pen strip-chart recorder. The recorded curves were compared when operating dry and with jet-assisted cutting.

Transducers were installed on the shearer to measure cutting performance by monitoring the following parameters: drum speed, drum torque, tram rate, drum thrust, jet pressure, jet power, shearer power, drum vibration, drum position, and noise at the operator station. Figure 6 is a block diagram of the longwall instrumentation plan. The objective of the instrumentation plan was to measure gross system forces and performance variables of the shearer as it was operated under various cutting conditions. Electric motor variables were selected for measuring cutter torque and drum thrust. Cutter rotational speed and shearer tram rate were measured from the cutter and haulage motor shaft speeds.

CUTTING TEST VARIABLES

DRUM SPEED

The drum speed was relatively constant for all load conditions because the drive train is direct-gear driven with a low slip motor. A magnetic pickup was mounted to measure the gear tooth passing frequency and converted to a 10-V analog output.

DRUM POWER

Mechanical shaft power of the cutter drive motor was estimated by measuring the electrical power entering the motor, less motor loss due to windage, coil resistance, and core inefficiencies. A transducer was used to measure the electrical power used by the motor and, along with the motor speed, to calculate the motor torque.

TRAM RATE

A Hall effect transducer was used to measure the shearer traverse rate, because a passive magnetic pickup would not operate properly at slower speeds. The transducer was mounted next to a splined shaft exiting the commutator end of the haulage motor with speed ranging from 0 to 2,500 r/min. The output from the Hall effect transducer was converted to an analog voltage, which was proportional to the haulage speed.

SHEARER THRUST

Shearer thrust was estimated by relating the haulage motor current and voltage to shaft torque, along with the haulage transmission gear ratio and final sprocket pitch diameter.

JET PRESSURE

Jet pressure from the l0,000-psi triplex pump was monitored by a transducer placed in the feed line near the cutter head.

JET POWER

Jet power was calculated using the jet pressure and flow. The water flow to the shearer was calculated from the difference of the pump input flow measurement and the flow measurement of the outlet line of the pressure control dump valve.

SHEARER POWER

Shearer power was estimated by summing the power contributions of the left-hand cutter and haulage motors. Electric motors driving the hydraulic pump and right-hand cutter were neglected because their power consumption was not directly related to the coal cutting efforts of the water-jet drum.

CUTTING EFFICIENCY

Energy per unit volume (E_V) as a measure of rock boring rate efficiency was first proposed by Ross and Hustrulid (ref. 2) in 1972. Since the E_V approach is based on actual practice, it was felt that the E_V approach is valid for evaluating the rotary shearer tests.

The data for the rotary shearer tests were calculated on gross system parameters. No corrections or estimates were made to account for motor or gear train efficiency, or to predict the force loading on one bit during one revolution of the cutter drum. The test steady-state data are derived over a specific length and time. The respective parameters from each test are summed, and the averages are used for comparisons. Although the tram rate (depth of cut) is defined as a constant, it varied within the control limitations of the shearer.

TEST RESULTS

WATER-JET PRESSURES

The water-jet cutting drum had 32 cutting tools, with 1 jet nozzle per tool. The 0.6-mm nozzles were located in front of each cutter bit with a standoff distance of 4 in (fig. 7). The cutting drum was mounted on the left-hand cutter boom of the shearer; it was the leading drum for all tests. The trailing drum was positioned to follow in the cut path of the lead drum. Table 2 shows the actual pressures and jet power delivered to the cutter drum for each of the tests.

Test design water-jet pressure, psi	Actual water jet pressure, psi	Water flow per nozzle, gal/min	Jet velocity, ft/s	Jet power per nozzle, kW	Total jet power, kW
6,000	5,750	1.22	894	3.06	97.79
5,000	5,023	1.16	850	2.54	81.20
4,000	3,933	1.01	736	1.72	55.05
3,000	2,905	0.81	595	1.03	32.85
2,000	2,018	0.68	501	0.60	19.22
1,000	1,098	0.50	365	0.24	7.62

TABLE 2. - Water-jet test pressures, resultant flow, jet power, and velocity

The jet power actually assisting the mechanical bits in the cutting operation is less than what is shown in table 2. The water jets are operating continuously for each revolution of the drum, but the coal cutting occurs only during a one-half revolution. Additionally, only 10 pct of the volume of the coal is cut when the bit reaches 60 pct of the maximum depth of cut (ref. 3), which is equivalent to a drum rotation angle of 36.9° from the top of the cut. For a symmetrical cut, 80 pct of the volume of coal cutting would occur during a 106° bit arc. For example, at a test pressure of 5,750 psi, the jet power delivered to the drum is 97.9 kW. The jet power that assists in removing 80 pct of the volume of the cut for one revolution of the cutter drum is as follows:

$$\left(\frac{106}{360} \text{x} 97.9\right) = 28.826\text{kw} \quad (38.7\text{hp})$$

A phasing system that would direct the jet spray into the coal cutting area over a 120° arc would reduce the jet power and flow requirements by two-thirds for the water-jet-assisted cutting.

The jet nozzle leading each bit is expected to assist the mechanical cutting tool by:

- Lubricating the cutter tip and coal surface contact area;
- Removing smaller particles from the path of the cutter to allow more force to be imparted to the uncut material;
- Cooling the tungsten carbide cutter tip and thereby increasing bit life;
- Impacting, weakening, and propagating cracks in the mined material to ease removal by the cutting tool.

MACHINE PERFORMANCE

The preliminary machine performance data from pass 1 are shown in table 3. The average depth of web of the six data cuts is shown in figure 5. The depth-of-cut range is from 1 to 1.5 in. The shearer power requirements, as shown, decrease as jet pressure increases; the shearer power and E_V vary in proportion to the depth of cut, as indicated by the tram rate. The drum power requirements are reduced by 14 pct (fig. 8), and total shearer power is reduced by 13.6 pct with the addition of water-jet assist. The E_V, an indication of cutting efficiency, is increased by 2 pct when comparing the 6,000-psi and 1,000-psi tests.

Test jet pressure, psi	Shearer cutter power, kW	Haulage and cutter power, kW	EV, in-lbf/in^3	Tram rate, ft/min
TOTAL TEST COMPARISON				
5,750	98.66	101.58	747.7	4.17
5,023	101.16	104.31	644.8	4.70
3,933	110.00	113.52	644.9	5.42
2,905	110.75	113.86	672.8	5.05
2,018	108.04	110.94	686.5	4.88
1,098	114.77	117.59	732.6	4.80
SAMPLED AT 4 FT/MIN				
5,755	98.49	101.63	820.0	4.00
5,012	102.37	105.87	842.8	4.11
3,933	94.79	98.14	792.4	4.06
2,914	102.91	105.94	875.9	3.96
2,007	111.93	114.92	882.4	4.15
1,074	108.56	111.43	850.9	3.97

TABLE 3. - Shearer performance data

The coal shearer performance is affected by the types and sizes of haulage and cutter motors, gear trains, drum style, cutter style, cutter spacing, web depth, cut depth per revolution, applied jet pressure, and cut material hardness. The machine performance evaluation with water-jet assist is only affected by web depth, cut depth, and water jet pressure, with other aspects similar. At a shallow depth of cut, the cutting efficiency of the coal shearer is lessened and indicated by a higher E_V; more energy is applied to breaking and grinding small pieces of material. As the depth of cut is increased, the efficiency of a shearer will increase up to an optimum depth of cut until the vanes contact a kerf buildup, or the torque limitation of the cutter motor is reached. The coal shearer has sufficient thrust capabilities because of the chainless rack-and-sprocket drive that provides a positive engagement of the rack with a minimum of thrust power (2 to 3.5 kW) out of 110 kW total machine power. Drum power is the major contributor to the total machine power and the (E_V) cutting efficiency. With the addition of water-jet assist, the nozzle orifice distance from the coal surface decreases as depth of cut per drum revolution increases, allowing for greater force concentration of the water stream onto the cut material. The shorter distance permits a more coherent and faster stream of water, and possibly generates hydraulic pressure into the cut material. The interaction of E_V, machine power, and jet power with the depth of cut per revolution (tram rate) is not completely understood.

A second set of data from each data cut (tram rate somewhat constant at 4 ft/min and l-in depth of cut) was sampled to reduce the effect of the depth of cut on the performance parameters (table 3). The cutter and shearer power requirements were reduced by 9 pct (fig. 9), and EV was reduced by 4 pct for comparisons of the 6,000- and 1,000-psi tests. The largest power and E_V reductions occur at 4,000-psi test pressure (fig. 10).

The data sampled at 4 ft/min tram rate provide a means of equating the tests of various water-jet pressures. The E_Vs for all tests have increased, implying that 1 in is not the optimum depth of cut. There are also other factors (possibly depth of web or changes in hardness of the simulated coal block) that affect the data, as seen in the scatter of the shearer power and E_V results. Currently, further data reduction is being performed on a depth-of-cut basis for each of the jet pressures; this should provide a clearer indication of the benefits of water-jet-assisted cutting.

SIZE DISTRIBUTION

After the test sequence, coal cuttings were sampled at the center of each of the respective test areas. A section the width of the cut was sampled. Approximately 4 lb of simulated coal cuttings was sampled for each test pressure. The samples were dried and screened for size. Figure 11 shows the simulated coal chip distributions as cumulative percent passing a common screen opening. The consistency of these distributions can partially be attributed to pre-sized coal particles that have undergone successive degradation from sizing, transporting, mixing, and casting operations prior to the cutting tests. The limited samples taken indicate that 90 pct of the cuttings pass through a l-in screen opening and are not indicative of run-of-mine coal. However, the size distribution comparison of the water-jet test pressures for the water-jet-assist cutting trials suggests a trend of increased particle size with increased jet pressures.

DUST SAMPLING

Two cyclones ("exterior") were hung about 40 in from the drum at a level about even with the top of the cut. Inlets were positioned away from the drum to prevent the entry of large water droplets. Two additional cyclones ("interior") were placed in a dust sampling box. Dust was drawn into the box at constant airflow through 2-1/2-in plastic tubing; the inlet to the tubing was located approximately 24 in from the drum.

The action of the Bureau of Mines shearer cutting in coalcrete is similar to that of a shearer cutting coal on an underground longwall face. However, the airflow patterns on a longwall face, which have a great effect on the transport characteristics of airborne dust, could not be simulated during testing. No attempt was made, therefore, to relate dust levels measured during testing to those that would be generated during actual underground mining. However, it is assumed that the relative changes in dust level that occurred with changes in water pressure during testing are representative of the relative changes that would occur in dust levels underground when water-jet cutting was used.

Sampling results are shown on figure 12 for externally and internally mounted cyclones. The relative dust levels decreased significantly (88 to 72 pct) as the water pressure was raised from 1,000 to 3,000 psi. Further reductions in dust were small as the pressure was raised from 3,000 to 6,000 psi. Strip-chart recordings at l,000-psi water pressure show that concurrent measurements made inside and outside the sampling box were similar in magnitude and pattern. Although the areas under the curves were about the same, individual peaks for dust measured outside the box were more distinct. The shape of the individual peaks for external cyclone data is probably due to the influence of drum rotation on ambient airflow.

The purpose of the sampling performed was to quantify these reductions. The reasons for these reductions are not completely understood, although an increase in the quantity and pressure of water used during mining is expected to have affected the dust levels generated. The use of sprayed water is a common dust-control technique used in underground mining. Generally, it would be expected that the quantity of airborne dust generated during mining would be reduced as the quantity of water is increased. When sprayed water is used, some of the dust particles are removed from the air stream by impacting water droplets. Primarily, however, intimate mixing of water with the mined material must be achieved to reduce the quantity of dust suspended at the cutting face and dust resuspended during subsequent transport. The quantity of water that can be delivered is limited by the supply system, the maximum allowable moisture content of the mined material, and the residual water that can be left in the work area.

As a general guide line, it can be assumed that no more water should be supplied to a shearer drum equipped with a jet-assist system than is supplied to a typically low-pressure spray system. A longwall drum spray at a pressure of 100 to 125 psi, would typically have a flow of approximately 1 gal/min. The spray nozzles used during testing of the shearer had orifice diameters of 0.6 mm and (at a pressure of 3,000 psi) supplied 0.9 gal/min. Therefore, a water-jet system operating at 3,000 psi and a typical spray system operating at 125 psi would deliver approximately the same quantity of water. Any reductions in dust resulting from the use of a water-jet system operating at 3,000 psi, versus a system operating at 100 to 125 psi with conventional nozzles, could not be attributed to water quantity.

Increasing the pressure of sprayed water will have an effect on dust control. Several studies have been conducted to determine how pressure affects airborne capture by water droplets. It is difficult, however, to quantitate how higher pressures will affect the moisture content of material as it is mined. To be thoroughly mixed with material being removed by the bit, the water should penetrate some distance into the broken and crushed material. This penetration must take place quickly, before the material is exposed to the air. In some materials, water jets penetrate the solid rock before it is cut, which may result in wetting of "inherent" dust located in the rock cleavage planes. Although to what extent is not known, the high-pressure water-jet does penetrate the material as it is cut. This allows wetting of crushed material immediately in front of and beneath the bit.

Material will circulate with the vanes of the cutting head for a short time before being exposed to the ambient airflow. During this time, efficient mixing of the cut material with the water is important for improved dust control. High-pressure water jets have the advantage of better mixing during this time and can, therefore, reduce dust levels farther.

SUMMARY

The longwall shearer cutting of the simulated coal block provides a reasonable simulation of actual cutting activity occurring on a working longwall face.

The shearer laboratory tests to date show an improvement in shearer performance and significant reductions in respirable dust levels by the addition of water-jet-assisted cutters.

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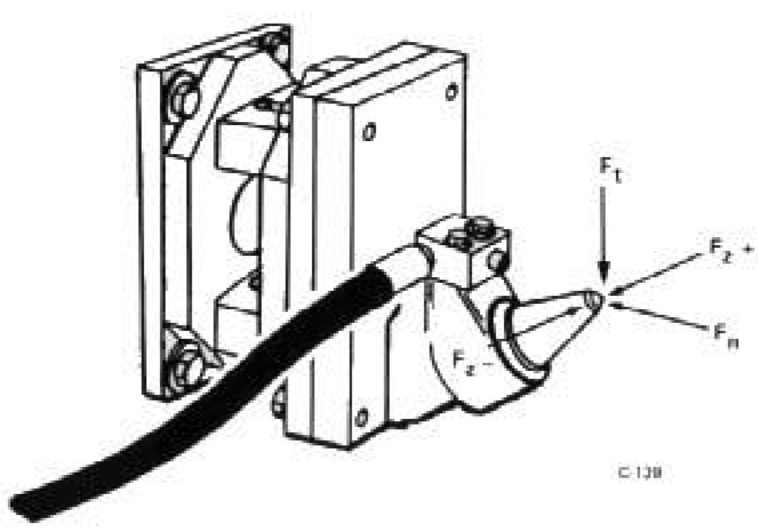


Figure 1. Heay-duty inseam tester

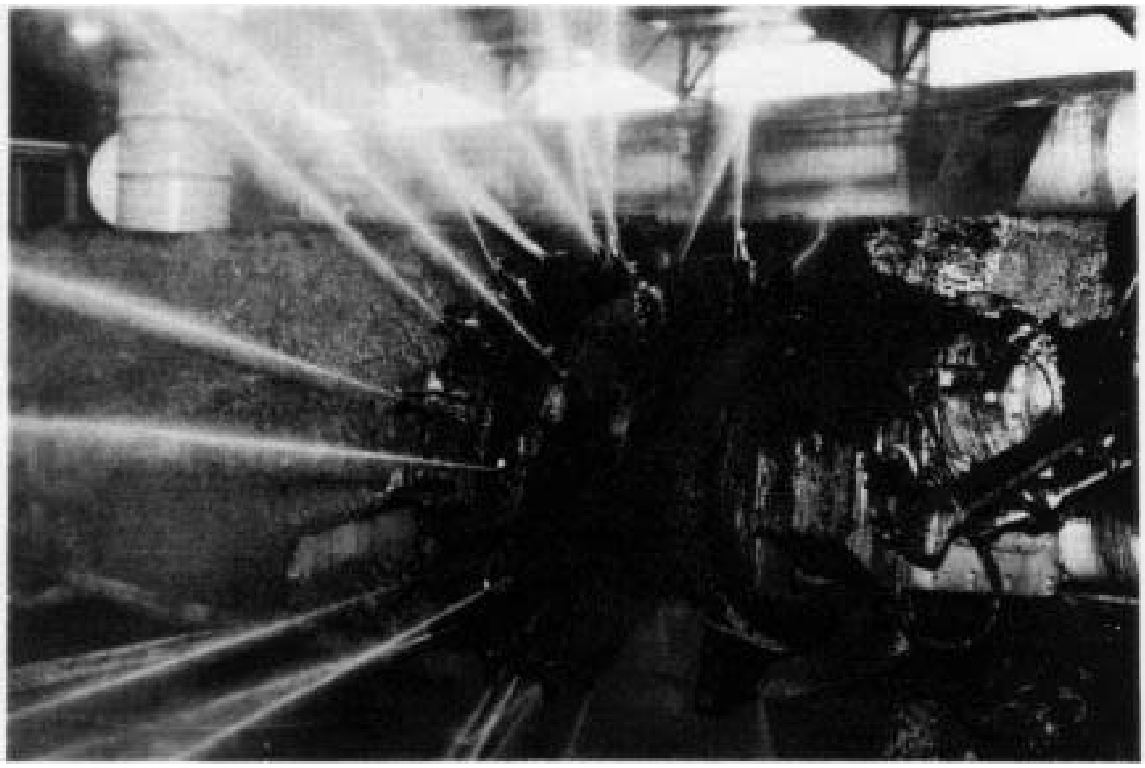


Figure 2. Force reaction on a cutter bit.

Figure 3. Water jet-assist cutter drum

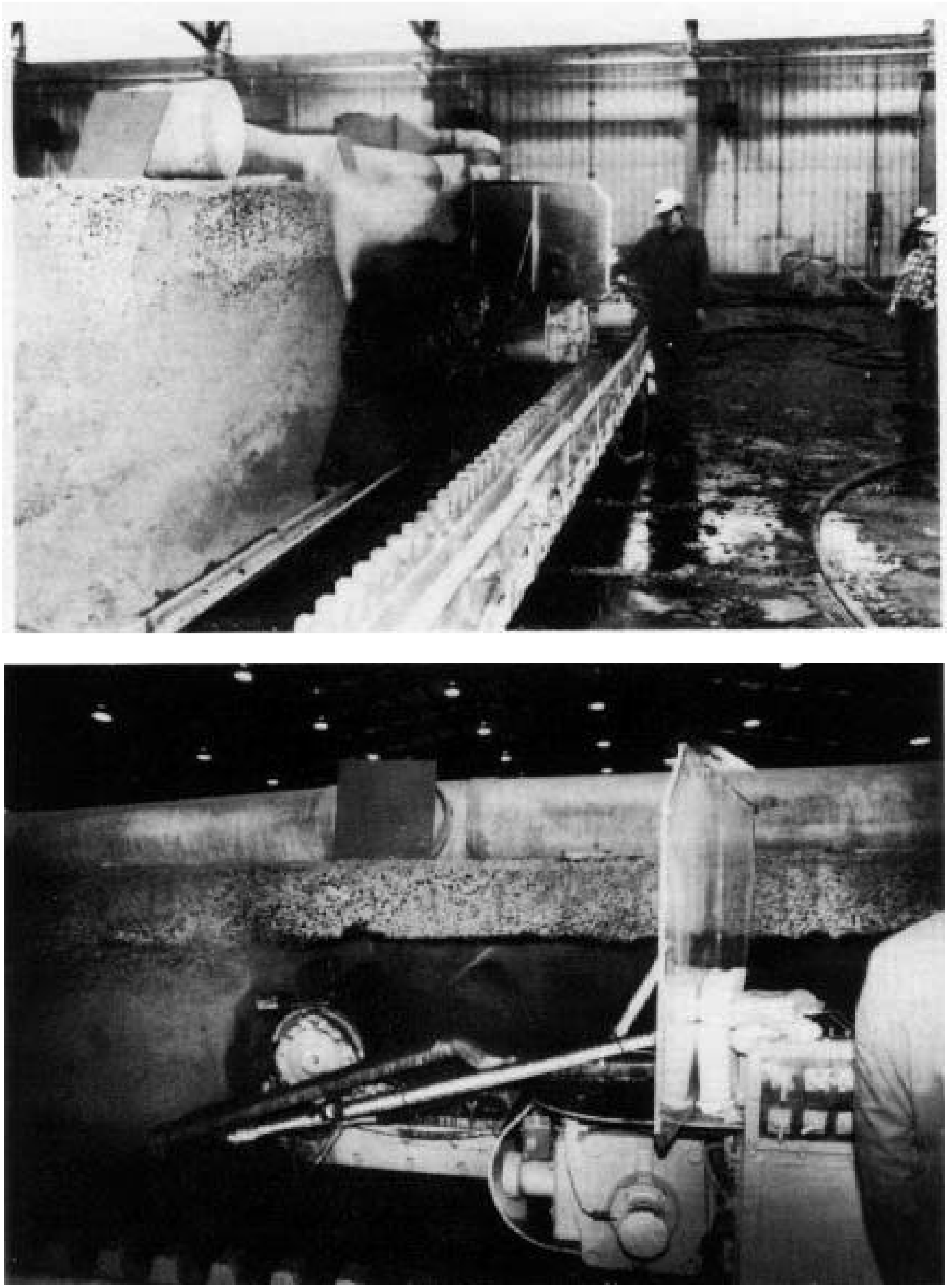


Figure 4. Coal shearer cutting on simulated coal block

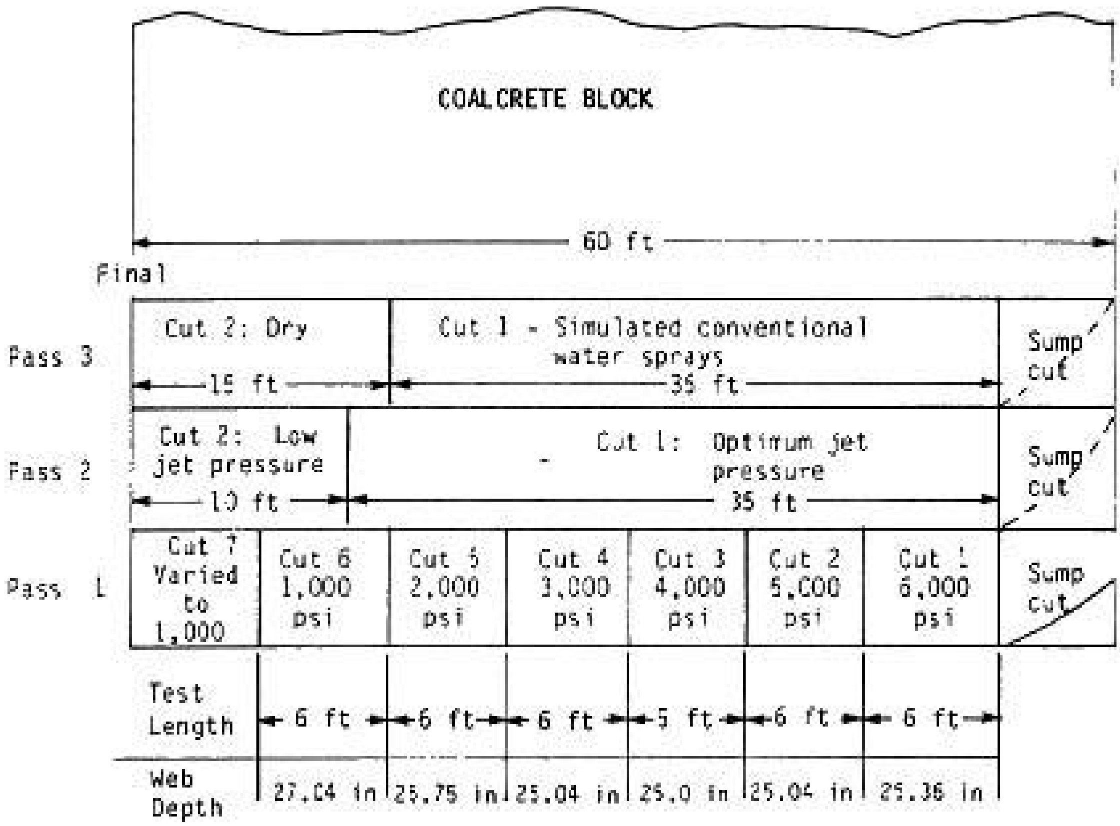


Figure 5. Coalcrete block utilization

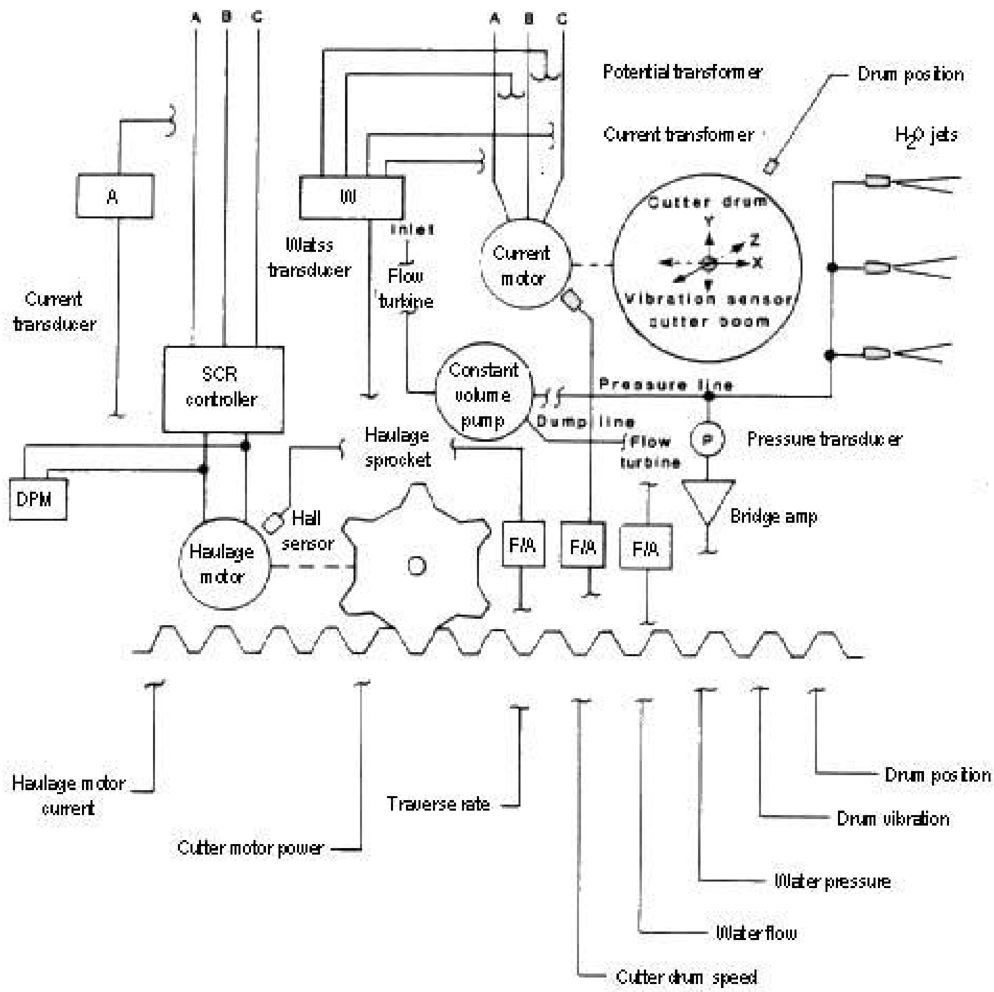


Figure 6. Longwall shearer instrumentation block diagram.

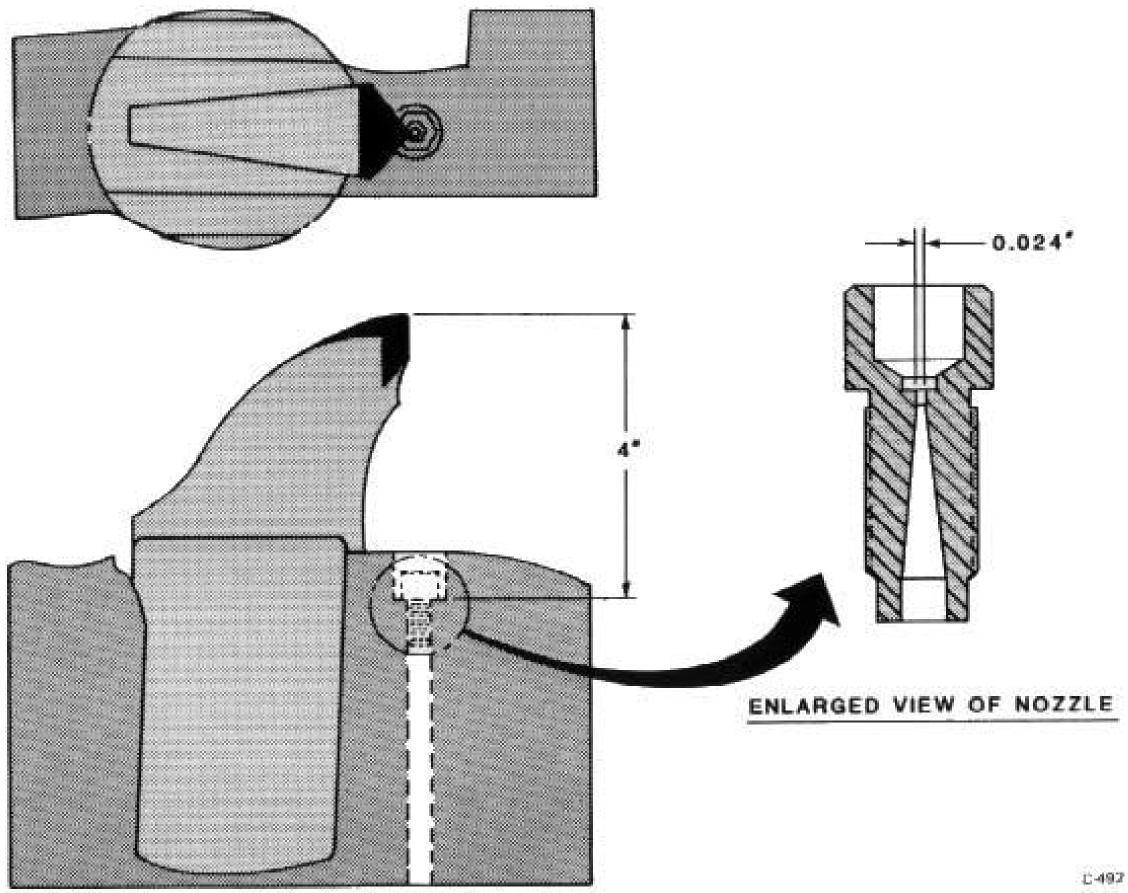
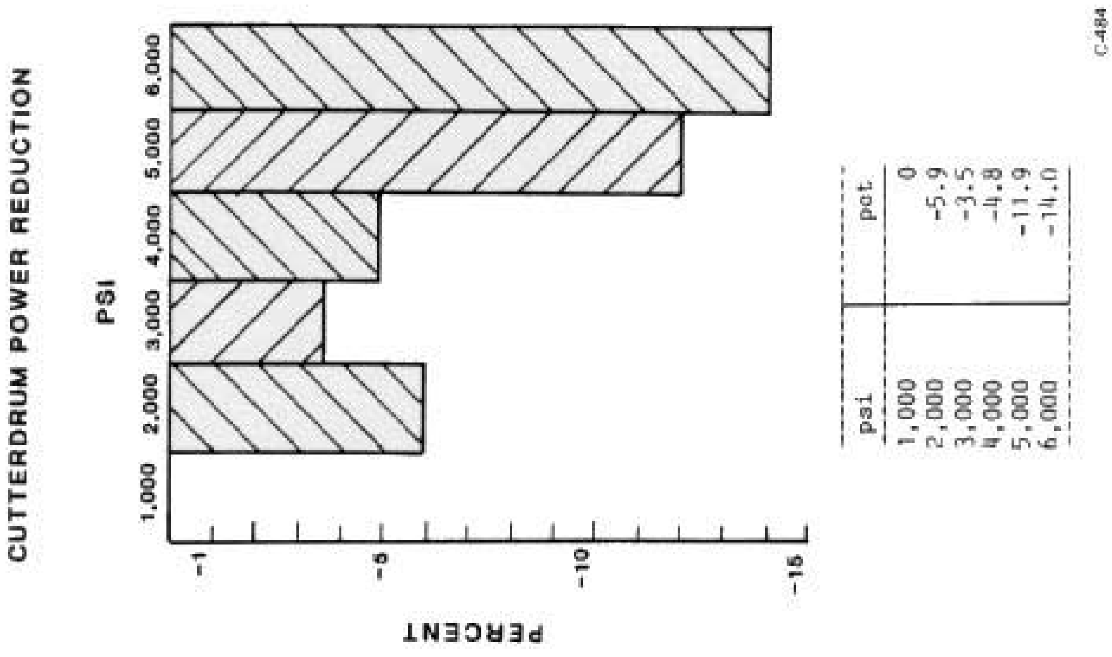


Figure 7. Bit mounting and jet nozzle used on the longwall coal shearer.



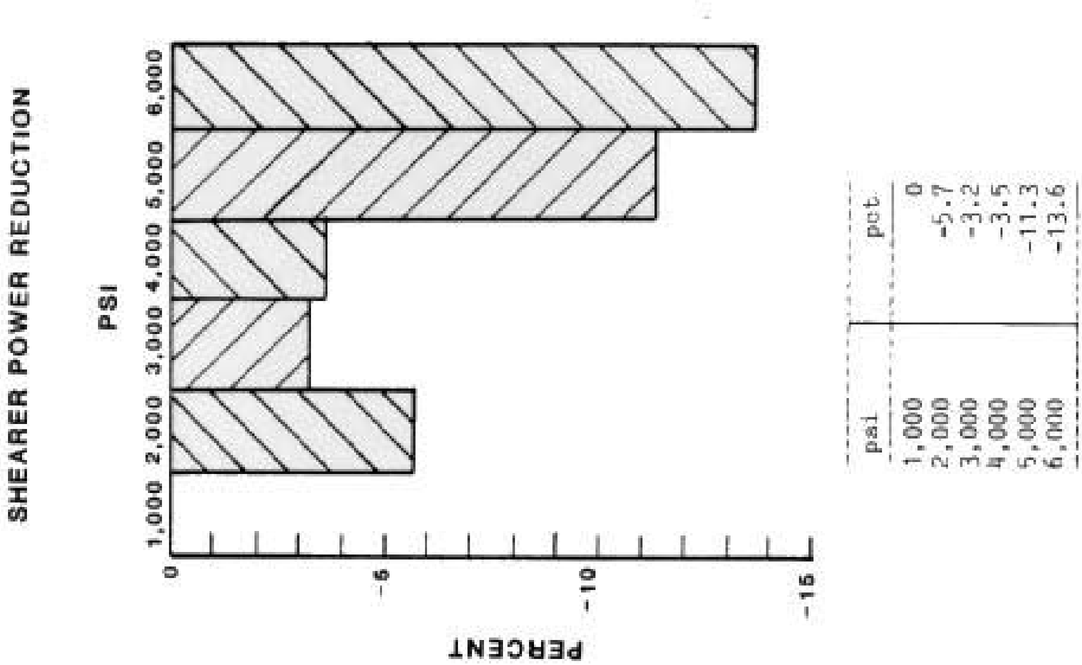
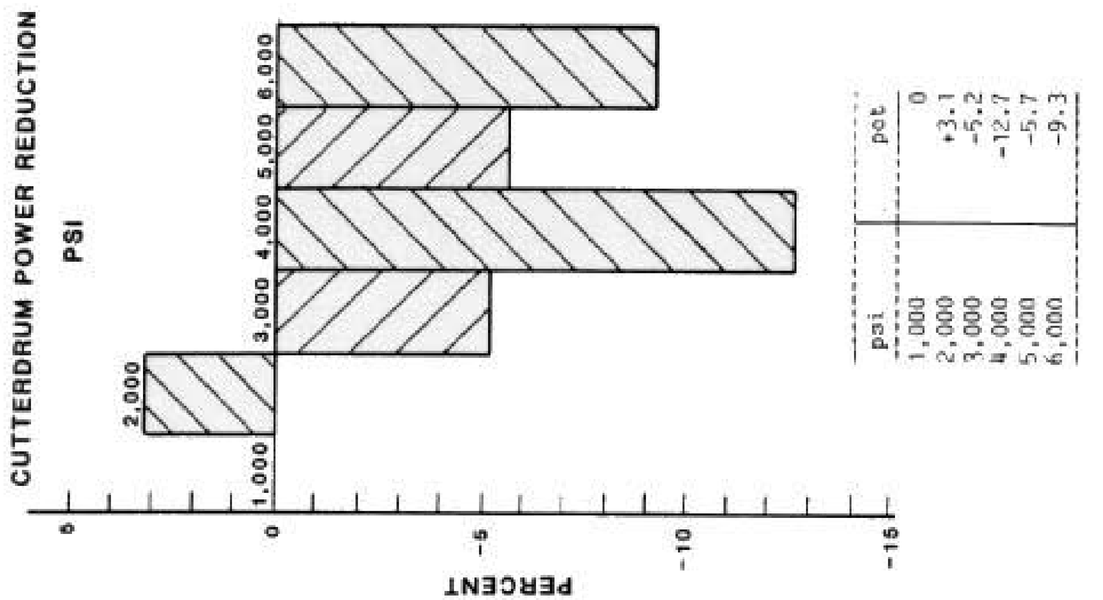


Figure 8. Whole test comparisons referenced to 1,000 psi jet pressure.



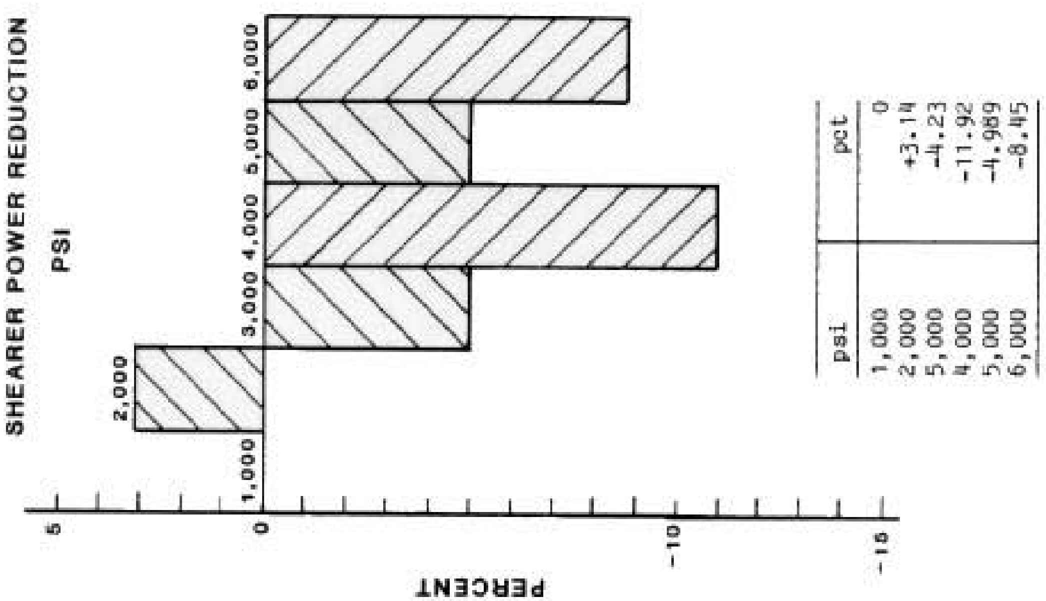


Figure 9. Test comparisons at 4 ft/min tram rate referenced to 1,000 psi jet pressure.

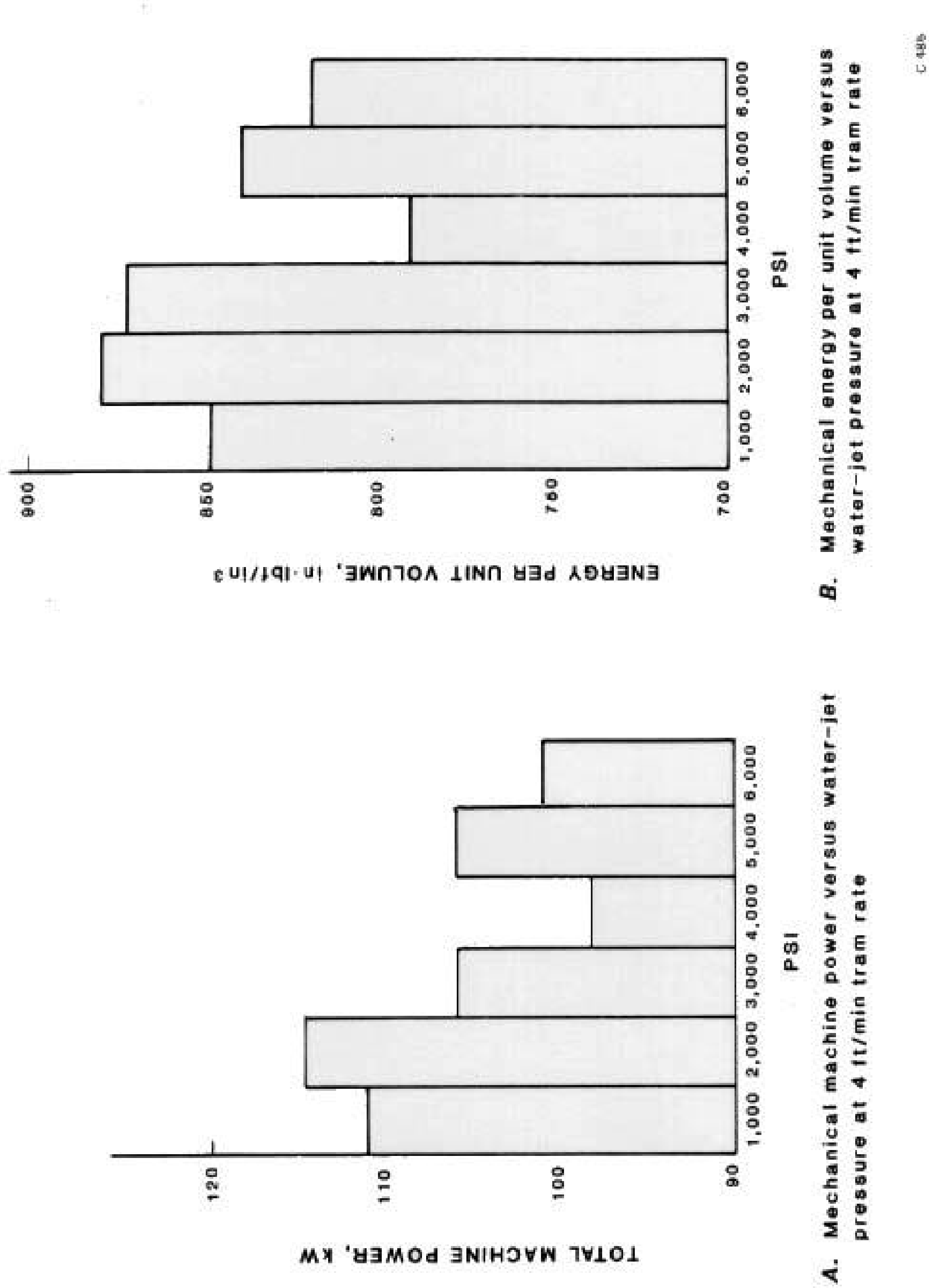


Figure 10. Shearer power and E_V versus waterjet pressure.

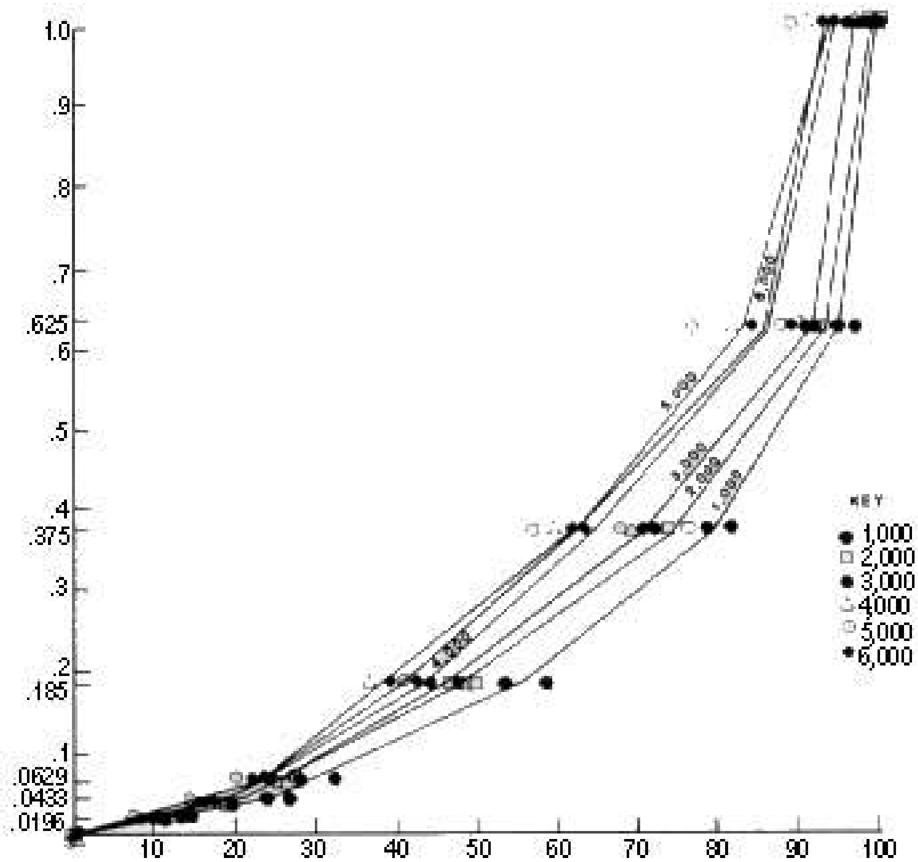


Figure 11. Simulated coal chip distributions of 1,000 to 6,000 psi jet pressures.

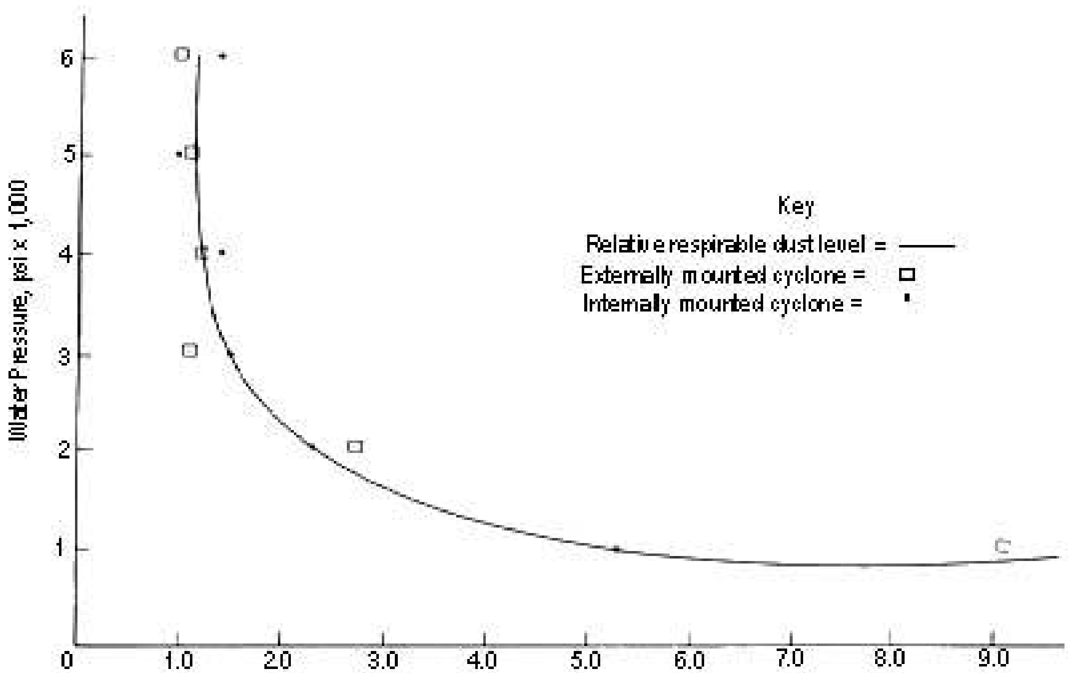


Figure 12. Applied waterjet pressures versus relative dust levels.

TECHNICAL AND TECHNOLOGICAL CONSIDERATIONS IN THE CARVING OF GRANITE PRISMS BY HIGH PRESSURE WATERJETS

by

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INTRODUCTION

In February, 1985, the University of Missouri-Rolla was recognized by the National Society of Professional Engineers for the creation of a new version of the Ancient British Megalith known as Stonehenge. This recognition took the form of being named one of the Ten Outstanding Engineering Achievements in 1984. Given that the budget for this project was considerably less than 1% of the budget for any other of the other nine programs recognized at the same time, and given that the work was in large measure carried out by undergraduate and graduate labor at the University of Missouri-Rolla, the receipt of this award has considerable significance to the University.

The UMR Stonehenge stands some 5 meters high on the northwest corner of the University of Missouri campus (Fig. 1). At this location, the monument comprises some 53 stones, each carved in their entirety from granite prisms transported from Elberton, Georgia, using high pressure waterjets The entire cutting operation was completed with a single rotary swivel and, in large measure, with one high pressure pump, given that an earlier piece of equipment which had been in University service for some 6 years had had to be replaced early in the program. Although the monument has artistic and symbolic significance to the campus, this paper will deal with the technology required to carve it.

The process of using High Pressure waterjets to cut granite is not in fact new. It was the ability of high pressure water at a pressure of 70 MPa (10,000 psi) to cut a granite with the compressive strength of 210 MPa (30,000 psi) remarked during a test in the 1960's which gave the first indication of the considerable potential which waterjets have for excavation (Ref. 1). In order to achieve the cutting of the UNR rock which required exposure of some 2,700 square feet of new surface, it was necessary that the waterjet system, however, be considerably improved from the penetration rates achieved in that first test when it took approximately 30 minutes to cut a hole through some nine inches of granite. While much of this development work took place some years ago, it is perhaps of interest to review those findings at the present time and demonstrate their application and use in the construction of the current monument.

EARLIER EXPERIMENTATION

The first experiments in cutting granite at the University of Missouri-Rolla were undertaken under funding from the U.S. Bureau of Mines (Ref. 2) in order to determine whether high pressure waterjet cutting would be sensitive to small changes in the rock microstructure identified by a slight change in the physical properties of the rock quite easily detectable to a skilled stonemason.

Results from that initial research which was largely carried out using a single impact of a jet on each sample surface, indicated that while rock could be cut in all three directions, that the alignment of the impact direction relative to the rock fabric made a difference in the volume of rock removed of up to some 20 times for equivalent waterjet impact energy. The reason for this is that the orientation of the microcracks in the granite tends to increase the extent of the damage caused by the jet in one direction and limit it in another.

During the 1970's evaluative testing of the waterjet ability to cut a variety of different rock samples and to establish correlations between cutting conditions, rock properties and effects, included granite (Ref. 3). Thus, by the late 70's, it had been established that high pressure waterjets, at pressures below 140 MPa, could cut through granite and create a hole 2.5 cm. in diameter at the rate of approximately 70 m. per hour. That work was funded by what was then the Energy Research and Development Administration (Ref. 4). Then, in 1977, the National Science Foundation, which had previously funded investigation investigating the use of a wide variety of different techniques for cutting rock, collaborated with the Elberton Granite Association in a study to determine if a more advanced technique could be developed for cutting over that currently in practice.

CONVENTIONAL CUTTING OF GRANITE

Current methods of cutting granite have centered around two techniques. The first of these is the application of a very high temperature generated when a fuel oil-air powered burner is played over the rock surface. Generally the torch is some it feet long and is held against the rock by a laborer who moves it over the rock surface, at a relatively slow rates rapidly heating the immediate surface. This rapid and restricted heating of the rock to a temperature above $1,000^0$ C. causes a differential expansion of the quartz and a phase change to occur. This induces very high stresses within the surface, causing localized fractures, and at that point very small particles of rock are broken from the surface as a very fine cloud of dust. Using this technique, it is possible to achieve a cutting rate of around 11 square feet an hour with a fuel oil consumption rate of 7 to 10 gallons per hour. This procedure is, however very noisy since the burner can be equated to a very small jet engine. It has the further disadvantage that it is difficult to use where the ground is wet, where the rock is weathered, or where there are large inclusions of quartz, mica, or feldspar, the major constituents of the granite. This technique requires that a relatively wide channel be cut, on the order of five to six inches, to allow the head of the lance to remain close to the rock surface, and has the further disadvantage that the rock surface on either side of this cut has been considerably weakened by the application of heat.

In contrast, the second method is the more historic and requires that one drill holes in the rock and then create a split in the rock between adjacent holes. This is achieved by setting small wedges in these holes and then sequentially striking each wedge along the line which outlines the shape of the block to be excavated. With this method, it is possible to induce a fracture to grow between the drill holes to the point that a block of rock is broken from the solid. This technique requires the presence of adjacent free surfaces, in order to work, and the control on the direction of the fracture once it leaves the guiding line of the pre-split holes is not always that required. In order to create initial free surfaces in a new layer of the rock, or to cut a plane which does not lie in the direction under which the granite would normally split, this technique is somewhat modified. Pneumatic or hydraulic drills are used to drill a sequence of holes to the full depth required and are spaced only 5 cm. or less apart along the required length of the block. The intervening rib of rock between adjacent holes is then removed with a mechanical tool. While effective, this method has proven too uneconomical in recent years for the practice to continue. Thus, the industry was faced with the need to find a new method of cutting granite.

WATERIET CUTTING TRIALS

As a result of the arrangement between the National Science Foundation and the Elberton Granite Associations the University of Missouri-Rolla was one of two concerns contracted to demonstrate the capability high pressure waterjets for granite cutting. Tests were arranged to take place in quarries in Elberton under funding from the Elberton Granite Association, through Georgia Institute of Technology. The University team demonstrated that, cutting at a pressure of approximately 100 MPa. with a flow rate of 60 liters per minute, it was possible to cut a slot in the rock some 6 m. long and up to 1 m. deep. The dimensions of this cut are important because the largest rock subsequently to be carved in the UMR Stonehenge was some 5 m. long and would need to be cut to expose a side length of some 1 m. Excavation rates during the demonstration program in Georgia had been between 1.2 and 2.5 square meters per hour, depending upon the rock condition. This rate exceeded that which had been developed in the preliminary tests carried out, prior to going to Georgia, at the University test facility using blocks of rock which had been supplied by the Granite Association and also by quarries from Barrie, Vermont and Millbank, South Dakota.

The primary objective of these initial trials, since it had already been established that waterjets could cut granite at the pressure considered, was to find the angle best suited for the waterjets as they were directed out of the original nozzle body. Because the waterjets would only be capable of cutting to a depth of approximately 6 to 14 mm. on each pass over the rook's surfaces the nozzle must follow the jet into the cut. In order for this to occur, the slot must be wide enough for the nozzle body and feed pipe to the nozzle. Optimization of flow conditions to the nozzle required that the feed pipe have a diameter of 2.5 cm., which in turn meant that the nozzle body would be approximately 37 mm. in diameter, and so the slot required should be at least 50 mm. wide. Preliminary tests with a variety of nozzles angles indicated that if the included angle between nozzles was less than 22 degrees, the waterjet would not cut a slot of consistent width, but with so much of the force of the water being directed forward and less being directed to the side, it would bounce off the rock inward and over a sequence of three or four passes, the slot would get significantly narrower.

For this reason, results indicated that the included angle between the two jet orifices be maintained in excess of 30 degrees and ultimately an angle of 45 degrees was chosen

(Fig. 2). The two nozzles in the program were sized according to the volume flow rate of water achievable from the pump at 100 MPa, and this came out to be approximately 0.94 mm.

SYSTEM DESIGN

In order to cover all the surface in the slot, it was decided to spin the waterjet nozzles. This insured that the water would in fact cover the entire surface (as a function of advance rate and rotation speed) while at the same time the use of two nozzles of equivalent size, inclined at the same angle from the axis of the feed pipe, meant that the cutting lance would be balanced. It is important that the nozzle body and the resulting jets be symmetrically oriented around the axis of rotation of the feed pipe, since as the jet cut down to a distance of some 1.2 m. into the rock, it would be very difficult to provide support for the nozzles and thus a stiffening to insure that the slot maintained alignment. Some guidance would come from the walls of the slot previously cut, but it was of major importance to establish a balanced jet system to insure that the slot being cut was straight. As an aside it might be mentioned that this balance was sufficiently good and that it proved possible to remove less than 12 mm. of material from the edge of a rock sample during the course of the program. The spinning lance moved down and along the granite block without contact with the surface and maintained a straight cut as it moved down. This capability which the waterjet has, is not matched with conventional mechanical systems where the lack of resistance to the cutter on the free edge of the rock would likely mean that as the cutter (either a saw or pick-cutting device) would be deflected from the vertical by the imbalance in the forces on the cutting surface.

In order to design the proper rotational apparatus for the waterjet system, it is necessary to understand the role which the property known as incremental distance has on the efficacy of waterjet cutting. Waterjets will cut through rock, in general by an extension of the microcracks which exist around the grain boundaries of the material. In a very granular rock this confines the jet action to a slot of approximately three times the diameter of the jet. In contrast, in a crystalline rock, the grain boundaries are not as clearly defined and the crack will frequently propagate some additional distance further out from the impact zone before it reaches the surface, thereby widening the effective cut. By establishing how far away two adjacent jet paths could be set while concurrently removing all the intervening rib of material, one could achieve the most effective rate and slot creation rate for the jet cutting system. Following a series of experiments, it was decided to set a rotational speed of approximately 100 rpm for the equipment. This number was chosen based on a lance advance rate of approximately 3 m. per minute down the rock length which would mean that the distance between adjacent passes of the water over the rock surface in the central part of the slot (the maximum increment distance) would be approximately 6 mm.

The efficiency of jet cutting was determined by the depth to which the jet would cut on adjacent passes. This was controlled to a degree by lowering the jet a fixed amount on each pass. Normally the jet cut approximately 25 mm. ahead of the nozzle. The rotation of the lance was achieved by a hydraulic motor attached to the platform which carried lance support.

The mechanism to move and rotate the lance (Fig. 3) was extremely simple since all that was necessary was that a high pressure hose carry water from the pressurizing pump to a coupling and a rotary seal from which the water passed to a cutting lances, on the bottom end of which were the two waterjet nozzles which directed the jet at a rock surface. In order to construct this device certain requirements were obvious. Since the rock had to be aligned correctly in order to be cut and moving 17 ton blocks at frequent intervals was likely to be expensive, it was much easier to build a degree of flexibility into the cutting mechanism.

To achieve this, four towers were constructed at the corners of a rectangle. The towers were constructed of 8" by 8" wood ties (Fig. 4) and provided a firm foundation upon which the frame work which comprised two longitudinal triangular frames were established. Running between these two lateral frames was a cross piece which could be moved along the laterals at either end to accurately position the lance. This connecting triangular member was some 6 m. long, thus allowing cutting of the largest blocks which would be received from the quarry. The triangular member was made of a radio antenna mast which had a flat surface to which a traveling platform could be attached. This platform was used to mount the drive mechanism for the lance rotation and the feed mechanism for the lowering of the lance. The connection was through a sliding Teflon block to give rigidity of position to the lance platform, and four rubber tires allowed the free movement of the block up and down. It should be mentioned that the passage of the block was to a degree inhibited by the collection of very fine sand from the cutting operation which required that the passage be frequently cleaned.

The lance was fitted to the platform on the cross member through a threaded rod (Fig. 5). A small hydraulic motor was connected to the top of the threaded rod and flow to this motor was geared so that the operator could observe the rotation of the rod which had been set up so that one revolution lowered the nozzle 2.5 mm. Thus, in cutting the rock, the operator could stand back from the cutting operation (he was located outside the boundaries of the cutting zone established by the four pillars) and by watching and counting the number of passes of small white mark on the threaded section, could control the lowering of the lance. Generally, three rotations were achieved at the end of each pass of the lance over the rock's surface. The feed rate of the nozzle down the cross member was achieved by connecting the mounting platform to a gear drive through a bicycle chain (Fig. 6). The bicycle chain proved sufficient to pull the platform back and forth during the operation. It was, however, found necessary because of wear problems to increase the size of the chain somewhat over that originally used.

Concluding Remarks

It was not always possible to achieve the exact precision of the cut that had been originally anticipated. This was because of the fact that the rook came from fairly close to the surface of the granite quarry, thus there was some weathering of pieces of rock and this in turn meant that the jet could cut a slightly wider path in the weathered section than it did where the granite was fresher. This had the effect of causing as much as a centimeter of deviation from the original alignment that had been established and this meant that when the rocks were finally assembled they were not quite as precisely aligned as had originally been hoped.

Nevertheless, the primary objective of the exercise was achieved and the 53 rocks comprising the monument were all carved to shape and the monument completed in time for the June 20, 1984 dedication of what is now officially known as the UMR Stonehenge. During the course of the cutting operation, the equipment operated for a total of some 1,000 hours, although much less time than that was required to actually cut the rock and this time is taken from the hour meter on the equipment. This indicates the length of time that the equipment was running whether cutting or not. It does, however, indicate that a set of equipment is now available which can cut on a regular basis, steadily for an acceptable lifetime of equipment with an acceptable level of maintenance. The major component of wear once the pump had been changed and the operating procedure had been firmly established, came from the erosion of the nozzles. Because conventional carbide tips were being used for the operation (on the basis of their relatively low cost) and the fact that the nobles were held within 3 cm. of the surface, they were exposed to the rebound of the water. This rebounding water contained very sharp granite pieces so that the holder and the outer edges of the nozzles were relatively rapidly eroded. The thickness of metal surrounding the jet path gradually became so thin that the internal pressure of the jet blew the nozzle out. This normally did not occur until after some 20 to 30 hours of cutting had taken place, and given that the nozzles cost on the order of $20 to $30, this meant that the nozzles' replacement cost for the program was on the order of $1 an hour which was considered to be quite acceptable and the cutting lifetimes of the nozzles were considered effective.

One additional point might be made. That is that while cutting rates were considered quite acceptable in most of the cutting operation where the jet was cutting along the grain of the rocks the slots tended to be slightly narrower and more difficult to cut. In order to gain the slightly additional cutting advantage required to maintain an acceptable cutting rate (on the order of 1.8 square meters per hour) a long chain polymeric additive was used in the water. By allowing this to properly hydrolyze before the entry into the jet, an improved jet stream was achieved and, in consequence, it was possible to maintain the rate of advance which was achievable without the additive, on more favorably oriented sides.

ACKNOWLEDGMENTS

This work was inspired by Dr. Joseph Marchello, Chancellor of the University of Missouri-Rolla, and the monument was designed by Dr. Joseph Senne, Chairman of the Civil Engineering Department. The cutting program was carried out by a team of graduate and undergraduate students, with major advice and assistance from Mr. L. J. Tyler and Mr. J. Blaine of the Rock Mechanics and Explosives Research Center staff. Without all their help, none of this work could have been accomplished and it is gratefully acknowledged.

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Figure 1. Stonehenge Monument -- East View

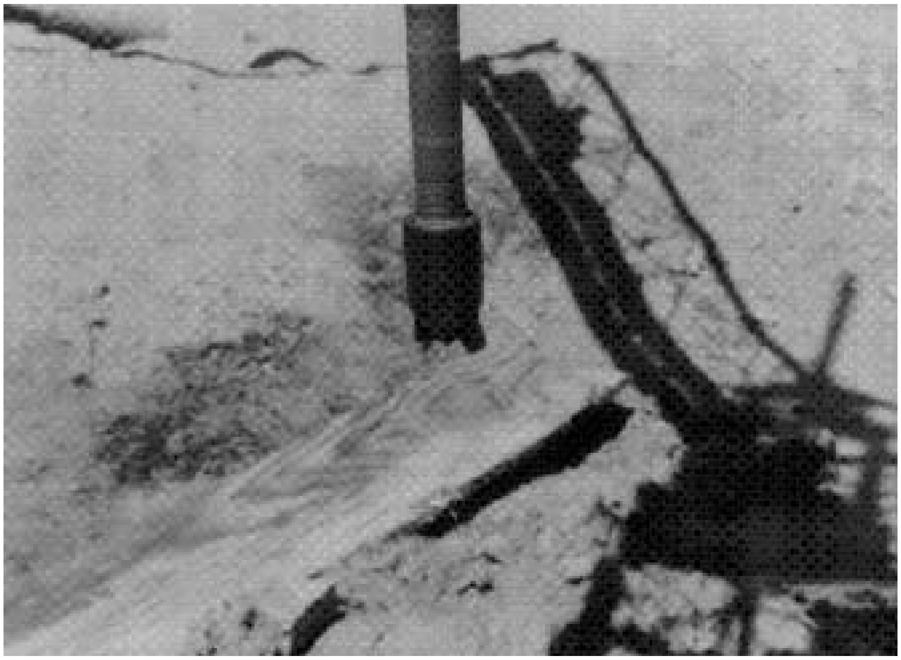

Figure 2. Nozzle holder with two nozzles with 45 degree included angle.

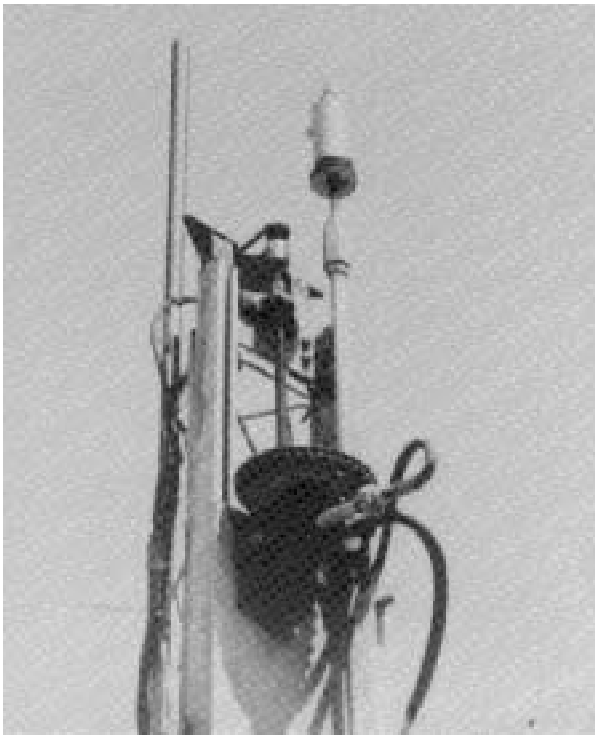

Figure 3. Lance feeding device and lance running gear. Rotary coupling on the top of high pressure pipe.

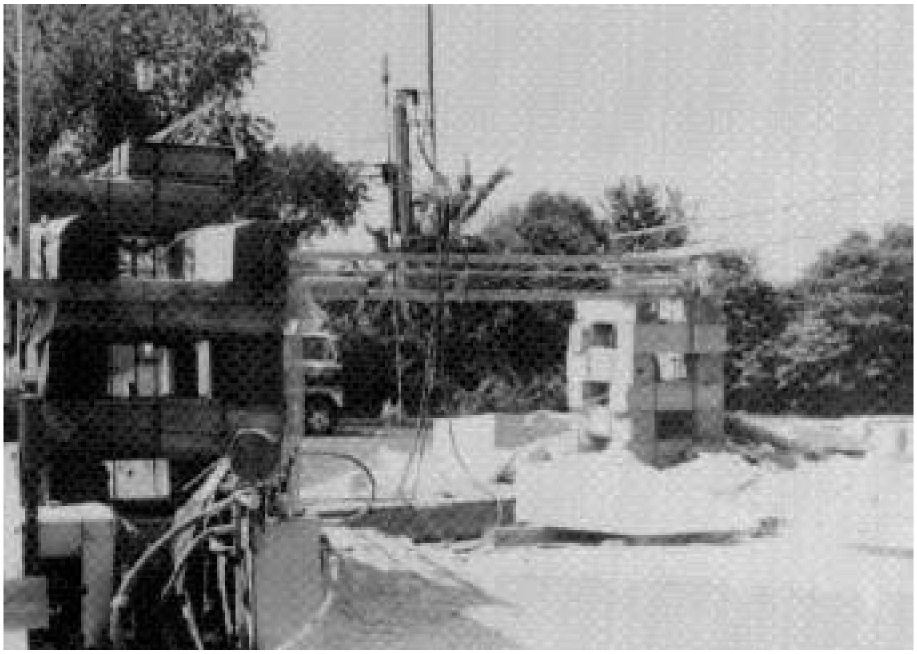

Figure 4. General area of cutting rig.

Figure 5. Main platform view with horizontal bar, high pressure pipe and feed screw.

Figure 6. Cutting lance horizontal feeding mechanism.

INVESTIGATION OF ULTRA—FINE COAL DESINTEGRATION EFFECT BY HIGH PRESSURE WATER JET

by

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ABSTRACT

The advantages of using high pressure water jets as a new technique for breaking coal into particles of an ultra fine size distribution are reviewed in this paper. The results of a preliminary investigation of the technique, carried out in the High Pressure Waterjet Laboratory of the University of Missouri—Rolla, are presented. These data indicate that the lower levels of energy required of this comminution process, in contrast to that required by conventional technology, improve the practical viability of using fine powdered coal in industrial processes, when the waterjet comminution is included in the grinding cycle.

The compressive stress mode of failure induced in conventional milling processes is changed in waterjet comminution to a tensile failure by extension of existing particle boundaries, and the benefits of this change in failure mode are reviewed.

INTRODUCTION

Comminution may be defined as either a single or multistage process during which mineral ores, or other materials are reduced from a variety of original sizes by crushing and grinding, to a narrowly defined product size range required for subsequent processing. In many industries this is the only way available to provide materials in the proper size range in order to achieve a final product having the required properties. Current comminution technology is both energy—intensive and inefficient. Up to 99% of the energy consumed during the operation of conventional size—reduction devices goes into non—productive work, with only 1% of the energy input available to create new surfaces in the fragmentation process. Comminution requires a vast quantity of energy and is thus an appropriate target for significant energy savings. This is particularly true given the large tonnages of materials involved in the size reduction operations, which are so great that even small improvements in the efficiency of comminution would provide considerable savings in both energy and mineral resources.

The low efficiency of the grinding process is frequently not the result of the inability of the machine to break particles effectively, but rather of the application of stress where there are no particles. In tumbling bail mills, for example, both the positioning of the particles to receive the stress and the stress application itself are provided by a random tumbling motion of the balls. The result is that much of the energy input to the mill is wasted in nonproductive contact between balls or between the ball and the wall and the overall energy—efficiency of such processes is accordingly low.

Current pressures for improving the efficiency of comminution are driven, not only by the increasing cost of energy, but also by demands for finer product sizes from both the mineral and coal industries, requiring that expanded research must take place into more novel methods of fragmentation and that a more scientific basis for understanding of the comminution process, in these new modes, has to be created. Among those areas requiring special attention are the application of ultrasonic energy, vibrational energy, cryogenic pulverization and the use of differential rock expansion moduli which induces rock weakening by extending the pre—existing microcracks at grain boundaries.

In all these new developments, however, there is an overall movement toward changing the primary mode of material failure. Substantial progress in ultra—fine comminution can be achieved if the compressive crushing principle used to date is changed to one where particles are fragmented by the tensile growth of pre—existing internal flaws. One effective and proven method of doing this is if the flaws are internally pressurized to rupture by using high energy concentrated water or oil jets.

Previous work performed at the Rock Mechanics and Explosives Research Center (RMERC) has examined the process of water jet crushing of rock, coal and wood (Ref. 1, 2, 3).The benefits from the use of the jets arose in two major modes, it was found that waterjets have the capacity for reducing the material to an ultra—fine size range with a relatively low level of input energy, and. because of the fluid penetration along the grain boundaries, a much better separation of the different components of the material was achieved, at a much larger particle size than has historically been the case. Such a beneficiation which can easily be translated into an efficient separation of pyrite, silica and other dirt from coal, is brought about by the waterjet ability to penetrate along pre—existing microcracks on the individual particle boundaries. The water will extend the microcracks by hydraulic fracturing and differential compression due to the change in modulus, and thereby establish a separation of the two minerals. This process can be effectively used to break the raw material into its different constituents, often at the original particle size of the component. This may then lead, in some cases, as it did with lead ore, to an ability to separate the components on the basis of the particle size. The very localized application of pressure, when combined with a chemical additive in the water may also allow an enhanced removal of the organic sulfur component of the coal.

MILLING PRINCIPLES

From the basic mechanistic point of view in order for a particle to be fractured, a stress must be induced which exceeds the fracture strength of the material. The mode of fracture and the path which it follows depends both on the material, and the structure of the particle and on the way and rate at which the load is applied. The way in which the load is applied will control the stresses which induce fracture extension within the particle. The force to induce this growth can be either simple compression, which causes the particle to fracture in tension, applied at either a slow or fast rate. The applied load could be in shear, such as is exerted when two particles rub against each other. Or the load may be applied as a direct tensile force on the particle. There are many terms which have been used to describe the mechanisms of single particle fracture, such as abrasion, cleavage and shattering. A relation between them is suggested in Figure 1. Typical postulated stress conditions for the particle are shown in Figure 2.

Size reduction involves the rupturing of chemical bonds within the material in order to generate new surfaces, thus the chemical processes which are associated with the fracture process will have a significant effect on the energy required to induce this fracture. This extends beyond the actual bonds themselves to include the surrounding environment. For example, the presence of water at the crack tip, reduces the forces required for propagation and thus results in improved breakage efficiency especially where the water contains inorganic ions and organic surface-active agents (Ref. 4,5) . One explanation for the effect of these additives is that they are able to penetrate the micro—cracks in the solid, ahead of the major crack front and, take part in the highly reactive events occurring during breaking. Because the capillary flow of these fluids into the material ahead of the main front runs at the velocity of crack propagation it provides a means of transmitting energy more easily within the crack area, and because the material surrounding the reaction zone is less effected, there is a degree of confinement of the energy to the crack tip zone. The high pressure water jet containing chemical additives creates extremely dynamic conditions where micro-cracks grow ahead of the surface and become pressurized, thereby enhancing any chemical changes which might occur (Figure 3),

ULTRA—FINE MILLING TECHNIQUE CHARACTERIZATION

In order to produce a homogeneous, pumpable suspension of coal, which does not settle in the delivery lines, and which burns at the required rate, coal for use in "liquid" power plants must be ground from the "standard plant size" down to below 10 microns in size. Among the many milling methods used for this process the finest product is achieved by the use of autogenous attriting machines. The distinguishing feature of these machines is that size reduction is effected by particles impacting on each other, having acquired the necessary energy to cause fragmentation in the process from a solid or fluid impeller. Included in this class are the following systems:

Buhrstones—which causes comminution through an abrasion action
Colloid Mills- where comminution occurs by collision between particles
Fluid Energy Mills—where again particles interact upon one another
Sand Grinder—particles are reduced by contact with sand particles

The advantage of this range of equipment is that material can be broken into very small sizes (less than 10 microns) with the size distribution controlled within a narrow size range. The equipment, however, can only operate, at any one time, with small quantities of materials, and the initial feed size of particles lies in the range from 10 mm to minus 150 microns depending on type. For the sand grinder, for example, the feed stock should be already crushed to below 70 microns. A much greater disadvantage for this type of machine is the very high power consumption required for the crushing. For example, the energy required to grind Montana Rosebud coal using no. 2 fuel oil to a size of 90% less than 4 microns is 1940 kWh/ton, and with no. 6 fuel oil 10,250 kWh/ton

(Ref. 6). Predicted energy requirements for producing 10, 3 and 1 micron product sizes are represented in Table 1.

The energy required to achieve a given size reduction increases as the product size decreases. (Ref. 7) This increase is due to many factors and is a consequence not only of the type of mill and the microscopic condition, but also of the mechanisms of failure that operate at the individual particulate level. This is obvious because fragmentation in a chamber is partly brought about by an interaction between the particles and the chamber wall. The behavior of single particles under stress cannot adequately represent the effects undergone in stressing large numbers of particles differing in size, shape, orientation and mechanical properties as well as stress history.

In such situations, the treatment of individual particles requires special attention and Griffith's hypotheses (Ref. 8) initially derived from an investigation of an elliptical crack on the surface of an elastic, isotopic, and homogeneous material must be drastically modified to describe the behavior of coal which is anisotropic, heterogeneous, and extensively pre—cracked. In this material, physical properties vary as a function of the degree of metamorphism of the coal particle. Under such a situation an analytical approach to coal fragmentation is very complex.

Recent experiments conducted with shaped explosive charges to investigate fracture formation in coal (Ref. 8) showed that there is intense fracturing of coal near the jet path, which is usually bounded by joints, bedding planes and cleat planes. The coal breaks into large and small pieces, usually parallelepiped, following natural cleavage planes. Beyond the intensely crushed zone, some large fractures are observed to cross joints and travel long distances, while fractures originating at the base of the jet penetration also travel across bedding planes and run deeper into the samples.

No plastic zone was observed for jet penetration in coal, but a crushed zone did exist immediately around the jet penetration hole. A scanning electron microscope comparison of the finest particles produced both by high pressure waterjet and by mechanical projectile impact show the same brittle pattern of fracture. Also the high speed photography (see Fig. 4) did not detect plastically deformed coal particles produced during high pressure water jet collision with coal chunk (v=1000 m/s).

NOVEL TECHNIQUE

Conventional mechanical methods of crushing and grinding have a very low efficiency so that a search for new novel grinding techniques is of great importance. A variety of novel techniques of comminution is currently being explored. Among the new techniques being examined high pressure waterjets, currently used to cut and drill rock, are now being considered for their disintegration potential.

In almost 20 years of experience at the RMERC in wide ranging experimental and theoretical studies of high pressure waterjet applications for rock excavation, material cutting and surface cleaning, it has been found that the high pressure waterjet has also excellent ability for material disruption. Such a capability is due to the following features:

- A Waterjet of 69.0 MPa pressure is moving with a velocity of around 400 m/sec, and its narrow jet diameter provides a very high energy flux input to the target.
- The jet's high energy density is concentrated in a very small impact zone, while intense differential pressure across the jet leads to microcrack generation and growth.
- The jet, upon impact, creates a stagnation pressure which forces water into the cracks and microcracks. It develops a hydromechanical jet action in these cracks and creates an increasingly dense network of cracks in the wall of the cavity created (Fig. 3).
- Rapid jet penetration into the pre-cracked coal can be enhanced by the use of surface active agents, which will also work to further comminute the coal and pretreat any contaminants in the coal.
- For COM preparation, the waterjet can be changed to an oil jet to eliminate the intermediate drying process.
- The separation of dirt from coal is improved by use of pressurized waterjets with separation, on occasion, being possible purely on the basis of a size differential.
- A crushing machine based on use of a high power wateriet is expected to be an extremely simple, cheap unit operating as a one step open circuit machine which can be introduced into any existing comminution flow path.
- There is a reduced expectation of mechanical wear or process contamination of the product.
- There is a tremendous opportunity to improve process technology, not only for coal comminution, but for all kinds of minerals by introducing a comminuting jet or by combining existing milling equipment with waterjets in a synergistic relationship.

Based upon these initial considerations, and the results of earlier work, a program of research was undertaken at UMR to gain data on the levels of energy required for waterjets to comminute coal. Results of the first tests were much greater than had been anticipated. It proved possible to reduce conventional, run—of—the—mine Missouri coal, from an original size of 1 inch down to an average particle size in the 1—5 micron level in a single stage operation. The energy required in order to achieve this was supplied through jets of less than one millimeter diameter, and was calculated to be considerably less than half that used in conventional crushing.

While the liberation potential of the technique could not be determined in that particular test series, it had been demonstrated previously (Ref. 2) when a 70 MPa pressure waterjet was traversed over the surface of both dolomite and sandstone samples containing galena. Because of the penetration of the water along the grain boundaries of the material, it proved possible to achieve greater than an 85% liberation of the galena from the host rock, and, of equivalent importance, to make this separation using a simple size separation, given the difference in the individual grain sizes of the components of the rocks. In passing it should be mentioned that the ability of the jet to make this penetration, based in part on the differential compression of the two different particle types under the impacting jet pressure is an explanation of the reason that it is easier for a high pressure waterjet to cut through a 200 MPa compressive strength granite, than it is for the same jet to cut through a 100 MPa compressive strength marble.

Given the favorable results of these initial evaluative experiments and a climate wherein a need for a more effective comminution process is required, it was decided to construct a dedicated coal comminution device, based on the waterjet crushing technique.

EXPERIMENTAL PROCEDURE

The experimental procedures were carried out in the High Pressure Waterjet Laboratory of the RMERC, using a Kobe Size 3 pump, driven by a 60 kW motor, and capable of providing a flow of 20 liters/min. at a pressure of up to 100 MPa. Samples of coal were obtained from a Missouri mine, measuring up to 60 cm. a side, and processed in an "as received" condition. During the course of the test series the cutting action of the jet was recorded using a Spin Physics high speed video—tape camera, operated at a framing rate of some 2,000 frames/sec. Smaller coal samples, in the size range from 1.33 to 1.9 cm., were obtained, both from an original reduction of the larger coal blocks, and as a separate supply. As a results of some photographic studies of the waterjet action on the coal particles, a special test frame was constructed for the work to be undertaken (Figure 5).

The smaller coal samples were fed under a rotating, 0.96 mm. diameter waterjet, operated in the pressure range from 20 to 85 MPa, and turning at 600 rpm., with samples being confined, either in a closed, or perforated container. Test runs of 15, 30 and 45 seconds were used, at operating jet pressure levels of 20, 40, 60 and 80 MPa. Following each test the coal was removed from the containers cried and then sieved. The total energy input to the coal was calculated for each test. The calculation was derived from a measurement of the volume of water used for the given nozzle diameter, as a function of operating jet pressure, and derived from the equation:

$$P = 5.0484 \times 10^{-4} \times Q \times \Delta p$$

with an assumed coefficient of power efficiency for the pump motor of 0.85. For the calculation Q was the volume flow through the nozzle in GPM, and Δp, the jet pressure is measured in psi. The power (P) resulting is measured in kilowatts.

Results from the test program are presented ion Figure 6. It is necessary to distinguish between the results of the test carried out in the perforated vessel and those where the vessel was closed. In the case where the container was perforated, the jet had only one opportunity to impact on the coal particles as it fell down from the feed hopper into the container. In the case of the closed container, because the water provided a support mechanism, the particles could pass under the jet influence a number of times in a very short time frame. Further the induced turbulence around the jet, as it became submerged provided the opportunities for strong turbulence and cavitation to be generated in the container, enhancing the destructive power of the jet. The turbulent conditions would also cause impact between coal particles, leading to an autogenous phase of the grinding process. The induction of cavitation in the process is extremely significant since the very high pressures generated by the implosion of the individual bubbles on particles, can induce additional comminution of those particles beyond the impact zone of the main jet. This is significant because it is often difficult to capture a small particle of coal, one suspended in the water, and to retain it long enough for the full force of the jet to hit it. What usually happens is that as the higher pressure fluid around the destruction zone of the jet reaches the particle, the particle is pushed away from the jet before the force is high enough to cause fracture.

In contrast, the overall force generated by the cavitation bubble is not present until the moment of collapse, which occurs in a very short time interval, and with pressure (which can be as high as 1,400 MPa) which has only a very limited range. The intensity of the bubble collapse pressure, and the large number of these bubbles generated during the cavitation event, mean that a tool exists for further break—up of even the smallest particles resulting from the main jet action. Unfortunately, at the present time there is insufficient knowledge of the best techniques to optimize this phenomena for its most effective use, and a large experimental and theoretical program would be required for this purpose.

RESULTS AND DISCUSSION.

Results from the test series are given in Figures 6 & 7. The initial tests were carried out with a perforated container which produced coal in two very distinct size fractions. It was considered that this was because of the single exposure of the coal to the jet. A relatively small amount of fine coal, below 400 mesh, was created from the localized crushing action, similar to that described above and identified around shaped charge jet impact. Larger particles were generated by the growth of the smaller number of large cracks which propagated from the impact point to a free surface, at the same time. The energy requirement was still at an acceptably low level.

In order to achieve a single size range with a more uniform and finer product, it was necessary thus to get the larger particles re—exposed to the jet. In order to lift the particles back into the jet stream, the container was closed. This trapped the jet water and provided the required lifting mechanism. Serrendipitously it also caused a cavitation and turbulent action within the chamber, which enhanced the destructive action of the jet. When this was changed, between 55 & 75% of the coal was reduced to an average size of 5 microns. This is plotted with the power consumption calculated for that portion of the product which was crushed to below 100 mesh (Figure 7).

Because this is just a preliminary report on some very crude initial experiments absolute conclusions on operational conditions and final power consumption data should not be drawn. Results to date are, however, sufficiently significant as to allow a comment. Within a single 30 second test exposure one kilogram of Missouri coal was reduced from an original mean size of 2.5 cm. to a product with an average particle size of 5 microns. The power consumption for this test extrapolates to an energy consumption of 652 kWh/ton.

If it is recognized that the attrition devices used in conventional operations require a feed stock of maximum size 200 mesh (Table 1), but even with that initial advantages energy requirements run from 4 to 70 times greater than that discovered for this process (Figure 8). Fine coal has an increased potential market, both in power generation and other industrial processes, of great size. The size of that market is a function of the cost of the processing required to achieve the coal fragmentation necessary. Energy and plant cost savings of the magnitude which appear possible as a result of this program suggest that the market may be quite a lot larger than currently envisaged.

ACKNOWLEDGMENTS

The authors wish to express their gratitude to Dr. G. Galecki and Mr. P. Olko for their very considerable assistance in the carrying out of the experimental work and for their input in discussion of this program. The encouragement received from Dr. Charles Beasley, Acting Director of the RNIERC is also gratefully acknowledged. None of the work in the Research Center is accomplished without the help of Mr. John Tyler and Mr. James Blaine, and this is also gratefully acknowledged.

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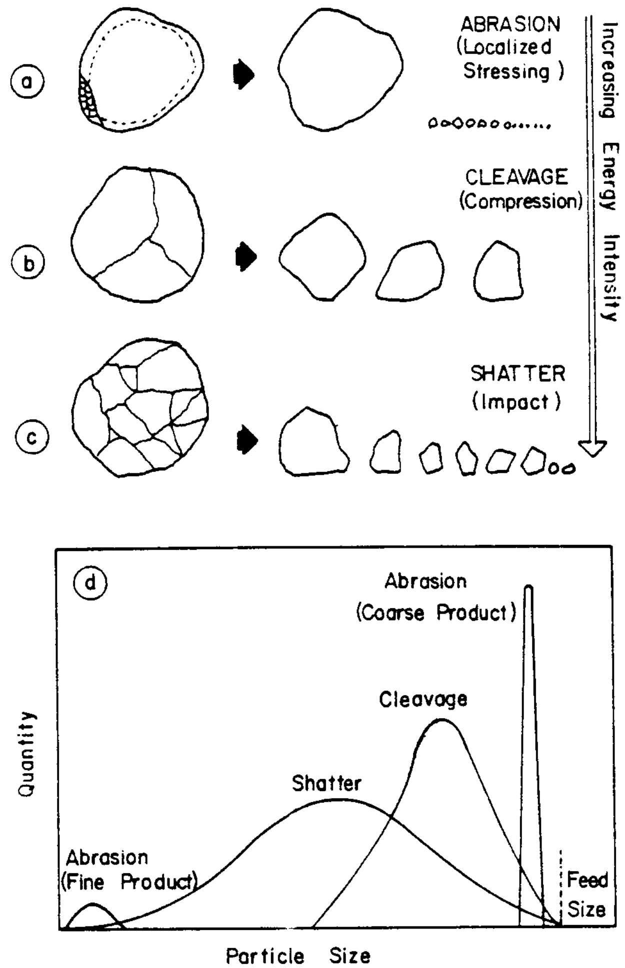

Figure 1. Representation of the mechanisms of particle fracture and the resulting product size distributions.

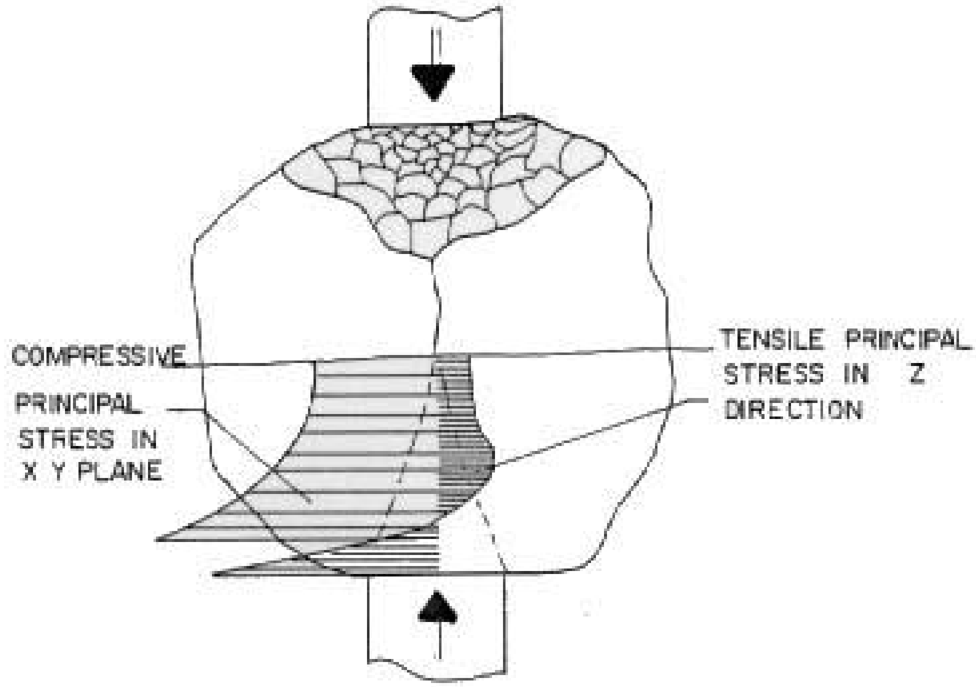

Figure 2. Combination of fracture mechanism and distribution of principal stresses under compressive loading.

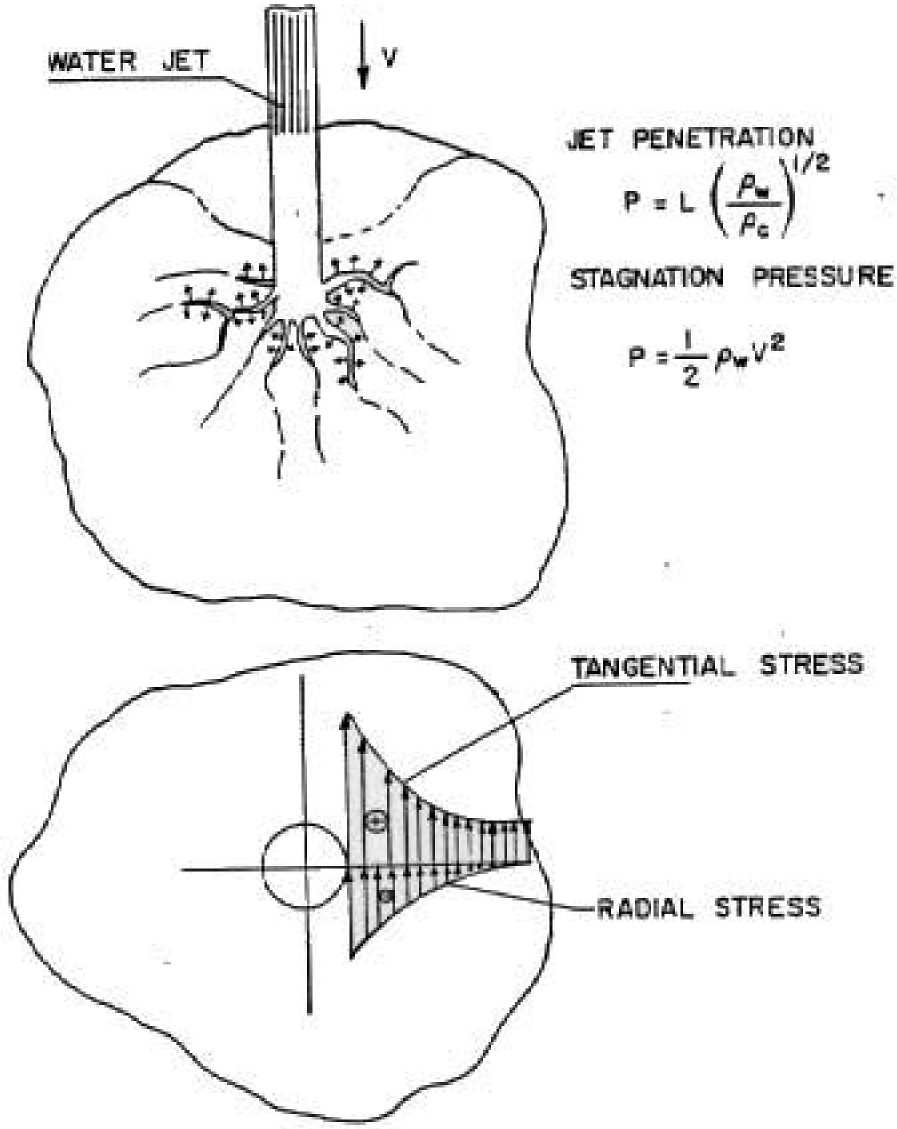

Figure 3. Mechanism of coal grain separation by high pressure waterjet.

TABLE 1. Prediction of Energy Requirements (kWh/t) for Crushing to 10, 3 & 1 Micron.				
		Median Size		
		10µm	3µm	1µm
Montana Rosebud	#2 Fuel Oil	106	880	6082
	#6 Fuel Oil	203	3515	47503
Lower Freeport	#2 Fuel Oil	26	217	1500
	#6 Fuel Oil	35	599	8100

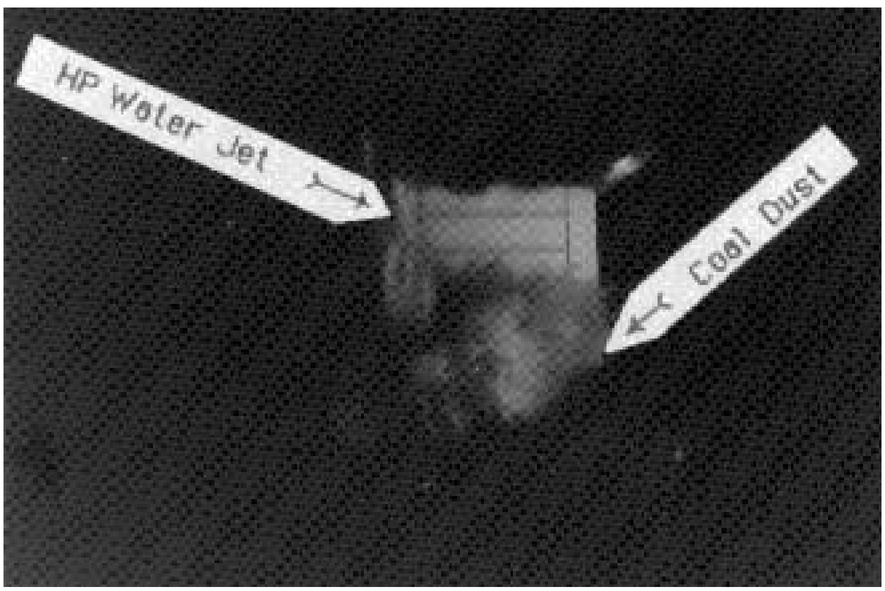

Figure 4. High pressure waterjet collision with coal chunk sequence.

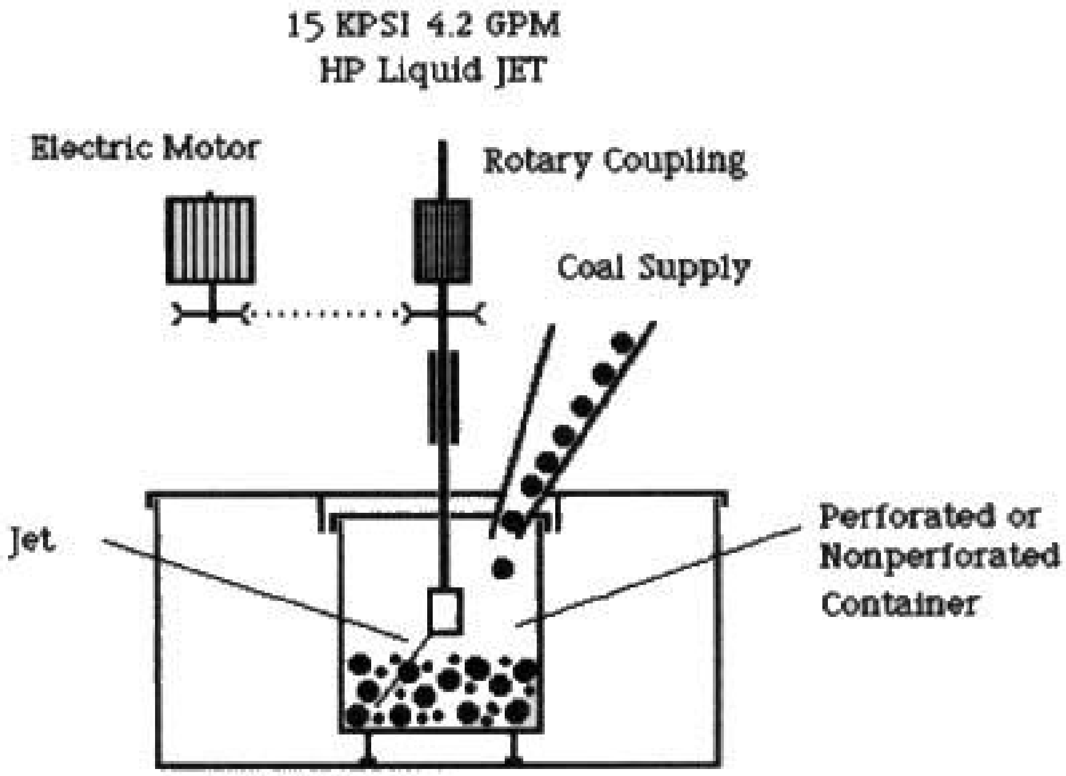

Fig. 5 Coal disintegrator

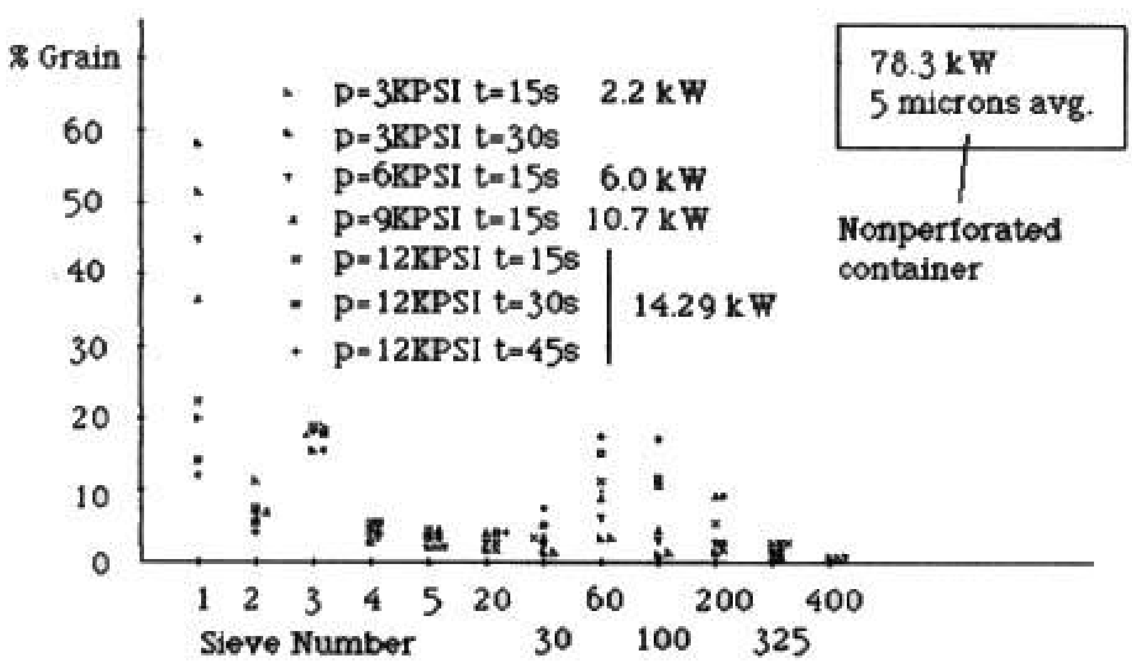

Fig. 6 Coal particles distribution under high pressure water jet action for specified time and jet power. Feed size .742 to .525 of inch.

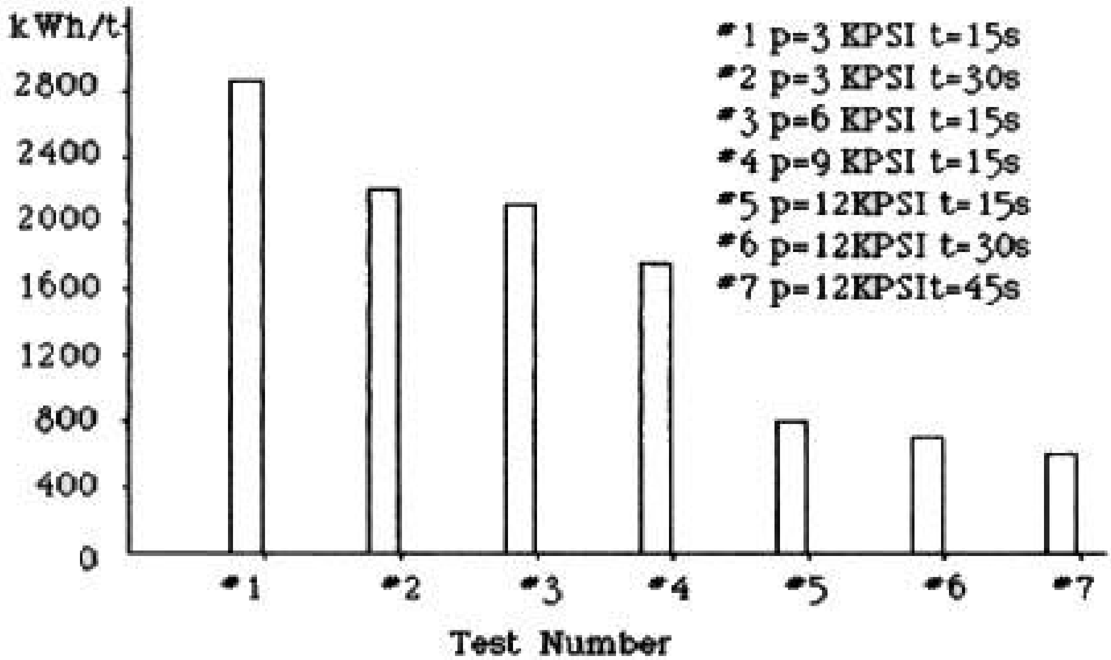

Fig. 7 All power consumed for coal communication by high pressure waterjet refer to fraction 100 mesh. Feed size .742 to .525 of inch.

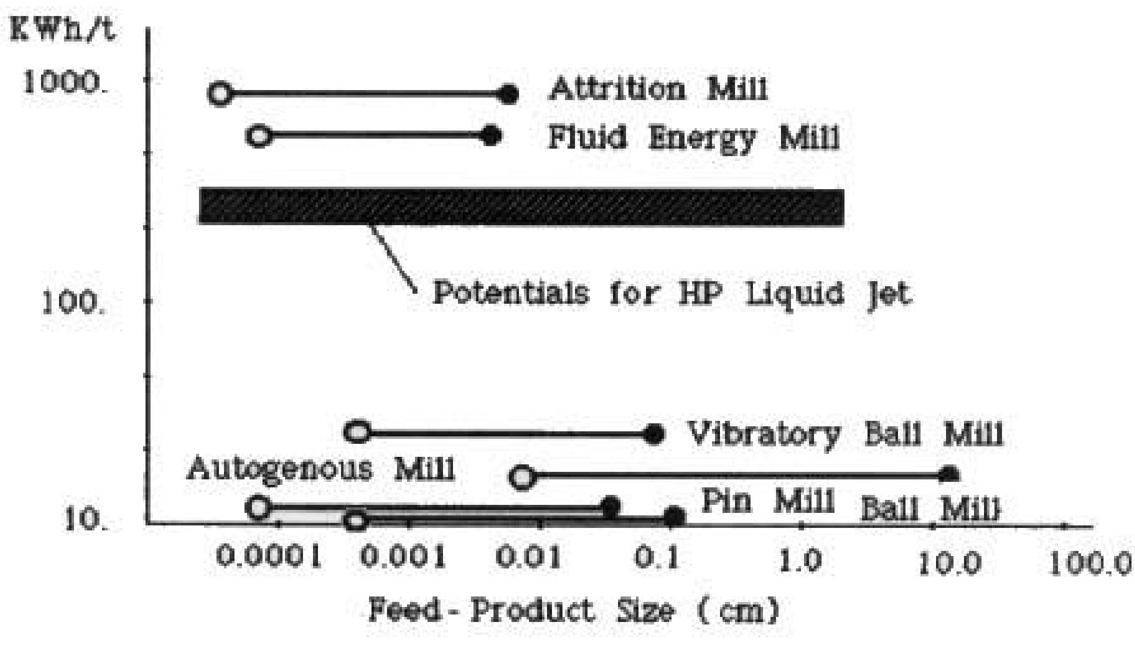

Fig. 8 The reported average energy requirement for several devices.

RELATIONSHIP BETWEEN WATER CANNON DESIGN, PUUSED WATERJET ANATOMY AND ROCK BREAKING EFFECT

by

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ABSTRACT

This paper describes experimental investigations on a pulsed water cannon conducted by the China Institute of Mining with the cooperation of the Zao-Zhang Coal Mine Administration (Shengton Province, China) and Fu-Dan University (Shanghai, China). Water cannon design, its major parameters and dynamic measurements of the pressure pulses are demonstrated. Based on the test results, a relationship between water cannon design, pulsed waterjet anatomy and rock breaking effect could be derived in this report.

BASIC TYPES. PERFORMANCE AND FEATURES OF THE EXPERIMENTAL WATER CANNON

1. Basic types of water cannon

On the basic of the detailed analysis of papers (1,2,3) current devices generating the pulsed water jets (the so-called water cannon) could be classified into two basic types according to the working principle of piston impact (5):

1) The extrusive water cannon;
2) The percussive water cannon.

Various physico-mechanical processes of water pressurization are experienced in the above-cited experimental models. Let us throw further light on their distinctive features. The extrusive water cannon is characterized by a smaller jet diameter, a greater jet length, a larger ratio of jet length to diameter (generally more than 1,000 to 5,000), a longer duration of peak pressure and less technological difficulties, because the device under operation is almost set in the quasi-static state of extrusion. By contrast, features of the percussive water cannon could be defined by a larger jet diameter, a shorter jet length, a smaller ratio of length to diameter (less than 1,000 to 5,000) and greater peak pressure pulses. In view of occurrence of repeatedly impulsive loads, demand for the impact toughness of materials used for manufacturing the device has to be higher, and technological difficulties may be more than that of the extrusive water cannon.

Test results indicate that geometric and dynamic features of pulsed jets have to meet following requirements for obtaining the better rock breaking effect:

1. Nozzle exit diameter should be enlarged to 6mm or larger as far as possible;
2. Jet length should be as short as possible, and the ratio of length to diameter should be in the range between 1,000 to 5,000;
3. Impact pressure should be higher than rock compressive strength as far as possible.

2. Design features and basic performance of the experimental water cannon

In order to cater for requirements mentioned, a percussive water cannon was chosen as the experimental set-up. Except for the energy cumulation process produced in the nozzle, two pressurization processes have to take place in the water cannon sections right ahead of the nozzle. The latter result from the cross section area changes of pistons and the acceleration process of the striking piston. The percussive water cannon must be also constructed with a cushion chamber. Based on this principle, an experimental percussive water cannon of type 79 was designed for the purpose of investigation on interrelation between cannon design, jet anatomy and rock breaking effect. Its major parameters are as below:

1) Nozzle exit diameters:	4mm, 6mm
2) Peak impact pressure:	7400 bar (nozzle dia. 4mm) 6500 bar (nozzle dia. 6mm)
3) Maximum nitrogen initial pressure	100 bar
4) Nitrogen compression ratio	3:1
5) Ratio of cross-section areas of the striking piston (large dia.) to that of the compression piston (small dia.)	10:1
6) Flow rate per shot	0.51 liter
7) Oil pressure (hydraulic power supply) up to	350 bar
8) Stroke of the striking piston	250 mm
9) Stroke of the compression piston	170 mm

INVESTIGATION ON JET ANATOMY AND CHARACTERISTICS OF PULSED WATER J ETS

Utilizing high-speed photography equipped with square-grid scenery and recorder of pressure waveform, the anatomy and the kinematic and dynamic characteristics of the percussive water jets were studied.

1. A thorough study of typical photographic films of pulsed-jet anatomy

1) Major parameters of the pulsed-jet No. 9906061

a) Fundamental parameters of the pulsed-jet

Nozzle exit diameter	6 mm
Impact pressure	5300 bar

b) Major parameters of photography

shot frequency	4,000 fps (frame per second)
Film size	4.5 x 22 mm
Timer frequency	1,000 1/sec
Shot distance	2,000 mm
Scale	100 millimeter per grid

2) Anatomy of the pulsed-jet No. 9906061 is shown in Fig. 1.

2. Typical impact pressure wave of the pulsed-jet

Two typical impact pressure waves recorded by a light oscillograph (Fig. 2) are selected and the preliminary analysis is discussed in this paper. Photographic and hydrodynamic parameters are the same as the above-mentioned. Fig.2 shows that each pressure wave can be subdivided into several section as follows:

1. Transition section bridging the gap between initial (may be zero) and well-developed values of the pressure wave;
2. Main section;
3. Cushion section;
4. Tailing section representing the remainder of pressure wave.

The above-cited parameters are identical for both nozzle exit diameters 4 mm and 6 mm. By comparison, there exist somewhat distinctive features between two pressure waves due to different nozzle exit diameters.

It is evident that the maximum value of main pressure wave in the nozzle exit diameter of 4 mm was greater by about 7% than that in the nozzle exit diameter of 6 mm. Some other findings are:

- In both transition sections duration of d_o = 4 mm and d_o = 6 mm was almost same, approximately of 5 to 6 ms;
- In main sections duration from forming to vanishing (exactly attenuating) was approx. 7 ms for d_o = 4 am, and 5-6 as for d_o = 6 mm;
- In cushion sections the vanishing pressure lasted 3 to 4 ms with d_o = 4 mm, but no apparent interval was displayed during d_o = 6 mm;
- In tailing sections the peak impact pressure was repeatedly demonstrated three times and lasted 23 ms during d_o = 4 mm, and no pressure fluctuation took place and lasted 16 ms with d_o = 6 mm.

3. Kinematic characteristics of pulsed jets

High-speed photography makes it possible to visually examine the anatomy of pulsed jets and calculate the instantaneous velocity of the pulsed-jet precursor at any moment. Based on the quantitative analysis of each frame one after another, velocity curves of the pulsed-jet precursor can be drawn as shown in Fig. 3.

Fig. 3 shows some points of interest as below:

1. The initial velocity just spurted from the nozzle was very low, and was 20 to 25 times less than the well-developed peak velocity of the pulsed-jet precursor;
2. Velocities of the pulse-jet precursor increased by leaps with time and was the same as the pressure waveform of the pulsed jet;
3. After a moment the precursor velocity increased rapidly at an extremely short time, and reached a peak value;
4. In Table 1 are listed major parameters of pulsed-jet precursor during its peak value as displayed in the films.

In addition, when air was entrained or dissociated (as seen in the case of the cyclonic water injection), jet anatomy would be changed as shown in Fig. 4. This type of jet anatomy is characterized by prompt diffusion and fatter haloes. An abhorred noise popped when water jet issued from the nozzle due to the accompanying dilatational process of air. The rock-breaking effectiveness, however, was dramatically decreased (5).

EXPERIMENTAL STUDIES OF ROCK-HREAKING EFFECTIVENESS

Aimed at evaluating the rock-breaking effectiveness of the water cannon device in all details, a further test program of breaking rocks was arranged. Limestone (type XI) blocks (size 500 x 300 x 500 mm) were chosen as the standard specimens in tests, and the rock-breaking process was recorded by high-speed photography and was kept in a file.

1. Effect of variations of major parameters on the rock-breaking effectiveness

a) Effect of variation of nozzle exit diameter, d_o is listed in Table 2.

b) Effect of variations of P (impact pressure) and S (stand-off distance) are also listed in Tables 5 and 4 for d_o = 6 mm.

When stand-off distance, S. is greater than 450 mm, the rock breaking effectiveness will be considerably decreased, as shown in Table 4.

2. Typical test data (photographic films) are briefly described as below:

1) Film frame No. 9986063 and its working conditions

a) Nozzle exit diameter	6 mm
b) Impact pressure	5850 bar
c) Stand-off distance	350 mm
d) Rock specimen	Ordovician limestone
e) Shot frequency	1,000 fps
f) Film size	18 x 22 mm
g) Timer frequency	1,000 1/sec

2) Film frame No. 9986063 systematically demonstrated the rock breaking process as shown in Fig. 5.

PRELIMINARY CONCLUSIONS

1. Using high-pressure pulsed-jets, the process of breaking brittle rocks takes place in an extremely short interval. The better design makes it possible to accumulate

the greater part of the energy within the jet precursor which plays an important role in the rock-breaking process.

2. Under all other parameters being equal, the increase of nozzle exit diameter, d_o would lead to an increase of the rock-breaking effectiveness. Therefore, to break brittle rocks, it is desirable to employ a shorter waterjet impulse characterized by a larger diameter and a smaller length.
3. In the well-developed percussive waterjet the rapidly accelerated dense cores of the jet will each, in turn, overtake the slower downstream flow so that a series of haloes surrounding the jet core will be formed at the expense of the rock-breaking effectiveness of the jet precursor.
4. On the other hand, haloes would also be formed during the ejection of a pulsed water jet entrained with air. Consequently, it would considerably worsen the pulsed-jet performance.
5. Prevention of air from entraining and reduction in the retarded water flow would improve both the pulsed-jet performance and the rock-breaking effectiveness.
6. With the above-cited aim, it is preferred to use water cannon of the percussive type instead of using the extrusive type.
7. The cushion chamber has to be constructed in the water cannon device. As a result, the straight strike of the piston to the barrel wall would not take place at the end of the striking stroke. The energy consumed in the cushion chamber must be as small as possible.

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4.
Куткин И.А. " Высокоскоростная Киносъемка Импульсной Водяной Струи " 1972

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Film frame No.		9901062	9906061
Impact pressure	bar	1,250	5,300
Peak velocity*	m/s	485	816.5
Duration after issue	ms	18.0	9.1
Jet Length	m	0.9	1.1
Dia. of low-speed halo,	mm	240	400
Dia. of dense core of precursor	mm	18	33
Duration of acceleration of leading edges of precursor	ms	0.8	0.6

*Being calculated according to films.

Table 1. Major parameters of pulsated-jet precursor (during the peak velocity was reached)

Nozzle exit dia.	mm	4	6
Nitrogen filling pressure	bar	50	50
Nitrogen commpression pressure,	bar	145	145
Impact pressure of pulsed jet,	bar	4,950	4,600
Stand-off distance	mm	350	350
Specimen		Standard specimen of limestone type XI	
Test result		produced a crate of 30 (dia.)×40 mm (depth)	divided into two after three shots

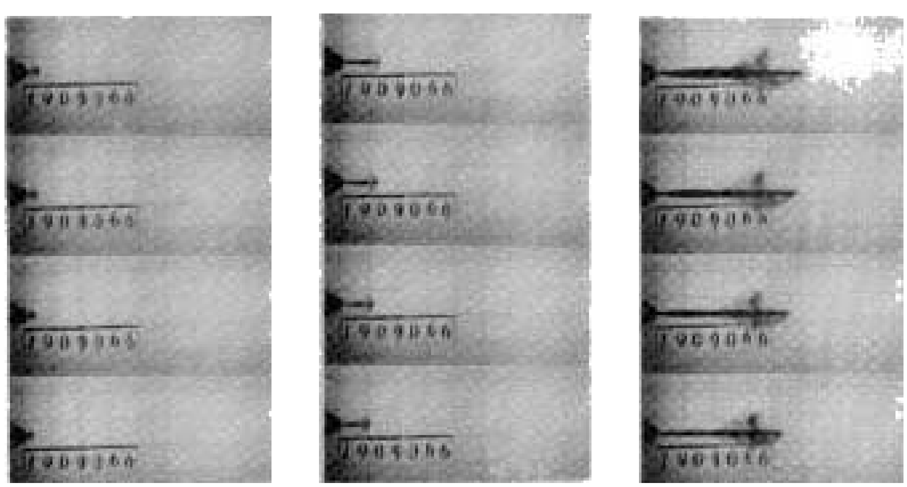
Table 2. Effect of variation of nozzle exit diameter on rock-breaking effectiveness

Nozzle exit dia	mm	6	6
N_2 filling pressure	bar	40	60
N_2 compression pressure	bar	120	180
Impact pressure	bar	3,900	5,650
Stand-off distance	mm	300	300
Specimen		stundard specimen of limerock type XI	
Text result		not apparent	divided into two

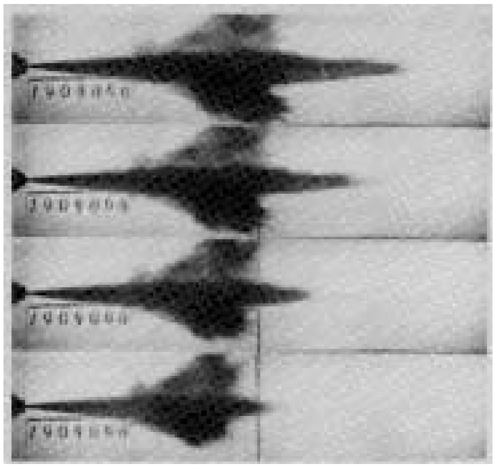
Table 3. Effect of impact pressure on rock-breaking effectiveness

Nozzle exit dia.	mm	6	6	6
N_2 filling pressure,	bar	60	60	60
N_2 compression pressure	bar	180	180	180
Impact pressure,	bar	5,650	5,650	5,650
Stand-off distance	mm	300	350	450
Specimen		Standard specimen of limerock type XI		
Test result		dividen into two	divided into two	produced a crater

Table 4. Effect of stand-off distance on rock-breaking effectiveness



1 2 3



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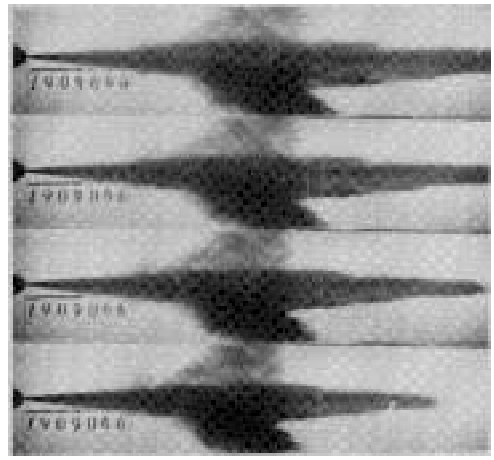

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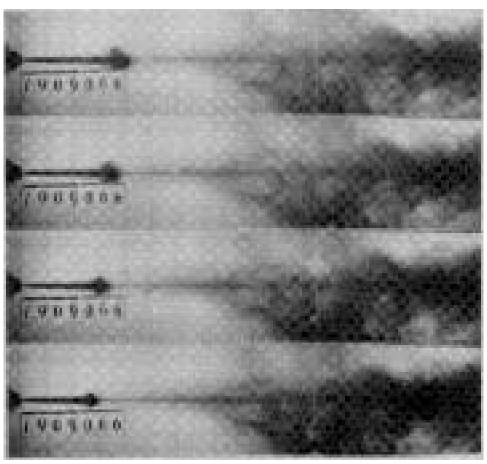

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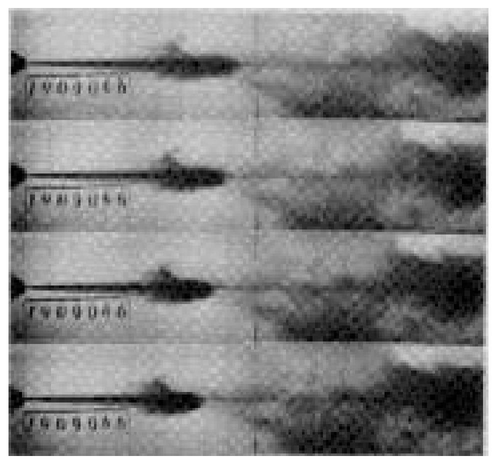


7

Fig. 1 Typical Photos of Pulsed Jet Anatomy by High-speed Photograph no. 9906061
1. low pressure jet when just ejected;
2. after 2 ms, high pressure jet, a cap-shaped precursor;
3. after 8 ms, dense core appears;
4. at 9 ms, peak pressure appears;
5. after 16 ms, jet diffuses and speed slows;
6. after 16 ms, low pressure jet formed by tailing wave is ejecting;
7. after 19 ms, there appears a cap-shaped precursor again.

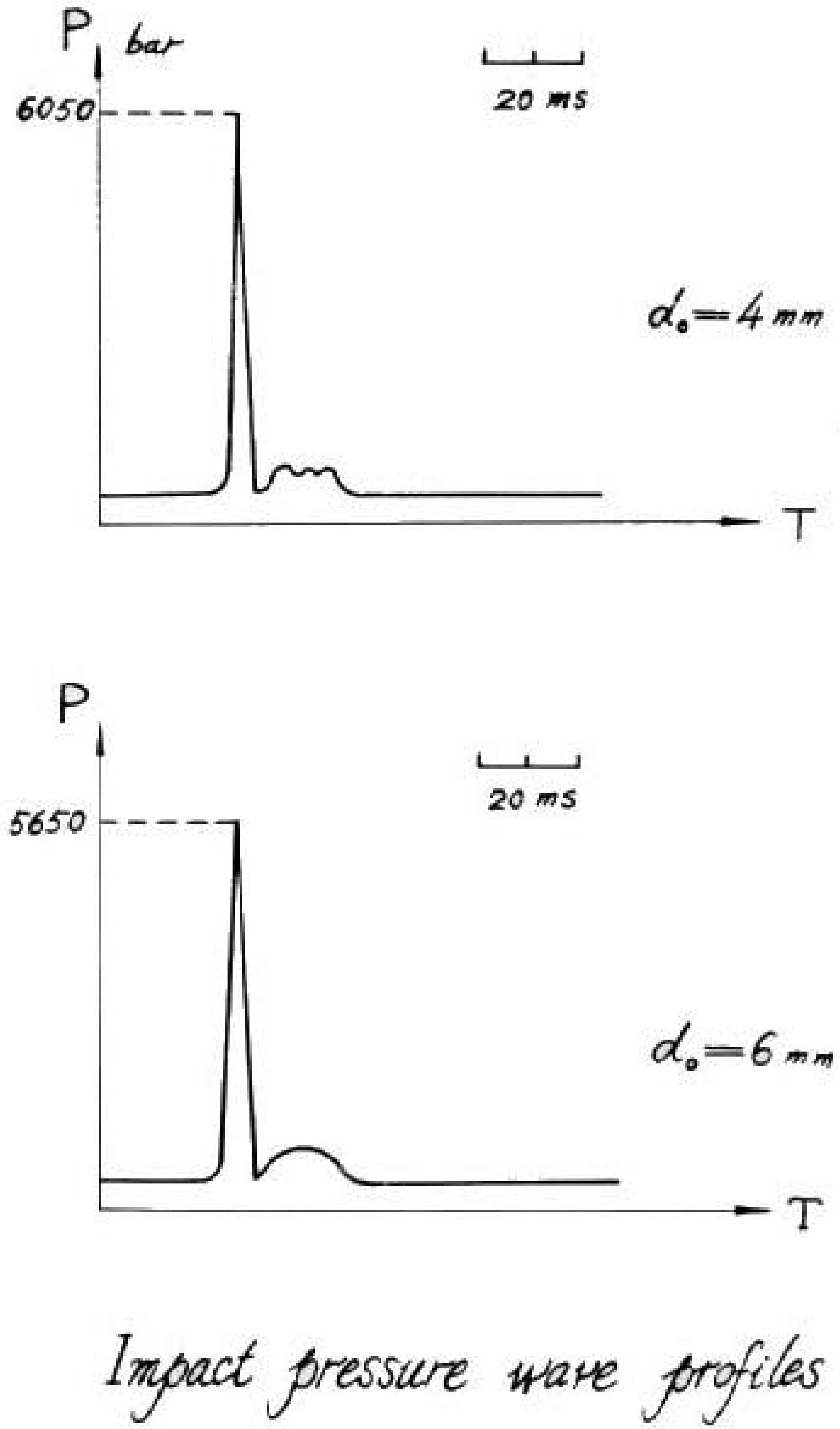


Fig. 2 Waveform of impact pressure of pulsed jet

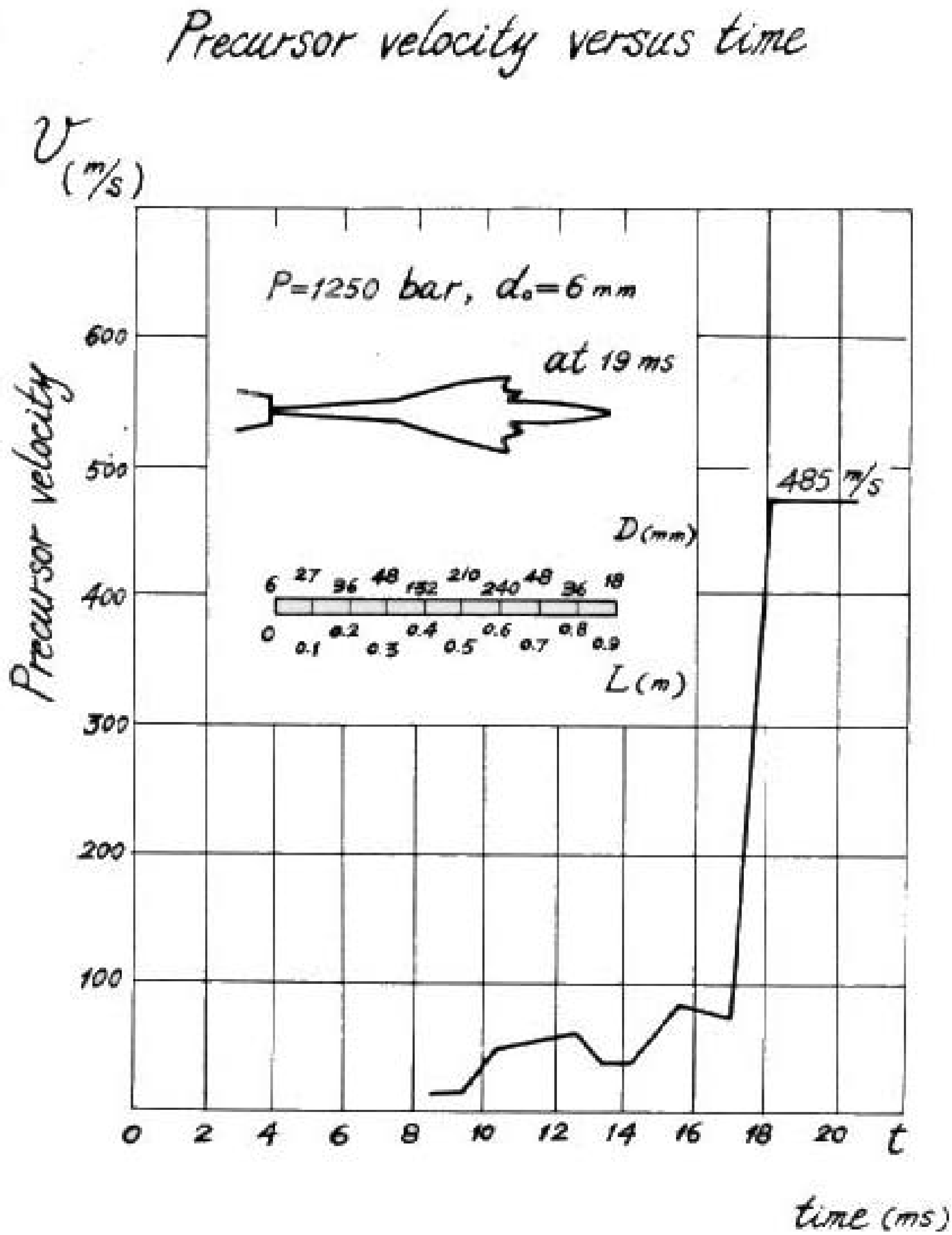


Fig. 3 Precursor velocity curves of pulsed jet

a) P=1250 bar, d_o= 6 mm

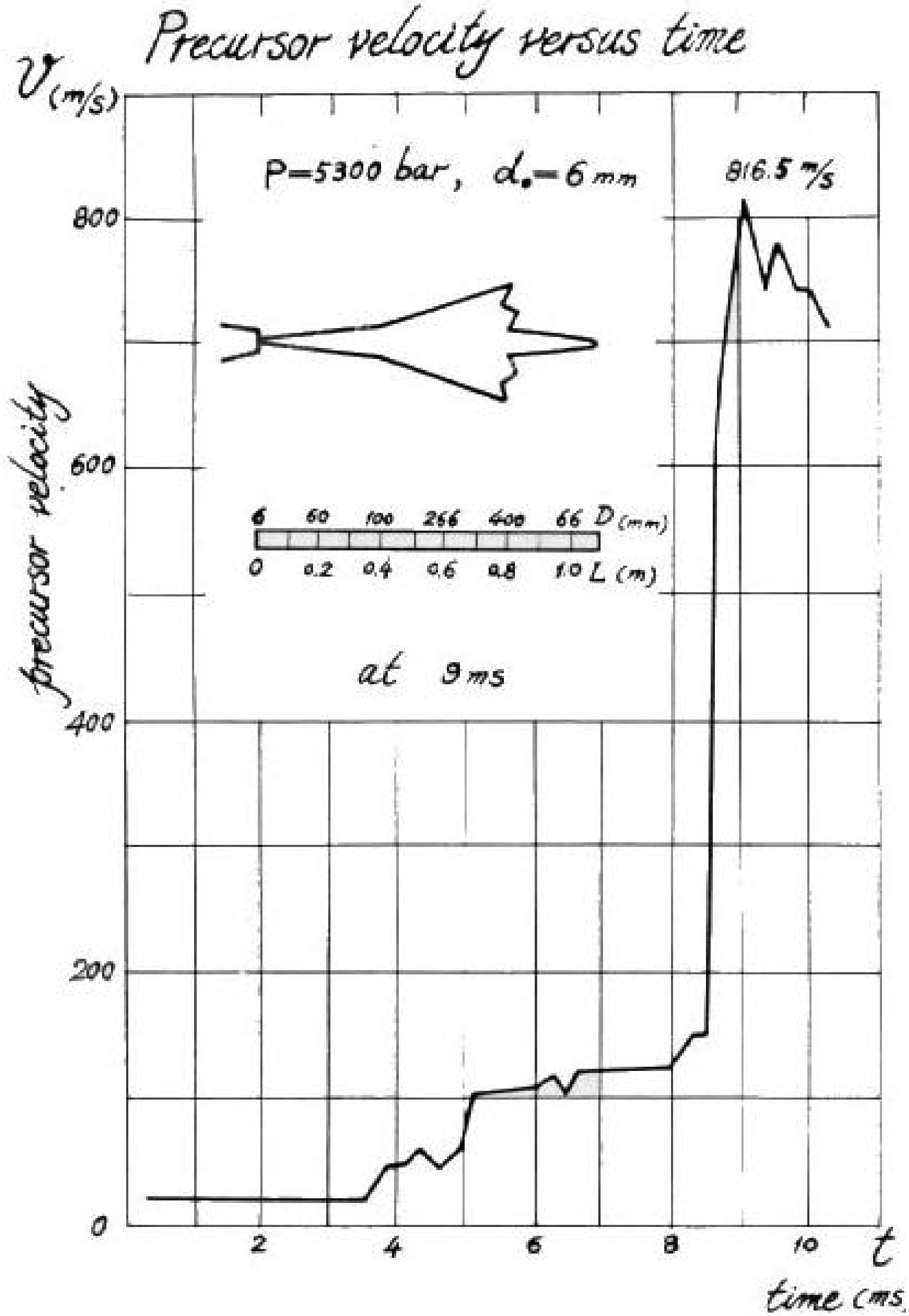
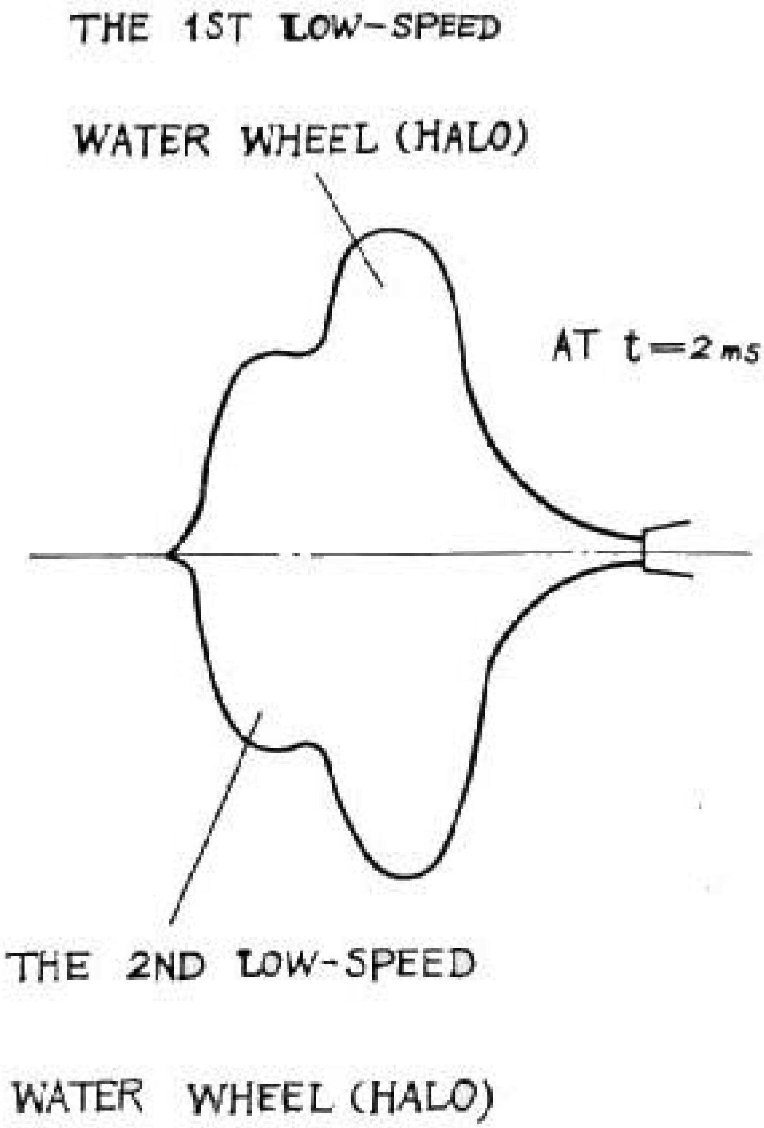


Fig. 3 Precursor velocity curves of pulsed jet

b) P = 5300 bar, d_o = 6 mm



Anatomy of percussive jet
affected by air and low-speed water stream

Fig. 4 Pulsed jet anatomy affected by air low-speed water stream

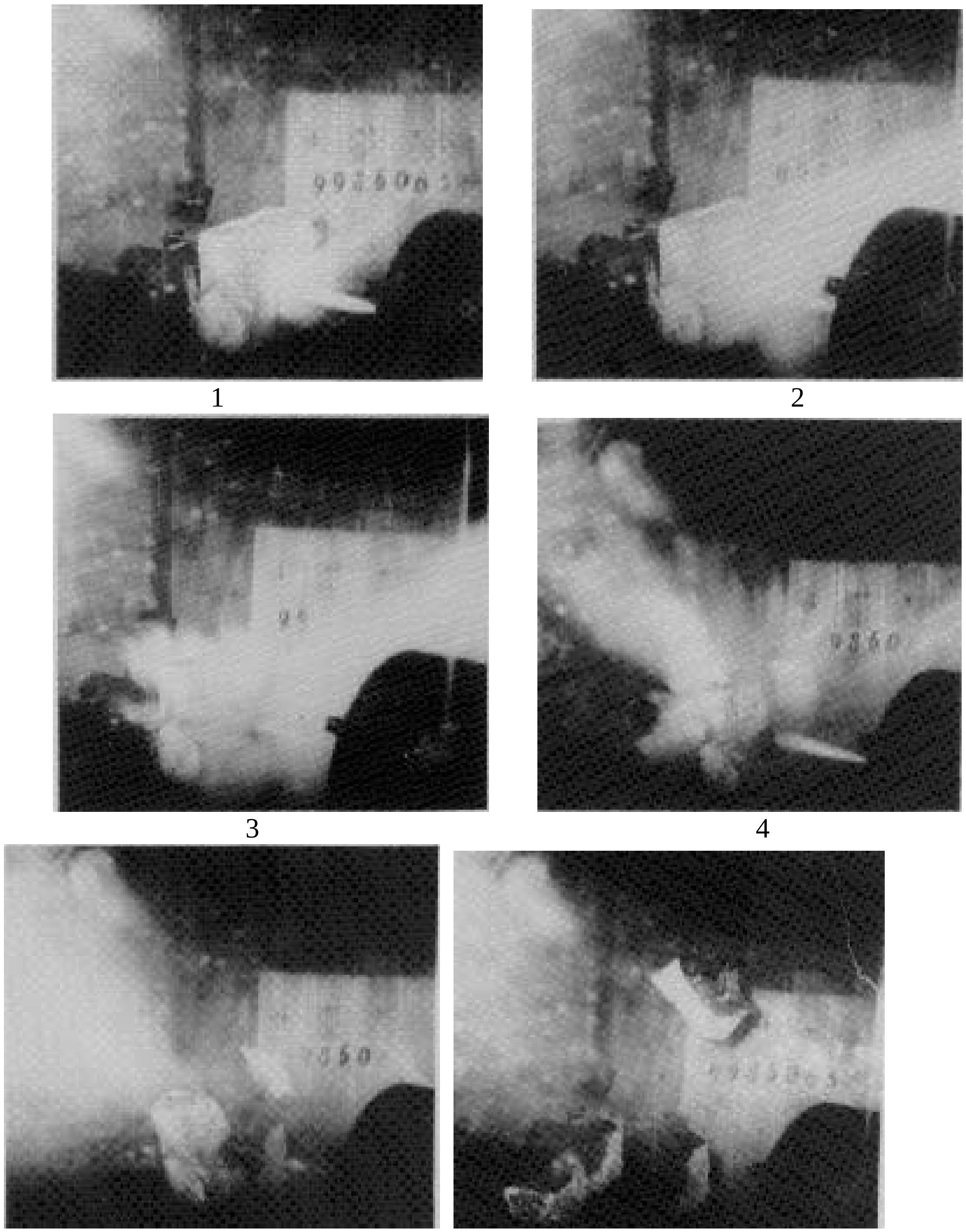


Fig. 5 Typical Photos of Pulsed Jet Breaking Rock High-speed Photograph no. 9986063
1. at 0 ms, the precursor of high pressure jet just reaches to rock;
2. at 2 ms, rock is strongly impacted;
3. at 5 ms, rock broken and jet penetrates the rock;
4. at 13 ms, the jet formed by tailing wave impacts the broken rock;
5. at 29 ms, the tailing wave vanishes and the left-side rock block is titting;
6. at 133 ms, the right -above rock flies away and the right-below rock keeps in the rest.

COMPUTER AIDED ENGINEERING AND DESIGN OF CUMULATION NOZZLES FOR PULSED LIQUID JETS

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ABSTRACT

The paper presents a simplified analysis and a computer graphics program which can be used on a personal computer for designing exponential cumulation nozzles, calculating performance of the nozzles, and calculating and plotting the static pressure distribution of liquid along the nozzle. The analysis is based on incompressible one dimensional liquid flow, neglecting the effect of air pressure in the nozzle prior to a shot.

The interactive computer program is written in IBM PC Advanced BASIC. It allows the user to select design input parameters, draw the nozzle to scale, and provide output data on performance and pressure distribution along the nozzle.

Possible applications of this type of unsteady flow nozzle are water cannons for tunneling, mining, or metal forming. Also discussed is a new and efficient type of hydraulic shock absorber using this type nozzle which can in principle maintain a constant maximum force during the deceleration stroke.

INTRODUCTION

The paper presents an interactive computer graphics program which permits design, computer drafting, performance analysis and plotting performance data for exponential cumulation nozzles. It permits rapid exploration of a wide range of design and performance parameters. These nozzles can produce pulsed high pressure liquid jets for application in rock excavation, mining or metal processing. As proposed in this paper, they may also be considered as components for use in a new type of hydraulic shock absorber which would have improved performance over existing designs by operating at the maximum permissible chamber pressure during the entire deceleration stroke.

The computer aided design program is written in advanced BASIC and runs on an IBM or compatible personal computer. It is hoped that this paper may aid in the analysis of this unique type of hydraulic component which may show potential for future applications.

DISCUSSION OF CUMULATION NOZZLES

A cumulation nozzle is used to produce pulsed liquid jets in which high velocity and high kinetic energy per unit mass is accumulated in a small portion of the liquid by extracting energy from the remainder of the liquid mass during a pulse.

As shown in Figure 1, the class of cumulation nozzles we consider has a flow area which decreases exponentially with distance along the nozzle. This type of nozzle was invented by Voitsekhovsky (ref. 1). A water cannon based on this type of nozzle was developed under the author's direction by Terraspace Inc. in the period 1970 - 74 (ref. 2,3).

In this single shot water cannon, the nozzle is connected to an inlet chamber where a piston makes impact on a volume of liquid, extrudes it into the initially empty nozzle, and converts piston kinetic energy to liquid kinetic energy stored within the nozzle, as the piston is stopped. The first liquid to exit may have a very high velocity, depending on design and operating parameters. The remainder of the liquid coasts out of the nozzle as the jet velocity drops. Typical design parameters for the Terraspace water cannon are shown in Table 1.

The maximum jet velocity actually measured (9800 ft/sec) corresponded to a dynamic pressure ($1/2\ \rho V^2$) up to 650,000 lb/in^2(4460 MPa) when a vacuum of 1 to 3 Torr was pulled on the nozzle prior to a shot. The jet pressure was only ~ 350,000 lb/in^2 (2413 MPa) when air at atmospheric pressure filled the nozzle.

Field test results of the water cannon showed it to be effective in excavating dolomitic limestone and also granite gneiss (ref. 3). The laboratory tests demonstrated cratering and splitting of Indiana limestone and Barre granite blocks.

3. ANALYSIS

Since the design of the Terraspace water cannon in 1970 - 71, papers have been published on the theory of cumulation nozzles by Ryhming (ref. 4) and by Glenn (ref. 5). A schematic diagram of the geometry is shown in Figure 1.

For the present analysis we consider a simplified version of the Ryhming analysis based on the following assumptions:

1. Prior to a pulse, the liquid is located in the region ahead of the nozzle and its volume is sufficient to fill the nozzle completely during a piston stroke.
2. The nozzle is initially empty of liquid and the presence of air in the nozzle is neglected, so there is zero pressure on the advancing liquid face.
3. The impacting piston crosses a gap of gas of negligible pressure and impacts on the liquid volume in a collision which preserves both the momentum and energy of the impact.
4. The flow of liquid is incompressible and continuous, with no cavitation or shock wave effects.
5. The nozzle flow area decreases exponentially with distance. The area decreases by a factor of 2.718 in a length K.
6. After the piston impacts the liquid and during its deceleration, the chamber liquid remains at constant pressure as a consequence of the constraints placed on nozzle size, shape and piston mass. The effects of shock waves and cavitation of liquid are neglected.

The basic equation derived by Voitsekhovsky (1) requires the nozzle e - folding length (K) to satisfy the following equation:

$$K = \frac{(AI)(MP)}{(RO)(AP)^2}$$

where AI = area of nozzle inlet
MP = piston mass
RO = density of liquid
AP = area of piston

The nozzle independent variables are:
DP = piston diameter
DI = inlet diameter
DE = exit diameter
L = nozzle length
PW = maximum allowable wall static pressure in nozzle
RO = liquid specific gravity

The dependent variables are:
K = e - folding length of nozzle
PE = maximum jet stagnation pressure ($(1/2)\,(RO)\,(UE)^2$)
UP = piston impact velocity required
MP = piston mass required
UE = maximum nozzle exit velocity
VW = volume of liquid required to fill nozzle
SP = stroke of piston against liquid
EP = kinetic energy of piston

The main equations are the continuity, energy and momentum equations.

4. THE COMPUTER PROGRAM

The computer program permits the following menu selections:

1. See the Instructions
2. See a Nozzle Design Example
3. Design and Draw a Nozzle
4. Calculate Performance of a Nozzle
5. Plot Pressure Distribution along the Nozzle
6. Terminate Program

Examples of the main analytical portions of the computer program are shown in Figure 2, which includes performance analysis and pressure distribution.

The static pressure distribution along the nozzle is based on a closed form solution derived by Voitsekhovsky (ref. 6) which may be written:

$$P(X) = \frac{1}{2}(RO)(VP)^2((\frac{DP}{DI})^2(e^{X/K} - e^{(2X-L)/K}))$$

This equation is line 50050 in the program in Figure 2. The shape of the pressure distribution along the nozzle is shown in Figure 3.

The wall static pressure remains low at the nozzle inlet, rises to a maximum near the liquid front and then drops to zero at the liquid front. The maximum wall static pressure is reached at the instant the liquid front reaches the nozzle exit.

In theory the maximum jet stagnation pressure is four times the maximum wall static pressure. In practice, with air at one atmosphere pressure in the nozzle, peak jet pressure can be as low as one half the theoretical, depending on nozzle area ratio and initial piston velocity.

The computer program called "NOZZLES" is stored on one single - sided floppy disk. It runs on an IBM PC or IBM - compatible personal computer with 64K memory.

5. POSSIBLE APPLICATIONS

The program can be used to explore nozzle designs and to calculate performance data for a wide variety of possible applications. The most likely uses are for water cannons to produce pulsed water jets for tunneling or hard rock mining. Another possible use is for design of small diameter pulsed jets to be used in punching or forming metals or plastics (e.g. nozzle diameter as small as 0.010 inch, 0.025 cm).

Another possible application for cumulation type nozzles may be in improved shock absorbers. A characteristic of the exponential nozzle is that it holds constant pressure on the chamber during piston deceleration. Therefore a shock absorber can be designed to give a very efficient deceleration stroke without the sharp pressure spike which can cause premature failure in shock absorbers subject to impact forces. Possible applications may be visualized for helicopter and aircraft landing gears, seismic shock snubbers, auto bumpers, crash barriers, etc.

For example, a helicopter oleo - pneumatic landing gear shock absorber may be visualized schematically as shown in Figure 4. This design incorporates a hydraulic chamber, nozzle and jet energy dissipation chamber which are mounted on the structure of a helicopter. The outer oleo cylinder with internal piston confines a volume of hydraulic fluid (oil) in the chamber and in turn it receives the shock load applied by the wheel of the helicopter in case of a crash landing at vertical impact velocities up to 30 ft/sec (9.1 m/s). The upper chamber is preloaded with nitrogen to form a spring and to assure that the oil is returned rapidly by gas pressure and gravity to the pressure chamber during normal ground maneuvering loads.

In case of a crash landing, the piston velocity is suddenly raised by tire impact. This forces oil upward into the nozzle, storing kinetic energy equal to the product of piston force times the stroke. This deceleration impulse must be designed in combination with plastic yielding of the helicopter structure to limit the deceleration of the passenger compartment to a safe level to assure passengers survival. Peak deceleration loads not exceeding 4 to 8 g are desirable.

Design studies and analysis of such passive hydraulic shock absorption systems are needed to explore their merits and to compare their performance with alternate systems which would incorporate active microprocessor control of hydraulic fluid pressure.

6. ACKNOWLEDGEMENTS

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7. REFERENCES

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Typical Design Parameters for the Terraspace Water Cannon

Piston mass	143 lb (64.8kg)
Piston diameter	7 in (17.8cm)
Piston impact velocity	200 ft/sec (61m/s)
Volume of nozzle	38 in (623cm)
Nozzle inlet diameter	2.43 in (6.17cm)
Nozzle exit diameter	0.329 in (0.836cm)
Nozzle length	48.9 in (124cm)
Nozzle area ratio	54.5
Maximum internal pressure	200,000 psi (1380MPa)
Theoretical maximum jet dynamic pressure	800,000 psi (5520MPa)
Theoretical maximum jet velocity	10,000 ft/sec (3322m/s)

TABLE 1

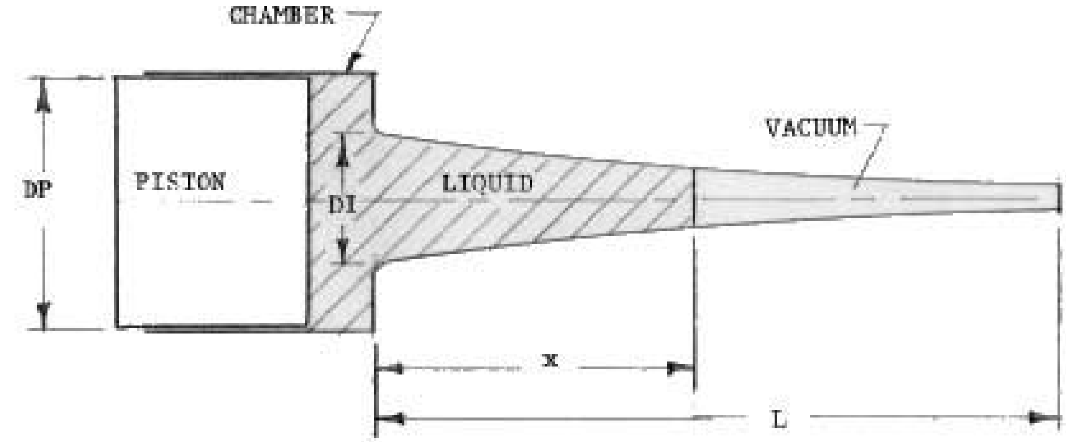

Fig. 1 Schematic Diagram of a Nozzle with Liquid Front Located at Distance x from the Nozzle Inlet

Fig 2 Portion of Computer Program for Nozzle Performance Analysis

```
40000  REM *** 4. PRFORMANCE ANALYSIS ***
40010  SCREEN 0,1: CLS: COLOR 14,1,9
40020  LOCATE 2,5: PRINT "ANALYZE PERFORMANCE OF NOZZLE"
40030 LOCATE 4,3: PRINT "ENTER THE FOLLOWING DESIGN VALUES"
40040 LOCATE 6,2: PRINT "MAX. NOZZLE WALL PRESSURE "
40050 LOCATE 7,2:lNPUT "(FROM 50000  TO 300000 PSI) = "; PW
400060 LOCATE 9,1:PRINT "SPECIFIC GRAVITY OF LIQUID (.1 to  20)1"
40070 LOCATE 10,7:INPUT "(USE 1 FOR WPTER) = ";RO
40072 LOCATE 14,2:INPUT "PISTON DIAM. (l-7 IN.) = ";DP
40074 LOCATE 16,2: INPUT "INLET DIAM (.1-3 IN.) = ";DI
40076 LOCATE 18,2:INPUT "EXIT DIAM. (.05-.5 IN.) = ";DE
40078 LOCATE 20,2: INPUT "NOZZLE LENGTH (5-5O IN.) = ";L
40080 LOCATE 22,2:INPUT "ARE THESE VALUES CORRECT?(Y/N)"; E$
40090 IF LEFT$(D$, l)="N" OR LEFT$(D$, 1)="n," THEN GOTO 40030
40091 IF LEFT$(D$, l)="N" OR LEFT$(D$, 1)="n," THEN GOTO 40030
40100 IF LEFT$(E$, 1) ="Y" OR LEFT$(E$, l)="y" THEN GOTO 40,200 ELSE GOTO
2000
40190 LOCATE 22,2 INPUT "ARE THESE VALUES CORRECT (Y/N)"; E$
40191 IF LEFT$(E$,1)="N" OR LEFT$(E$,l)="n," THEN GOTO 40030
40192 IF LEFT$(E$,1)="Y" OR LEFT$(D$,1)="y" THEN GOTO 40200  ELSE GOTO
2000
40200 COLOR 12,1,9
40210 LOCATE 24,6: PRINT "PRESS SPACE BAR TO CONTINUE"
40215 IK$=INKEY$: IF IK$="" THEN 40215
40220 CLS: COLOR 14,1,9
40230 LOCATE 2,5: PRINT "ANALYZE PERFORMPNCE OF NOZZLE"
40240 PE=4*PW
40250 LOCATE 6,2:PRINT "MAX. JET EXIT PRESSURE (PSI) IS";PE
40260 UE=SQR(148.6*PE/RO)
```

```
40270 LOCATE 8,2: PRINT "MAX. JET VELOCITY (FT/SEC) IS "; UE
40280 UP=(UE*DE*DI)/((DP)^2)
40290 LOCATE 10,2:PRINT "REQ'D PISTON VELOCITY (FT/SEC) IS"; UP
40300 K=L/(2*(LOG(DI/DE)))
40310 LOCATE 12,2:PRINT "e-FOLDING LENGTH OF NOZZLE (IN.) IS";K
40320 VW= K* .785*(DI)^2
40330 LOCATE 14,2: PRINT" VOLUME OF LIQUID REQ'D"
40340 LOCATE 15,2: PRINT "TO FILL NOZZLE (CU. IN.) IS ";VW
40350 SP = VW/(.785 * (DP^2))
40360 LOCATE 17,2:PRINT "THE PISTON STROKE ON LIQUID (IN.) IS";SP
40370 MP=(K*RO*48.98*(DP^4))/(1728*(DI^2))
40380 LOCATE 19,2:PRINT "REQ'D WEIGHT OF PISTON (LB.) IS"; MP
40390 EP=.5*MP*(UP^2)32.2
40395 LOCATE 21,2:PRINT "REQ'D PISTON ENERGY IN FT.LB. IS";EP
40400 COLOR 12,1,9
40410 LOCATE 23,15: PRINT "PRESS SPACE BAR TO RETURN TO MENU"
4042O IK$=INKEY$: IF IK$="" THEN GOTO 40420 ELSE GOTO 2000
49000 GOTO 2000
50000 SCREEN 1,0: COLOR l,l:CLS
50010 LOCATE 2,4:PRINT "PLOT PRESSURE DISTRIBUTION ALONG NOZZLE"
50025 REM ***TO KEEP PRESSURE ON SCREEN UP TO 300000 PSI ***
50028 DEF FN P(X) = .0067*RO*(UP^2)*((DP/DI)^2)*(EXP(X/K)-EXP((2*X-L)/K))):
50030 K = (L/2)/LOG(DI/DE)
50031 HSF=2
50033 XC=l00:YC=150
50050 DEF FN P(X) = .0067*RO*(UP^2)*((DP/DI)^2)*(EXP(X/K)-EXP(-(L/K)
        +2*(X/K))):
50060 X1=(L/100) :Y1=FN P(X1/500)
50070 X2 =(L/l00)+l:Y2=FN P(X2)/500
50075 LINE (XC+HSF*X1,YC-Y1)-(XC+HSF*X2,YC-Y2), 2
50077 FOR X=((L/100)+2) TO L STEP 5:Y=FN P(X)/500
50080 LINE -(XC+HSF*X,YC-Y), 2
50085 NEXT X
50090 REM *** PLOT AXES ***
50110 LINE (l00,YC)-(319,YC), l
50120 LINE (XC,40)-(XC,150)), 1
50130 FOR X=105 TO 319 STEP 10
50140 LINE (X,YC)-(X,YC-5),1
50150 NEXT X
50160 FOR Y=40 TO 150 STEP 10
50170 LINE (XC,Y)-(XC+5,Y), 1
50180 NEXT Y
50185 COLOR 1, 1
50190 LOCATE 22,3: PRINT "PRESS SPACE BPR TO RETURN TO MENU"
50200 IK$=INKEY$: IF IK$="" THEN 50200 ELSE GOTO 2000
59000 GOTO 2000
```

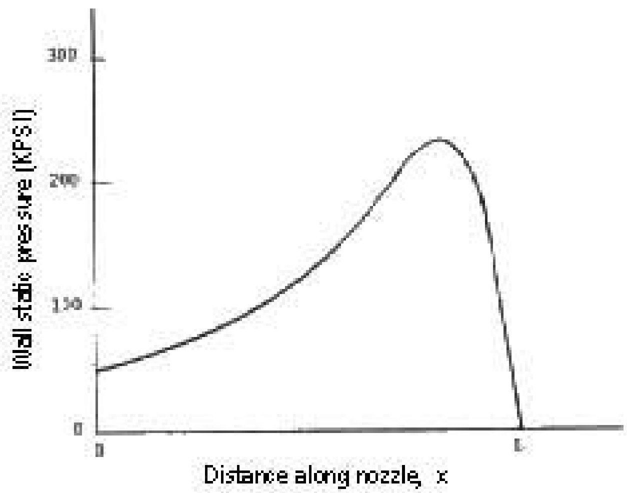

Fig. 3 Example Plot of Pressure Distribution Along Nozzle at Instant When the Nozzle Becomes Filled with Liquid

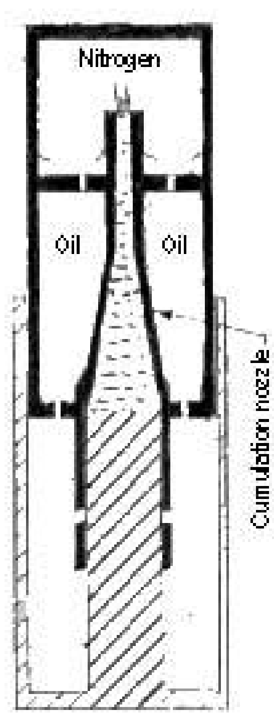

Fig. 4 Schematic of a Helicopter Oleo-pneumatic Shock Absorber to Withstand Crash Landing

WATER JET CAVITATION PERFORMANCE OF SUBMERGED HORN SHAPED NOZZLES

by

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ABSTRACT

It has been reported that the cavitating water jets which apply the eroding action of cavitation are effective for cleaning ships hulls, digging coal, debarking wood etc.. It is also pointed out that heavier eroding effect is obtained in water than in air.

In this paper, the authors report on the tests of a simple horn shaped nozzle that would promote the generation of cavitation in water. The promotion of cavitation is made by intensifying the shearing action between the jet current and surrounding liquid through adapting a divergent horn shape at the outlet of the nozzle (ref. 1). Under the conditions of the discharge pressure range of 4.9 - 29.4 MPa and a nozzle diameter of 1.0 mm, the brick chosen as a specimen was eroded. Then, the volume of erosion in the water was measured for various horn shaped nozzles.

As the result of the experiments, it is noted that the horn shaped nozzle removed approximately 14 times that of the volume eroded when compared with a hornless nozzle, which confirmed the effectiveness of the proposed nozzle. In addition, we can prove that a shape of horn which gives the maximum erosion does exist.

NOMENCLATURE

Symbol	Description	Unit
A	Eroded area	cm^2
D	Diameter of nozzle	mm
H	Water depth	mm
l	Length of nozzle throat	mm
L	Length of tapered portion	mm
P	Pressure	MPa
S	Standoff distance	mm
T	Exposure time	s
V	Eroded volume	cm^3
x c	Length of potential core	mm
θ	Angle of taper	o
σ	Cavitation number	-

INTRODUCTION

Surface cleaning with a water jet has been widely applied to chip removal after machining, bark removal, descaling the inside of pipes, paint removal and so on. Since the range of application will be enlarged if the cleaning capability of the water jets increases, the discharge pressure and the flow rate tend to increase, and at present, a discharge pressure up to about 70 MPa is being used in practice. However, increasing the discharge pressure leads to shortening the lives of the packings and check valves of the pumps and increasing the flow rate leads to enlarging the pump size inevitably causing a cost increase. Therefore, some other methods for increasing the cleaning capability have been sought.

It has been attempted to add the eroding action due to cavitation to the water jets, as one of the methods for increasing the cleaning capability. Cavitation often generates vibrations, noises and erosion in fluid machinery such as pumps and hydraulic turbines, but this attempt is to utilize the erosion in a positive way. These attempts have been made by devising the shape of nozzles which are the Centerbody Cavijet type in which separation of flow is caused by a center body placed at the outlet of the nozzle (ref. 2), and the Pulser Cavijet type and the Organ Pipe Cavijet type (ref. 3) which discharge the jets in a pulsating manner by self-resonating at the same frequency as that of the vibration which is generated in the shear layer of the jet. There are also reports stating that if these nozzles are used in water, higher eroding action is gained than in air as the cavity generates more effectively due to the shearing action between the jets and the surrounding water (ref. 2), (ref. 4).

In recent years, the cleaning application with water jets has been increasing not only in air, but also in water where it is used for removal of shellfish sticking to the legs of offshore structures and coral growing on the reef or cleaning the submerged surface of ships. Under submerged conditions, the cleaning capability tends to drop due to the damping action to the jet by the surrounding water. However, since the jet applying cavitation displays greater eroding action in water than in air, it is particularly suitable for submerged cleaning. Furthermore, if a nozzle that could generate cavitation more effectively is applied to a submerged cleaning operation, higher cleaning capability would possibly be obtained than that in air with conventional nozzles that do not cause cavitation.

Professor K. Yanaida, who is one of the authors, invented the horn shaped nozzle for submerged use which utilized cavitation. This nozzle has a divergent horn at the outlet of the nozzle which positively induces a shearing action between the jet and the surrounding liquid. Therefore, such a nozzle increases the generation of the cavity. It is noted that the nozzle used in this paper should be the simple shape (ref. l).

The experiments are carried out by using a nozzle of 1.0 mm diameter and at a discharge pressure in the range of 4.9 - 29.4 MPa. Tests were made to investigate the influence of the divergent angle, length of the horn, standoff distance, and a comparison with the conventional hornless nozzle through the erosion testing of brick. As a result of this test, it was made clear which shape of horn shaped nozzle and standoff distance would give the highest eroded volume. Figure 11 shows that the eroded volume of the horn shaped nozzle is approximately 14 times that of the volume using a conventional hornless nozzle. Consequently, it is found that in spite of its simple shape, the horn shaped nozzle generates a great deal of cavitation. Furthermore, this horn shaped nozzle causes a greater volume of erosion in the water than that caused by the conventional hornless nozzle in air and the effectiveness of the submerged cleaning is made clear.

Hereunder, the report is presented on the experiments performed in order to fully know the characteristics of the horn shaped nozzle.

EXPERIMENTAL APPARATUS AND PROCEDURE

The horn shaped nozzle used in this experiment is shown in Fig. 1. As the shape of the nozzle, the length of its throat l varies over the range of 3 - 8 mm, the length of the tapered portion is O - 20 mm, and the angle of taper O is 20^o – 90^o and the nozzle diameter is kept at 1.0 mm. In this paper, the particulars of the nozzle are coded in the form of L - θ - D - l. For instance, in the case of the nozzle with the length of the tapered portion 8 mm, the angle of taper 30^o , and the length of throat 8 mm, the particulars are represented as model 8-30-1-8. And, in the case of the conventional type of nozzle, L=O and θ = 90^o are to be used. As the specimen that is subject to eroding action, bricks with dimensions of 60x207x97 (Height) mm are used in this experiment. The compressive strength of the brick is in the range of 16.7 - 19.6 MPa in dry conditions. This value does not change even if it is kept in the water for three hours. As for the pump, a Sugino Jet Cleaner, model JCM-30060, which is capable of discharging up to the maximum of 60 l/min and pressurizing up to 29.4 MPa is used.

For the purpose of the submerged tests, the water tank shown in Fig. 2 is used. In order to keep the distance between the water surface and the surface of brick constant, a drain valve is fitted to the tank. The inside diameter of the feeder pipe to the nozzle is 15.6 mm and the standoff distance can be adjusted by moving the nozzle vertically. Th e conditions of the experiment are illustrated in Fig. 3.

In the preliminary tests, no change in the quantity of erosion was observed when the water depth was varied from 100 to 400 mm. When a water jet is discharged in water, fine air bubbles are generated in the water and sometimes they may restrain the erosion of brick at the next discharge of the jet. Therefore, the change of the eroded volume of brick due to the length of interval between one jet discharge and the next is investigated. It is found from the result that no change can be observed if the interval time becomes 20 seconds and over. Therefore, subsequent tests are made with the water depth H of 300 mm and over 20 seconds time interval between each jet discharge. At this time, the water temperature in the tank was 12^o – 14^o C.

For the calculation of the eroded volume, first iron powder of mesh size 80 with a bulk specific gravity 2.92 is filled into the cavity made by erosion in the brick. Secondly, it is taken out to be weighed. The eroded volume is then converted from the measured weight. The erosion test is performed three times under the same conditions and the mean value of the results is taken up as the eroded volume.

RESULTS OF EXPERIMENTS

Angle of Taper and Standoff Distance

In order to find the best shape of the horn shaped nozzle which displays the maximum eroded volume in water, the eroded volume of the brick is investigated at a discharge pressure of 14.7 MPa taking the angle of taper θ, standoff distance S and the length of tapered portion L as the parameters. The results are shown in Fig. 4,5 and 6. From these results, it is known that the 30^{o} angle of taper achieves the maximum eroded volume with every length of tapered portion and gets smaller in the sequence of 20^{o} , 40^{o} and 60^{o} . It is also made clear that the maximum eroded volume can be gained at the standoff distance of 40 - 60 times of the nozzle diameter regardless of the nozzle shape.

Length of Taper Portion

The effect of the length of tapered portion L is shown in Fig. 7. The maximum eroded volume can be obtained at the length of tapered portion of 8 - 12 times of the nozzle diameter and the trend is common for every standoff distance.

Length of Throat

Since it is made clear from the aforementioned results that the shape of horn shaped nozzle which gains the maximum eroded volume has the angle of taper of 30^{o} , and the length of tapered portion of 8 - 12 mm, the influence of the nozzle throat length is investigated as the next step. As shown in Fig. 8, the eroded volume is measured for three different lengths, which are 3,5 and 8 mm, with the length of tapered portion 12 mm and the angle of taper 30^{o} . The influence of the length of throat to the eroded volume is small, (around 20 mm standoff distance) but in the range of standoff distance between 40 and 80 mm, the longer the throat length becomes, the larger the eroded volume. By the way, the standoff distance at which the maximum eroded volume appears is within the range of 40 - 60 mm for every throat length.

Pressure of Discharge

The eroded volume when the pressure of discharge is changed over the range of 4.9 - 29.4 MPa is shown in Fig. 9. The eroded volume rises sharply until the pressure reaches around 10 MPa for every length of tapered portion and its increasing rate tends to decline when the pressure exceeds around 20 MPa. Besides, when the pressure approaches 20 MPa, the kind of nozzle which displays the maximum eroded volume is coded 8-30-1-8 instead of 12-30-1-8 at the pressure of 14.7 MPa. Therefore, the length of taper gaining the maximum eroded volume is changed due to the discharge pressure.

Time of Exposure

The change of the eroded volume due to the different length of exposure time is shown in Fig. 10. When the exposure time exceeds somewhere between 120 and 180 seconds, the increasing rate of the eroded volume tends to decline for every angle of taper. The declining rate of increase is the smallest for the angle taper of 30^{o} . Little increase of the eroded volume for the nozzles with the angle of taper of 20^{o} and 40^{o} can be observed after the exposure time exceeds around 180 seconds.

Horned Shaped Nozzle and Conventional Hornless Nozzle

In the next experiment, a comparison is made between the horn shaped nozzle, coded 12-30-1-8, and the conventional hornless nozzle, coded 0-90-1-8, with regard to the eroded volume at a discharge pressure of 14.7 MPa in both air and water.

From Fig. 11, it is clear that the horn shaped nozzle can generate an eroded volume approximately 14 times as large as the conventional hornless nozzle can, and it has an excellent eroding power in water. In air, the nozzle coded 12-30-1-8, creates nearly as much eroded volume as that of 0-90-1-8. Any of these nozzles gains a larger eroded volume in water than in air. The standoff distance at which the conventional type of nozzle exhibits the maximum eroded volume is around 10 times the nozzle diameter which is far smaller than that of the horn shaped nozzle coded 12-30-1-8 which shows the maximum erosion at approximately 60 times the diameter.

Erosion Area

The eroded area on the surface of brick in the previous experiment is shown in Fig. 12. The area is defined by the product of the lengths of the major and minor axes.

While the horn shaped nozzle obtains a larger area than that of the conventional hornless nozzle in water, the difference between them is not as large as the 14 times that is seen in terms of the eroded volume. From these facts, it is known that the horn shaped nozzle erodes brick deeper than the conventional hornless nozzle. It is also found that the standoff distance at which the maximum erosion area is gained is around 80 mm by the horn shaped nozzle and approximately 40 mm by the conventional hornless nozzle and each is larger than the respective standoff distance of 40 - 60 mm and 10 mm at which the maximum eroded volume appears

On the other hand, if the results of the experiments are compared between those in air and in water, the erosion area in water is approximately 5 times as large compared with that in air which means erosion in water occurs over a wider area.

DISCUSSIONS

Angle of Taper and Standoff Distance

The smaller the angle of taper becomes, the faster the inflow velocity of the entrained water gets. Thus as the shearing action between the water jet from the nozzle and the entrained water increases, the cavity is generated more effectively. Accordingly in Fig. 4, 5 and 6, the maximum erosion should be given at the angle taper of 20°, but in fact it appears at 30°.

Two reasons are considered for this phenomenon. One is the frictional force working between the horn wall and the entrained water. The frictional force on the horn wall gets larger as the inflow velocity of the entrained water increases and causes energy loss of the entrained water or energy loss of the jet. The loss of energy lowers the shearing action and leads to a decrease in cavity generation.

The other is the diffusion of the jet ejected from the nozzle. Since the half spreading angle of a submerged jet flow is approximately 12° (ref. 5), it is supposed that due to diffusion of the jet, the surrounding water is prevented from flowing into the tapered portion, thus generation of cavity drops.

The standoff distance at which the maximum eroded volume S_o is gained is largely influenced by the cavitation number σ. According to paper (ref. 6), there is the relation of $So/D \quad \sigma^{-m}$ and as σ becomes larger, So/D gets smaller.

The value of σ in this experiment at P=14.7 MPa is approximately 0.007 which would give S_o/D = 40 - 60. In order to get a firm knowledge on the relationship between S_o/D and σ, further experiments should be done by changing the pressure of the jet and the nozzle diameter.

By the way, the value of S_o/D of the horn shaped nozzle is 4 - 6 times as big as compared with that of the conventional hornless nozzle. As nozzles with high values of S_o have the advantage that they can wash the objects to be cleaned from afar without coming close to them, the horn shaped nozzle is more effective than the conventional one.

Length of Tapered Portion

In Fig. 7, the reason that the maximum erosion appears at L/D = 8 - 12 is presumed because the length of the potential core X_c is 8 - 12 times the nozzle diameter. In the potential core region, the velocity gradient along the radial direction of the mean velocity of jet is larger than that in the main region of the jet (ref. 7). Therefore, in the core region, the shearing stress of both laminar and turbulent flows becomes larger than that in the main region. If X_c is almost equivalent to the length of tapered portion L, the shearing action between the water flowing into the tapered portion and the jet, and the shearing action within the jet are mutually multiplied resulting in more effective generation of the cavity. Thereby, intensive erosion would be gained.

Time of Exposure

One of the causes for the fact that the rates of increase of the eroded volume drops is supposed to be that the hole depth made by erosion becomes larger causing an increase in the standoff distance, thus the eroding power is reduced. In some cases, a part of the cavity generated may restrain cavitation. This is because, in the course of formation of cavitation, while some of the cavities generated collapse to form cavitation, the rest of them remain in their original state.

In Fig. 11, it is indicated that the horn shaped nozzle generates far more effective cavitation than that of the conventional type. Since larger volumes and the areas of erosion are gained in water than in air not only by the horn shaped nozzle but also by the conventional hornless nozzle, surface cleaning using eroding action, for example, removal of shellfish on the ship hull, de-scaling the inside surface of pipes, casting sand removal etc., works more effectively in water.

If cleaning is done in water, the noise level of the jet is reduced. Besides, it has the merit that the removed objects simply drop without scattering and are easily collected.

Therefore, if cleaning at present done in air could be converted to in water, the cleaning effectiveness would possibly rise in some cases. Because the horn shaped nozzle generates extensive cavitation in water, in spite of its simple structure, it will be widely used in the cleaning operation in the water which is expected to become popular in the future.

5. CONCLUSIONS

For the purpose of improving the cleaning power of a water nozzle, the authors' attention has been directed to the method of cleaning that applies the erosion action due to cavitation in the water.

The horn shaped nozzle which was invented by Professor K. Yanaida, one of the authors, is the nozzle that has a horn of divergent shape at the outlet part of the nozzle, positively generates shearing action between the jet and the surrounding water, and effectively causes cavitation (ref. 1). Professor K. Yanaida has made a theoretical analysis of this nozzle in detail and is going to publish it in the near future.

In this paper, the results of experiments made to investigate the performances of the horn shaped nozzle are reported. The principal results with the nozzle diameter of 1.0 mm and at the pressure of 14.7 MPa are as follows:

- With this horn shaped nozzle, the maximum eroded volume can be obtained at the angle taper of 30^{o} and the length of tapered portion 8 - 12 mm.
- In spite of its simple structure, the horn shaped nozzle has a capability approximately 14 times that of the eroded volume of the conventional hornless nozzle.
- The standoff distance at which the maximum eroded volume is recorded is in the range of 40 - 60 mm for the horn shaped nozzle, which is 4 - 6 times that of the conventional hornless nozzle's distance.
- Both horn shaped and conventional hornless nozzles show larger eroded volumes in water than in air.

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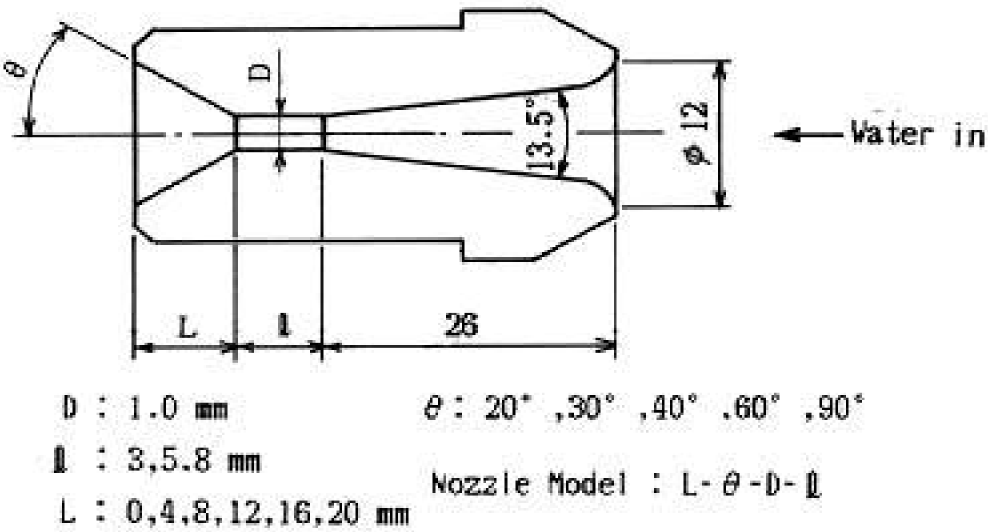

Fig. 1 Horn shape nozzle

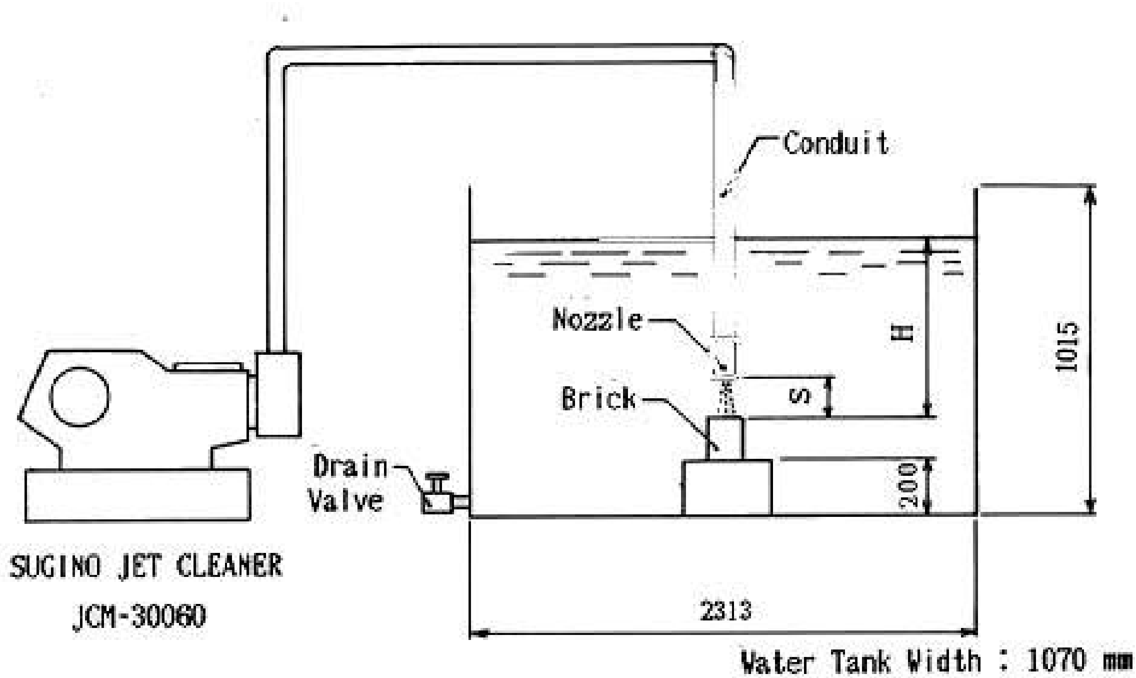

Fig. 2 Schematic of test apparatus

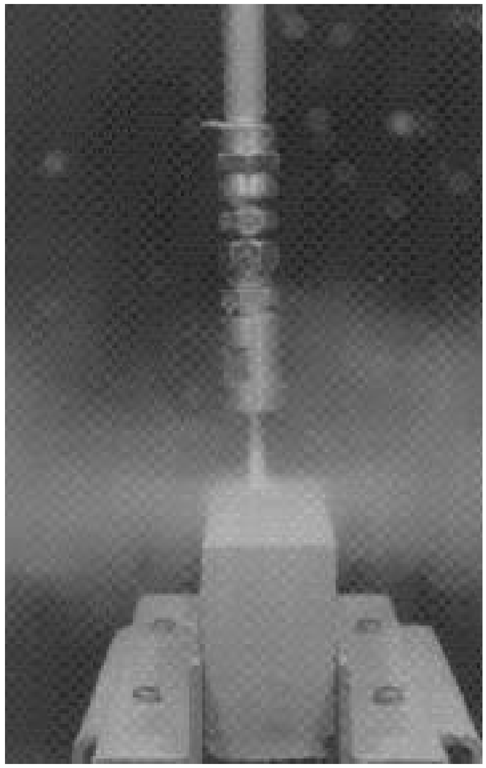

Fig. 3 Erosion of brick in the water

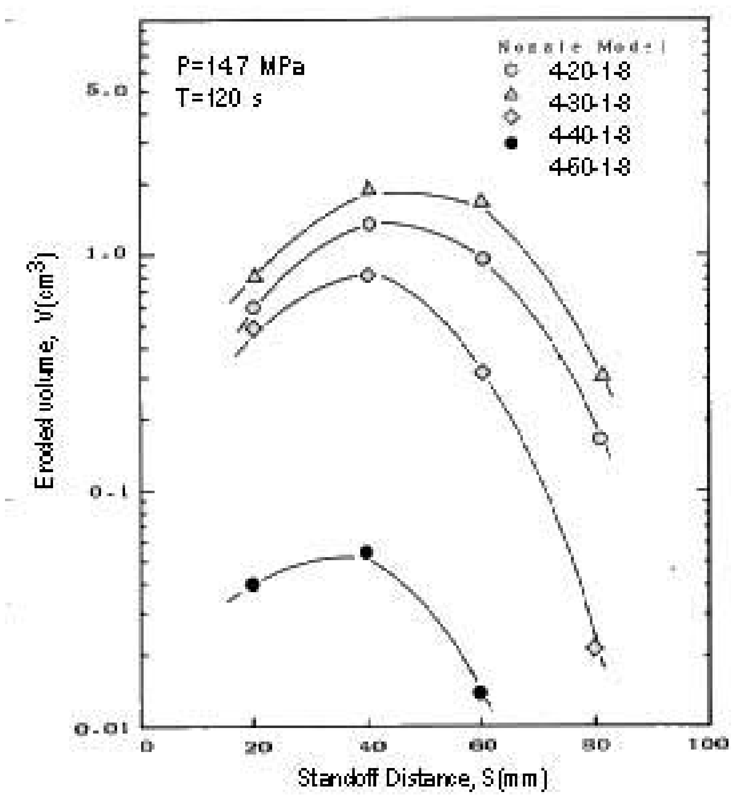

Fig. 4 Effect of standoff distance

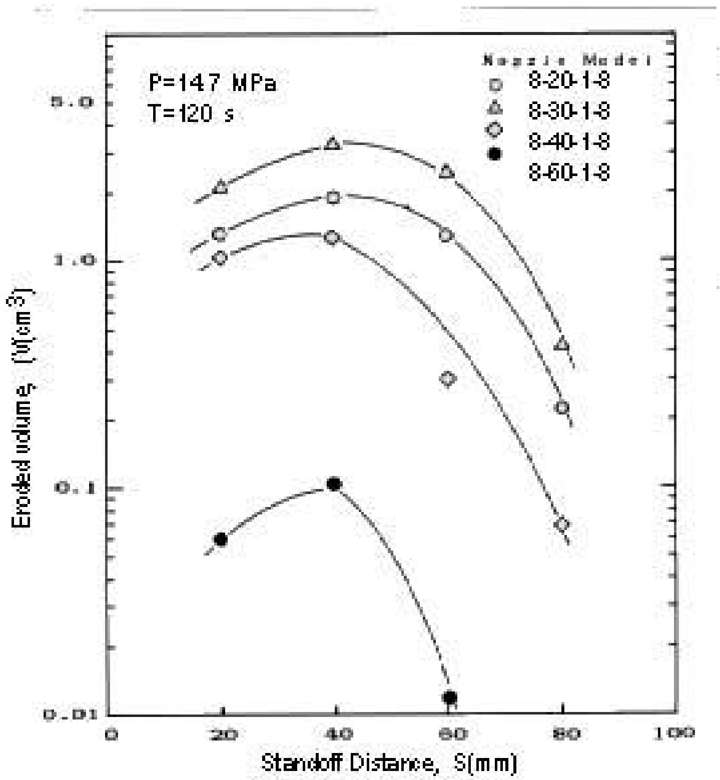

Fig. 5 Effect of standoff distance

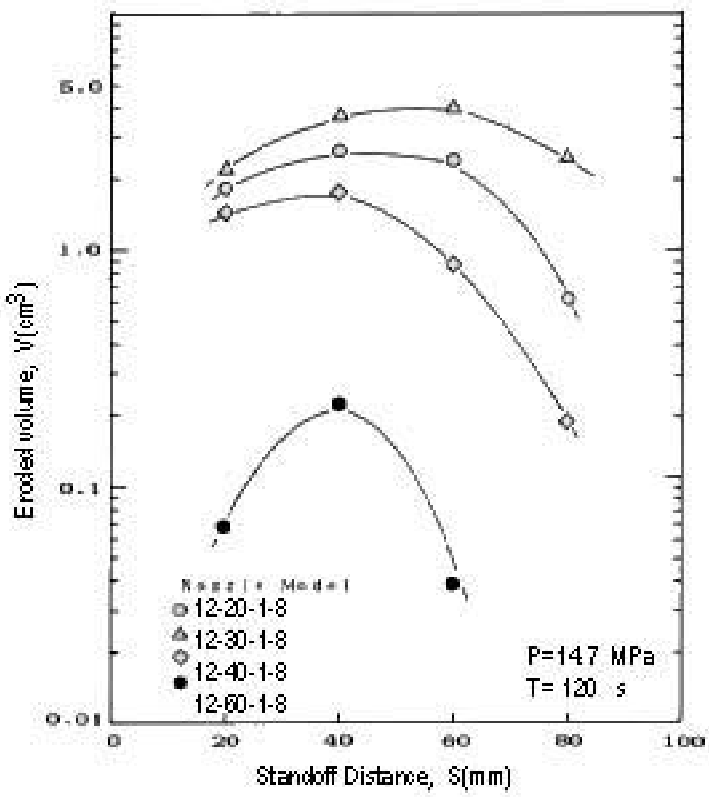

Fig. 6 Effect of standoff distance

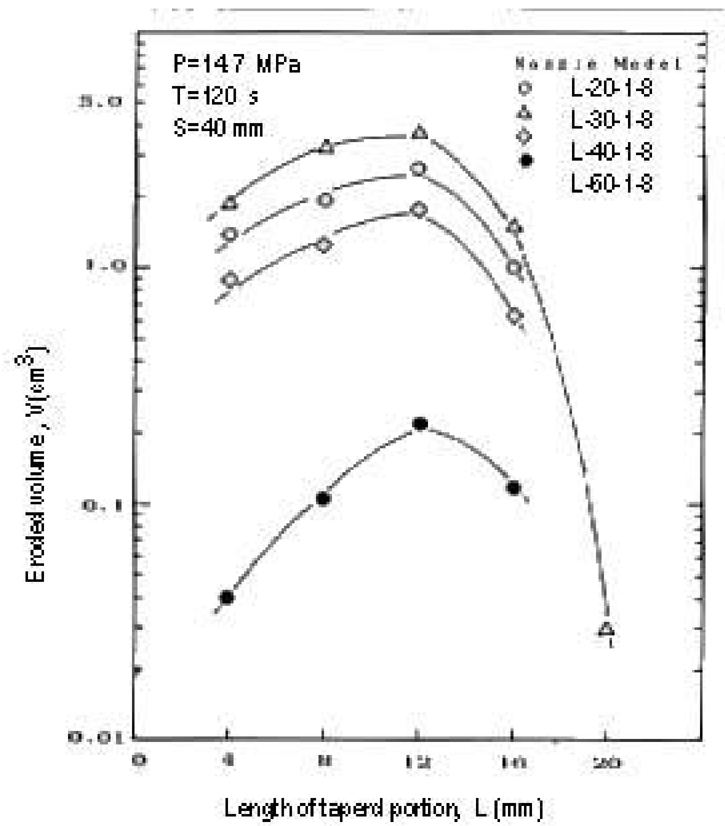

Fig. 7 Effect of tapered portion

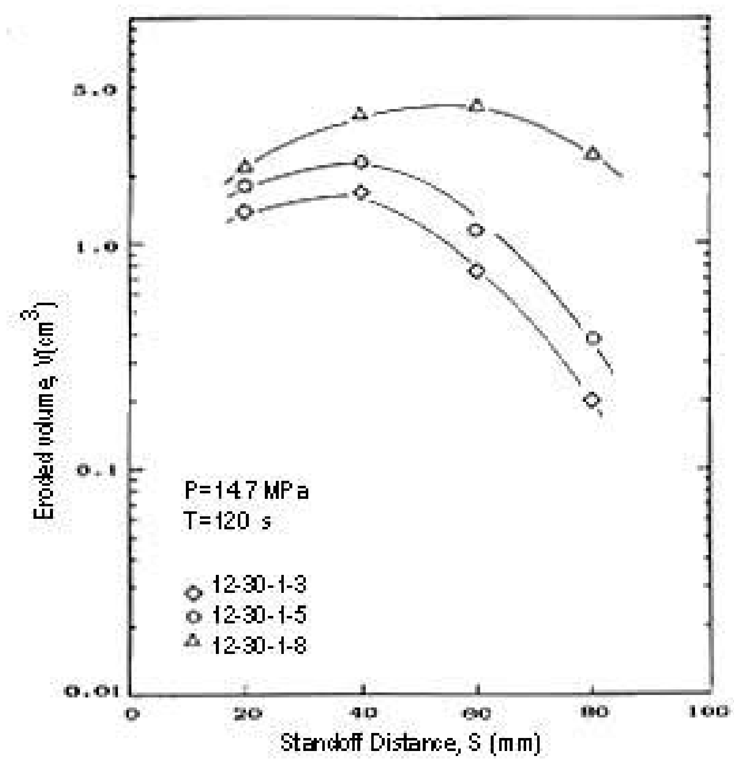

Fig. 8 Effect of nozzle throat

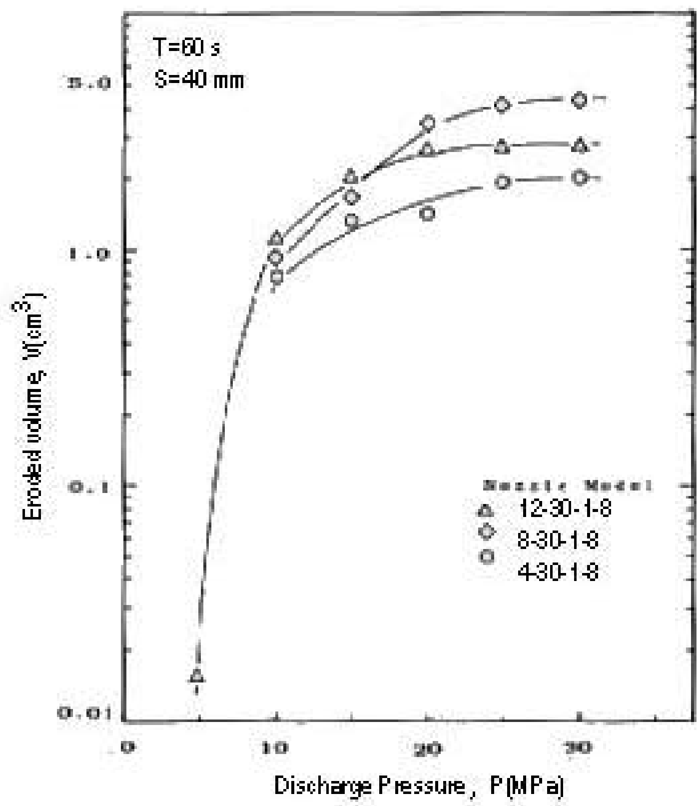

Fig. 9 Effect of discharge pressure

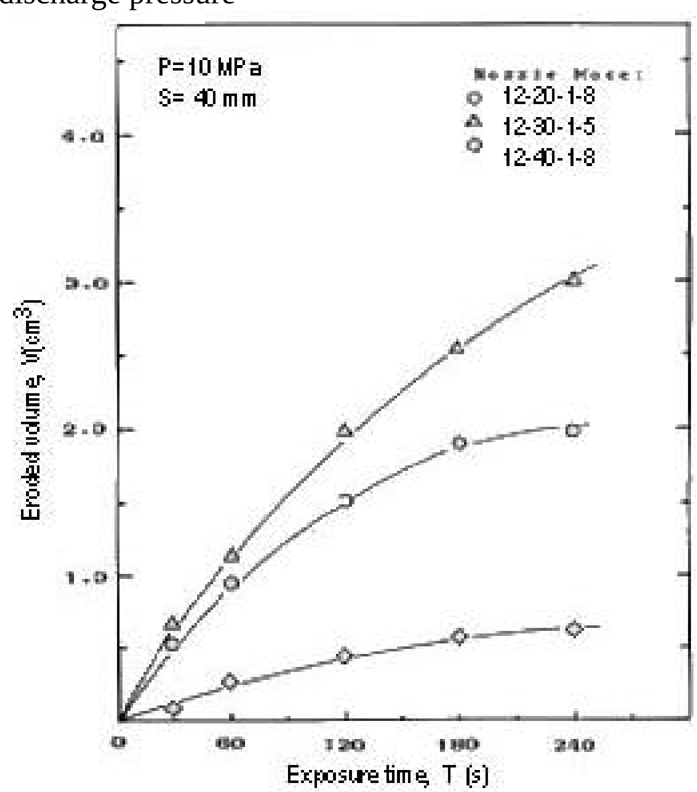

Fig. 10 Effect of exposure time

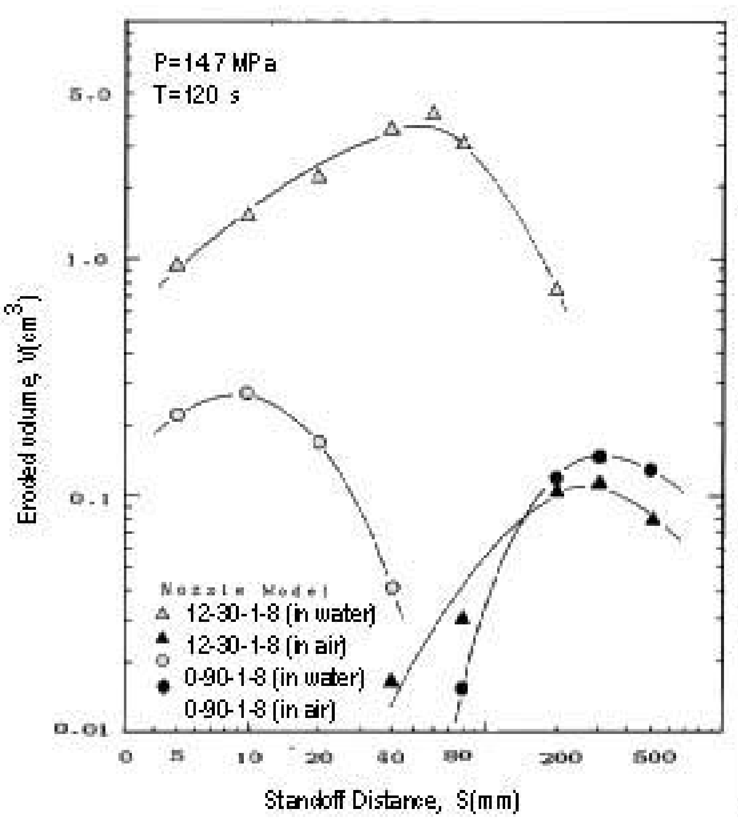

Fig. 11 Comparison of eroded volume between in water and in air

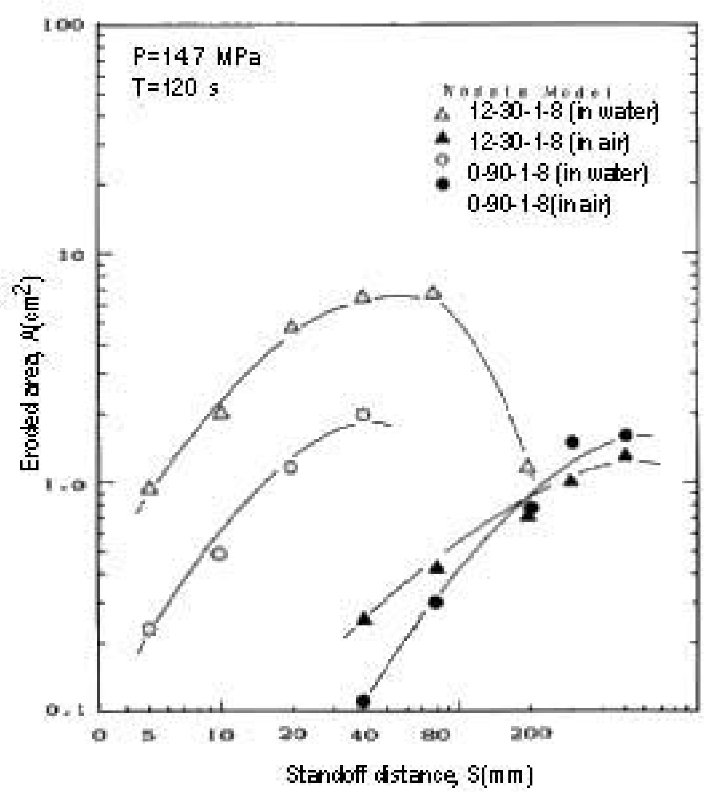

Fig. 12 Eroded Area

WATERJET ASSISTED DRILLING FOR TIEBACK ANCHORS

by

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ABSTRACT

The drilling of holes for installing grouted tieback anchors has been accomplished using waterjet assisted rotary drills. These new tools offer significant advantages over conventional equipment, in terms of easier handling, faster drilling, low noise, and belling capability. This paper will describe two projects and the different equipment used on each.

INTRODUCTION

The development of waterjet assisted tools has bean very active in recent years. The basic hardware pumps, hose, valves, and fittings, have become more reliable and readily available. The successful applications in waterblast cleaning and industrial cutting have laid an excellent foundation for expanding this new technology. The equipment has established a good track record for productivity and safety. Based on these successes new applications are continually being examined.

StoneAge, Inc is active in developing tools that utilize high velocity waterjets. Typically these tools operate in the range of 8,000 to 12,000 psi (500 to 800 Bar) and up to 200 hp (150 kw) We are continuing to find beneficial applications in the mining, oil & gas, industrial cleaning, and civil engineering industries.

A recent development is a waterjet-assisted drill for installing tieback anchors Tiebacks are used to tie a structure to an anchor in stable ground. Their use is somewhat new but is expanding in acceptance for several civil engineering applications. Tiebacks consist of three basic elements; anchor, tendon, and connection to the structure. A common use of tiebacks is to secure a retaining wall in excavation projects. Typically the procedure is to drill a hole 3 to 12 inches (8 to 30 cm) in diameter, at least 15 ft (3 m) deep, place a steel rod in the hole and then fill completely with cement grout.

This paper will describe two recent tieback projects that waterjet equipment was used on. The first was a small excavation project that required tiebacks to hold soldier beams in place for a permanent supporting wall. The second project used tiebacks to stabilize a failing earth slope.

StoneAge, Inc. and Soil Engineering Construction, Inc. cooperated as partners in developing the equipment described in this paper. In addition Soil Engineering conducted on-site experimentation, contracted jobs and used the equipment on a production basis. StoneAge designed and fabricated all of the equipment.

EXCAVATION PROJECT

This job was to stabilize the sides of an excavation 20 ft (6 m) deep for a recreational structure. The Mall job size and remote location made it very expensive to bring in a large drill rig. A handheld waterjet assisted drill was tried for the hole drilling and proved capable of handling the task.

The soil composition consisted of decomposed granite, gravel and aluvium. Below a depth of approximately 10 ft (3 m) the composition increased in cohesiveness and strength. The site was first excavated to a shallow depth of 3-4 ft (1 m). Then a series of 26 augured holes, 24 ft (7.3 m) deep were made around the perimeter. In these holes were placed 14 inch (35 cm) wide-flange I-Beams to serve as soldier beams for the wooden lagged retaining wall. Holes were torched in the beams at the proper location for the tiebacks to pass through.

The tieback holes were to be approximately 2 inches (5 cm) in diameter to a depth of 20 ft (6 m) at a 2 to 1 slope down into the strata. Attempts at using a wash-boring/auger drill were unsuccessful. The drill had a 5 hp rotation motor; weighed 400 lbs (I80 kg) and was hand fed. The drilling rate was slow and the drill became stuck on the first hole. Apparently gravel fell in behind the bit and could not be washed out. Setup time for the drill was approximately 1 hr per hole and production was expected to be 3 to 4 holes drilled and tiebacks installed per day, based on previous experience.

<u>Equipment</u>

A handheld waterjet assisted rotary drill system was assembled to drill the tieback holes. (Figure 1) The system consisted of the following major equipment.

1. SA 1750 Drill: air powered, 300 rpm, self dumping high pressure swivel
2. Hydroblaster Pump (2 ea): diesel driven, 4.5 gpm (I7 L/min.), 10,000 psi (700 Bar)
3. Rotary bit: carbide spade cutter, 3 sapphire nozzles, 2 in (5 cm) O.D.
4. Drill rod: 3/8 ID, 3/4 OD, X 10 ft (1 cm x 9 cm x 3 cm); NPT connections

SLOPE STABILITY PROJECT

The waterjet assisted drill was ideally suited to this project. The rock/soil type, location, and access constraints were all favorable to this new technology

The project consisted of stabilizing a 60-70 ft (20 m) high slope, 100 ft (30 m) in length. Private residences were at the top of the slope and a river was at the bottom. The entire slope was a massive sandstone of approximately 2000-3000 psi (170 Bar) compressive strength. Drainage off of the residences had caused significant erosion and failure slips of the slope. Development at the top and the river at the bottom prevented any large drilling equipment being brought in.

The project design called for tiebacks to be installed to a depth of 40 ft (12 m) at a 15 to 30 degree dip from horizontal These would be placed on 10 ft (3 m) centers and provide the support for a lattice of steel rebar wired together. A layer of gunite would then be applied to obtain a 6 to 8 inch (15 cm to 20 cm) thick covering over the entire slope.

<u>Equipment</u>

A lightweight waterjet assisted drill rig was designed and fabricated specifically for this job. It employed a recently developed rotary drive and swivel, and made use of those lessons learned on the handheld tieback drilling job. (e.g. large OD, quick coupling rods.) The drilling system consisted of the following major equipment.

1. Drill Rigs: steel frame, 12 ft (3.7 m) overall, 150 lbs (70 kg), hand crank feed, air motor through gearbox rotation, 350 rpm, high pressure swivel.
2. Duplex drill rod: 3/4 ID, 2 in OD X 10 ft (1.9 cm x 5 cm x 3 m), taper thread connection.
3. Drill Bit: 2 3/4 in OD (7 cm) w/ carbide cutter, 2 carbide nozzles.
4. Pump: diesel powered, triplex 10 gpm (38 L), 10,000 psi (700 Bar).
5. Belling Nozzles 2 3/4 OD (7 cm), 2 nozzles 180 degrees apart and 90 degrees to hole axis.

The pump and compressor were stationed at the top of the slope. One man stayed with that equipment while two men operated the drill rig.

<u>Procedure</u>

The drill was used in two configurations; stand alone and sled mounted. The bottom row of holes were drilled first with the rig set up on a dirt berm with the rear end supported by two cables from bolts on the slope (Figures 2 & 3). The footing was awkward but adequate for operating the rig and making drill rod changes. A heavier or larger drill would have been very difficult to handle. After drilling the holes the belling nozzle was put on the drill rod and ran to the bottom of the hole. By rotating the belling nozzle and moving forward and back in the hole a chamber was created. This chamber was roughly 2 ft (6 m) in diameter (based on tests close to the surface) and provided ample volume for anchoring. The holes were then cleared of water by a few blasts of air from a blow pipe. Two tiebacks rods 20 ft (6 m) in length were coupled together and placed in the holes. Grout was prepared on the top of the slope and pumped down through hose and tubing into the tieback holes.

The drilling of the holes on the slope face required that the drill rig be mounted on a sled with work platform (Figure 4). It was too difficult for the operators to be supported only by ropes and still handle the drilling and rod changing functions. The sled was supported by cables tied to anchors at the top and raised and lowered by an air winch. Moving and orienting the drill was greatly simplified. The sled also provided storage for storing the extra drill rod. Tieback installation and grouting the holes followed the same procedure as before. Use of the sled eliminated any need for a scaffold. That resulted in a great economic savings and improved the access for other operations on the slope.

Performance

The drill rig surpassed most all expectations. With scaffolding and a 700 lb (320 kg) rotary drill, the estimate was to achieve only 2 holes per day on the average. It was hoped that the waterjet drill could double that. Once the bugs were worked out of the equipment and operating procedure, 8 to 10 holes per day were completed. Some problems were encountered with sand and debris getting into the hoses and drill rods. This caused some bit plugging, and fouling of the swivel. No damage occurred and thorough flushing of the system corrected the situation. Increased operator awareness and care in handling the hose and drill rod eliminated this problem. The ease of setup, good hole flushing characteristics, and belling capability were significant benefits. In addition, this project demonstrated some very good drilling rates. Even though the drill thrust was hand powered, 10 ft (3 m) of hole was drilled in about 30 seconds on the average. No holes collapsed and the drill was never stuck.

RELATED WORK

To date the waterjet assisted drills have been used successfully on four tieback installation projects in California. In addition to rotary spade bits, waterjet assisted roller cone bits have also been used very effectively. (Figure 5)

This equipment has found usefulness in several other civil engineering applications. Vertical holes can be effectively produced and used for driving pipe piles, installing caissons, and providing grout injection points. The waterjet assisted drills are also very well suited for creating horizontal drain holes.

As experience is gained and the equipment continually improved it is expected that many more applications will be found.

SUMMARY

Waterjet assisted drills have been job proven and show that in the right situation they offer significant advantages over conventional equipment. Hole drilling for tiebacks appears to be a well matched application. Tiebacks are often used in rock/soil compositions that are amenable to waterjet assisted drilling.

Their advantage is most significant on projects that cannot make use of large conventional drills. Light-weight, belling capability, fast drilling, low noise, and good hole flushing can benefit many job situations. The equipment is safe, durable, and easily maintained. The necessary high pressure pump and hose equipment are available for rental in all major cities. For smaller projects or those with equipment access problems this new technology offers attractive alternatives.

Figure 1 Handheld waterjet assisted drilling of tieback anchors.

Figure 2   Waterjet assisted drill rig, hand fed.

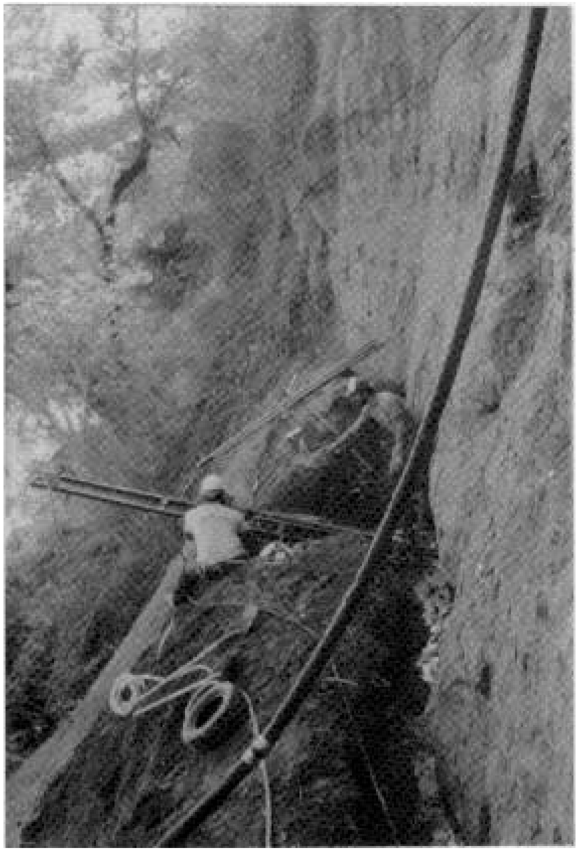
Figure 3 Tieback drilling rig, waterjet. assist.


Figure 4 Sled mounted, drill rig, waterjet assist.

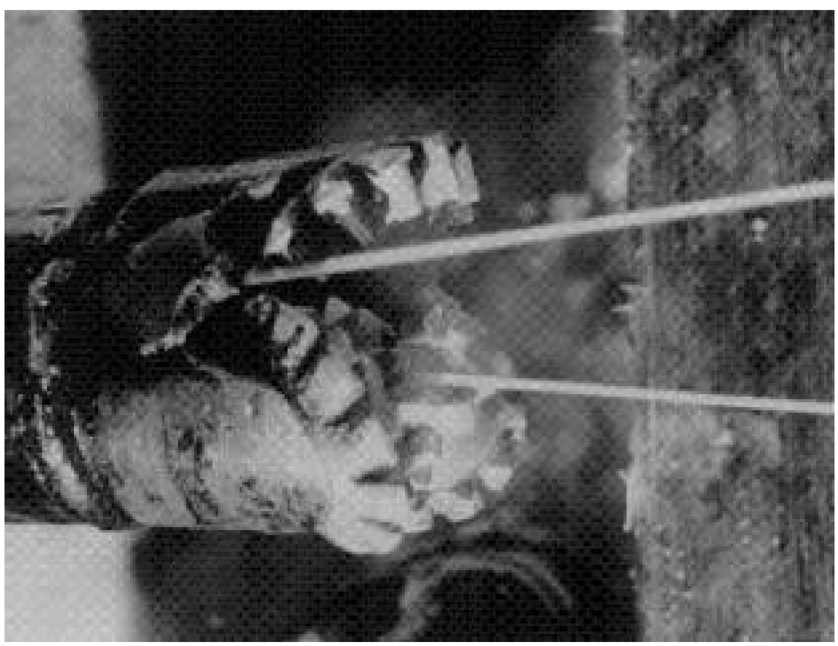

Figure 5 Roller cone bit, waterjet assisted, 3 in (7.6 cm) OD.

AN EXPERIMENTAL INVESTIGATION OF CUTTING AND FRACTURING OF ROCKS WITH SINGLE AND MULTIPLE CONVERGING JETS

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ABSTRACT

Results pertaining to fracturing (breaking) of rocks with high speed single and multiple converging or rotating jets are presented. The technique of converging two continuous jets to a point is termed "Jet Accumulation Technique". The purpose of the investigation was to compare the effectiveness of this technique for fragmenting rocks with that of single or multiple rotating jets. Several nozzle designs were used for converging the jets to a point. Both stationary and traversing tests were conducted on a variety of rocks at pressures up to 69 MPa. No phenomenal breakage or fracture occurred with the converging jets. Single or multiple rotating jets appeared to perform better than the converging jets for fracturing rocks. An attempt is made to point out the drawbacks of the "Jet Accumulation Technique" for fragmentation of rocks.

NOMENCLATURE

d	Orifice diameter,	mm
P	Nozzle pressure,	MPa
S	Standoff distance,	cm
V_o	Velocity of converging jets,	m/s [Fig. 4]
V_{tr}	Traverse speed of jet over rock specimen,	cm/s
V_{fj}	Velocity of the fast secondary jet,	m/s [Fig. 4]
V_{sj}	Velocity of the slow secondary jet,	m/s
α	Angle of convergence of jets,	deg. [Fig. 4]
γ	Angle of inclination of the jet front to the primary jet axis,	deg. [Fig. 4]
τ	Duration of exposure of the rock sample to the jet,	sec.
ρ	Density of the liquid,	kg/m^3
β	Angle of collapse of the conical liner,	deg. [Fig. 5]

INTRODUCTION

There is a great deal of interest in fragmentation of rocks in such areas as tunneling, quarrying and mining. Although high speed water jets have been investigated for these applications, most of the work reported in the literature has dealt only with the pulsed jets (Refs. 1 and 2). Pulsed jets are capable of producing stagnation pressures as high as 4,000 MPa and impressive results have been obtained in breaking (shattering) of rocks with them. However, there are a number of problems in the systems producing pulsed jets and these need to be overcome before the pulsed jets could be fully exploited for commercial applications. Relatively little information exists, on the other hand, on the use of continuous (cavitating or noncavitating) water jets (Ref. 3). This is probably due to the fact that if low pressures (~ 70 MPa) were used, the time required for fragmentation were rather high (>1 min) and attempts to reduce this time required unacceptably high pressures (>400 MPa). It is in this regard that a phenomenon, called "Jet Accumulation Technique", described by Mazurkiewicz, et. al. (Refs. 4 and 5) appeared very interesting. Simply stated, the technique consisted of converging two identical jets to a point (see Fig. 1). Based on the well known concept of jet formation (Munroe or Neumann jets) by charges with lined conical cavities (Refs. 6,7 and 8), the authors argued that if the jets were made to collide (converge) at a point within the interior of a block of rock, the entire block would then fracture or split open within a few seconds. The effectiveness of the technique was demonstrated by splitting an unconfined block of Berea sandstone (15 cm in diameter and 30 cm in length) in 10 to 15 seconds with converging jets at a pressure of 69 MPa (Ref. 5).

The investigation presented in this paper was undertaken to thoroughly test the technique for fragmentation of a variety of rocks. The technique was not satisfactory and no phenomenal breakage or fracture occurred in all the tests conducted in the laboratory. Often a single stationary jet or multiple rotating jets were found to be superior to that of converging jets for fragmentation of rocks. An attempt has been made in the appendix to explain why the "Jet Accumulation Technique" would not work for enhanced rock cutting or fragmentation.

EQUIPMENT AND PROCEDURE

The tests were conducted with the Union quintuplex pump capable of delivering 49.2 liter/min of water at 69 MPa (Power = 74.6 kw). Nozzles for converging the jets were designed in such a way that the angle of convergence for the twin jets could be varied from 10^o to 60^o (Fig. 1). Some tests were also conducted with triple converging jets as shown in Fig. 1. Nozzle body configurations for producing rotating jets were the same as those described in Ref. 10. The test depicted in Fig. 3 was conducted with the new Indescor pump which is capable of delivering 76 liter/min at 103.5 MPa. The rock samples (up to 30 cm thick) employed were: Indiana limestone, marble, conglomerate, Quebec grey and pink granite, Berea sandstone and Barre granite. While most of the samples were confined in a matrix of concrete, some of them were unconfined.

The procedure was quite simple. When all the operating conditions were steady, the samples were exposed to the stationary converging jets up to a maximum duration of 1 min. If the rock samples did not break during this period of time, then they were traversed under the jets at 2 cm/sec, noting the mode of fracture, etc. As the comparison of results was only qualitative, no attempt was made to measure cutting rates or specific energies in the investigation.

EXPERIMENTAL RESULTS AND DISCUSSION

Experimental results relevant to the discussion in this paper are summarized in Table 1. The rock samples exposed to the converging jets are shown in Fig. 2 (A to J). For comparison, the fracture of Barre granite with a single stationary jet and a rotating jet system are depicted in Figs. 2K and 3 respectively.

An examination of the results in Table 1 and the corresponding photographs in Fig. 2 clearly indicate that no fragmentation of the rock specimens occurred with the converging jets for various test conditions investigated. For instance, one of the important test parameters is the standoff distance. If the fracture in the rock is initiated at the point of convergence of the jets, then fragmentation of the rock should occur at very short standoff distances. No such fragmentation was observed in the experimentation. By similar arguments, the smaller the angle of convergence, the greater should be its effect on fragmentation. Although the angle of convergence was varied from 10^o to 60^o, no fragmentation (of the entire block) was evident for any of the rock samples tested. The fragmentation of Berea sandstone reported by Mazurkiewicz, et. al. (Ref. 5) with 1^o and 2^o converging jets was probably due to a local defect in the sample (as discussed in the appendix,.

$V_{fj} / V_o = (\sin\alpha / 2 + \sin\gamma) / \sin(\alpha/2 + \gamma)$. For $\alpha = 2^o$ and assuming $\gamma = 90^o$,

$V_{fj} / V_o = 1.02$, the augmentation of V_o is indeed quite small).

Traversing the rock samples under the converging jets so as to distribute the point of convergence over a length within the rock sample did not show any promise. The results clearly indicated that converging jets were not effective for enhancing the cutting rates or fragmentation of rock samples. They had no particular advantage over single jets for cutting or fragmentation. This is obvious from Fig. 2K which shows a confined block of Barre granite fragmented with a single continuous stationary jet at 55 MPa (d = 1.78 mm). A line of fracture developed in 30 seconds and the sample completely split in 60 seconds. Figure 3 shows another example of fracturing the rock with rotating jets (two diverging jets). Of course such fragmentation depends entirely on the structure of the rock. It is believed that fractures shown in Figs. 2K and 3 were due to the presence of planes of weakness within the rock sample. It appears, however, that it is feasible to fracture rocks in general with rotating water jets. This can be achieved by drilling a hole partially through the rock, sealing and pressurizing it with high pressure water. This technique deserves further study.

CONCLUSIONS

The main conclusion from the experimental results was that the converging jets were not effective for enhancing the cutting rates or for the fragmentation of rocks. If it were at all possible to fragment rocks with continuous water jets (a lot depends on the nature of rock), then single stationary or rotating jets should be the obvious choice.

ACKNOWLEDGMENTS

The author is grateful to Mr. W.H. Brierley for designing the nozzles and for conducting some of the tests in the laboratory. Thanks are due to Mrs. S. Farrell for typing this paper.

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APPENDIX

The model of formation of cumulative jets as hypothesized by Mazurkiewicz, et al. (Refs. 4 & 5) is shown in Fig. 4. They assumed that the mechanism of formation of secondary jets (cumulative jets) is similar to the well known phenomenon of the formation of ultra high speed (2000 to 10,000m/sec) metallic jets produced by the exploding charges behind conical (or wedge shaped) metal liners (Fig. 5). They then extended the principles behind the formation of metallic jets to their own concept and showed that when two identical water jets collide (Fig. 4), secondary jets are formed. These jets move in opposite directions along the line bisecting the angle between the original jets. By applying the conservation principles to the problem (the jets were assumed to be squares in cross-section), they arrive at the following equations for the velocities of the secondary jets [the same equations would not apply for circular cylindrical jets.

$$V_{fj} / V_o = (\sin\alpha / 2 + \sin\gamma) / \sin(\alpha / 2 + \gamma) \qquad (1)$$

and

$$V_{sj} / V_o = (\sin\gamma - \sin\alpha / 2) / \sin(\alpha / 2 + \gamma) \qquad (2)$$

The fact that the velocities of the secondary jets depend only on the angles of convergence (α) and the jet front (γ) casts some doubt on the validity of the derivations. For instance, for $\gamma = 90^o$ (a reasonable value), V_{fj} tends to infinity as α tends to 180^o. This is physically impossible. The author of the present paper believes that the principles in the phenomenon of metallic jets (Refs. 6, 7 & 8) do not apply to the present problem. The metallic jets are formed due to the high dynamic pressures behind the liner. As discussed by Birkhoff, et al. (Ref. 6), the charges can produce, under favourable conditions, dynamic pressures, on small areas and small periods of time, higher than a quarter of a million atmospheres. The maximum pressure at the point of collision of two steady water jets, on the other hand, is only $\frac{1}{2}\rho V_O^2$ (for V_o = 1200 m/sec, which is quite high for steady jets, the dynamic pressure is only 6,800 atmospheres!). The formation of shock waves at the point where the jets meet in the interior of any material is highly unlikely.

What really happens when two cylindrical jets impact at a point is described in great detail by Taylor (Ref. 9). As shown in Fig. 9, sheets of water are formed when two jets meet at a point. The shape and the direction of flow of the sheets depend upon the angle of convergence. If this is the case, then the energy available in the jets is simply dispersed over a large area and therefore the possibility of breaking or fracturing the rock is reduced considerably. This observation tends to support the experimental results presented in this paper.

Test No.	P (MPa)	S (cm)	τ (sec)	V_{tr} (cm/sec)	Comments
1	55.2	2.27	-	2.0	α = 20°; twin jets; limestone; simple linear cutting. No fragmentation.
2	"	"	60.0	-	" " "
3	"	7.62	60.0	-	" " "
4	"	"	-	2.0	" " "
6	"	0.64	60.0	-	" " "
7	62.1	0.64	-	2.0	α = 20°; three jets; limestone. No fragmentation.

Table 1. Summary of Experimental Results

Test No.	P (MPa)	S (cm)	τ (sec)	V_{tr} (cm/sec)	Comments
8	"	"	60.0	-	" " "
9	"	"	-	2.0	α = 20°; twin jets (center and outer); no fragmentation
11	55.2	"	-	2.0	α = 60°; twin jets; no fragmentation
12	"	"	60.0	-	" " "
13	"	5.10	-	2.0	" " "
14	"	"	60.0	-	" " "
15	69	"	60.0	-	" " "
16	"	"	-	2.0	" " "
17	"	2.54	-	2.0	" " "
18	"	"	60.0	-	" " "
19	"	"	60.0	-	α = 60°; three jets; no fragmentation
20	"	"	-	2.0	" " "
21	69	2.54	-	2.0	" " " marble
22	69	2.54	60.0	-	α = 60°; three jets; marble; no fragmentation
23	"	"	-	2.0	α = 60°; twin jets; marble; no fragmentation
24	"	"	60.0	-	" " "
25	69	2.54	-	2.0	α = 20°; twin jets; d = 1.17 mm Aggregate. No fragmentation
27	"	"	-	"	" " d = 1.70 mm
28	"	"	30	-	" " "
29	69	2.54	-	2.0	Single jet. d = 1.70 mm. Grey granite. Deep cut.

Table 1 (cont'd)

Test No.	P (MPa)	S (cm)	τ (sec)	V_{tr} (cm/sec)	Comments
30	"	"	30	–	" " "
31	"	"	–	2.0	α = 20°; two jets; d = 1.17 mm No fragmentation
32	"	"	30	–	" " "
33	69	1.27	–	2.0	" " Berea sandstone
34	"	"	30	–	" " "
35	"	"	–	2.0	Single jet. d = 1.70 mm "
36	"	"	30	–	" " "
37 to 41	"	0.5 & 1.27	15	2.0	α = 60°; single and twin jets; d = 1.17 and 1.70 mm. Berea sandstone. No fragmentation
42	69	0.635	60	–	α = 60°; twin jets; d = 1.17 mm; Unconfined pink granite. spalling only
43	"	"	–	2.0	" " "
44	"	1.27	–	2.0	Single jet. d = 1.70 mm. Spalling only
45	69	5.7	15	–	α = 60°; twin jets; d = 1.17 mm Berea sandstone. No fragmentation
46	"	"	–	2.0	" " "
47	"	"	–	10.4	" " "
48	"	"	–	"	single jet; d = 1.70 mm
49	"	"	–	2.0	" " "
50	"	"	15	–	" " " No fragmentation

Table 1 (cont'd)

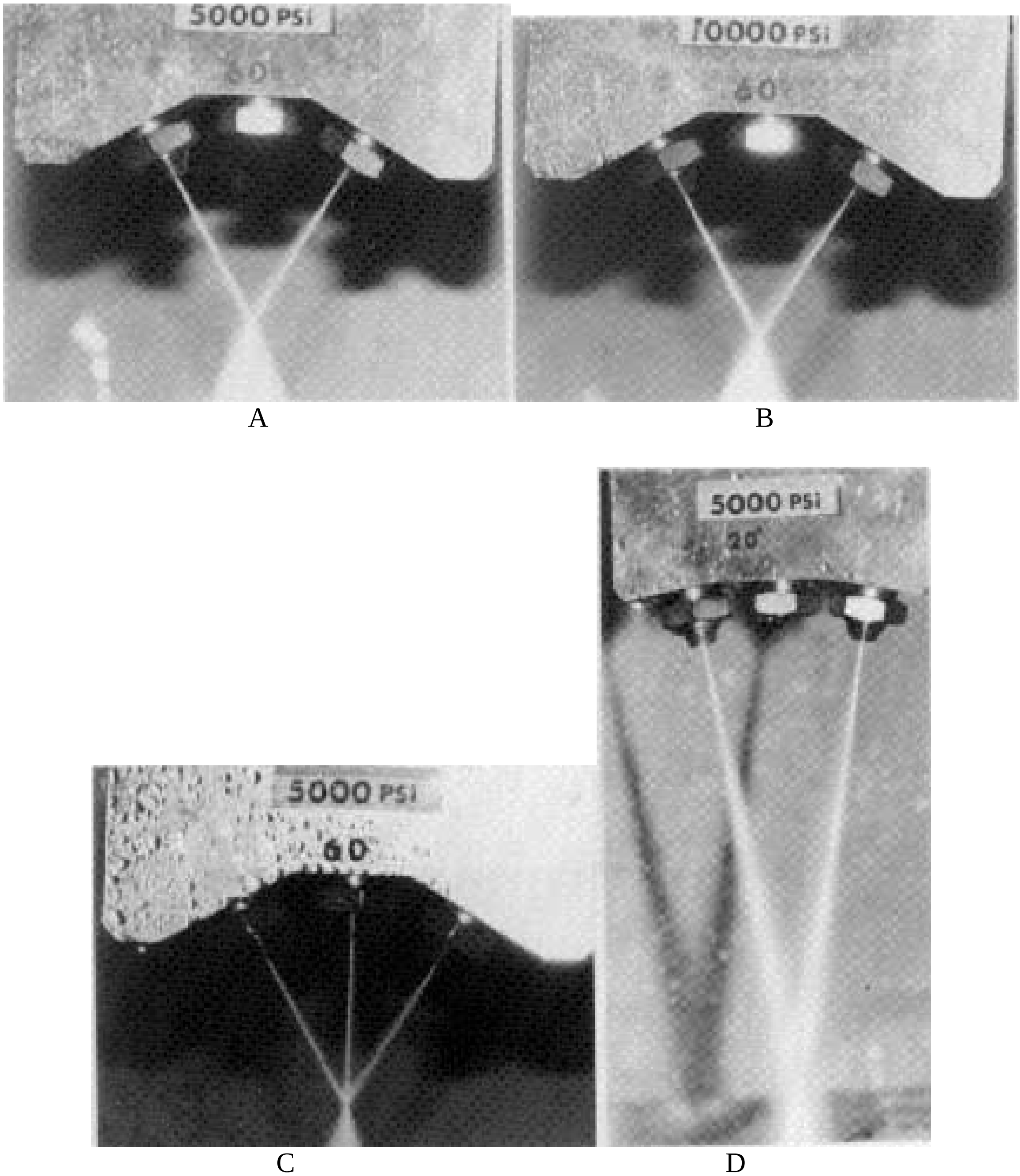
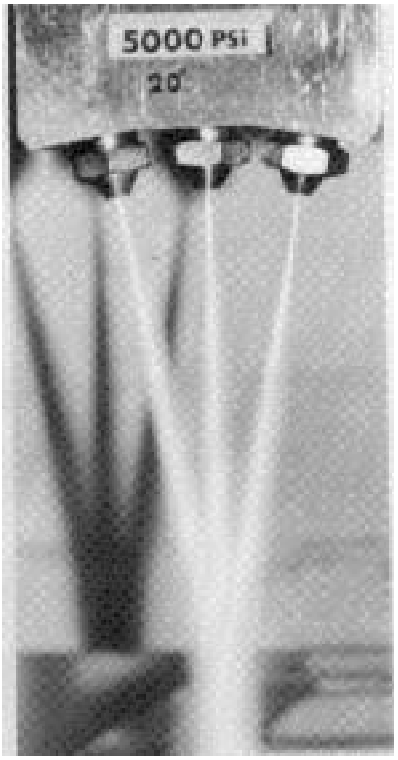
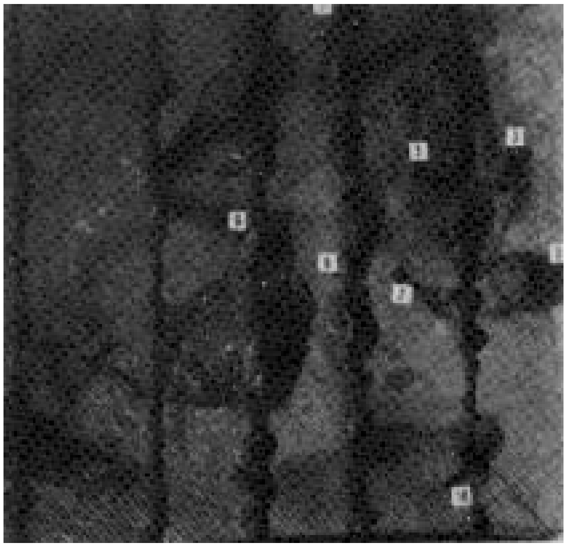


Figure 1 General view of converging jets.

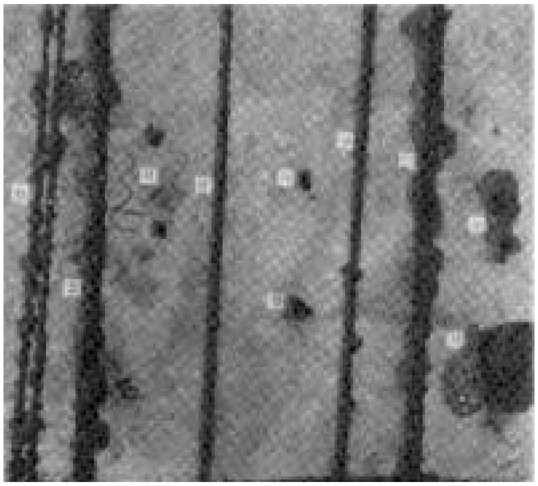


E

Fig. 1 General view of converging jets (continued).

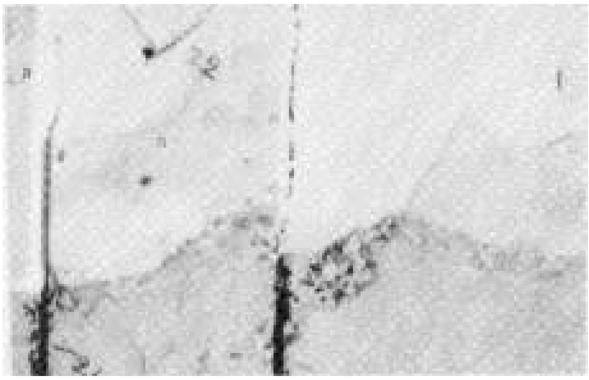


A Indiana Limestone

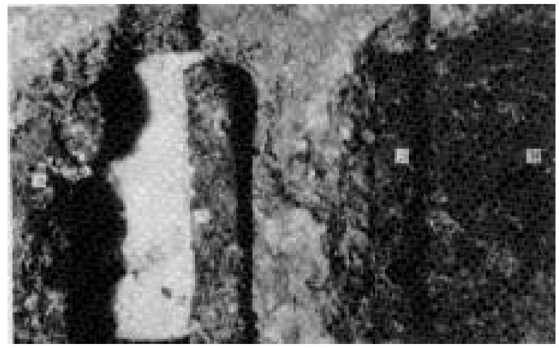


B Limestone

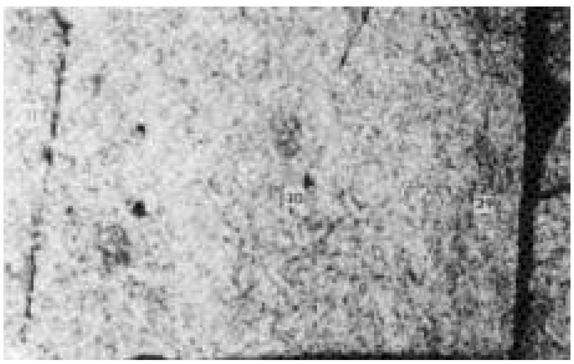
Figure 2 Rocks cut by single and converging jets



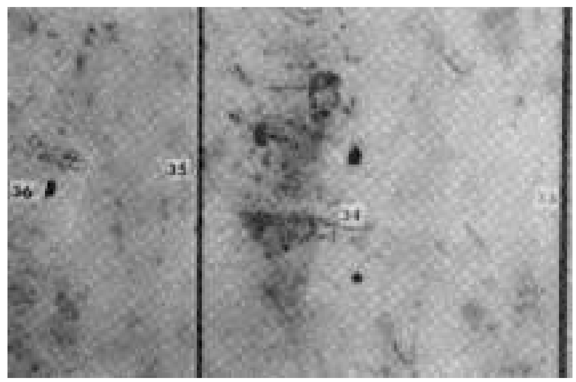
C Marble



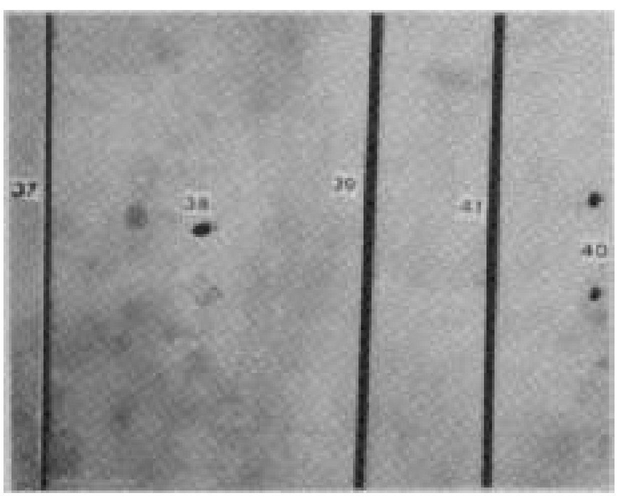
D Conglomerate



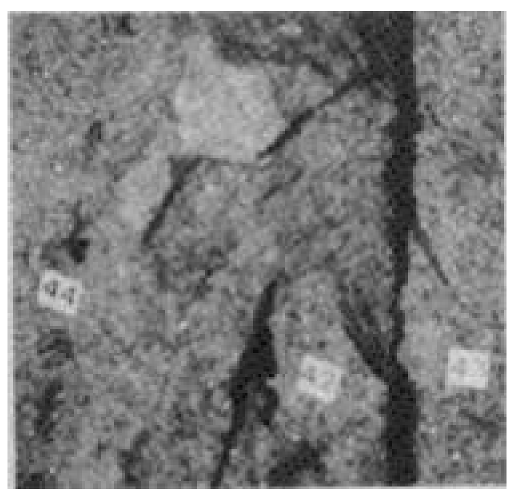
E Gray Granite



F Berea Sandstone

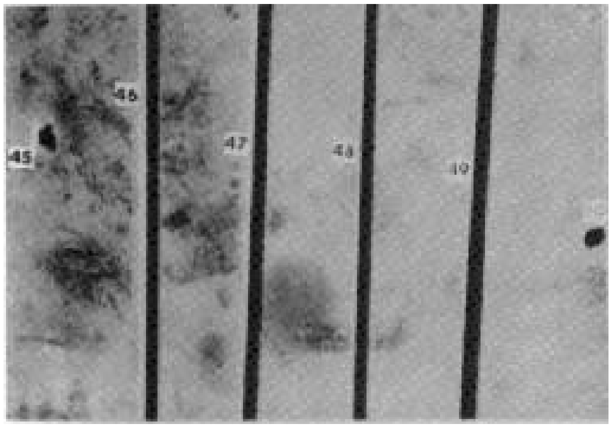


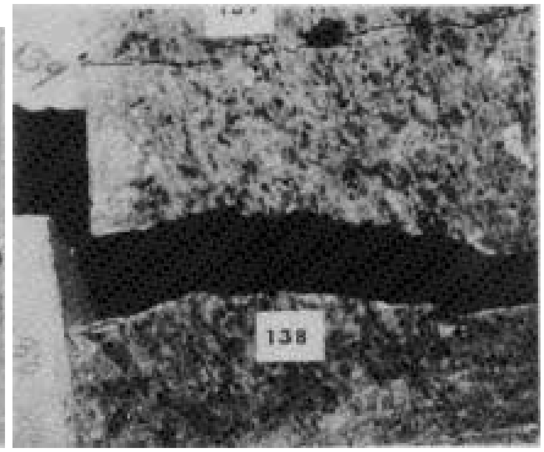
G Berea Sandstone



H Pink Granite

Fig. 2 Rocks cut by single and multiple converging jets (continued)





J Berea Sandstone

K A confined block of barre granite

Fig. 2 (Continued)

Fig. 3 A block of Barre granite completely fractured with rotating water jets at 100 MPa

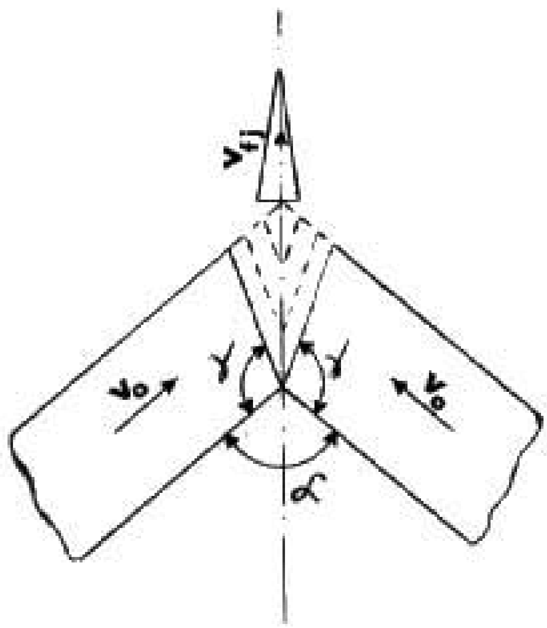


Fig. 4 Model of formation of cumulative jet (ref. 5)

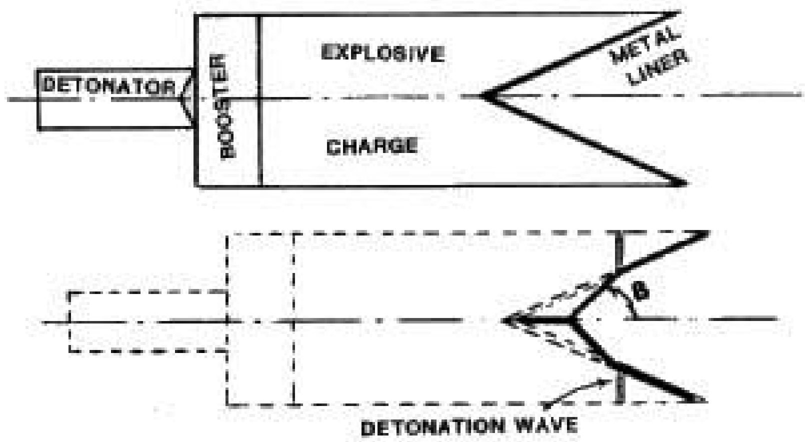


Fig. 5 Jet formation by charges with lined cavities (ref. 6)

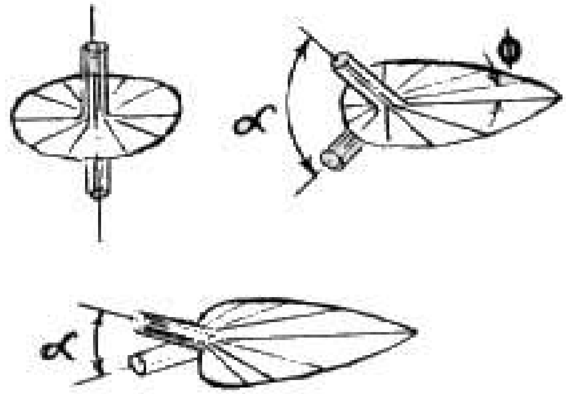


Fig. 6 Sketch of sheets formed by impact of two cylindrical jets (ref. 9)

FURTHER STUDIES IN WATER JET ASSISTED DRAG TOOL CUTTING IN ROCK

by

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ABSTRACT

Due to the economic limitations of rock cutting tools in terms of the strength and abrasivity, studies into water jet assisted drag tool cutting of rock materials began in the University of Newcastle upon Tyne in 1978. In common with the majority of laboratory studies traverse speed was severely limited to a magnitude of order slower than those used by roadheaders, continuous miners and shearers.

Following the encouraging initial studies conducted by the N.C.B. and at Newcastle, in 1981 further studies commenced with the conversion of an existing instrumented rock planing machine to cut at greater than 1 m/s. The results and hypotheses discussed in this paper forms the basis of a current study into fundamental aspects of water jet assisted drag tool cutting under realistic conditions of traverse speed and mechanical depth of cut.

A generalized conclusion in testing four different rock materials using a 'standard' experimental program was that a high pressure water jet can significantly assist cutting through force reductions, particularly in the 'normal' direction. However care must be taken to identify the correct pressures needed to maximize these reductions especially in the weaker materials where jet penetration is significant.

When these force reductions are considered together with the other recorded advantages of decreased wear, dust suppression and incendive ignition prevention, high pressure water jet assisted cutting may well prove both economic as well as desirable in future applications.

INTRODUCTION

An increasing amount of coal and rock excavation is being carried out mechanically using machines equipped with arrays of drag tools. The need to excavate more effectively and economically in harder strata has been met by an increase in the size and weight of machines from 16 tonnes to 100 tonnes (ref. 1). The Dosco (in Figure1) is typical of a modern medium duty roadheader. However one of the limiting factors when cutting hard strata is the amount of energy that can be put through a standard tungsten carbide tipped drag tool before rapid degradation of the tool occurs. Thus independent of machine size, drag tools are limited in their application by the strength and abrasivity of the rock materials to be excavated.

Research into the use of high pressure water jets to assist the process of drag tool cutting in strong and abrasive strata has been carried out both in the laboratory and in-situ. Reports from South Africa (ref. 2), U.S.A. (ref. 3), (ref. 4), Switzerland (ref. 5), France (ref. 6), Germany (ref. 7) and Great Britain (ref. 8), (ref. 9), indicate that water jets can assist significantly in improving the cuttability of the drag tool system.

In the University of Newcastle upon Tyne, a pilot study in water jet assistance to rock cutting with drag tools was started in 1978 (ref. 10). The main conclusions from this study were: the cutting and normal force components were reduced; and that optimum reductions were obtained with the jet impinging immediately in front of the tool (1-2mm). However in common with the majority of the laboratory investigations previously referenced this study was carried out at a comparatively slow traverse speed (typically 0.165 m/s), which is an order of magnitude slower than the average tool velocity used by roadheaders, continuous miners and shearers.

It was therefore essential to establish such speeds within the laboratory. As a comparison the Dosco Mk 2B roadheader (ref. 11) operates at quoted peripheral tool speeds of 1.05 m/s and 1.83 m/s. Further, in-situ studies (ref. 8) have indicated that more than a 50% increase in performance can be obtained in hard strata using water jet assistance.

EXPERIMENTAL FACILITY

Available in the University of Newcastle upon Tyne was a 50 tonne rock planer previously used for disc cutting studies (ref. 12). Modification to the ram allowed the cutting speed to be increased to a maximum 1.1 m/s, although the force capability was reduced to 4 tonne in the cutting direction, (Figure 2).

Samples of approximately 1000 mm x 800 mm x 700 mm can be mounted giving therefore up to lm length per single cut. Jet pressures of up to 70 MPa are provided by a 75 kW intensifier type unit supplied by Preswell Engineering giving a maximum of 45 l/min at 70 MPa. An alternative intensifier can be mounted on the basic unit to give 256 MPa of pressure. Jet pressures are monitored by an in-line transducer.

Force measurement was carried out using a three component dynamometer, recording the analogue signals on an S.E. 3000 series instrument magnetic tape deck with visual replay through a UV recorder. Jet pressures are similarly recorded and examined. Analysis of the data is by a microcomputer following A/D conversion of the filtered recorded signal.

The tool used throughout the studies reported in this paper, was a Wimet Swiftsure HW SS41/2, assisted by a tungsten carbide nozzle made to an M.R.D.E. design The most common tool/nozzle geometry employed is shown (in Figure 3), where the water jet impinges approx. 1 mm ahead of the tool from a stand-off distance to the tool tip of 65 mm.

EXPERIMENTAL PROGRAMME

The main aspect of this research co-sponsored by E.C.S.C./N.C.B. and S.E.R.C. is to investigate fundamental aspects of water jet assisted drag tool cutting in rock materials. In order to satisfy the requirement that the research be applicable to drag tool equipped machines in the field, not only must the traverse speed be comparable, but also the rock materials tested. Therefore four rock materials, three sandstones and a limestone have been subjected to the 'standard' cutting program given in Table 1. Typical rock material properties can be found in the Appendix. It is felt that these rocks cover the majority of the 'hard' strata conditions encountered by drag tool equipped machines, by virtue of their relative combinations of strength and abrasivity, (Figure 4).

The major variables after the rock material itself were the mechanical depth of cut and the jet pressure. Only two traverse speeds were used; 0.27 m/s in order to provide the lower speed link with previous laboratory studies at both the University of Newcastle upon Tyne and other establishments; and 1.1 m/s in order to model field conditions.

Additional studies are currently being undertaken using different nozzle positions and tool types, including worn tools. However these studies are designed to support the following hypotheses based on the results from this 'standard' cutting program. When considering the results presented here, all data points are averages of four repetitions at each experimental condition.

ROCK FAILURE MECHANISMS IN HYBRID CUTTING

Hurt and Laidlaw (ref. 13) have pointed out that the tool tip macroscopically performs two functions:

i) to break the rock ahead of the tool
ii) to clear a path for itself through the remaining material.

The first function is termed "chipping" while the second function is termed "profiling". Chipping is the major rock removal process and corresponds to the peak forces during cutting. Profiling contributes most to the mean forces.

However, microscopically drag tool cutting can be regarded as an indentation process by Hood (ref. 2) and Lindqvist (ref. 14). Whilst it was observed by Lindqvist and Lai (ref. 15) that a zone of highly fractured and inelastically deformed rock is created beneath the tip of rock cutting tools. This crushed zone develops very early during the indentation and transmits the main force into the rock. When a critical stress which is a function of tool geometry and rock type is attained a dominant crack will be initiated. The sudden release of strain energy in the tool will then eject the chip violently.

The crushed zone can be regarded as an "effective bluntness" to the tool. It acts as an interface between the tool and the rock and controls the efficiency of energy transfer from the tool to the rock.

A model of hybrid cutting is proposed from the laboratory investigations undertaken and observations made. The water jet reduces the size of the crushed zone by flushing and decreasing the "effective bluntness" of the tool. Therefore the jet can preserve the tool tip as an effective stress concentrator thus maintaining the "theoretical minimum level" required to achieve the critical stress for crack initiation.

This flushing also accounts for a more rapid build-up of force prior to the chipping event. Idealized force/time traces for non-assisted and assisted cutting are shown (in Figure 5). These illustrate the advantages to both absolute peak force magnitudes and to the efficiency for energy transfer.

Profiling in turn may be assisted by water jets through both flushing of debris and lubrication of tool/rock interfaces.

RESULTS

Penetration tests for the water jet running on a smooth rock surface were carried out on all rocks. Immediately the rock materials divide into two groups, those where significant jet penetration occurs at all levels of speed and jet pressure, and those where no significant penetration can be measured even at slow traverse speeds and high jet pressures. Figure 7 shows the jet penetration characteristics for the Dumfries and Penrith Sandstones where significant penetrations could be measured. Whereas the Pennant Sandstone and the Middleton Limestone belong to the latter group. The following discussion of the influence of water jets on the forces experienced by the tools is considered for each of these groups. (Figures 8 to 11 show these results for each rock tested.

The Effects of High Pressure Water Jets

Rock with Jet Penetration (Penrith and Dumfries Sandstone)

With different depths of cut and traverse speeds, for the jet configuration used, there exists a pressure that results in minimum cutting and normal forces. This optimum situation usually occurs when the jet penetration alone is in the region of 20-50% of the mechanical depth of cut.

When the jet penetration is greater than about 60-70% of the mechanical depth of cut, the chipping events are greatly reduced. Laboratory observations reveal that the number of saucer shaped chips produced decreases and the groove left on the rock surface suggests that the chipping process was heavily suppressed and the profiling action was dominant. When the jet penetration was greater than the mechanical depth of cut, the chipping events were completely absent and the cutting is essentially a wedge profiling action.

Chipping is the efficient way of rock breaking as it utilizes the low tensile strength of rock. Profiling, however, involves a greater proportion of rubbing and secondary debris comminution which wastes energy. At the optimum jet pressure minimum forces are theoretically required because the jet penetration just reaches the tool tip and is able to flush away the crushed zone. Therefore the tool tip acts as an effective stress concentrator and crack initiator. However, when the pressure is higher than the optimum pressure, and the jet penetration is deeper than the mechanical depth of cut, then the tool tip becomes situated in the jet slot. Hence, under these conditions the cutting is essentially an inefficient profiling action and the force components acting on the tool rise from those experienced at optimum pressure. Figure (6) illustrates the above mechanisms for a hypothetical situation.

Rock without Jet Penetration (Middleton Limestone and Pennant Sandstone)

The similarity between the geometry of the large chips created in tool-only cutting and hybrid cutting suggests that the chipping event is unaffected by the jet. The jet helps with chip forming but does not alter the process.

Although there is some scatter in the results obtained, which is normally expected in rock cutting, the general trend is that the higher the jet pressure, the greater the force reduction. However, as was pointed out previously, a diminishing benefit is anticipated when the pressure is further increased until an optimum pressure is reached. Beyond this optimum pressure, an increase of force is to be expected.

The Effects of Mechanical Depth of Cut

The magnitude of the "effective bluntness" of the tip relative to the depth of cut is important as different modes of rock cutting were observed to be dominant at different depths of cut. The following dominant modes of cut were observed for:

5 mm - profiling dominants
10 mm - profiling and chipping
15 mm - chipping dominants

Domination of the profiling mode at 5 mm depth of cut explains the absence of an optimum pressure as described in the previous section. Generally, the higher the pressure, the greater the force reduction obtained. The reason is that profiling involves rubbing and debris comminution in which the higher jet pressures more easily remove the loosened debris.

The Effects of Traverse Speed

Although it is claimed that the cutting force is independent of tool speed (Nishimatsu (ref. 16, 1979), O'Doughty and Burney (ref. 17, 1963), there is an unmistakable trend that the speed has affected the normal force. While the high normal force at high speed in the Pennant Sandstone can be explained through tool wear, the results in the low-abrasivity rocks such as Middleton Limestone and Dumfries Sandstone suggest that it is a common phenomenon. One possible explanation is that the debris is more easily removed during slow cutting than in fast cutting, where more debris is crushed under the tool resulting in the rise in normal force with cutting speed.

The Effects of Abrasivity

Water jet assistance could find its best application to be in cutting abrasive rocks at fast traverse speeds. Tests show clearly that the fast wear rate in abrasive rock is a limiting factor preventing the greater application of drag tools in hard rock cutting. A careful examination of the wearflat shows temperature wear striations which supports the findings that at high temperature, the hardness of the tungsten carbide at the tool tip is less than the hardness of quartz which is a main constituent of hard sandstone (Roxborough (ref. 18, 1973). Most of the force reduction therefore is attained in the normal direction where 52% reduction at 15 mm mechanical depth of cut in Pennant Sandstone and 58% and 30% reductions at 10 mm and 15 mm mechanical depths of cut in Penrith Sandstone have been recorded.

CONCLUSIONS

The results of laboratory linear cutting studies using unrelieved cuts up to 1.1 m/s with high pressure water jet assistance up to 70MPa, have shown that a reduction of both cutting and normal force components in the rocks tested was obtained with most reduction in normal force components.

A model that describes the role of the water jet in hybrid cutting has been proposed which predicts the existence of an optimum pressure. Any jet pressure greater than this optimum will result in an increase of forces and wasteful expenditure of energy due to reduced incidence of chipping and increased profiling action. Laboratory findings support this model in two of the rocks tested where the optimum pressure was below 70MPa.

It was found that the normal force component increased with speed whilst the cutting force was found to be less sensitive in the speed range 0.27 - 1.1 m/s

The rock materials tested divide into rocks which can be significantly penetrated by the jet and those where jet penetration is very low even at the highest pressure and slowest traverse speed used. In this latter case reduced friction, lubrication and debris clearance accounts for actual but often marginal force reductions.

The testing program described in this paper has concentrated only on the forces experienced by the tool. The increased cutting life of tools, reduction of respirable dust (ref. 19) and methane ignition hazard makes the application of high pressure water jet assistance to drag tool machines both economically desirable and in some applications essential for safe operation.

ACKNOWLEDGEMENTS

The authors wish to acknowledge the support and facilities provided by Professor J.F. Tunnicliffe, Head of the Department of Mining Engineering, University of Newcastle upon Tyne.

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APPENDIX

Rock Material Physical Properties for the three standstones and one limestone tested in the 'standard' experimental programme given in Table 1.

Rock Type/Name	C.S.[1] MPa	T.S.[2] MPa	C.A.[3] 1/10mm	S.E.[4] MJ/m^3	B.D.[5] g/cm^3	A.P.[6] %
Dumfries Sandstone	50.4	2.7	1.98	11.3	2.15	19.04
Penrith Sandstone	75.1	6.8	2.86	17.5	1.48	12.68
Pennant Sandstone	161.3	10.1	2.27	44.5	2.72	8.79
Middleton Limestone	120.1	9.0	1.48	23.5	2.68	2.87

1 Unconfined Uniaxial Compressive Strength
2 Brazilian Disc Indirect-Tensile Strength Test
3 Cerchar Abrasivity
4 Cutting Specific Energy
5 Bulk Density
6 Apparent Porosity

Operational Parameter	Levels	Values
Pressure	4	0, 23 MPa, 46 MPa, 69 MPa
Depth of cut	3	5mm, 10mm, 15mm
Speed	2	1.10 m/s, 0.27 m/s

TABLE 1

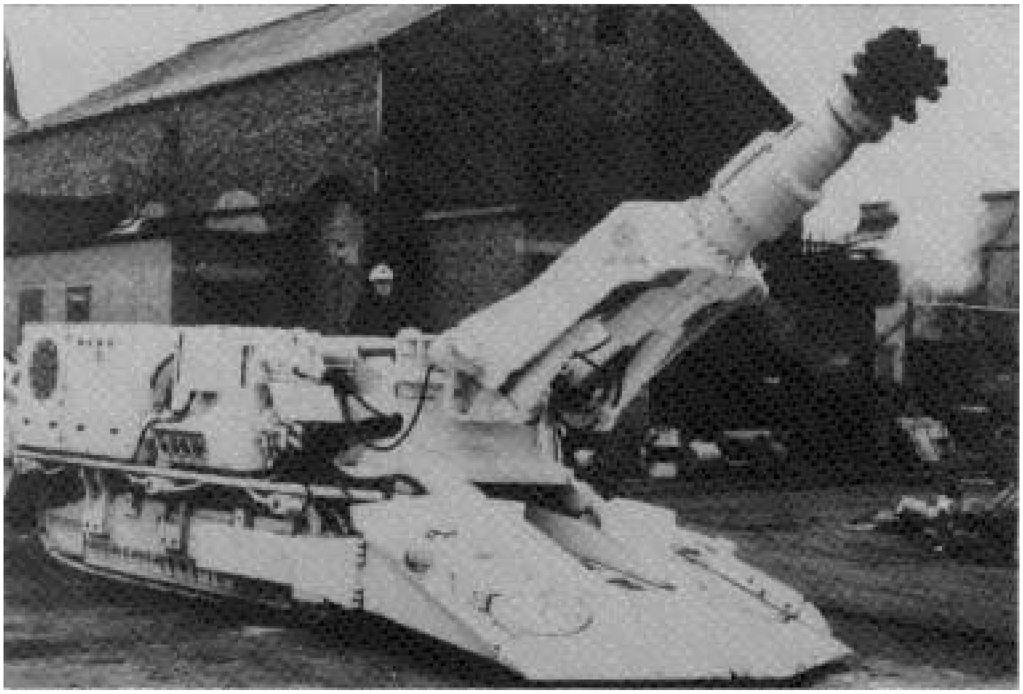


Figure 1

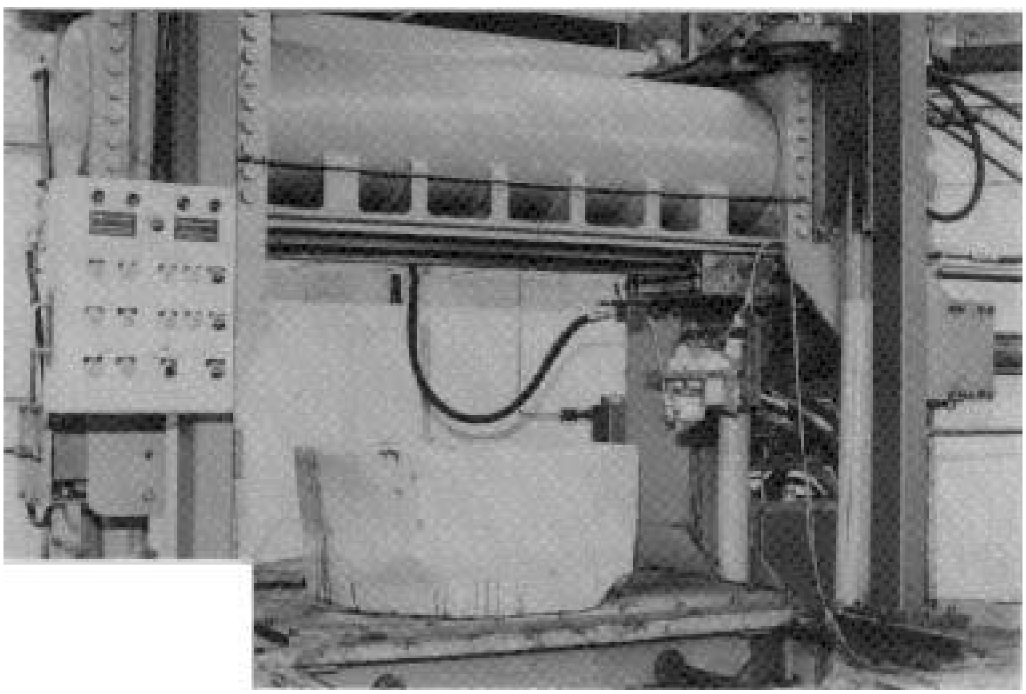


Figure 2

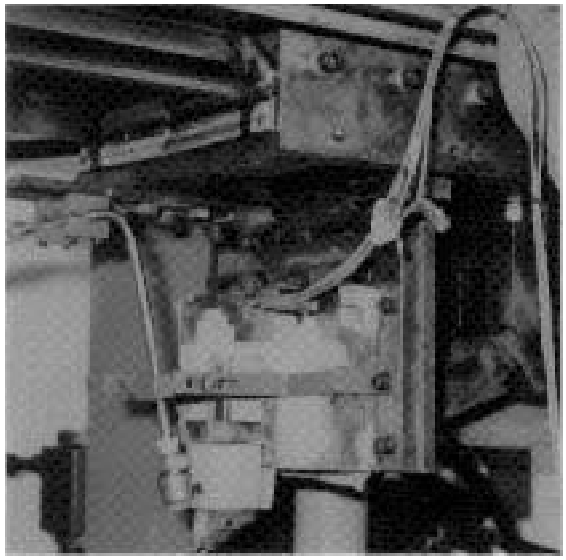


Figure 3

Strength		Low Abrasivity	High Abrasivity
	Weak	Dumfries Sandstone	
	Medium	-	Penrith Sandstone
	Strong	Middleton Limestone	Pennant Sandstone

Figure 4

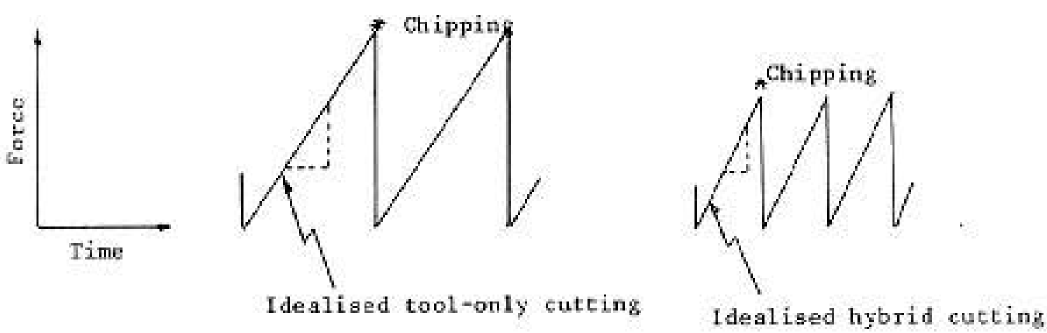


Figure 5

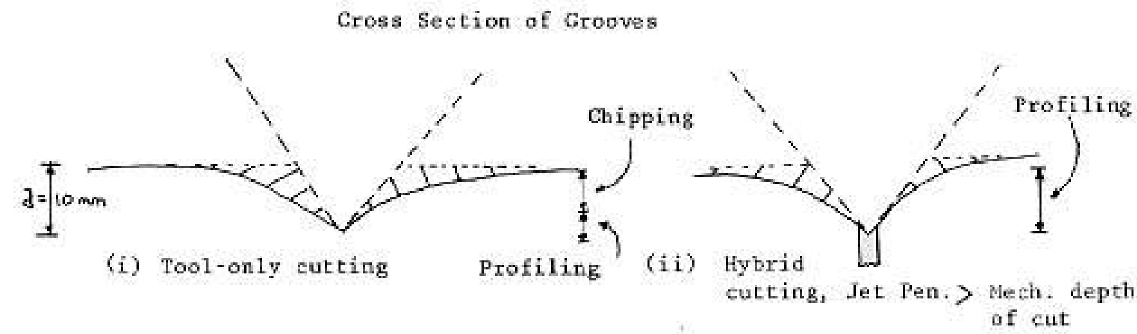
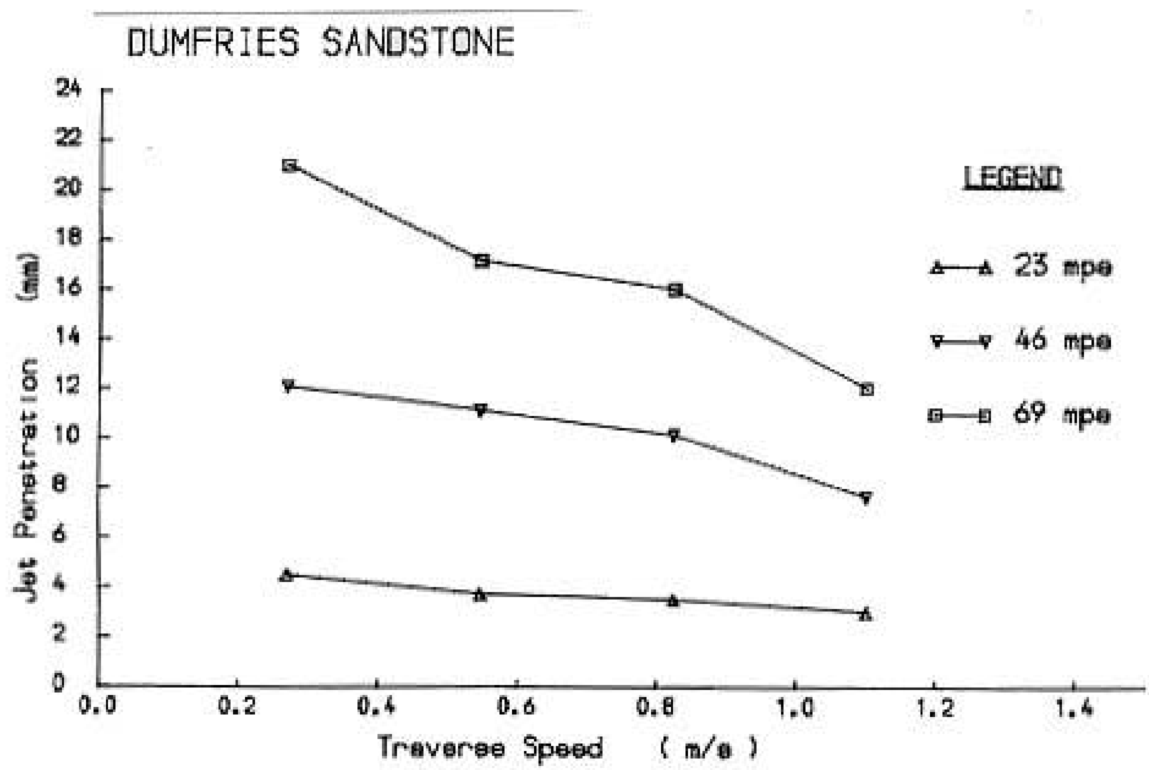

Cross Section of Grooves
Chipping
Profiling
d = 10 mm
(i) Tool-only cutting
Profiling
(ii) Hybrid cutting, Jet Pen. > Mech. depth of cut

Figure 6


DUMFRIES SANDSTONE
Jet Penetration (mm)
Traverse Speed (m/s)
LEGEND
23 mpa
46 mpa
69 mpa
0
2
4
6
8
10
12
14
16
18
20
22
24
0.0
0.2
0.4
0.6
0.8
1.0
1.2
1.4

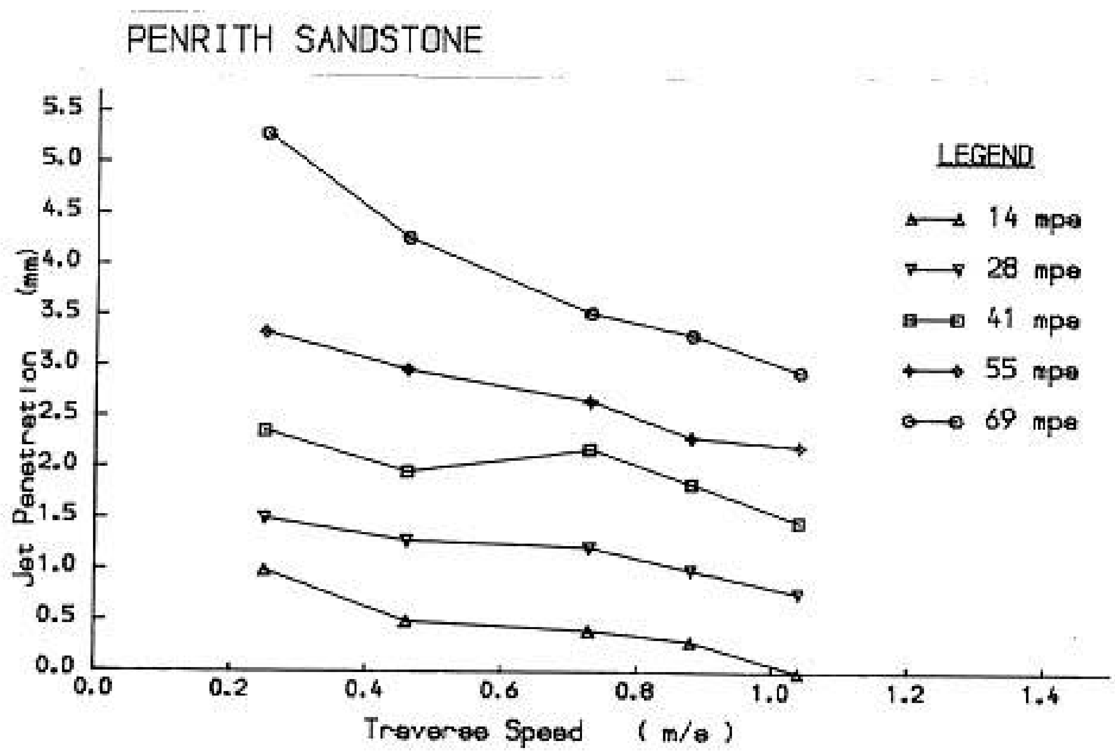


Figure 7 Jet penetration characteristics for Dumfries and Penrith Sandstones

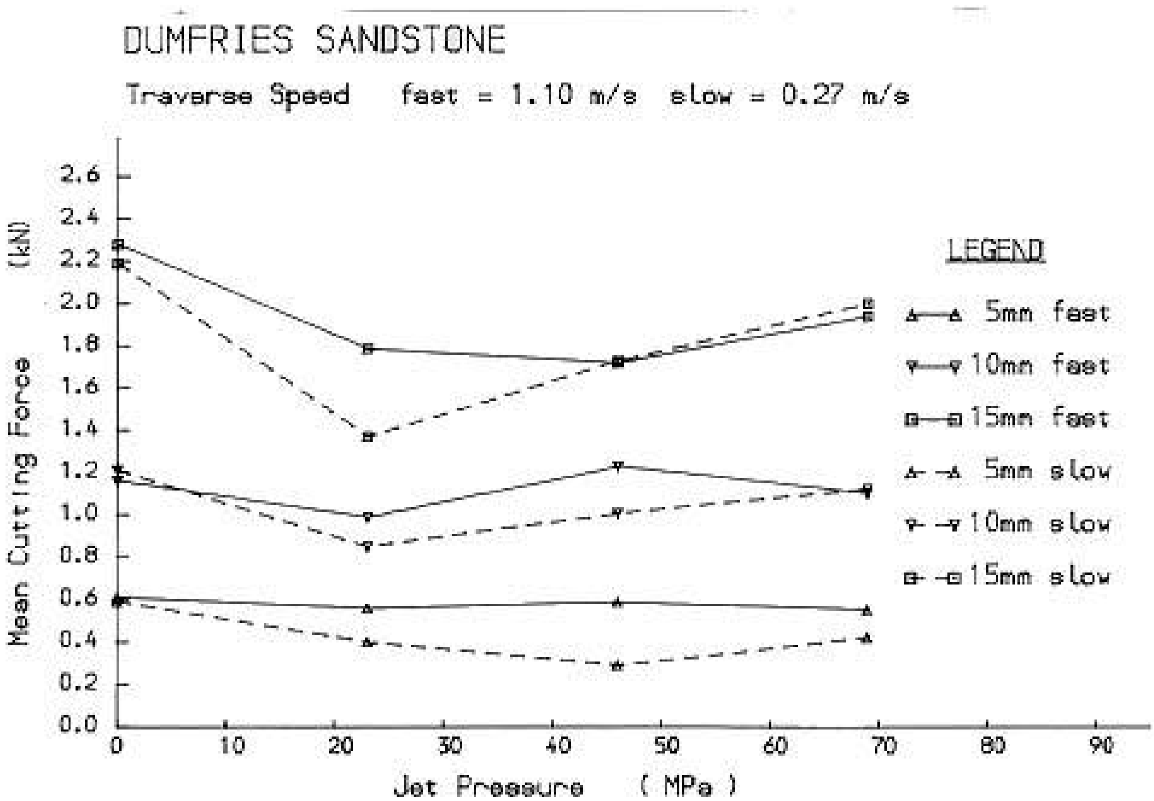


Figure 8(a)

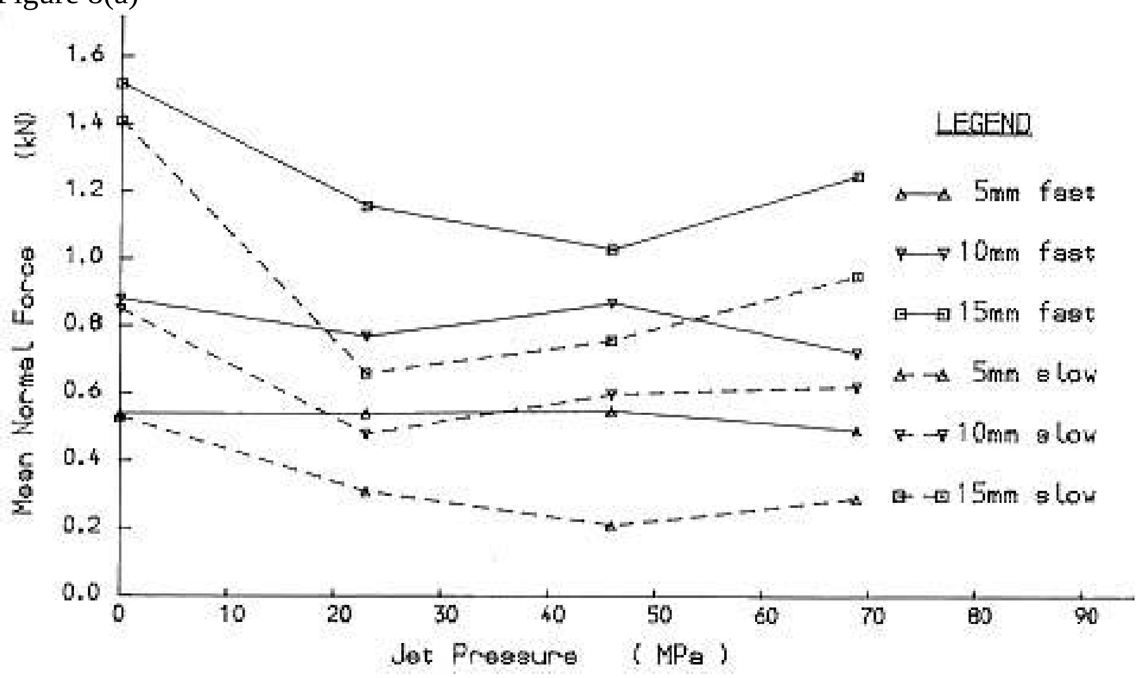


Figure 8 (b)

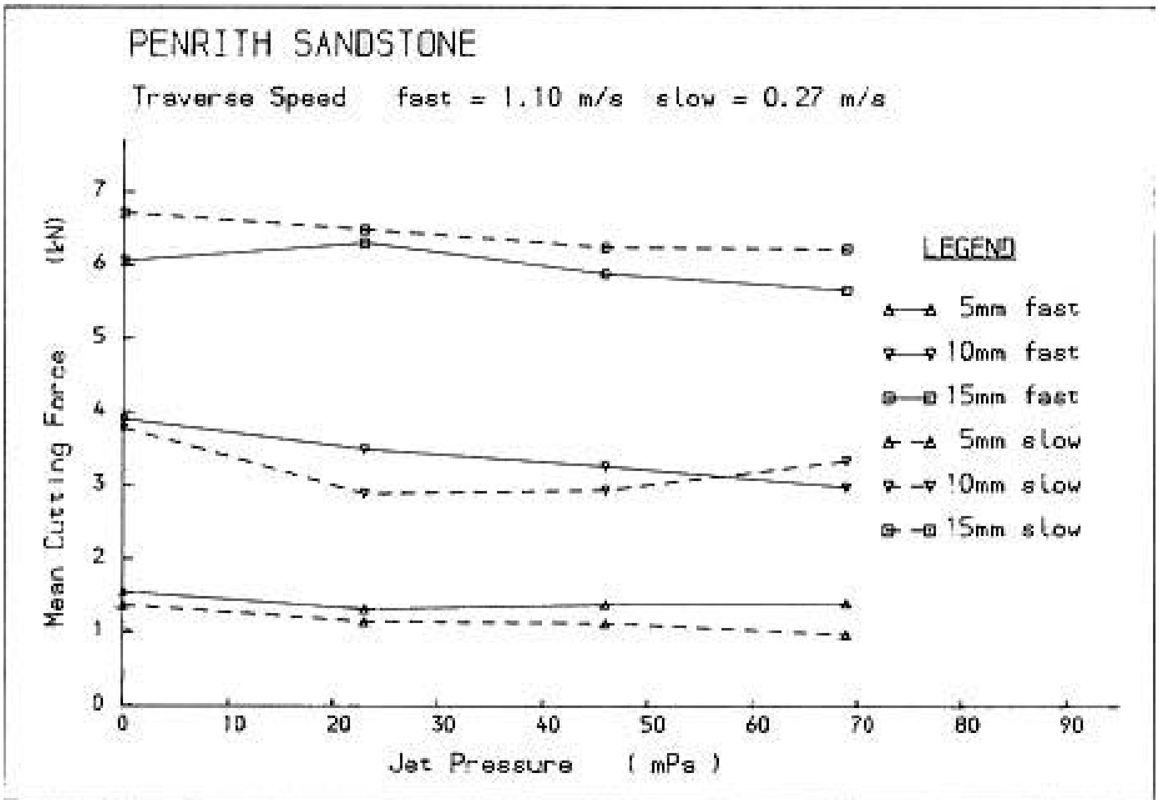


Figure 9(a)

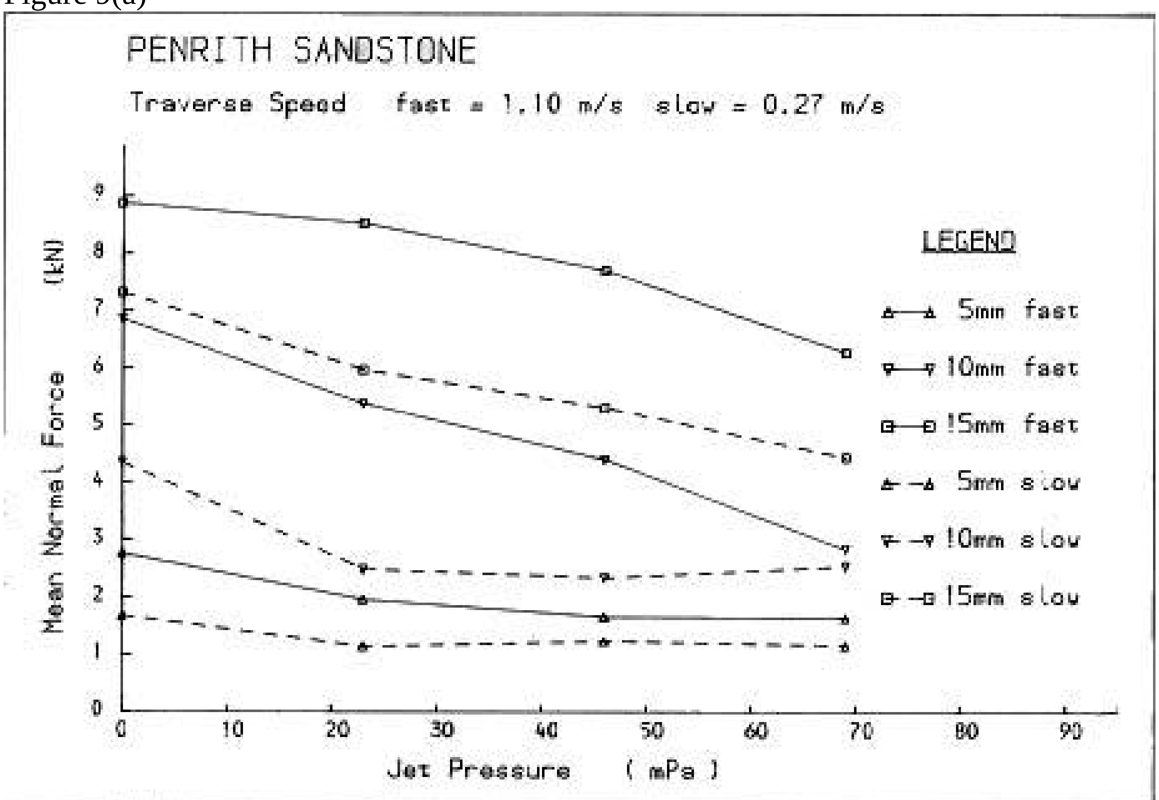


Figure 9 (b)

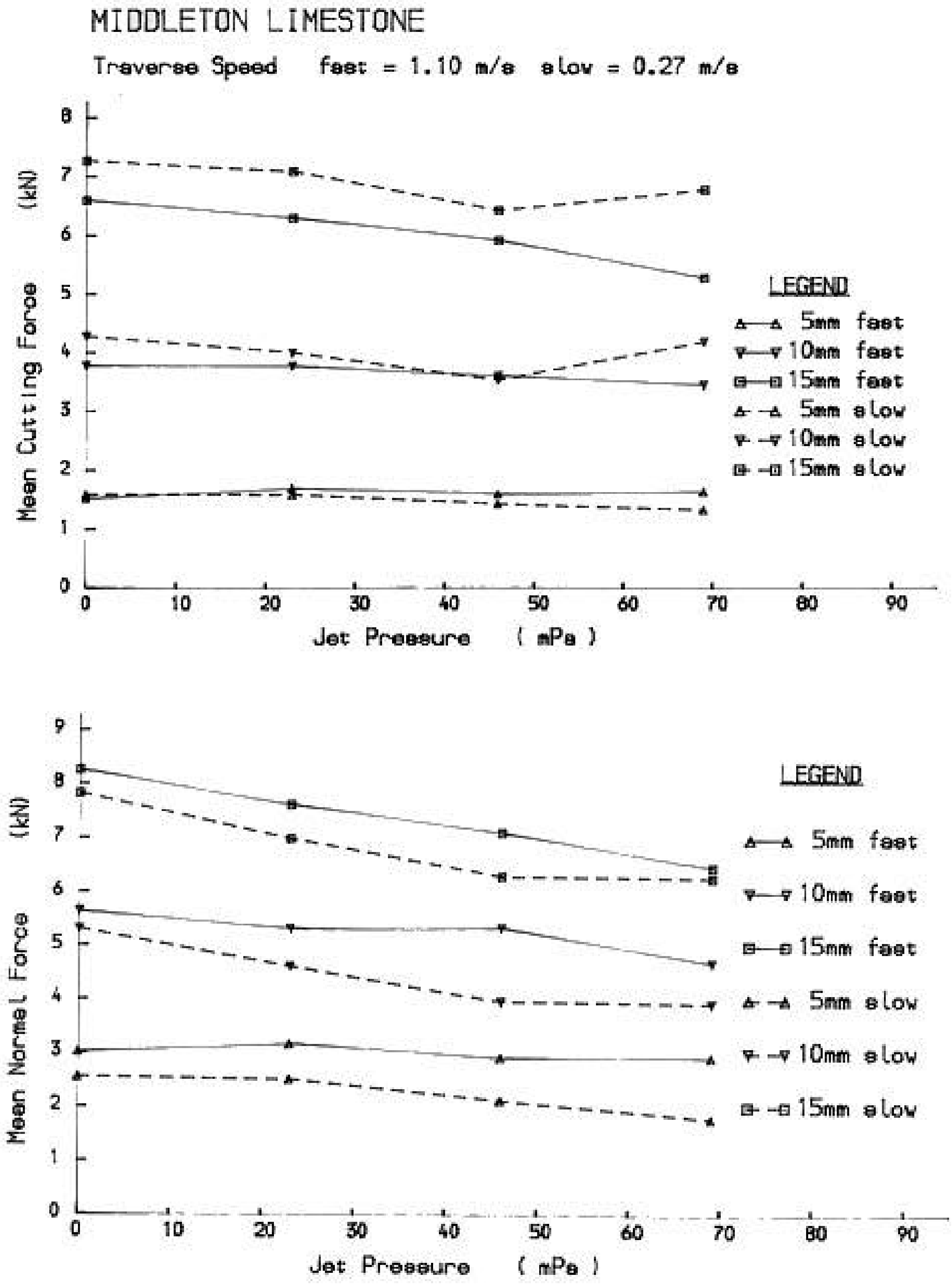
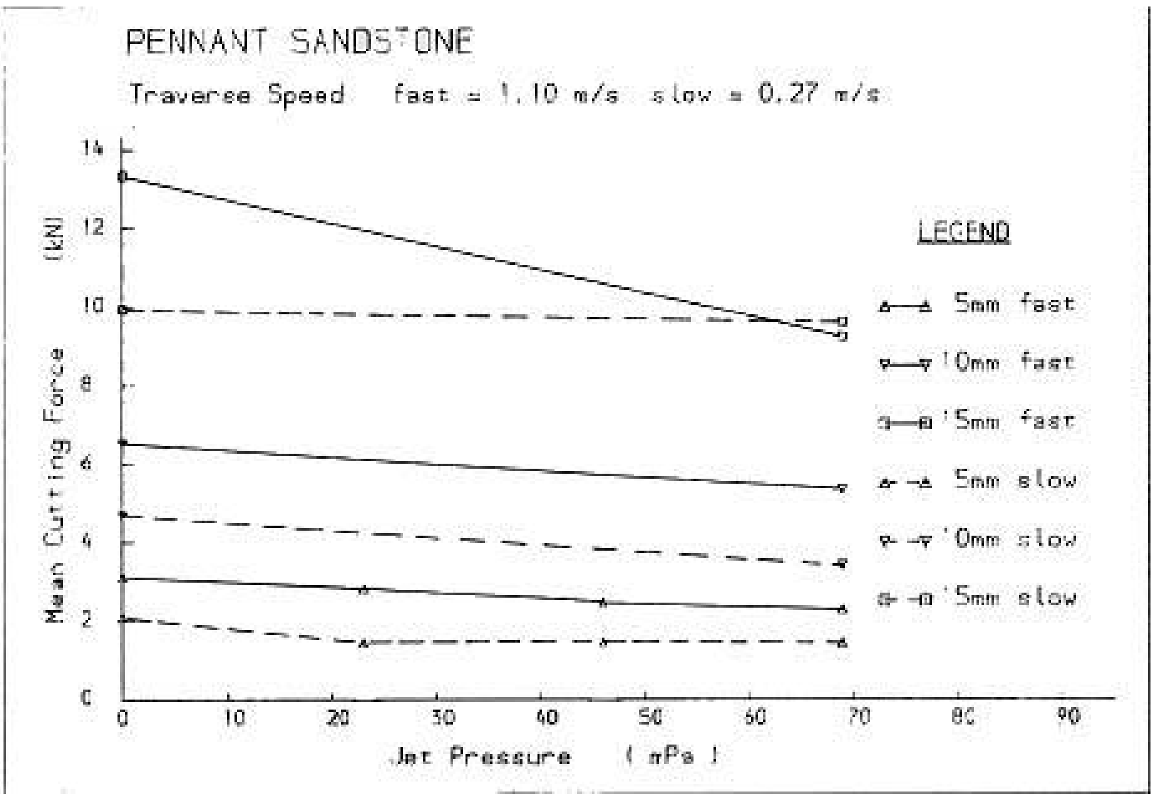

MIDDLETON LIMESTONE
Traverse Speed fast = 1.10 m/s slow = 0.27 m/s
Mean Cutting Force (kN)
0
1
2
3
4
5
6
7
8
0 10 20 30 40 50 60 70 80 90
Jet Pressure (mPa)
LEGEND
5mm fast
10mm fast
15mm fast
5mm slow
10mm slow
15mm slow
Mean Normal Force (kN)
0
1
2
3
4
5
6
7
8
9
0 10 20 30 40 50 60 70 80 90
Jet Pressure (mPa)
LEGEND
5mm fast
10mm fast
15mm fast
5mm slow
10mm slow
15mm slow

Figure 10


PENNANT SANDSTONE
Traverse Speed fast = 1.10 m/s slow = 0.27 m/s
Mean Cutting Force (kN)
Jet Pressure (mPa)
LEGEND
5mm fast
10mm fast
15mm fast
5mm slow
10mm slow
15mm slow

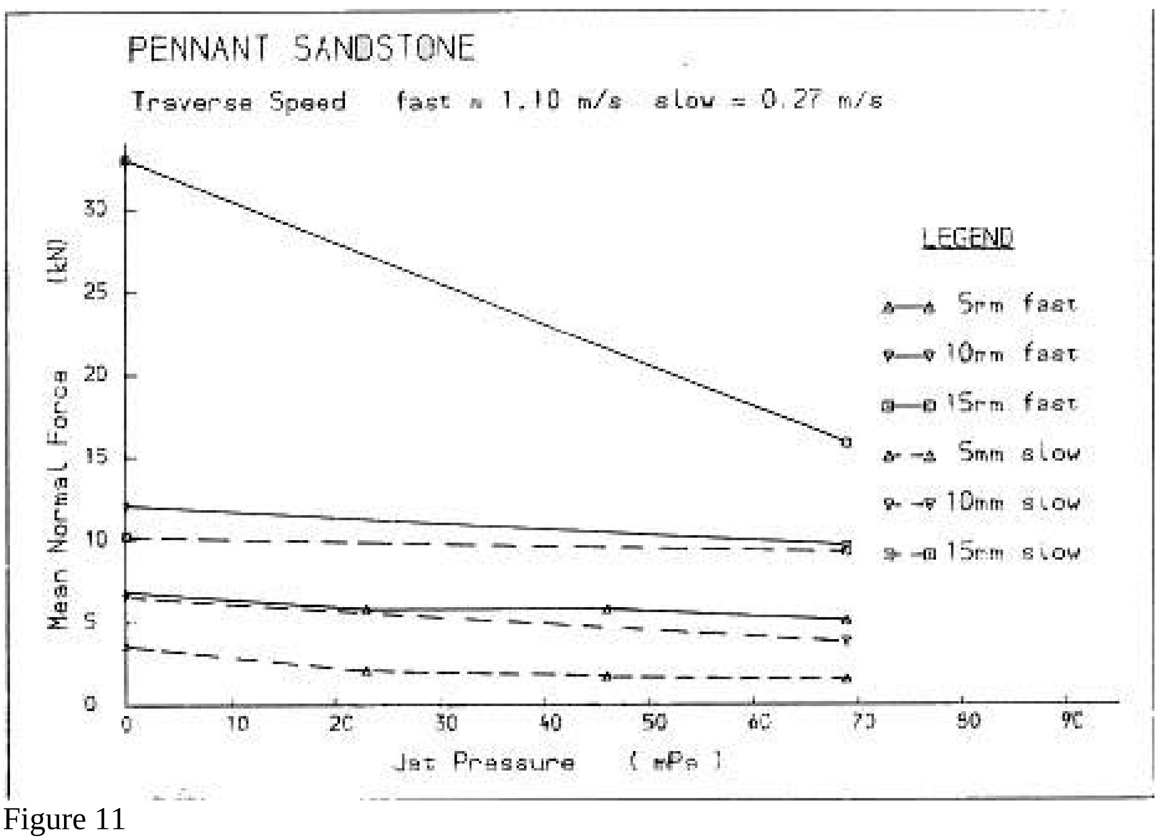

PENNANT SANDSTONE
Traverse Speed fast = 1.10 m/s slow = 0.27 m/s
Mean Normal Force (kN)
Jet Pressure (mPa)
LEGEND
5mm fast
10mm fast
15mm fast
5mm slow
10mm slow
15mm slow

Figure 11

RETROFITTING SHEARERS AND CONTINUOUS MINERS WITH WATER JET ASSIST CUTTING

by

Roger F. J. Adam, P.E.
KETRON, Inc.

ABSTRACT

Cutting coal, rocks and other materials by using a high pressure water jet is a well proven possibility. However, the use of this new technology in mining machinery is restricted by limitations in availability of reliable equipment such as pumps, seals and valves. Water jet assist cutting combines the action of a cutting tool and an associated water jet thus presents most of the advantages of water jet cutting with less limitations because it uses a lower water pressure (up to 10,000 psi). The action of the water jet when assisting a mechanical tool is complex. It still involves a jet cutting effect but it is possible that the main action of the water jet is to clean the rock debris ahead of the tool, improving its contact with the material and cooling the bit tip. It is most likely a combination of these different processes which results in such an amazing difference with dry cutting on cutting forces and dust generation.

INTRODUCTION

The reduction of the amount of respirable dust generated is a sufficient reason to promote and develop water jet assist cutting. The solution we propose is based on the following assumptions:

- Retrofitting existing machine is the only quick way to introduce a new technology.
- Water jet assist cutting reduce the cutting forces acting on the bits.
- Dust generation is considerably reduced by use of water pressure in the range of 4 to 6,000 psi.
- High pressure components are still in the development phase.

The logical approach is:

- to use the power saved on the cutting head and to increase the water pressure with a built-in drum intensifier,
- to provide high pressure water to the bits only when they are cutting.

The water pressure can be intensified in other locations.

1. At some distance from the machine by a power pack. This solution creates some risk due to high pressure pipes or flexible hoses. Machines will require a double pressure distribution when low pressure water for motor or power pack cooling is necessary as for shearer loaders.
2. On the machine by a built in power pack. This second location eliminates the water supply with two different pressures and eliminates most of the risk created by high

pressure distribution. It is very suitable to special cases such as drilling machines and roadheaders for which there is no need for sectorial distribution or pressure variation.

These two first systems require high pressure rotating seals for the water to reach the cutting head. This requirement is easily achieved when small diameter rotating seal can be used (roadheaders, shearer loaders, drilling machines). It is much more difficult for existing continuous miner cutting heads when rotating seal diameter ranges from 250 to 400 mm (10 to 16 inches). Low pressure, large diameter rotating seals are just recently undergoing tests in underground mines.

3. In the cutting head itself by a built in pressure intensifier. A drum intensifier eliminates the need for a high pressure rotating seal and there are also additional advantages when a phasing system is required which will be discussed later.

WATER SUPPLY

To save water and energy, high pressure water should be provided to the bits only when they are cutting. Distribution of water limited to bits which are cutting may be obtained by several ways.

1. One valve per bit actuated by the cutting reaction. However, when the water pressure is high, a valve actuated by the cutting effort of a bit may not open if the material is soft or the cut of depth shallow, two cases where dust and or risk of ignition are still present when no water is provided.
2. One valve per sector actuated by cam action.
3. A phasing system.

None of these two solutions is really satisfactory when they have to distribute high pressure water.

The optimum solution for shearer loaders and continuous miners is to supply the machine and the cutting head with low pressure water which is technically feasible with reliable components and to intensify the pressure in the drum. Under the assumption that water jet assist reduce the forces on the bit, the energy saved is available to intensify the pressure in the drum itself.

The solution of pressure intensification in the cutting head could be implemented with a built-in pump and a phasing system. A drum intensifier can also provide the possibility of limiting the supply of water to the cutting bits and better regulate the supply of water in accordance with the needs, thus saving water and energy.

Considering a water jet directed at the tip of the bit, the maximum efficiency of the water jet is obtained when the depth of the groove created by the jet is equal to the depth of cut of the cutting tool. In any given material this groove depth is function of the water pressure whereas the groove width is more dependent on the nozzle diameter.

When a shearer advances along a longwall in solid coal (Figure 1), the depth of cut of drum bits is a sine function of the rate of advance per rotation as represented by the curve 21 of Figure 2. The drum intensifier should provide water at a pressure regulated accordingly. Figure 3 illustrates the solution for a shearer drum fixed to a planetary gearbox including the usual low pressure water supply.

The pressure intensifier is composed of independent cylinders, one for each sector, six being a practical minimum. These cylinders are parallel to the axis of the drum and rotate with the drum. They are actuated by a stationary cam. If the cam is planar as represented which is the simplest way to built it, the flow will vary as the depth of cut but the jet pressure which depends on the square of the flow will vary as represented on curve 20 (Figure 2) which could still be satisfactory. The cam design can be adapted to have similarity of the two curves at the cost of some complication of the intensifier. Regulating the jet pressure as represented results in a energy consumption by the intensifier equal to 1/2 of what it will be if the drum intensifier was a pump with a sectorial distribution on a 180^{o} sector and one quarter of what it will be without sectorial distribution. The flow is limited to 602 of the flow with constant pressure on a 180^{o} sector, or 30% of the flow with a constant flow and pressure on the drum periphery.

The importance of this reduction of the power requirement is increased by the fact that the intensifiers create an axial force on the drum which translates in a lateral force on the drum ball bearings system.

If the cylinder were connected as they are in a pump, it would be better to orient them radially, a classical solution which requires a phasing system.

The sectorial distribution has to be shifted in accordance to the cutting direction because shearers work in two opposite directions. In the proposed solution, on a shearer drum, the cam can be fixed to the cowl and the movement of the cowl when the shearer changes direction will move the cam and change the water distribution accordingly.

If we consider a shearer with a drum speeds of 48 and 68 rpm, a water flow per drum of 20-28 gpm for the 48-68 rpm drum speed can be obtained by seven cylinders with 50 mm (2") diameter and 114 mm (4.5") stroke. The nozzle diameter, if the bit number is 42, will be .75 mm to provide 420 or 840 bars (6000 or 12000 psi) according to the gear ratio.

The solution of drum intensifier is applicable to continuous miner by using a similar approach. There is two important differences. The water supply to the drum requires a rotating seal of large diameter (a problem easier to solve under low water pressure). The distribution system does not need to be modified during the operation. Figure 4 shows that the optimum distribution is used during sumping which is important because sumping operations generate more than 70% of the respirable dust.

Figure 5 shows how a continuous miner can be retrofitted with water jet assist cutting. The design include a 54 bit lacing, 18 cylinders, 6 for each drum section, supplying water for 3 nozzles each. With a flow of 76 l/min (20 gpm) the energy for intensifying the pressure to a peak of 700 bars (10000 psi) is only 55 kw with a 0.8 efficiency coefficient.

However there is some inconvenience with the proposed system but this inconvenience is limited to lack of flexibility in the bit lacing design and to the unavoidable increase of complexity.

The bit lacing is not as flexible as it is with a normal drum. Bits should be theoretically located along a generatrix for each sector. Each sector should have the same number of bits to obtain the same flow and pressure per bit. Figure 6 shows such a bit lacing.

A drum intensifier can be very simple and the increase of complexity due to water jet assist cutting can be compensated by a decrease of the maintenance cost of the continuous miner.

The proposed system will be easy to retrofit on many existing machines by changing only their drums with a limited investment.

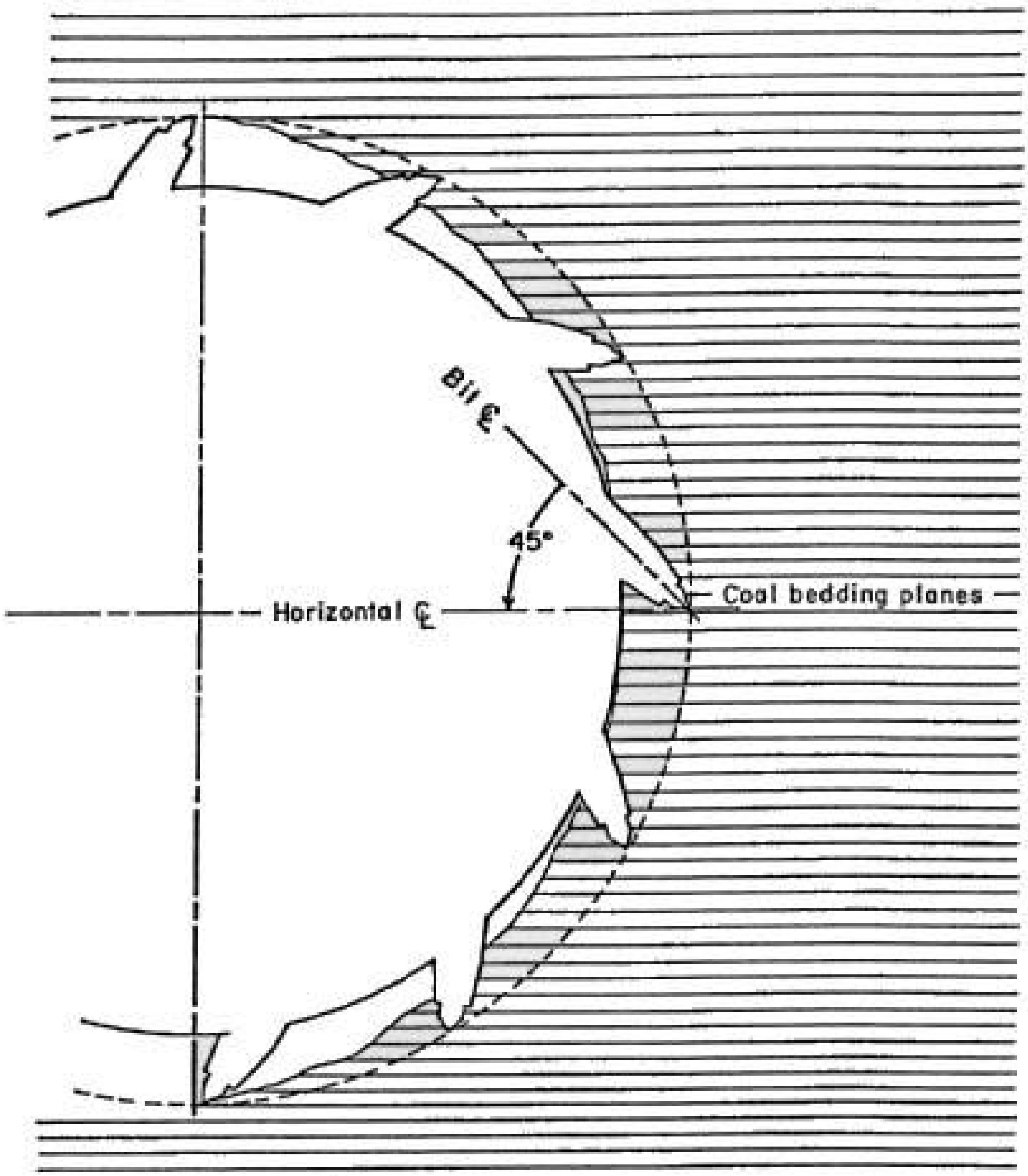
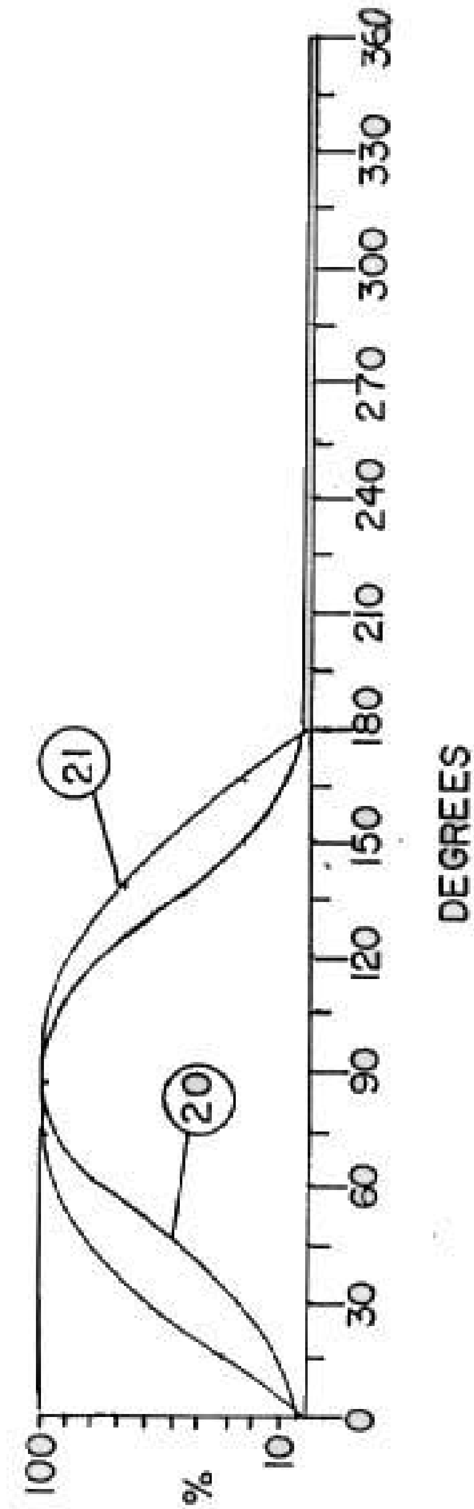


Figure 1: Bit Path at the Coal Face for Rotary Drum Head Mining Machine



100
%
10
0
30
60
90
120
150
180
210
240
270
300
330
360
DEGREES
20
21

Figure 2

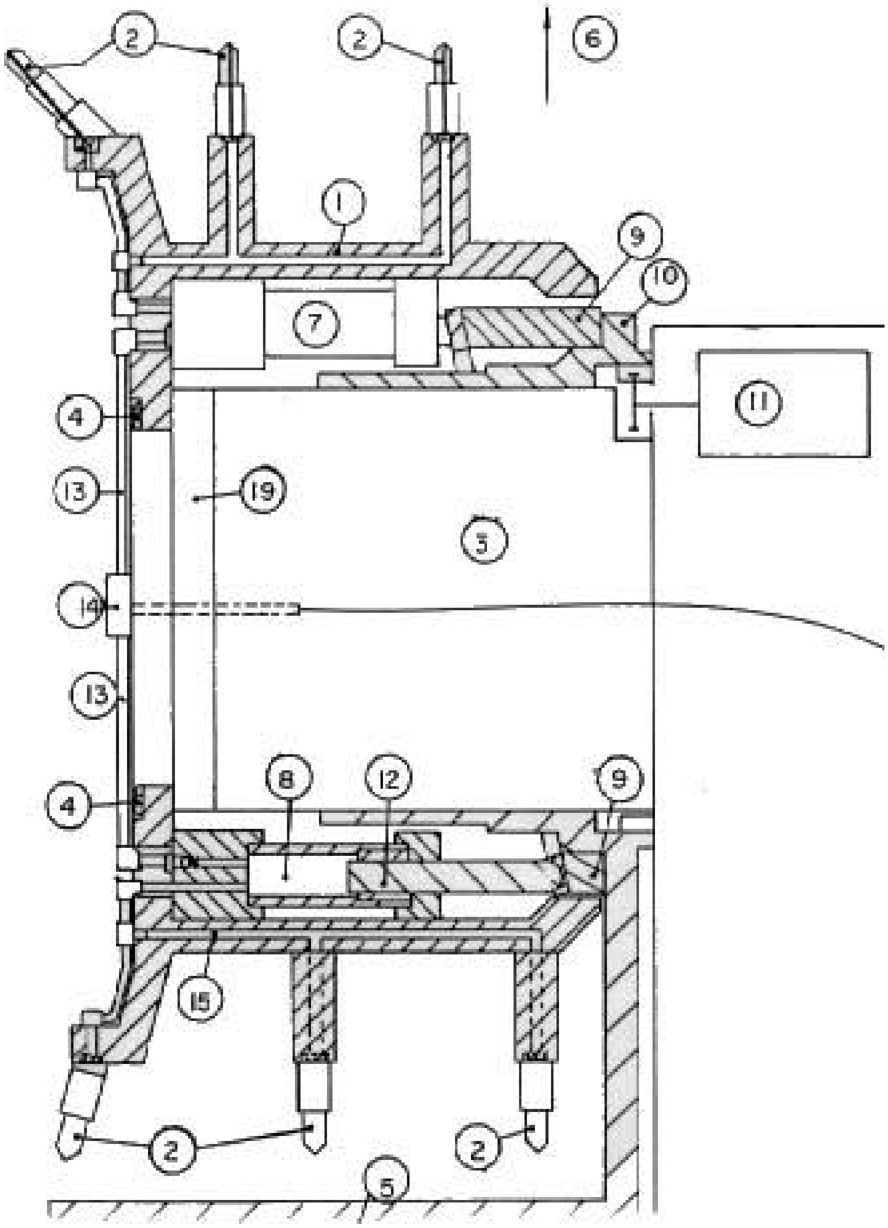


Figure 3

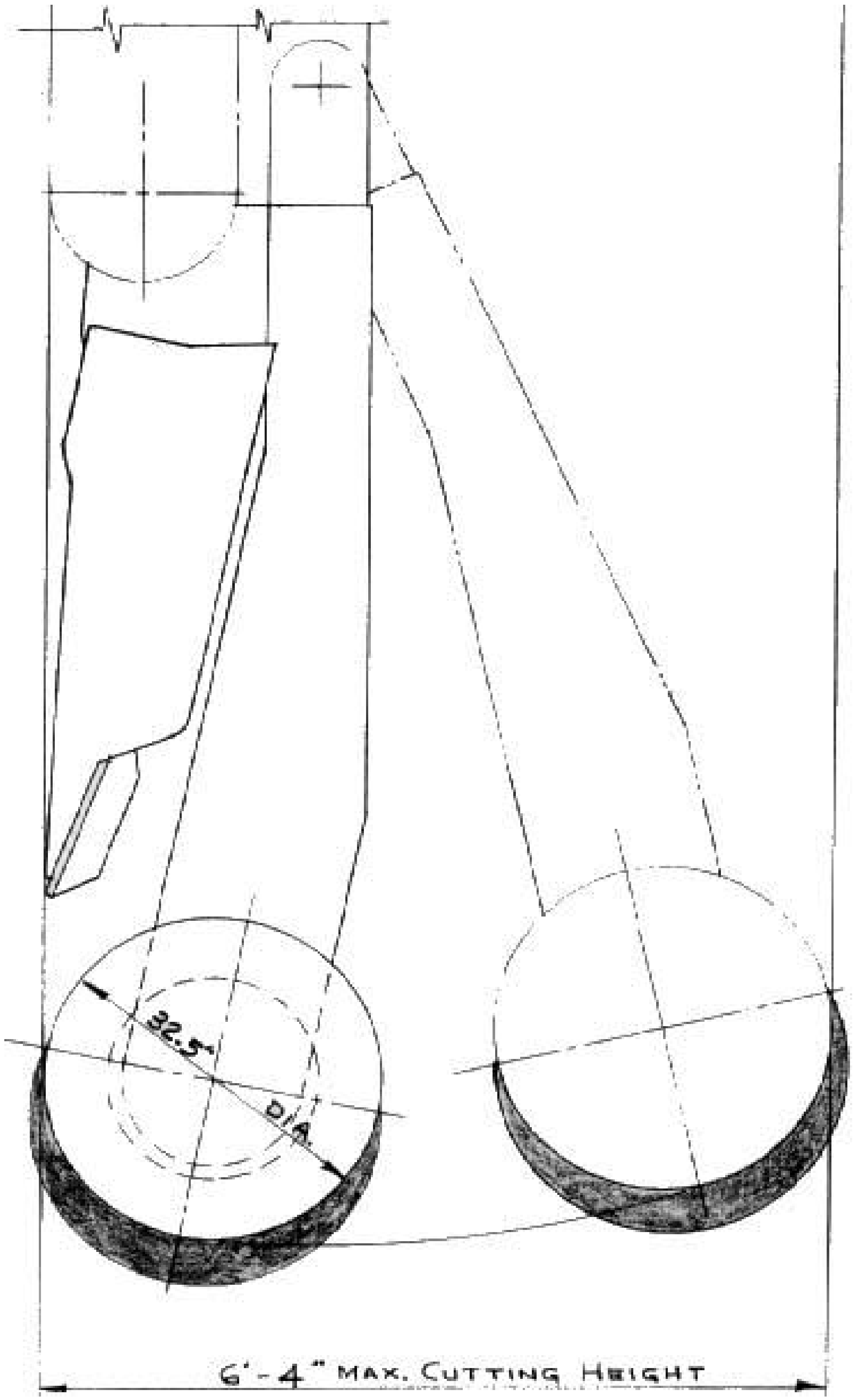


Figure 4

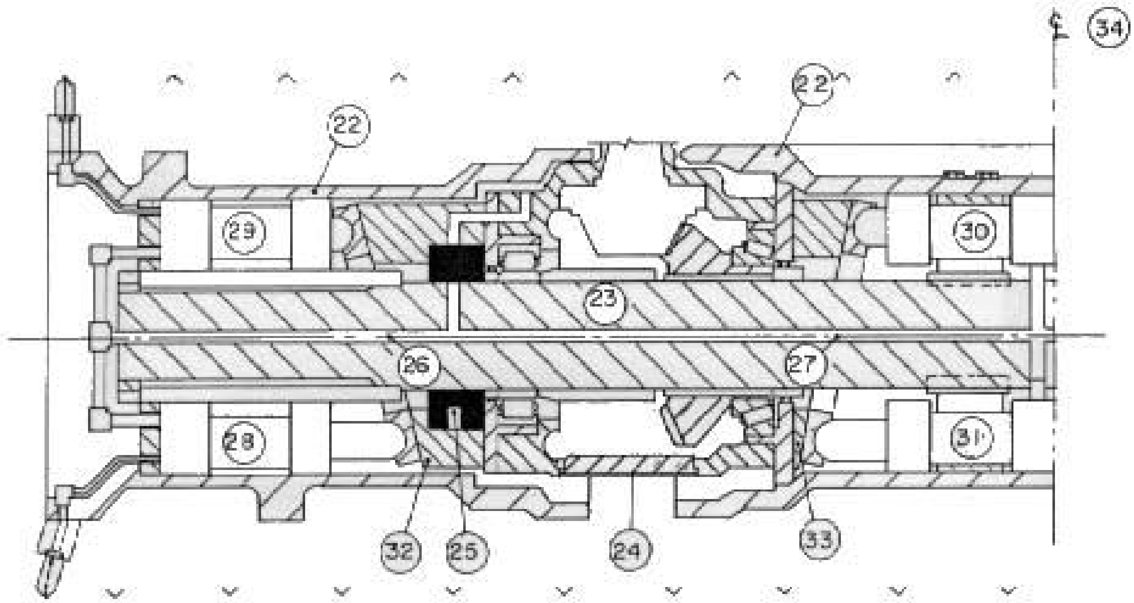


Figure 5

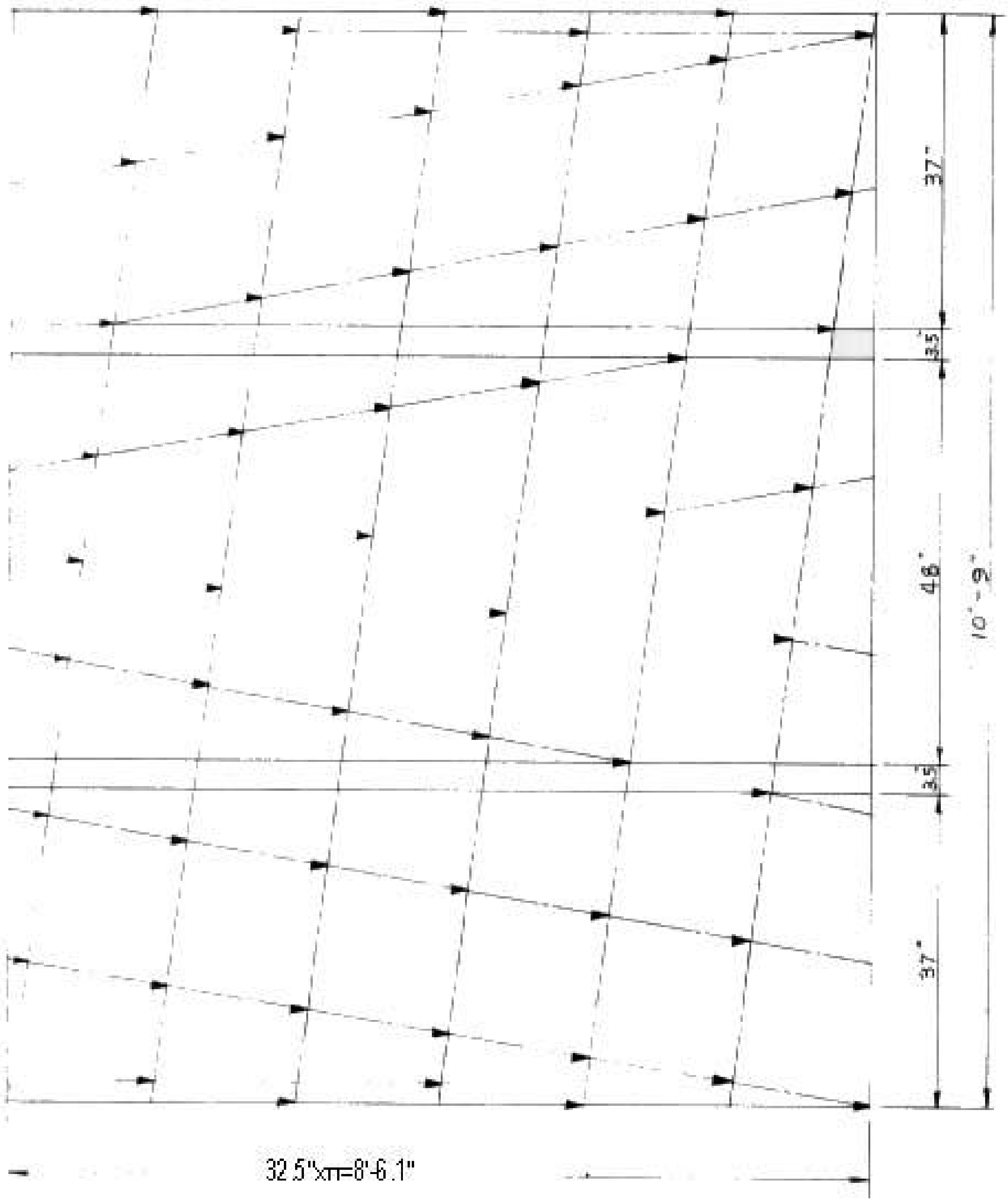


Figure 6

SAFER AND MORE PRODUCTIVE MINING AND TUNNELING WITH COMPUTER-CONTROLLED, WATER-JET-ASSISTED ROADHEADERS AND SHEARERS

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ABSTRACT

This paper describes recent European and North American developments in water-jet-assisted and fully automated mining and tunneling systems.

High-pressure water jets increase the cutting capability (materials of higher uniaxial compressive strength can be cut) and the rate of production (tpm) in mining and tunneling. The use of high-pressure water jets affords considerable cost savings through increased bit life, additional benefits from the reduction of dust levels and methane ignition hazards.

Mining and tunneling with high-pressure water systems is a new technology. This new technology is the first significant development since the introduction of electrically powered mining machines in the 1930's and 1940's. The design of safe and cost-effective systems that combine water-jetting with microprocessor controls is the objective of special research efforts.

European engineers and manufacturers have successfully tested and produced equipment with new guidance and monitoring systems. These systems permit fully automated mining in which the operator uses an LCD screen to monitor the progress of a fully automated machine. This technology permits the operator to be located away from the face in a clean, dust-free, and safe environment.

Further developments include a new electronic diagnostic system. This feature detects potential breakdowns in electrical systems, hydraulic systems, and high-pressure water systems, before the breakdowns occur. The use of this technology yields benefits in machine availability and shift production.

Both management and labor benefit from the improved safety, working environment, and increased productivity made possible by computer-controlled and water-jet-assisted mining and tunneling equipment.

INTRODUCTION

Mining and tunneling with water-jet-assisted cutting systems is a new technology. Many experts believe this new cutting technology is the first significant development since the introduction of electrically powered underground mining machines in the 1930's and 1940's.

Although several high-pressure, water-jet-assisted machines are already in use in Europe, none have been installed in U.S. mines. The intent of this presentation is to familiarize American mining and tunneling engineers with this new technique, one which permits safer, more productive and more cost effective excavation of rock and minerals than other cutting methods.

This paper is divided into four chapters: (1) Standardization and safety, (2) Classification and selection criteria for water-jet-assisted cutting systems, (3) Roadheaders, shearers and continuous miners, and (4) Computer-controlled, water-jet-assisted machines.

STANDARDIZATION AND SAFETY

Standardization of Components

Water-jet-assisted cutting of rock and minerals is quite new. Production type water-jet-assisted roadheaders have been in use in underground mines for only two years (Ref. 1, Evans, et al., 1984). A coal shearer with MHP is being tested by the U.S. Bureau of Mines in Bruceton, Pennsylvania (Ref. 2, Evans et al., 1984).

In 1984 ETE Corp. of State College, Pennsylvania, a subsidiary of Eickhoff of Germany, was awarded a U.S. Bureau of Mines contract for development and testing of an MHP coal shearer. This machine is presently being developed by Eickhoff. It will later be tested in a U.S. coal mine (Ref. 3, USBM Contract No. J0145039, 1984).

In the U.S. some companies are working on LP and MP water-jet-assisted dust suppression systems for standard drum-type continuous miners and shearers. Also, several North American, European and Japanese companies are developing water-jetassisted drilling equipment.

Most of the aforementioned cutting and drilling equipment is in the prototype and pre-production stages. Every equipment manufacturer uses different components for its water-jetting system. One may expect that water-jet-assisted cutting and drilling will be widely used by the mining and construction industries within this decade.

The diversity of water-jetting system components presents the industry with the issue of standardization. Before a large population of machines is in the field, the components for water-jetting systems should be standardized. A successful standardization effort is certain to reduce spare parts costs, and maintenance and supply problems.

The author strongly recommends standardizing sizes (flow volume) and mounting dimensions (done with electrical and hydraulic motors), rotary water seals, nozzles and nozzle holders, tool holders (blocks), bits (picks) and bit-force-actuated water valves.

The manufacturers must get to work on standardization now, while the population of water-jet-assisted equipment is very small. Already, we are seeing a multitude of designs and sizes of the aforementioned components. If the manufacturers do not standardize their components soon, high cost and maintenance problems will be certain to slow down the introduction of water-jetting systems.

Safety

High pressure water systems are potentially dangerous if not properly maintained and operated. If people are hurt or fatalities occur through water jetting, this promising new technique will be slow in gaining acceptance. Mine operators, contractors, unions and insurance companies will be concerned about safety and liabilities.

Let us consider the following solutions to potential safety problems:

Standardization of Controls and Safety Devices

The control panels of all water-jetting equipment should be standardized. On each roadheader, shearer, continuous miner and drill the main control levers and buttons, deadmen switches, and emergency shut-offs should be in the same location at the operator's station and be actuated identically.

Likewise safety devices such as period of start-up delays, as well as start-up signals (e.g. warning horns), and optical and electronic safety barriers should be standardized.

We must strive to achieve the situation in which an operator trained on any water-jet-assisted machine will be familiar with the controls and safety devices of any other water-jetting equipment, regardless of where the equipment is manufactured.

Regulations for Water-Jetting

Water-jetting equipment has great potential to increase safety in mines and tunnels by reducing dust levels and preventing methane ignition.

However, the proper authorities such as MSHA and OSHA must establish guidelines and regulations for safe design and operation of water-jet-assisted machines. Again, identical standards worldwide seem advisable. American and European authorities have been cooperating in the design, manufacture and testing of water-jet-assisted mining machines. Hence our common sense tells us that the authorities of the Free World's major mining countries must get together and work out a uniform set of rules and regulations for water-jetting equipment. When electricity was introduced to mines, political and communication problems did not permit the standardization that we may achieve now. As a minimum, North America, Europe and Japan should be able to agree on common standards.

CLASSIFICATION AND SELECTION CRITERIA FOR WATER-JET-ASSISTED CUTTING SYSTEMS

Classification of Water Pressures

Water-jet-assisted cutting systems are discussed with respect to the following pressure classification:

	Water Pressure	
Classification	Bar	PSI (rounded)
Low pressure (LP)	0-5	0-70
Medium pressure (MP)	5-200	70-3,000
Moderately high pressure (MHP)	200-1400	3,000-20,000
Very high pressure (VHP)	1400-4000	20,000-60,000
Ultra high pressure (UHP)	Over 4000	Over 60,000

The following paragraphs are for the benefit of potential users of water-jet assisted equipment, such as mine operators and contractors. The systems and components recommended have proven themselves in underground operation and in long-term tests in simulated mining applications.

Water Pressure versus Flow Rate

Theoretically the most effective combination for water-jet-assisted cutting is an extremely high water pressure at a very small flow rate. However, this ideal condition cannot be achieved with presently available components such as rotary water seals (phased and non-phased), valves, pumps and filtration systems. In the following paragraphs, applications of machines with various pressure ranges are discussed.

Very High Pressure (VHP) and Ultra High Pressure (UHP) Systems

In the early 1970's VHP (1400-4000 bar, 20,000-60,000 psi) systems were tested both in North America (Ref. 4, Wang et al., 1978) and Germany (Ref. 5, Henneke, 1978). The systems were tested with VHP water-jet-assisted tunnel boring machines ("moles") with disk-type roller cutters and with drag bits. No production machines were spawned by these early tests because of the unreliability and short lifetime of rotary water seals, pressure-intensifiers (water over oil), nozzles and other components. Both the Robbins Company of Kent, Washington, and Wirth of Erkelenz, Germany, ceased work on VHP water-jet-assisted "moles".

No VHP roadheader shearer or continuous miner with drag bits was ever built and tested.

At present only one VHP roofbolting machine powered by a twin pressure intensifier (up to 4000 bar, 60,000 psi) and 0.06 micron filtration is being tested in a coal mine in Maryland. At the time this paper was written the underground tests were still going on. The equipment is not yet commercially available.

VHP and UHP water-jetting systems, powered by pressure-intensifiers, are job proven and reliable for cutting of Kevlar, shoe soles, diapers, cardboard, brake linings, graphite composites and cake in the controlled and clean environment of a factory.

However, it is doubtful that this sophisticated equipment will function in the hostile environment of mines and tunnels, operated and maintained by regular labor. The author knows from his over 20 years of mining experience that mining and tunneling equipment doesn't exactly get TLC.

Bringing 4000 bar (60,000 psi) water-jetting equipment underground at this time is like introducing 4160 volts motors at the beginning of electrification of mines in the 1930's.

Moderately High Pressure (MHP) Systems

Modern water-jet-assisted cutting and drilling machines utilize MHP water pressures in the range of 350 bar (5,000 psi) to 700 bar (10,000 psi). For cutting rock, the most effective pressure seems to be around 700 bar (10,000 psi). Cutting tests indicate that for coal cutting 400 bar (6,000 psi) is the best pressure (Ref. 6, Erhart, 1984).

Some research establishments and equipment manufacturers are presently experimenting with 700 to 1600 bar (10,000 - 20,000 psi) systems to increase the hard rock cutting capabilities of roadheaders.

Medium Pressure (MP) Systems

MP systems in the range of 5 to 200 bar (70 to 5,000 psi) are being used for dust suppression and for prevention of methane ignition. Water jets of this pressure range do not assist the rock cutting capability of a machine but they have shown significant increases in bit lifetime.

Low Pressure (LP) Systems

Since about 1972, low pressure pick-face flushing systems operating at approximately 4 bar (60 psi) have been available from British manufacturers of roadheaders. These LP systems produced a marginal reduction in dust. However, they did not improve rock cutting capabilities, nor did they sufficiently cool the bits to eliminate sparking and the attendant methane ignition risk (Ref. 7, Schenck, 1982).

Components of Water Jetting Systems

The most promising results for water-jet-assisted cutting of coal and rock were achieved with MHP systems. Hence let us review component selection for equipment in this pressure range.

Water Pumps versus Pressure-Intensifiers

Plunger-type triplex pumps are the preferred choice of manufacturers of waterjet-assisted roadheaders for rock cutting and of shearers for coal cutting.

It appears that triplex pumps are less complex and more reliable than pressure-intensifiers for use in the hostile environment of mining operations.

A typical pressure-intensifier is powered by a variable volume pressure compensated piston pump at an hydraulic oil pressure of approx., 200 bar (3,000 psi).

Such a hydraulic system has an efficiency of some 85 percent, i.e. 15 percent of the prime mover's energy is lost through friction and unwanted heat (See Fig. 1).

All modern mining machines of American, British and Russian design have been switched from high-pressure piston pumps to gear-type pumps that work at low hydraulic pressure, below 140 bar (2,000 psi). Field experience has demonstrated that low-pressure hydraulic systems with a gear-type pump are less susceptible to dirt and contamination and require less filtration of the working fluids (Ref. 8 anon., 1982).

There is a growing requirement for the use of flame-retardant hydraulic fluids, such as water-oil emulsions (Aquacent of Century Oil or Pyrogard of Mobil). Flame-retardant fluids require low-pressure systems.

The major benefits of low pressure systems are that they are generally more reliable, require less maintenance, and have longer lifetime than high-pressure systems.

The trend away from hydraulic drives can be seen in the following developments on mining machines: Hydraulic tram and gathering head drives on continuous miners were replaced with electric drives, and many manufacturers of coal shearers switched from hydraulic haulage drives to electric drives.

For the aforementioned reasons the majority of manufacturers of water-jet assisted cutting and drilling equipment uses plunger or ram-type pumps (See Fig. 2

Bit-Force-Actuated Valves

To keep the water consumption of water-jet-assisted cutting systems low, several European manufacturers developed medium-pressure (100-200 bar, 1,450 to 2,900 psi) bit-force-actuated systems in the late 1970's. These systems have a bit-force actuated valve in the water feed line of the bit block. The valve serves to reduce water consumption, which, when the valve is open, is about 40 to 60 l/min. (10 - 16 GPM). The bit-force actuated valve is only open while the bit is in contact with the rock or mineral (See Fig. 3).

Tests showed that the water flow adequately cooled what otherwise would have been red-hot rock particles and sparks behind the bit tip. The tests also showed an 84% reduction of respirable dust and 92% lower coarse dust.

Field trials at the Friedrich-Heinrich coal mine, Germany, showed that the valve system manufactured by VOEST-Alpine of Austria was too complicated to be reliable. The initial pair of cutter heads for the AM 100 Alpine Miner cost $114,000 and the bit-actuated valves (198 per machine) cost $295 each and lasted on average from two to four weeks. No increase in productivity or rock cutting capacity (rock strength) was reported for this medium pressure system, operating at about 150 bar (2,000 psi) (Ref. 7, Schenck, 1983).

Besides problems with cost and reliability, bit-force-actuated valves have the following disadvantages:

The valve opening forces, especially at higher water pressure, are so high that the valve might not open when cutting soft and friable materials (e.g. such as bituminous coal) which cannot provide sufficient reaction forces.

Sandvik Rock Tools of Sweden is presently developing a miniaturized bit-force actuated valve for water-jet-assisted cutting of coal and other soft materials. Tests with this miniature valve are proceeding well.

Another inherent problem of bit-force-actuated valves is the delay between the time the bit strikes the face and the valve opens. Thus, the bit is cutting dry for a short period before the water jet can hit the face. Ideally the water jet should strike the face in front of the bit to weaken and presplit the rock or mineral.

Rotary Water Seals

We differentiate between phased (segmented) and non-phased rotary water seals. Phased rotary seals hold the water consumption down because the water jets are only activated in the section of the head which is actually cutting.

The rotary water seal is probably the most important component of a water-jetassisted cutting system.

Roadheaders with in-line (milling-type) cutter heads can utilize non-phased rotary water seals although much water is wasted on jets aimed at non-cutting bits.

Only phased (segmented) rotary water seals are recommended for transverse (ripper-type) and drum-type cutter heads. This recommendation applies to continuous miners, shearers and ripper-type roadheaders (See Fig. 4).

Phased (segmented) rotary water seals are difficult and costly to manufacture and test. Research establishments in North America and Europe have been testing numerous designs of phased rotary water seals. So far none have had the reliability and required lifetime to be suitable for use on production-type mining machines. Some seals had no dry-running capability, and their lifetime was only a few minutes.

In 1984, Eickhoff of Germany introduced a new roadheader with a phased (segmented) rotary water seal. This ripper-type ET-160-K roadheader has been operating successfully since September 1984 in a German coal mine, where it is used for driving entries in coal and rock (See Fig. 5),

A phased rotary water seal, designed for 1000 bar (14,500 psi) for use on an Eickhoff coal shearer has been tested successfully for over 400 hours This phased (segmented) rotary water seal is suitable for use on shearers, roadheaders and drum-type continuous miners.

The location and serviceability of water seals is of great importance. The seal should be easily accessible. It must not be located inside the cutter head transmission where a seal failure would cause destruction of an expensive transmission through water leakage.

A well designed rotary water seal has replaceable sealing elements. These must be checked or replaced at regular service intervals. For this reason the rotary water seal must not be buried deep inside the cutter head. A simple seal change must not require pulling off the heavy cutting drum (See Fig. 5).

Nozzles for Water-Jet-Assisted Cutting

The nozzle diameter depends on required flow rate and water pressure. At present water-jet-assisted machines with drag-type cutter bits, nozzle diameters of 0.6 mm (0.024 in), 0.8 mm (0.031 in) and 1.0 mm (0.039 in) are common.

Stand-off distances vary between 25 and 100 mm (1 to 4 in). The water-jet impinges the face 1 to 2 mm ahead of the bit tip.

Stainless steel nozzles are being used for LP and MP applications. Nozzles made out of tungsten-carbide and synthetic sapphire and ruby are common in MHP applications. VHP and UHP water-jetting systems utilize sapphire and ruby nozzles.

Although sapphire nozzles cost far more than tungsten-carbide nozzles, their use seems to be more economical. Their longer lifetime requires fewer labor intensive nozzle changes and thus reduces a mining machine's downtime.

At 700 bar (10,000 psi) water pressure tungsten-carbide nozzles last approx. 200 hours and sapphire nozzles 1000 hours.

Bits (picks) for Water-Jet-Assisted Cutting

Cutter bits are the most costly consumable item in machine excavation with continuous miners, shearers and roadheaders. This is especially true for roadheader excavation of hard and abrasive rock and minerals, and for operation of continuous miners and shearers in coal seams with hard inclusions and rock bands. High bit consumption may raise the cost of extraction ($ per ton) to such high levels that mining operations become uneconomical.

Water-jet-assisted cutting has afforded significant reductions in bit consumption. Nevertheless, careful selection of bit types is required in order to utilize the great cost saving potential of water-jet-assisted cutting of rock and minerals.

Three basic combinations of bit and nozzle location are possible.

Nozzle is Integral Part of Tungsten-Carbide Bit Tip

The ideal location of the water-jetting nozzle is in the tip of a cutter bit. (Ref. 9, Roepke, 1981). This new cutting tool is depicted in Figure 6.

This design offers the following advantages:

1. Short stand-off distance of the jet for the lowest specific energy consumption and the most effective cutting rate.
2. The jet strikes the crushing zone directly for maximum dust reduction at lowest water consumption.
3. Reduction of normal cutting force and cutting torque through lubricating effects of water on the bit, and through the hydrostatic presplitting effects of water.
4. Most effective cutting assistance and dust suppression for depths of cut (DOC) exceeding 3 mm (1/8 in).
5. Significantly increased bit life through cooling of both carbide bit tip and steel shank (See. Fig. 8).
6. This design permits higher peripheral bit tip speeds since the water jet hits the face directly. The jet does not have to penetrate the rock or mineral layer in front of the bit tip as with other water jetting systems.

The disadvantages of this new system are:

1. It has neither been tested in the lab nor in the field on actual mining machines.
2. It is only suitable for conical, point-attack bits. It will be difficult and expensive to seal the stationary water supply line against the bore of the rotating bit.
3. The nozzle is the bit tip, thus it might plug up. The nozzle orifice may be destroyed through the backwash of excavated particles in abrasive or hard materials.

The author believes that the advantages of Roepke's water-jet-assisted bit outweigh it's disadvantages, and that this promising "wet" tool should be tested on a mining machine.

<u>Bits with Internal Water Routing</u>

Bits with internal water routing, where the water jet emerges in front or behind the bit tip, are commonly known as "through flush" bits. This system is mainly available for forward-attack (tangential) and radial bits. These bits are mostly used on coal shearers in Europe and in a few installations in the U.S. These "wet" tools are manufactured by the British companies Hoy, Padley & Venables and Green & Bingham (See Fig. 7).

The advantages of the "through-flush" concept over shearers with standard water sprays are longer bit lifetime and improved safety. As summarized in a 1982 report to the Mine Safety and Health Administration (MSHA), the main points were:

1. Longer tool life from internal cooling of the bit shank and carbide Lip. (See Fig. 8).
2. Low water flow rate and low pressure at nozzle.
3. Reduces dust levels and the absence of frictional ignition of methane.
4. Small, low pressure water pump.

The "through-flush" wethead system uses approximately 2 l/min. per bit (1/2 GPM) with water at about 20 bar (300 psi) pressure from the pump, and 6 bar (90 psi) at the nozzle. Measurements by MSHA personnel in October, 1982 confirmed a dust reduction of over 90% with the new "wet" shearer (average dust: after 0.8 mg/m3; before 10 mg/m3). The tests were conducted at a site in West Virginia (Ref. 7, Schenck, 1983).

The disadvantages of the "through-flush" system are:

1. A "wet" bit costs approximately five times more than a standard longwall bit, due to its complex design (Ref. 10, Wilson, 1984), It is doubtful if in coal cutting applications, especially in the soft and friable bituminous coal seams in the U.S., these higher cost bits can compete against standard bits.
2. The "through-flush" blocks and bits have rather large dimensions. This might cause difficulties in fitting them on the constraint space available at cutter heads of continuous miners and roadheaders.
3. The highest reported pressure at the nozzle of "through-flush" bits is 35 bar (500 psi). This is too low for water-jet-assisted cutting (Ref. 11, anon., 1984).
4. The jet impinges the face at a greater distance than 1 to 2 mm in front of the bit tip, the most effective range for water-jet-assisted cutting.
5. The nozzle is located close to the face where it could be destroyed through backwash of hard and abrasive particles.

<u>Water-Jet-Assisted Cutting with Standard Bits</u>

All water-jet-assisted mining machines in use in underground mines are tooled with standard bits, which are commercially available from numerous manufacturers. Both conical point-attack bits and forward-attack (tangential) and radial picks are being used. MHP water pressures in the range of 350 to 700 bar (5,000 to 10,000 psi) are common on water-jet-assisted roadheaders. However, no records exist to show that 350 bar (5,000 psi) water jets actually provide cutting assistance in both coal and rock.

This system depicted in Figure 9 offers the following advantages:

1. Standard bits (picks) and blocks (tool holders) can be used.
2. There are no space restrictions for the nozzle assembly. Therefore mass-produced, commercially available low-cost nozzles (tungsten-carbide and sapphire) can be utilized.
3. Water jets can be properly aimed to impinge 1 to 2 mm in front of the bit tip.
4. Nozzles are mounted on surface of cutter head or on vanes of cutting drums.
5. There is no need to disconnect the water supply line when changing bits, and there are no sealing problems between the water supply line and the bit.
6. No destruction of nozzles takes place through "backwash" of hard and abrasive cuttings. The relatively longer stand-off distance (in most applications approx. 100 mm (4 in)) affords this benefit.

The disadvantages of water-jet-assisted cutting with standard bits are:

1. High specific energy consumption due to large stand-off distance results in higher power requirement (kW, HP) for the water pump drive motor.

2. Higher water consumption (l/min., GPM) is needed for higher pressure due to a longer stand-off distance.
3. Reduced depth of cut (DOC) from the water-jet having to penetrate the layer of rock or mineral in front of the bit tip. This reduced depth of cut is of no significance when cutting coal and other soft and easily penetrable materials (Ref. 12, Summers, 1984). However, when cutting harder rock and minerals, say in excess of 70 MPa (10,000 psi) uniaxial compressive strength, the reduced DOC becomes very detrimental. Hard rock combined with a large DOC and a long stand-off distance might render the water-jet-assist system ineffective.

WATER-JET-ASSISTED ROADHEADERS, SHEARERS AND CONTINUOUS MINERS

Principal advantages of water-jet assisted cutting relative to standard cutting are higher production rates, lower dust levels, elimination of methane ignition from sparking (friction), and reduced fines. When installed on rock and coal miners, water jets mean that relatively smaller, lighter and lower-cost machines can be purchased. If properly applied, the combined effect is lower cost and significantly improved health and safety in mining and tunneling operations.

Water-Jet-Assisted Roadheaders (Boom-Type Continuous Miners)

The first water-jet-assisted roadheader, supplied by AEC, Inc., was tested in 1975 at the Bruceton Center for Coal Mine Safety Research near Pittsburgh, Pennsylvania. A light, ripper-type roadheader (10 tonnes with 33kW, 44HP cutter motor) was equipped with a 700 bar (10,000 psi) water monitor with a flow rate of 80 l/min. (21 GPM). The non-rotating water jets were aimed ahead of conical point-attack bits. This early water-jet-assisted cutting system doubled the roadheader's rate of production and produced a 70% reduction of respirable dust to levels less than 2 mg/m^3 (Ref. 13, McNary et al., 1976) (See Fig.10).

In October 1978 a collaboration agreement was entered by the U.S. Department of Energy (DOE) and the Mining Research and Development Establishment (MRDE) of the British National Coal Board (NCB) to design and develop a water-jet-assisted system for a standard roadheader (Ref. 14, Tomlin, 1981). A standard Dosco Mk 2A roadheader, equipped with an in-line (milling-type) cutter head, was retrofitted with a 700 bar (10,000 psi) water-jet-assisted system with a flow rate of 4 l/min. (1 Imp. gallon/min.) per bit. The machine, equipped with a 48.5 kW (65 HP) cutter motor, succeeded in cutting limestone with a uniaxial compressive strength of 108-137 MPa (15,560-19,920 psi).

Based on these successful underground tests by British and U.S. Government agencies, several European manufacturers commenced manufacture of water-jet assisted roadheaders. At present roadheaders, equipped with up to 700 bar (10,000 psi) MHP water-jet-assist systems, are available from United Kingdom (Anderson-Strathclyde and Dosco) and Germany (Eickhoff and Paurat) (See Fig. 11).

Results of Field-Trials of Pre-Production Roadheaders

The following is a summary of results of underground operations with roadheaders employed in driving of drifts and entries in both coal and rock (Ref. 15, anon., 1985).

1. Cutting rate (tpm) increased.
2. Specific energy requirement (kWh/ton) reduced.
3. Bit (pick) consumption greatly reduced.
4. Unwanted dust was minimized.
5. Machine vibrations due to transient cutting loads minimized (should reduce maintenance cost).
6. No frictional sparking evident.
7. A significant improvement in cutting performance observed in hard rock strata up to 165 MPa (24,000 psi) uniaxial compressive strength.

Reduced Bit (Pick) Consumption

Cutting tests in coal and in rock have shown three to ten times longer bit life for water-jet-assisted cutting when compared to standard "dry" cutting method using external water sprays only. The following table shows the potential for bit cost savings with water jetting.

Excavated Material	Typical Compressive MPa	Uniaxial Strength PSI	Typical Bit Cost for "Dry" Cutting $/ton	Bit Cost for WJ-Assisted Cutting High Range* $/ton	Low Range** $/ton
Clay (Hardpan)	4	600	0.06	0.02	0.01
Bitum. Coal	20	3,000	0.10	0.03	0.01
Uranium Ore	35	5,000	0.25	0.08	0.03
Concrete	41	6,000	0.50	0.17	0.05
Shale	50	7,000	0.30	0.10	0.03
Sandy Shale	70	10,000	1.00	0.33	0.10
Sandstone	83	12,000	1.50	0.50	0.15
Copper Ore	179	26,000	4.00	1.33	0.40

* at 3 times longer bit life
** at 10 times longer bit life

By applying these cost figures to a mining company producing five million tons of bituminous coal per year, the annual savings in bit cost would be in the range of $350,000 to $ 450,000.

A contractor cutting 200,000 tons of concrete per year would save $66,000 to $90,000.

A copper mining company excavating annually some 200,000 tons of hard copper ore could reduce its bit cost by $534,000 to $720,000. The potential savings in bit cost would quickly pay for a modern water-jet-assisted roadheader.

Experts forecast that water-jet-assisted cutting should reduce overall cost of tunnel excavation per meter of advance by up to 40% (Ref. 16, Ropchan et al, 1980).

Increased Application Range for Roadheaders

The advent of water-jet-assisted cutting will increase the strength of rock that boom-type continuous miners will cut to 175 MPa (25,000 psi) when the heaviest boom miners are equipped for water-jet-assist systems.

About 70% of all tunnels and probably a higher percentage of all underground mines are in rock below or just reaching the 175 MPa (25,000 psi) economic limit indicated for water-jet assisted cutting roadheaders. The increased capability of roadheaders could mean a market loss for conventional drill and blast underground excavation and for tunnel boring machines (Ref. 7, Schenck, 1983).

Water-Jet-Assisted Shearers

In July 1984, the U.S. Bureau of Mines awarded to ETE Corporation, a wholly owned subsidiary of Eickhoff, Germany, a contract for design, manufacture and underground testing of a 200 to 700 bar (3,000 to 10,000 psi) water-jet-assisted shearer within 30 months (Ref. 3, USBM Contract No. J0145039, 1984). In order to minimize water consumption, the machine is to be equipped with either bit force-actuated water valves or a phasing system (segmented rotary water seal) (See Fig. 12).

The project is going well and is ahead of schedule. This "wet" shearer will be manufactured to Eickhoff's high quality standards. For this purpose, Eickhoff has been conducting long-term tests of components for the water-jetting system such as plunger pumps, bit-force-actuated valves and phased rotary water seals supplied by numerous manufacturers. It was found that many components did not live up to Eickhoff's requirements and quality standards and had to be redesigned or modified.

The following objectives were set for this challenging project:

Productivity through Safety

Higher production rate at lower dust levels and reduced methane ignition hazard.

Less Fines and Coarser Coal

The "wet" shearer has lower drum RPM's at a higher torque for increased depth of cut (DOC) and wider bit spacing. The "big bite" bits and the slotting action of the MHP water jets will increase lump size and reduce amount of fines, thus producing a more valuable product. If the coal is later cleaned to reduce sulfur and ash, such cleaning will be less expensive because of the reduced fines.

Reduced Bit Consumption

Significant savings in bit cost are expected for this new water-jet-assisted coal shearer.

Reduced Maintenance Cost

Several underground trials with water-jet-assisted machines have demonstrated reduced cutting vibration. It is expected that the smoother running "wet" shearer will have lower maintenance cost than a standard "dry" shearer.

Longwalling is, in many applications, already the most cost effective extraction method. It is expected that the new water-jet-assisted shearer will make longwalling even more economical than it already is.

Water-Jet-Assisted Drum-Type Continuous Miners

Room-and-Pillar mining with drum-type continuous miners is the dominant underground coal extraction method in USA, Australia and South Africa.

At present, approx. 65% of the U.S. coal production from underground mines comes from continuous miner installations. The underground coal production share for continuous miners is expected to increase to 75% by 1990.

Although the continuous miner is the major coal extraction machine in America, no drum-miner has so far been equipped with a functioning water-jet-assisted cutting system. The reasons for this seem to be:

1. Only American companies manufacture drum-miners. No foreign competitor forces Americans to improve their mature designs.
2. The design of the cutter transmissions and cutter heads (drums) are not adaptable to current MHP water jet technology. With the present head designs, either a large diameter rotary water seal or a water seal buried deep inside the cutting drum would be required (See Fig. 5). Both sealing versions were tried and did not function satisfactorily even at low water pressures.
3.

At present, two model LN 265 continuous miners have been equipped with low pressure (LP) water systems exclusively for dust suppression and prevention of frictional methane ignition. These LP systems are not intended for water-jet assisted cutting. Both projects are conducted with U.S. Bureau of Mines participation.

LN265 Drum-Miner with Phased LP Water System

In early 1985, a low pressure dust suppression and methane ignition prevention system was installed on a LN265 continuous miner in the South Mine of Pennsylvania Mines Corporation near Ebensburg, Pennsylvania.

Water pressure:	14 bar (200 psi)
Flow rate:	53-76 l/min. (14-20 GPM)
Water seal, Number:	One only in R.H. drum (hollow shaft)

Type:	French-made, spring-loaded ceramic seal
Phasing:	120^o segment
Operating Time:	After twelve 8-hour shifts seal had to be replaced due to cracking of ceramic coating.
Cutting drum, Diameter:	825mm (32.5 in.)
RPM:	74
Cutter bits, Type:	Conical point-attack
Model:	Kennametal U43
Number:	60 bits
Nozzles, Location:	Behind bits
Number:	One nozzle per bit

The "wet" drum cut well in coal and showed good dust reduction. It bounced when cutting roof rock because original head lacing was reduced from 90 to 60 bits.

LN265 with Non-Phased LP Water System

Another second dust suppression and methane ignition prevention test will be conducted in 1985 at a mine of Eastern Associated Coal.

Drum miner:	LN265
Water pressure:	min. 10 bar (250 psi)
Flow rate:	1.9 l/min. (1/2 GPM) per bit
Rotary water seal:	Non-phased, British-made

Combination of Water-Jet-Assisted Cutting and Drilling

A combined MHP water-jet-assisted drum-miner with sumping cutter head and a fully automated water-jet-assisted drilling and roof bolting could prove a cost effective equipment combination. This extraction system would have a high machine utilization due to its simultaneous mining and roofbolting capability. Furthermore, the expandable-shell, Atlas Copco "Swellex" bolt (Ref. 17, anon., 1982) would use MHP water from the "wet" continuous miner for expansion of the bolt shell to provide high-speed roof bolting. This equipment combination has great potential for high capacity, low-cost mining in a safe, dust-free environment.

COMPUTER-CONTROLLED, WATER-JET-ASSISTED MINING AND TUNNELING MACHINES

Water-jet-assisted cutting is a cost-effective and safe extraction method. Its great potential should be matched and enhanced by modern control, guidance and machine health monitoring systems.

Conventional Machine Control and Monitoring

Operating a continuous miner or a roadheader is a repetitive task. The operator has to maximize the advance rate of the machine in a tunnel, drift or entry of a certain size and shape while maintaining line and grade. In most applications this task could be done faster and more accurately by a computer.

An additional task of the operator is to monitor the machine's mechanical, hydraulical, electrical and water-jetting systems. This is a very important task because machine downtime causes costly loss of production. For example lost coal production in an American longwall installation costs $125 per minute (Ref. 18, anon., 1985). Monitoring of the machine's systems takes both the operator's time and his attention, thus reducing his effeciency for production work. Monitoring systems can definitely be performed better and more cost-effectively by a computer.

Automated Machine Control, Guidance and Health Monitoring

Several European companies developed automated control, guidance and health monitoring systems for mining and tunneling machines (Ref. 19, Boldt et al., 1984). The system described in this paragraph was developed in 1981 by Eickhoff, Germany (Ref. 20, Weber, 1983) (See Fig. 13).

It is a computer-controlled system that is being used successfully in mining and tunneling applications. It has the following features:

Machine Guidance System

The machine, which need not be located in the centerline (axis) of the tunnel, is aligned for line, grade and cross-gradient by a laser beam which is aimed at a machine-mounted target equipped with phototransistor elements. Those elements signal the machine's deviation from the tunnel's axis to a microprocessor which calculates the machine's exact position and alignment.

Automatic Machine Control

An on-board computer is pre-programmed with the exact shape and dimensions of the tunnel's cross-section. The computer actuates via solenoid-valves the cutter boom for exact excavation of the cross-section. The outline of the cross section and the excavated portion of it is displayed on an LCD screen at the operator's station. The following cutter boom control cycles can be selected:

1. Automatic excavation of the complete cross-section.
2. Automatic excavation of the circumference of the cross-section.
3. The automatic excavation can at any time be manually overridden by the operator.

Tolerances and Cost Savings

The Eickhoff system has an accuracy of + 50 mm (1.97 in). Overbreak in an entry in a German coal mine costs approx. $16/m^3$. Additional costs are incurred when lagging and backfilling are also required (Ref. 19, Boldt, 1984). For a concrete-lined tunnel additional backfill costs approx. $100 per cubic yard of concrete in place. Therefore, the installation of a machine control and guidance system that reduces over-cutting can be amortized over a relative short tunnel length.

Monitoring System for Maintaining Machine in Good Condition

The monitoring and diagnostic system for maintaining machine in good condition detects potential breakdowns electronically before they occur increasing machine availability and production. A warning is flashed onto the LCD screen informing the operator which component or system is having problems. If the operator doesn't react, the machine shuts itself down automatically.

Remote Control

The operator can control the machine from a safe and dust-free location. This remote operation is being monitored on an LCD screen. Remote control is suggested for use with MHP water-jet-assisted cutting systems.

Shut-off Precaution for Water-Jetting System

The water-jetting system is only actuated when the machine is actually cutting. A sensor actuates via a solenoid valve the water supply when 70% of the cutter motor's nominal torque is reached. With this safety device the water jets cannot be turned on accidentally or be left on at idle.

Automated Computerized Mining Cycle

Contrary to its name, a drum-type continuous miner of present design does not even come close to continuous operation. In fact, a continuous miner in an American coal mine cuts and loads coal only 23.5% of the available shift time, while it is idle or delayed during nearly half of the shift. But not only continuous miners are under-utilized. Roofbolters are idle or delayed 63.9% of the available shift time (Ref. 21, Davis, 1980).

Both mining equipment and computer controls are commercially available in order to build a continuous mining system for simultaneous mining and roofbolting. Such a system is already in use with the AEC "Orebiter" in a potash mine in Saskatchewan (Ref. 22, anon., 1983).

A sumping-type drum miner or a roadheader, equipped with on-board automated roofbolters such as the Atlas Copco "Boltec", controlled by a computer and operated by one man only, has the potential to raise the productive time (cutting, loading and roofbolting) to 45% of the available shift time (Ref. 23, Kogelmann, 1982). Combined with the safety and cost effectiveness of water-jet-assisted cutting and drilling, including the use of "Swellex" bolts, this advanced system could double the productivity of a standard continuous mining section.

CONCLUSIONS

Use of water-jet-assisted cutting - up to 700 bar (10,000 psi) - has demonstrated its potential to provide the next major technical advance for excavation of rock and minerals. Extraction of coal with shearers and drum-type continuous miners, augmented with water-jets shows a high probability of increasing both productivity and safety in underground coal mining operations. Significantly lower bit consumption, coarser coal and less fines, combined with lower specific energy consumption (kWh/ton) will lower cost of extraction (less $ per ton of coal).

A sumping-type drum-miner, equipped with on-board automated roofbolter, controlled by a computer, and augmented by water-jetting has the potential to double productive time i.e. the profit production of a continuous miner section.

The most significant contribution of water jets to drifting and tunneling may be to increase the ability of roadheaders to cut rock 20% to 35 % stronger than the maximum strength of rock that can now be economically excavated by these machines. Water-jets may increase the rock cutting capacity of roadheaders up to about 175 MPa (25,000 psi) uniaxial compressive strength. If this happens, then boom-type continuous miners might be found suitable to meet the rock conditions in as much as 70% of all tunnels and mines (See Fig. 14).

Use of water jets on boom miners could also increase the productivity of machines now in use. Water-jet-assisted roadheaders have the potential to lower the overall cost of tunnel excavation by some 40%.

Computer-controlled guidance and control systems will increase production rates and reduce ground support cost due to reduced overbreak. Microprocessor controlled monitoring and diagnostic systems detect potential breakdowns before they occur thereby increasing machine availability and production. Remote control is suggested for water-jet-assisted machines.

A classification system for the various levels of water-pressure utilized for water-jetting equipment is proposed.

International standards should be adopted for design of the components of water-jetting equipment and, should be augmented by universal rules for training, operation and maintenance. With proper training and standardization water jetting will bring to the mining and tunneling industries "Higher productivity through safety".

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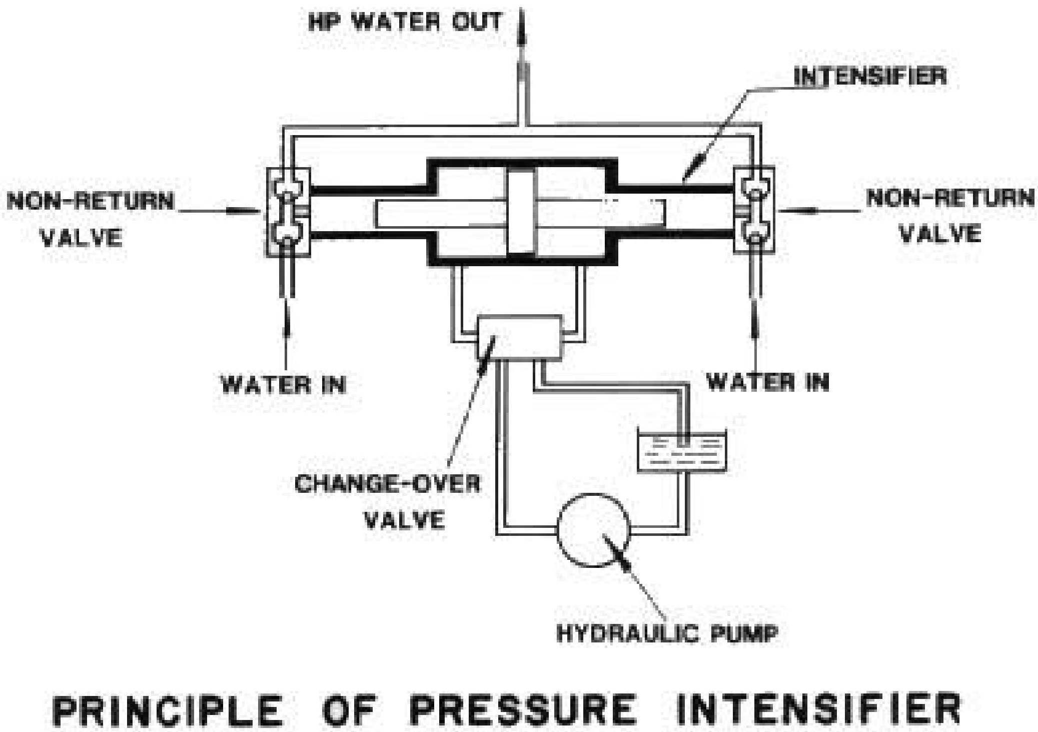


Fig. 1 Pressure intensifier (water over oil). For water-jet-assisted cutting and drilling machines twin-pressure-intensifiers, powered by pressure compensated hydraulic pumps are required.

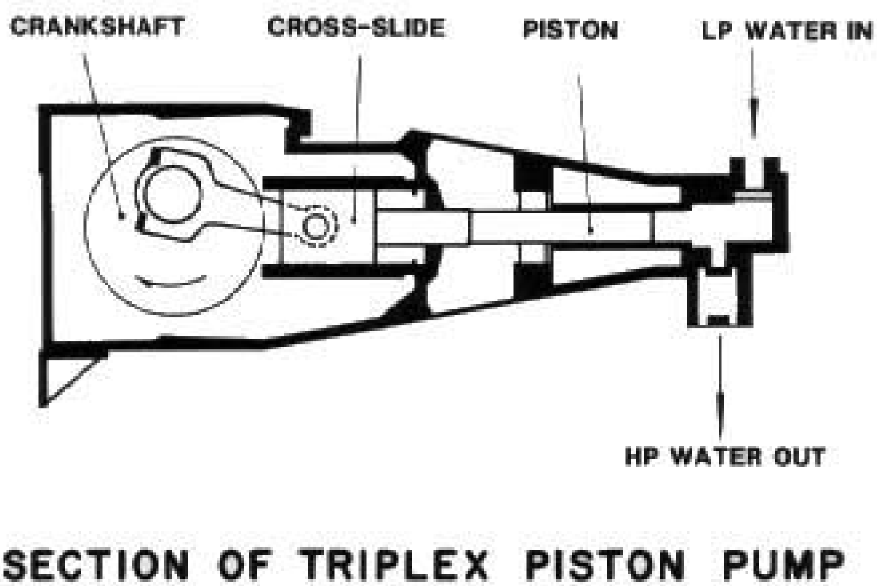


Fig. 2 Triplex plunger (piston) pumps are the preferred choice for water-jet-assisted cutting systems.

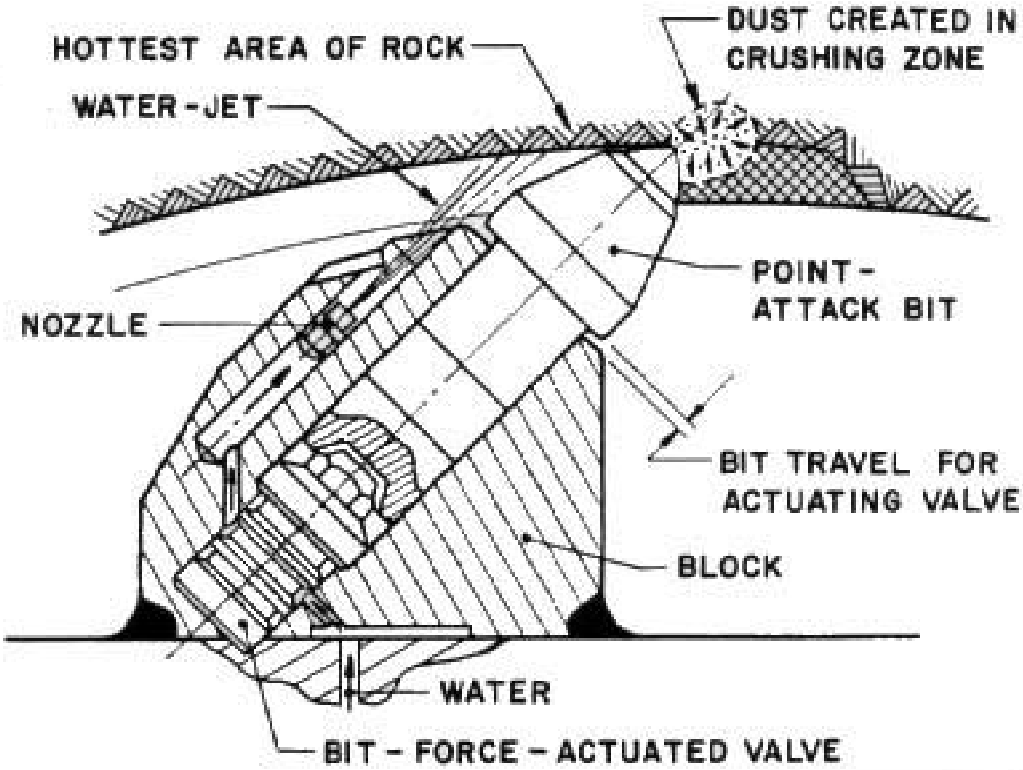


Fig. 3 Principle of European-made bit-force-actuated water phasing system.

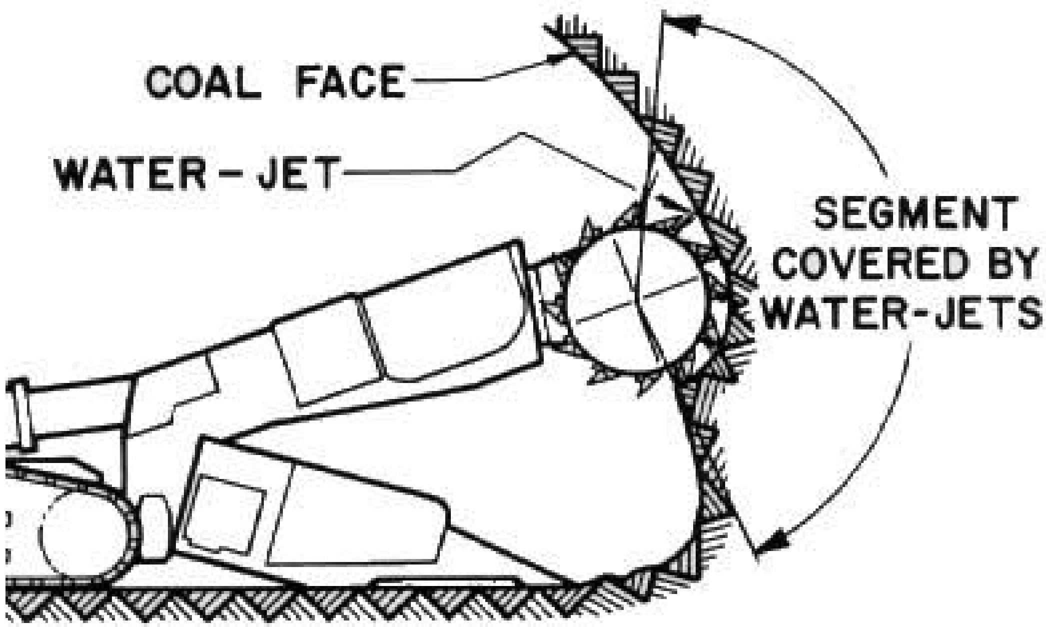
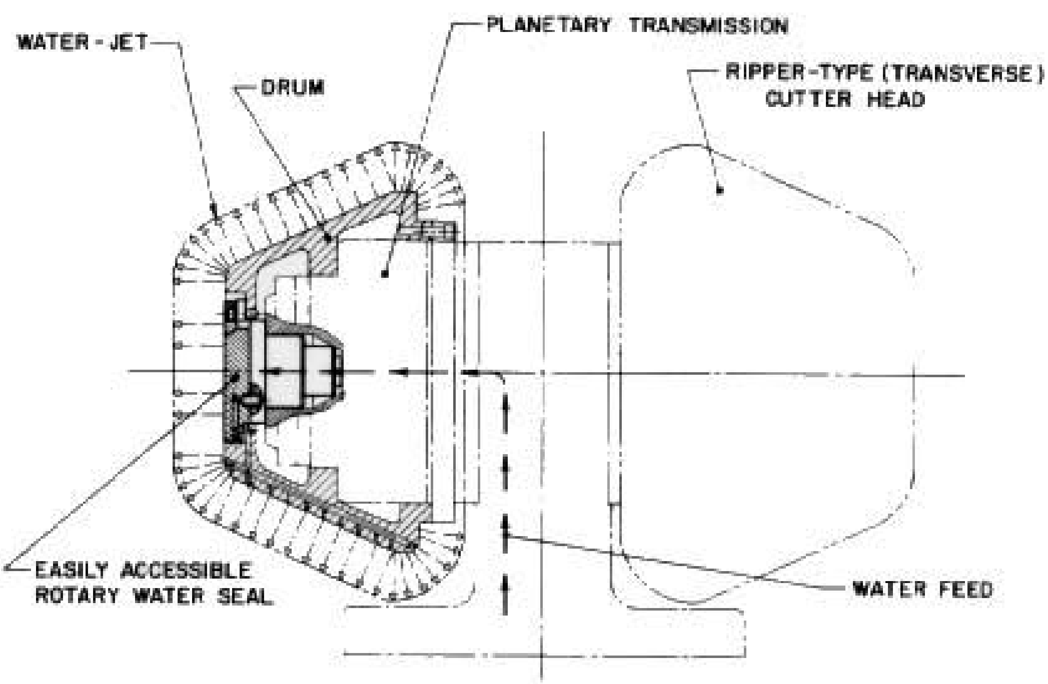
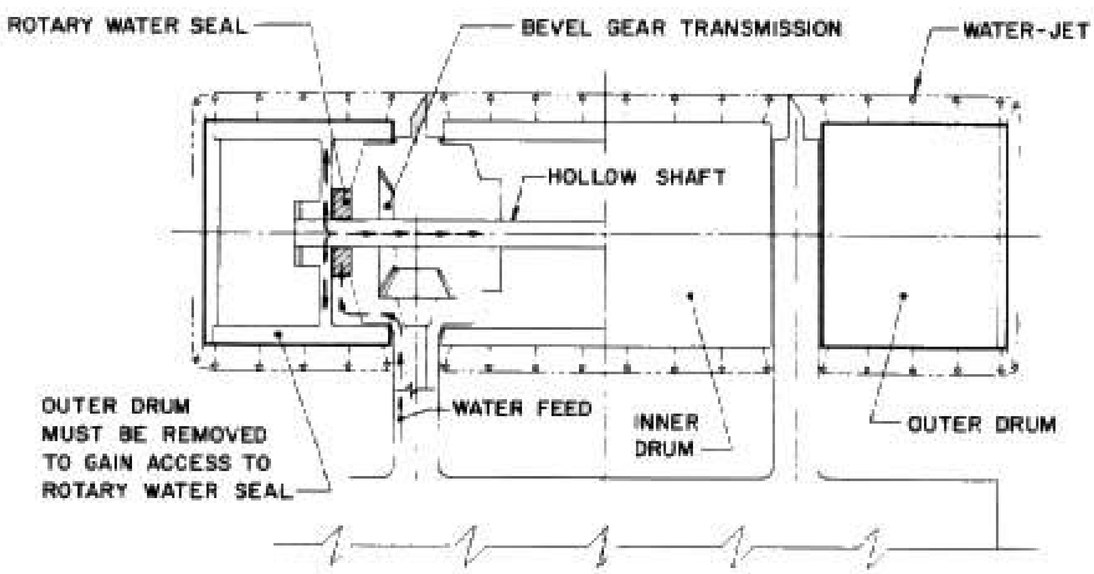


Fig. 4 Phased (segmented) rotary water seal for ripper-type (transverse) roadheaders and drum-type continuous miners reduce water consumption by directing jets only towards face.



PHASED (SEGMENTED) ROTARY WATER SEAL ASSEMBLY FOR EICKHOFF ET-160-K ROADHEADER



PHASED (SEGMENTED) ROTARY WATER SEAL ASSEMBLY FOR AMERICAN-MADE DRUM-TYPE CONTINUOUS MINER

Fig. 5

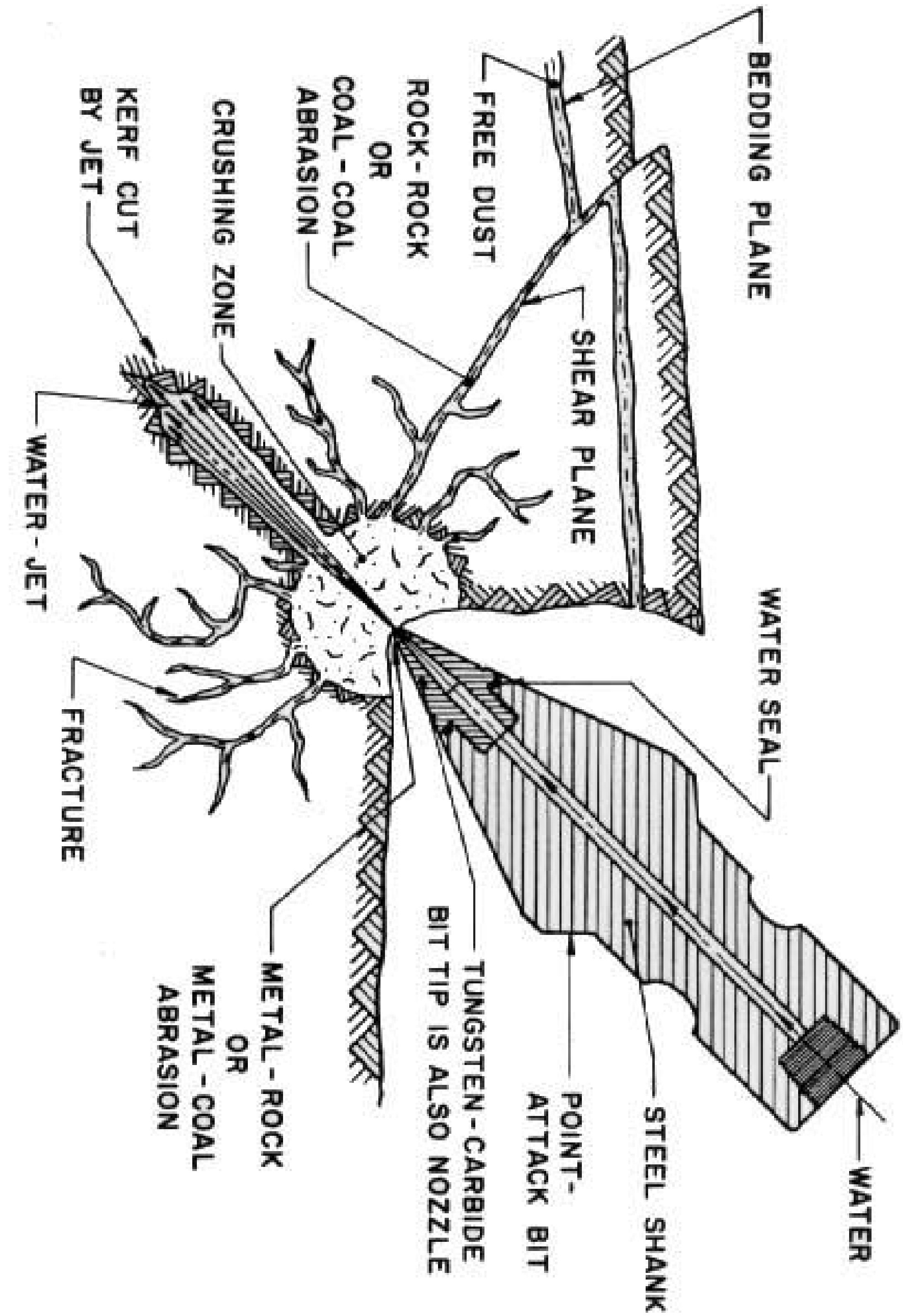


Fig. 6 Nozzle is integral part of the tungsten-carbide bit tip.

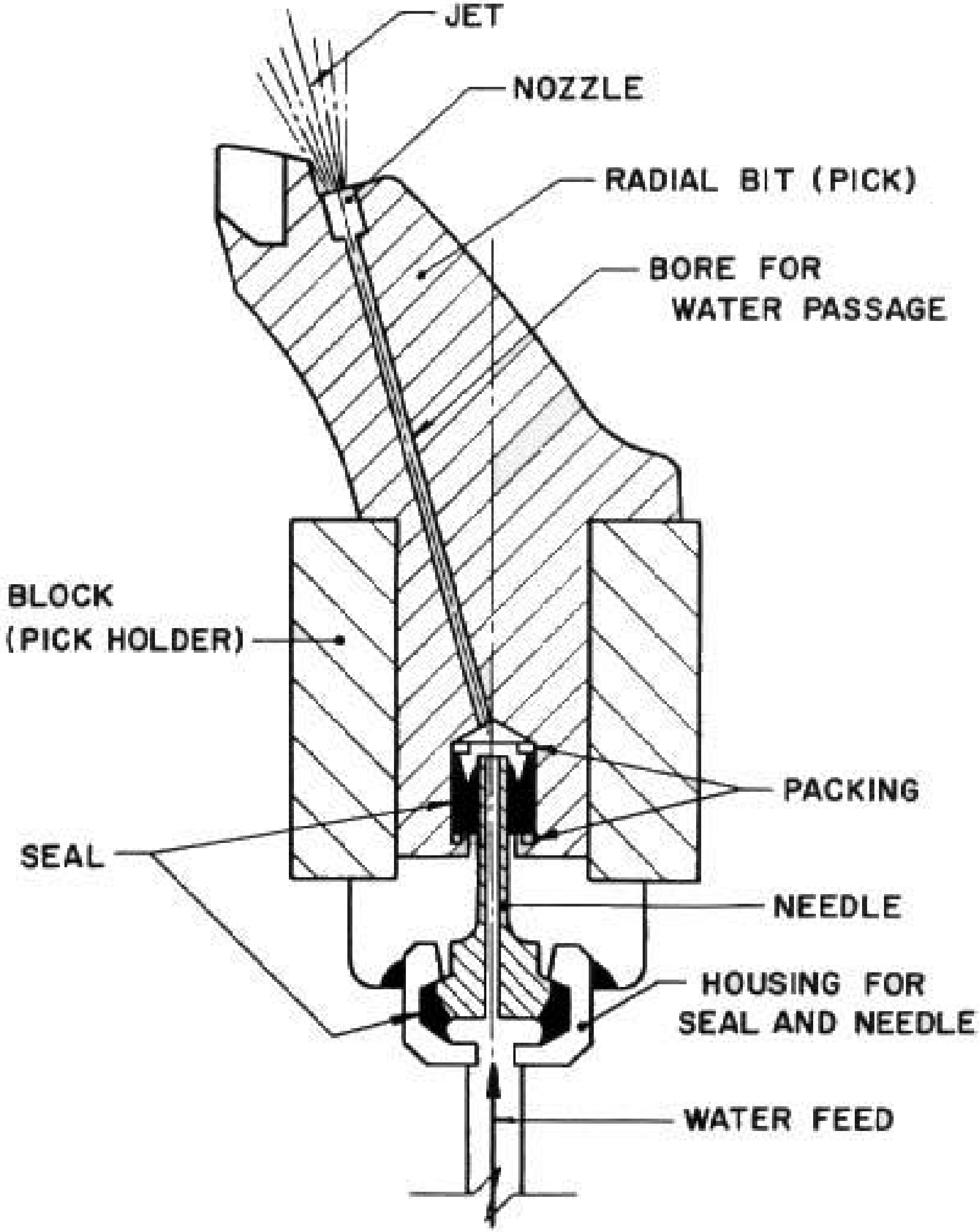


Fig. 7 Typical through-flush system.

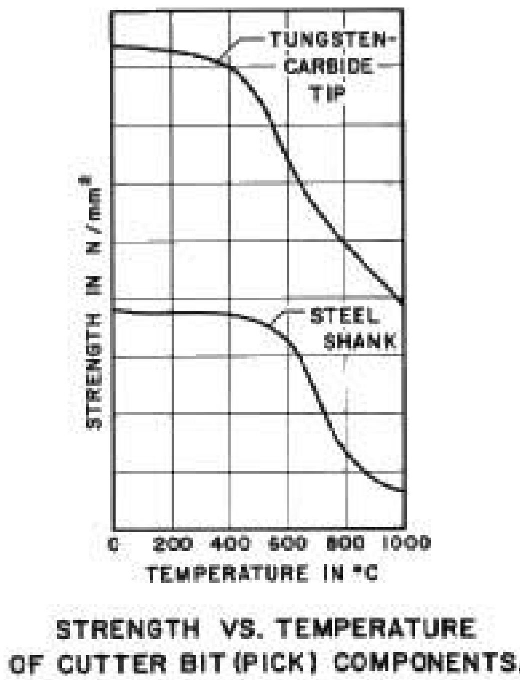


Fig. 8 Longer tool lifetime and reduced bit cost through water cooling of tungsten-carbide bit tip and steel shank.

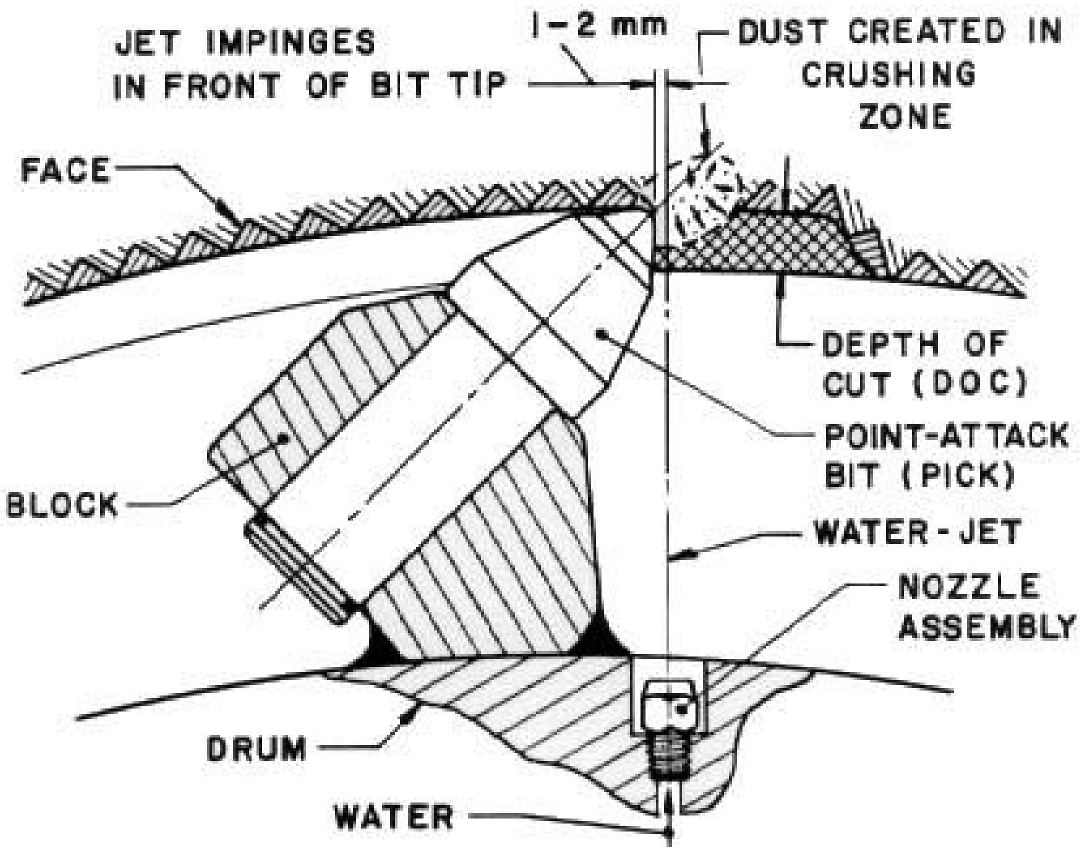


Fig. 9 Water-jet-assisted cutting system using standard bits.

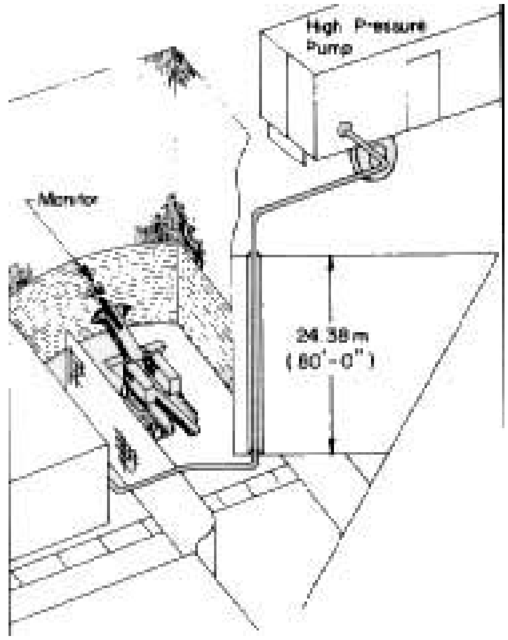


Fig. 10 The world's first water-jet assisted roadheader (700 bar, 10,000 psi) was tested in 1975 in the Bruceton, Pennsylvania coal mine of the U.S. Bureau of Mines.

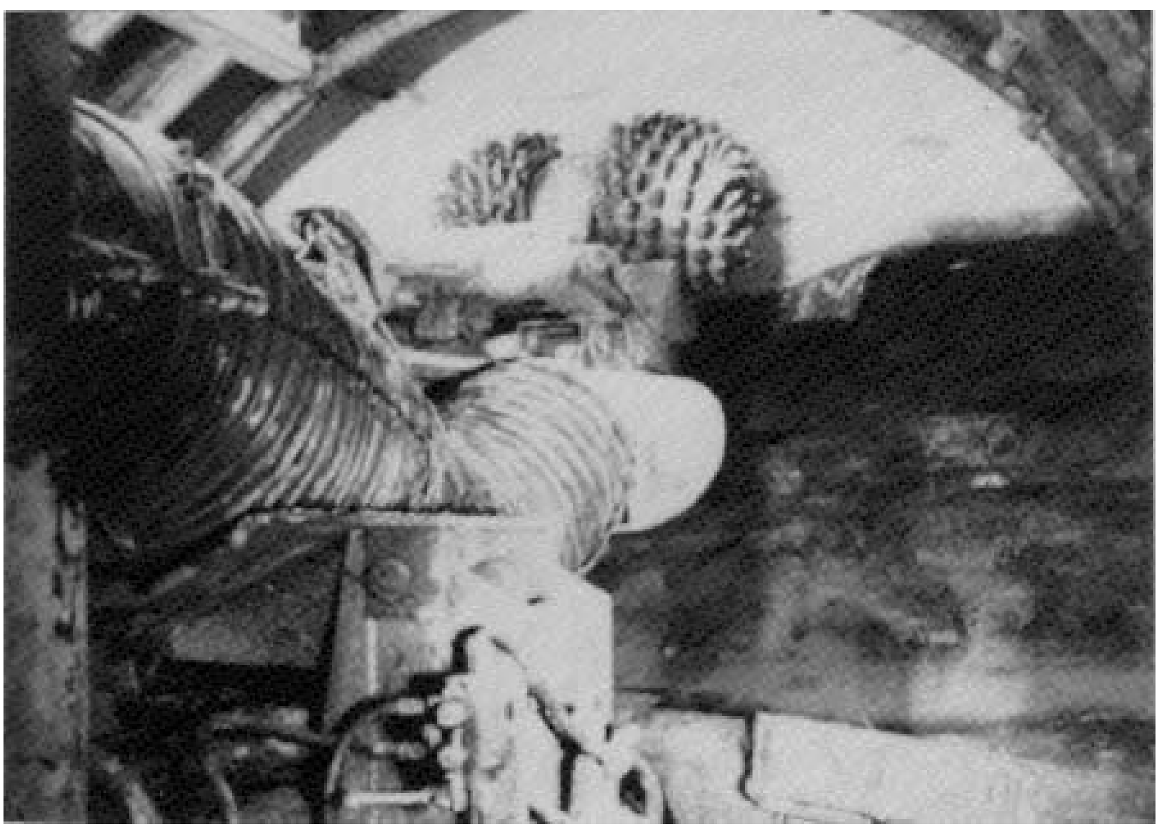


Fig. 11 Water-jet-assisted Eickhoff ET-160-K ripper-type (transverse cutter heads) roadheader driving an entry in coal and rock in a German coal mine.

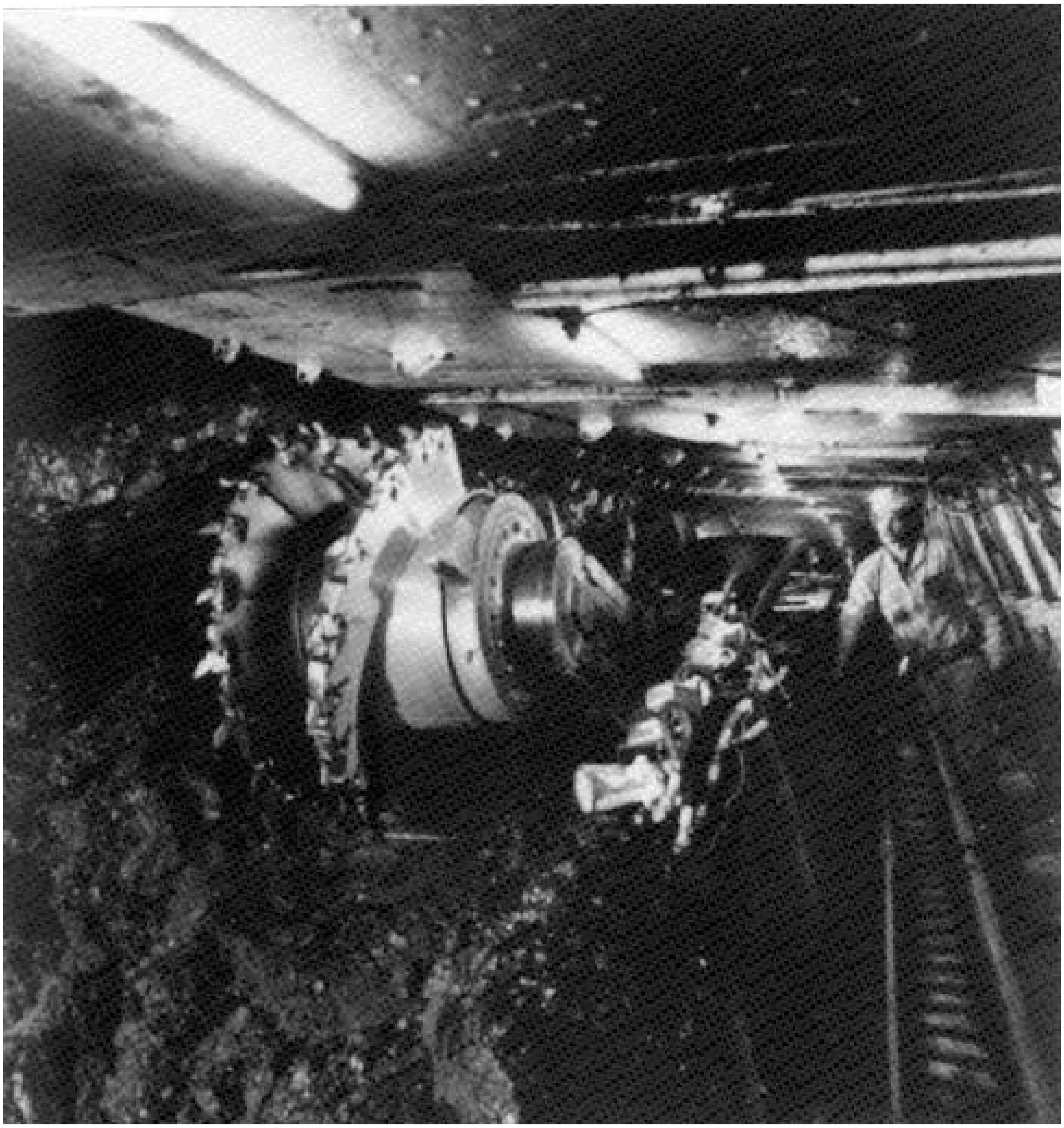


Fig. 12 HPW water-jet-assisted shearer (200 - 700 bar, 3,000 - 10,000 psi) with phase rotary water seal is being developed by Eickhoff, Germany for the U.S. Bureau of mines.

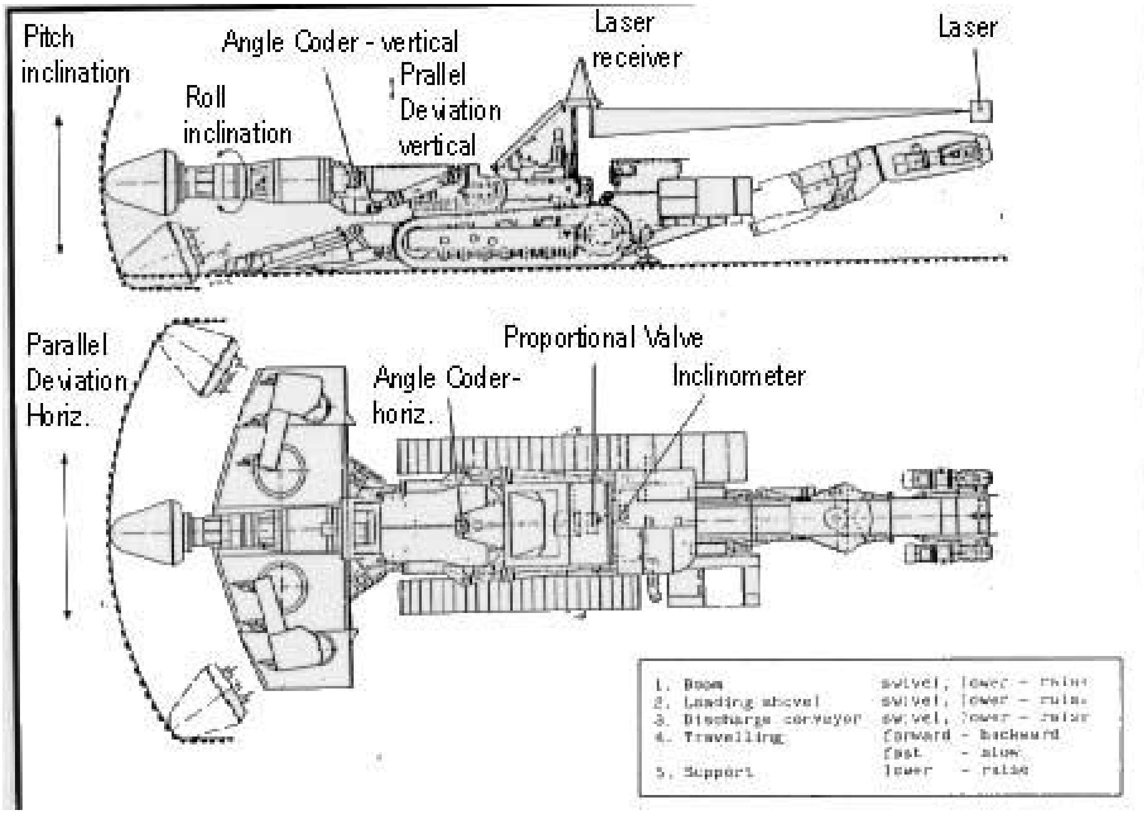
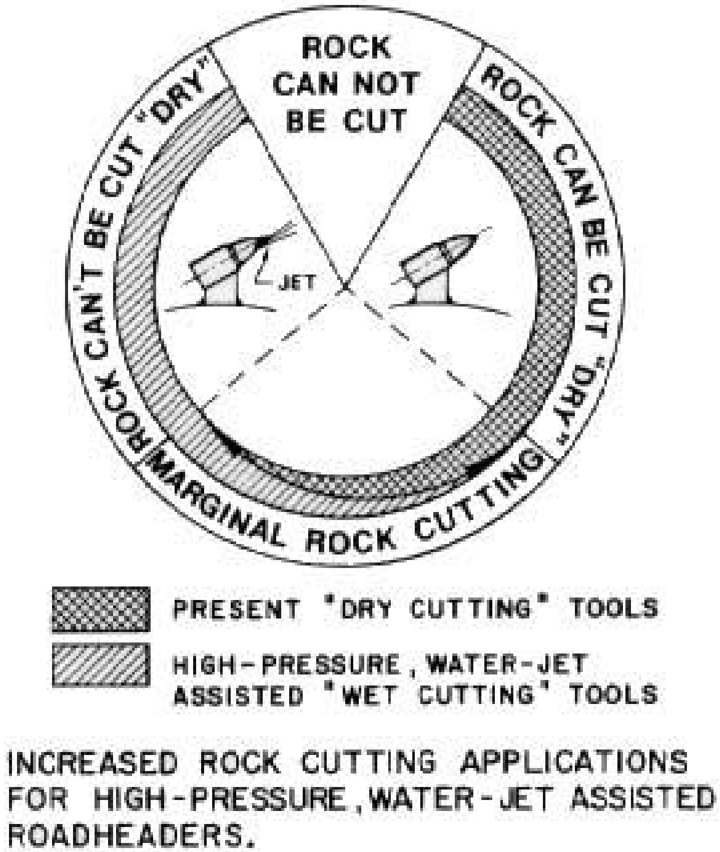


Fig. 13 The mining machine of the future: Computer-controlled alignment and profile control, machine health monitoring and diagnostic system of Eickhoff, Germany.



Cross hatching shows range of rock that can be cut by roadheaders without water-jets; rock that can be cut using water-jets shown by hachures.

Fig. 14 Water-jets have the potential to increase economical rock cutting capacity of roadheaders to 175 MPa (25,000 psi) uniaxial compressive strength. If this happens, then boom-type continuous miners will be able to meet the rock conditions in as much as 70% of all mines and tunnels.

AN OVERVIEW OF WATER-JET-ASSISTED ROCK CUTTING AT THE PITTSBURGH RESEARCH CENTER

By H. J. Handewith,[1] R. J. Evans,[2] and E. D. Thimons[3]

ABSTRACT

This paper presents an overview of the water-jet-assisted rock cutting research being conducted at the Pittsburgh Research Center. Findings indicate substantial reductions in airborne respirable dust using directed 3,000- to 9,000-psig water jets to assist mechanical cutting tools. In addition to dust reductions, there is a strong potential for:

- Longer cutting tool life attributed to jet lubrication and cooling of the cutting edge;
- Reduced tool forces at the same depth of cut attributed to jet lubrication, precleaning of the tool path, and lubrication and propagation of mechanically induced rock cracks;
- Reduced occurrence of frictional ignitions attributed to jet cooling of both the cutting tool and rock surface.

Research to date has established confidence in the concept of water-jet-assisted rock cutting, inferred some limitations, and highlighted specific areas requiring ongoing effort.

INTRODUCTION

Very high water-jet pressures, in the 20,000- to 60,000-psig range, have been used in rock cutting research for the past two decades. Based on the premise that water impacting at supersonic velocities will erode grooves in any rock, and that the jet cutting mechanism (water) is immune to wear, the potential for this exciting new high technology is only limited by one's imagination. The generation and distribution of such high fluid pressures have created an entirely new industry of fluid flow, fluid filtration, high-pressure intensification, static and dynamic sealing, strength of materials, and safety. Very high-pressure cutting with water jets has received wide acceptance in industry; however, the harsh underground mining environment has been slow to accept this innovative technology.

Ten years ago, the mining industry started to investigate the use of moderately high water-jet pressures in the 3,000- to 10,000-psig range to assist existing mechanical mining tools. Although lacking the glamour and excitement of "high tech- high pressure," the moderate jet-pressure concept offers more immediate underground implementation and requires less of a technological leap. Existing hydraulic systems are now rated up to 7,500-psig working pressure. Commercially proven triplex piston pumps, flexible hoses,

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fittings, filtration systems, static and dynamic seals, and safety are all within, or almost within, that which is proven and available today. In addition, such systems can be adapted to existing mining equipment with little impact on production.

The mechanisms of water-jet-assisted cutting, though poorly understood, have demonstrated the following advantages (ref. 1, 2, 3).

- Airborne respirable dust was reduced 60 to 90 pct;
- Reduced occurrence of frictional ignitions, attributed to jet cooling of both the cutting tool and rock surface;
- Longer cutting tool life, attributed to jet lubrication and cooling of the cutter edge;
- Pick cutting forces reduced by 20 to 50 pct in some materials, as compared with dry cutting at the same depth of cut. This is attributed to jet precleaning of the tool path, jet lubrication of the tool and/or rock interface, and water lubrication and propagation of mechanically induced rock cracks.

Figure 1 represents the range of water-jet pressures tested at the Bureau of Mines Pittsburgh Research Center (PRC). Water-Jet-Assisted Mechanical Cutting data were generated using the in-seam tester with water-jet pressures up to 9,000 psig (ref. 1, 4). Though testing has been conducted using jet pressures up to 40,000 psig at PRC, the Mechanical-Assisted Jet Cutting data were extrapolated from literature and normalized for a sandstone with an unconfined compressive strength of 10,000 psi (ref. 5, 6, 7). It is obvious that there is a salient rock-weakening effect (RWE) when the water jet is used to assist mechanical cutting. The question as to why the RWE decreases after reaching a maximum cutting force reduction of about 50 pct has not yet been answered. One proposed explanation is that with increased jet pressure, a point is reached where the water cuts a groove in the rock, providing a path for it to flow away from the cutting tool rock contact area, thereby reducing the lubrication and cooling effects. Addressing this question is a high-priority future research item at PRC. Many benefits of the very high-pressure (over 20,000 psig) water jets are also apparent for under 10,000-psig water-jet-assisted rock cutting range.

Benefits derived from use of the moderate-pressure, water-jet-assisted cutting tools are considerable, and research could easily result in additional benefits being discovered.

With this background, water-jet-assisted rock-cutting research at the PRC was initiated in 1981 by a U. S. Department of Energy (DOE) contract to Dr. M. Hood at the University of California at Berkeley. The development research was conducted under the guidance of an ad hoc committee chaired by DOE. This committee consisted of representatives from the University of California at Berkeley, Ingersoll-Rand Research Inc., Lee Norse, Kennametal, Inc., and Boeing Services International, Inc. An in-seam tester (fig. 2), developed under a separate contract of the Bureau of Mines (BOM) with Ingersoll-Rand Research, Inc., was acquired and used on a coalcrete block to evaluate a 3,000-psig-directed jet that interacted with a standard conical bit. Initial results proved promising, and a second test was conducted underground in the Safety Research Coal

Mine at Bruceton, PA. Though the results of both tests were inconclusive, they indicated a substantial potential for tool force and dust reductions when using the 3,000-psig jet assist.

In 1983, certain DOE functions were transferred to the Bureau of Mines. Since that time, the basic and developmental research programs have been expanded. All water-jet-assisted cutting research being conducted at PRC is under the Bureau's direction and is the subject of this paper.

OBJECTIVES

The general objectives of this expanded basic research are to investigate the application of 3,000- to 9,000-psig water-jet systems and to consider them for use with existing coal and/or rock mining machines. Specifically, this research is attempting to establish the optimum jet pressure, nozzle configuration, nozzle stand-off distance, water-jet-specific bit design and spacing, bit traverse rate, and depth of cut. Water-jet susceptibility is being evaluated for several coal measure rocks and coal seams. Methods of distributing the moderately high-pressure water through a rotating cutter head are being investigated, including rotary seals, phasing systems to supply pressurized water in the direction of cutting, and water filtration .

THE PROGRAM

Figure 3 outlines the water-jet-assisted rock-cutting research currently in effect at PRC. Principal elements consist of establishing the optimum working combinations, and parameters for cutting tool configurations, water-jets, and geologic materials.

The first phase is designed to identify the vast number of possible test combinations and parameters using a single cutting tool operating on an in-seam tester. This device can measure and record triaxial, bit-generated force reactions when cutting many synthetic and geologic materials. Once the initial tool configuration and water-jet parameters are identified, the selected tools can be configured into interactive arrays for full-scale load and speed testing on one of several large laboratory tools. A production prototype longwall coal shearer, set up to cut a large simulated coal block, is used for this testing. The second tool available is a raise boring machine, mounted horizontally to a frame supporting a test rock cube that is 4 ft by 4 ft in size. This device is used to test tool-over-rock traverse rates up to 360 ft/min.

CUTTING TOOLS

Drag bits are similar to the cutting tools normally used for cutting processed metals. Generally these tools have a tungsten carbide or hardened steel cutting edge. They seldom generate normal force loads in excess of 2-1/2 tons. The rotational or cutting force (torque per tool) reaction is 80 to 120 pct of the normal force. In theory, these are the most efficient of all mechanical rock cutting tools for soft rock (ref. 8, 9). However because of the high frictional force reaction, they have a relatively short working life. Water-jet assist has the best potential for increasing the life of mechanical rock cutting tools by reducing frictional force (ref. 3). Drag bits are available in a large number of configurations. Two specific bit configurations are in common use today: the flat longwall bit and the point-attack conical bit (fig. 4). To limit the scope of this research, one of each bit configuration is used.

WATER-JET PARAMETERS

The modern concept of moderate pressure water-jet assist for mechanical rock-cutting tools was first reported by M. Hood in 1976 (ref. 10), with follow-up reports in 1977 and 1978. Related research was reported by Ropchan et al., in 1980 (ref. 2). These studies have provided the basis for the present PRC research. The following test limits (fig. 5) have been identified (ref. 1):

Parameter	Test Limits (min. and max.)
Jet pressures	250 to 10,000 psig for rotary cutting research; 5,000 to 40,000 psig for drilling research.
Jet nozzle diameters	0.3 to 1.0 mm.
Jet nozzle configurations	Leach and Walker.
Jet nozzle standoff	100 to 300 nozzle diameters.
Jet traverse speeds	30 to 600 ft/min.

Monitored for:

Jet flow rates,
Jet velocity,
Jet impact force and stagnation pressure,
Jet power and efficiencies,
Jet depth of cut and groove width.

SINGLE-TOOL RESEARCH

To determine the many possible combinations and test parameters associated with water-jet-assisted cutting, an in-seam testing device was acquired and refitted with a triaxial force dynamometer on a sliding mast assembly, as shown in figure 6. The test cutting tool and water-jet nozzle are mounted directly on the dynamometer, and instrumented to record the three principal cutting-tool force reactions: normal force (F_n), tangential force (F_t), and plus or minus side force (F_z). This device can be operated underground in a coal seam or on the surface using a special rock holding fixture to cut 2- by 2-ft rock cubes. Cut spacing and depth can be varied to suit the material being tested. The testing parameters are as follows (ref. 1):

Parameter	Test limits (min. and max.)
Bit spacing to depth-of-cut ratio	2 nominal
Bit traverse speed over test sample	26 ft/min
Depth of cut depending on test sample	0.25 to 2.5 in
Jet pressures	250 to 9,000 psig

Monitored for:

Average and peak dry cutting force reactions,
Average and peak water-jet-assisted force reactions,
Tool wear rates,
Tool failure forces.

NOTE: Single-tool research is conducted at a constant depth of cut to conform to established laboratory procedures.

The above limits were established to expedite research for optimizing operating conditions. They also include research of various filtration methods; i.e., system cleaning and coating methods, static and dynamic sealing methods, and the applicability of triplex pumps and water-pressure intensifiers.

Single-tool research includes a study of the very high-pressure jet drilling, up to 40,000 psig, now being introduced to the coal mining industry. The first concern was high-pressure hose and fitting safety. A high-pressure water intensifier (fig. 7) was procured, and a handheld drill (fig. 8) was provided by Jarvis Clark Co., along with the drill bits (fig. 9). Experimental work will start in the near future.

INTERACTIVE ARRAY TOOL RESEARCH

To correlate the single-tool, rock-cutting force data with a full-scale mining machine, it is necessary to generate an understanding of cutting tool forces when they are working in interactive arrays. Two laboratory devices are being used for the intermediate interactive array research.

A longwall coal shearer was assembled as a laboratory tool to cut a simulated coal block, as shown in figure 10. The shearer was retrofitted with a 6,000-psig water-jet-assisted cutting system The left-hand cutter was replaced with one equipped to direct pressurized water jets just ahead of each cutting bit. Pressurized water was supplied by a triplex pump and distributed through a hydraulic hose to a rotary swivel mounted inside the cutter head. Six ports directed the pressurized water to internal cutter chambers, and each chamber was connected to five or six jet nozzles. The chambering system was designed to support a rotary phasing seal that would permit the pressurized water to be directed only at the working face. Testing with the coal shearer was reported by Kovscek, Evans, and Taylor (ref. 11) at this symposium

The second laboratory test device for evaluating the performance of interactive cutting tools is the horizontally mounted raise boring machine system (fig. 11). Three large sandstone blocks have been tested for the effect of bit traverse speed on the pressurized water jets (table 1). The RWE of moderate pressure (3,000 to 10,000 psig) jet-assisted cutting is a poorly understood physical phenomenon.

	In-seam tester (IST)			Traverse speed tests (TST) in sandstone			
	Indiana limestone	Coalcrete test block	Pittsburgh Seam coal	Berea	German	Lyons	Dakota
Density..........lb/ft³..	1/160	106	85	1/130	157	155	134
Unconfined compressive strength...psi..	1/8,100	898	2,090	1/8,100	22,900	11,700	3,610
Silica content..........pct..	3/0.07	NA	NIL	3/93.1	55	>90	53
Porosity..........pct..	1/18.8	NA	NA	1/19.8	NA	11	NA
Hardgrove grindability index..........	NAp	62	58	NA	NA	NA	NA
Dust reduction 2/..........pct..	**NA**	**80**	**61**	**NA**	**89**	**74**	**81**
Depth of cut..........in..	1.0	1.5	1.5	NAp	NAp	NAp	NAp
Bit spacing..........in..	2.0	3.0	3.0	2.0	2.5	2.5	2.5

NA Not available
NAp Not applicable
NIL Nothing
1/ Data from published literature (ref. 12); all other data are from testing at PRC.
2/ Using flat longwall bit model K-107 with 3,000 psig and 0.015 in diam nozzle.
3/ Data from published literature (ref. 13).

TABLE 1. - Rock properties and test results through March 1985

Hood (ref. 8) reported that "drag-bit-generated force reactions resulting from waterjet-assisted cutting were not significantly affected by changes in bit velocity from 12 to 48 ft/min." A study by Harris and Mellor (ref. 5) concerning water jets without mechanical cutting tools reported "a marked dependence of jet penetration on traverse rates below 40 ft/min; however, at higher traverse rates, jet penetration becomes essentially independent of traverse rate." Both reports suggested that once a traverse rate threshold was reached, the RWE would remain more or less constant at higher speeds.

Testing to date has verified that the RWE remains unaffected by bit traverse rates up to 260 ft/min (fig. 12). Testing on other equipment will be required to verify this finding for speeds in excess of 300 ft/min, although the RWE is apparently lost above 300 ft/min. Other results of interest are shown in figure 13A. When the cutter head rotational velocity (r/min) and thrust were kept constant and water-jet pressure was increased, the energy (required to mine a unit volume of rock) and torque requirements decreased. Another finding is shown in figure 13B, where the advance was unaffected by jet pressures below 3,500 psig in Dakota sandstone. This indicates that an RWE threshold may exist for various rock materials.

EQUIPMENT DEVELOPMENT

A high-pressure rotary phasing system could reduce water volume and jet horsepower requirements by two-thirds. Two approaches are being studied. Figure 14 shows the self- or force-activated bit fixture (SABF). This unit was designed to test a bit assembly that activates a water jet when it comes into contact with the rock or coal face. Such a unit is very desirable on roadheaders that are continually contacting the working face from different directions. To date, the force-activated bit has worked at 2,000-psig jet pressure for about 400,000 cycles (approximately 118 h) without a major interruption.

A second approach to system phasing is an internally mounted rotary timing valve (fig. 15). This valve acts as a moderately high pressure rotary seal and provides porting into a cutter drum, permitting pressurized water to flow over a rotational segment of 120°. Seal problems have not allowed full testing on this project phase; testing is expected to resume in the near future.

DISCUSSION

Considerable research has been conducted on novel methods of rock fragmentation and tools as reported by W. C. Maurer (ref. 7, l4). Though some of these methods appear promising, only the existing mechanical tools show economic viability. The moderately high pressure water-jet assist proposes to extend the viability of these tools.

To penetrate an infinitely thick plate of confined rock, normal or thrust force (F_n) is required to interact with tangential or cutting force (F_t). The commercial range of F_n is from several hundred pounds to over a 45,000-lb force reacting on a single cutting tool. F_n is required to provide initial tool penetration; this is generally done by crushing the rock that is in contact with the tool edge into fine particles or gouge. F_t is a function of both F_n and depth of cut. It can range from a few pounds to over a 6,000-lb force. Ft is required to sweep the rock face and to break off the mounds or kerfs between tool tracks. This sweeping operation produces the largest rock fragments and requires the least input energy. Energy distribution between crushing and kerfing (considering the effect that moderately high pressure water-jet assist may have on that energy distribution) is the principal concern of this research.

Research to date has established confidence in the concept of water-jet-assisted rock cutting, inferred some limitations, and highlighted specific areas requiring ongoing effort. Table 1 presents some early results on dust reduction.

At this writing, water-jet systems have been developed for the in-seam tester (fig. 6), the longwall coal shearer (fig. 10), and the traverse speed test (fig. 11). Preliminary testing was conducted in Pittsburgh Seam coal, in the coalcrete block, in various sandstones, and on one limestone sample. Development problems have centered around internally caused nozzle clogging that was attributed to two factors: contaminated water and internal scaling possibly caused by weld slag. For the 0.024-diameter jet nozzle, 10-micrometer filtration resolved the water problem. The internal scaling problem proved to be more pervasive, and was only resolved by continued purges under full pressure. Externally caused plugging was easily cleared by the force of the jet. Figures 16 and 17 show the two bit and jet nozzle configurations that were used. Once the internally caused clogging problems were resolved, the configuration used in the longwall coal shearer proved to be the most reliable (fig. 16).

Research problems have evolved around the fact that most laboratory research has been conducted using a constant depth of cut. Most mining equipment has a constant thrust force (F_n), and the depth of cut will vary with the hardness of the material. As the longwall coal shearer comes close to a fixed depth of cut, it has proved to be a reasonable tool for correlating laboratory and in-seam-tester generated data. The traverse speed test fixture, continuous miners, roadheaders, and tunnel boring machines all have constant deliverable force rates; the advance, or penetration will vary only with the rock properties. More research is needed to establish the rock properties most affected by the moderately high jet-pressure-assisted cutting tools.

One of the more interesting results from the shearer tests is that cut particle size appeared to increase with higher jet pressures. The implication is that the higher jet pressures can reduce the percent of fines mined with the coal. It was also established in cutting the coalcrete block that shearer power requirements decreased with increased jet pressures.

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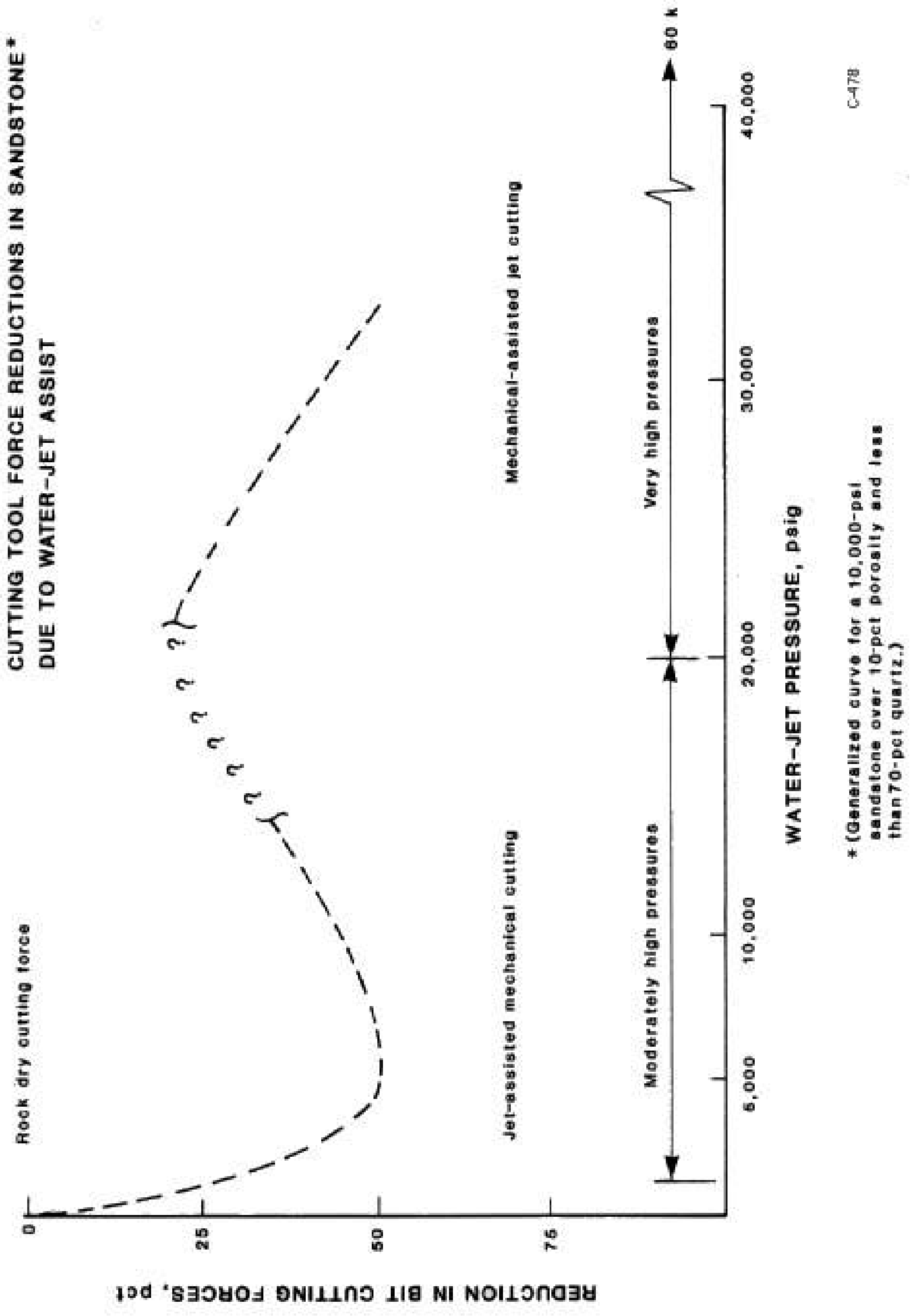


Figure 1. Cutting tool force reductions in sandstone due to water-jet assist.

Figure 2. Portable in-seam tester mounted on coalcrete block.

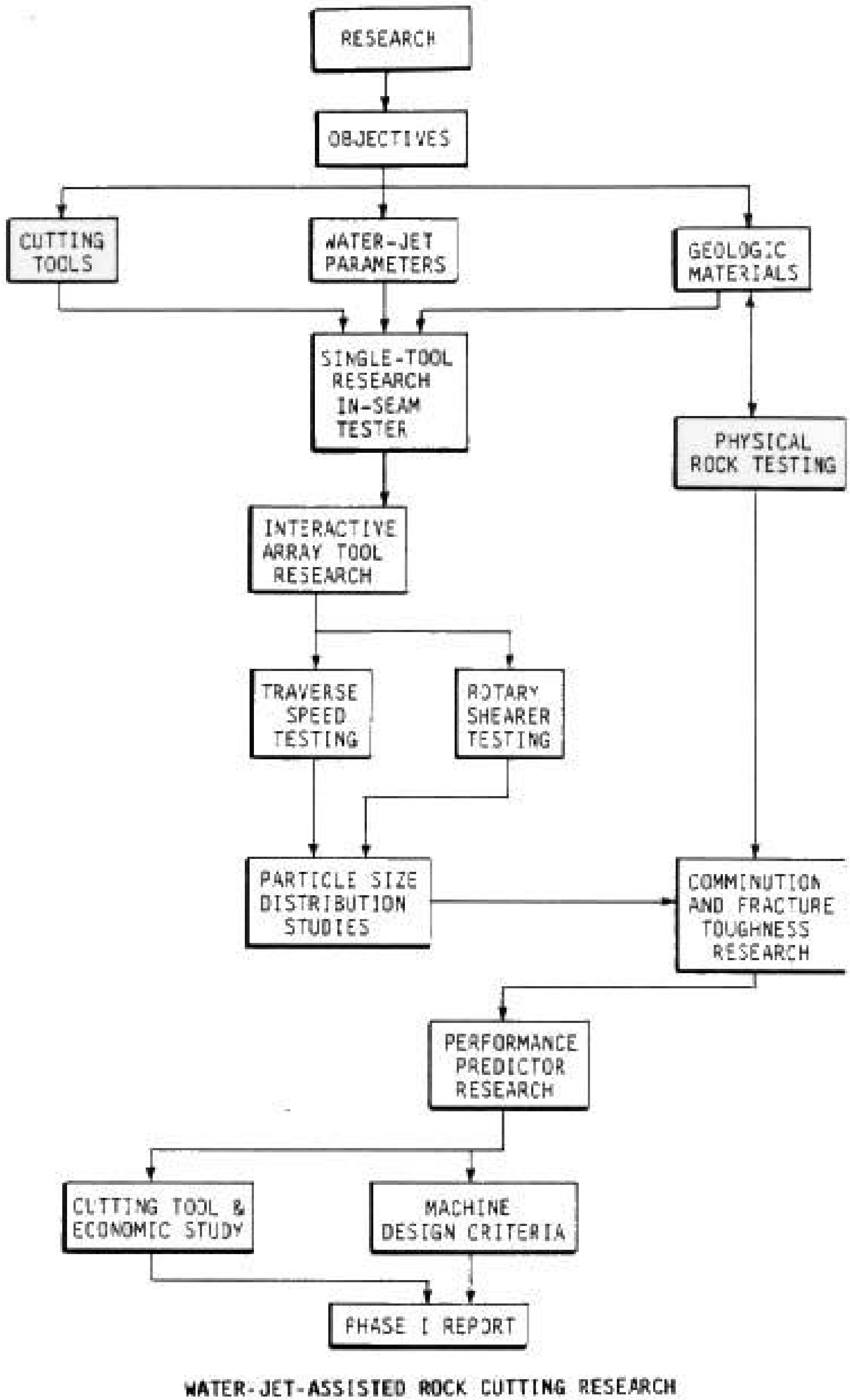


Figure 3. PRC program outline.

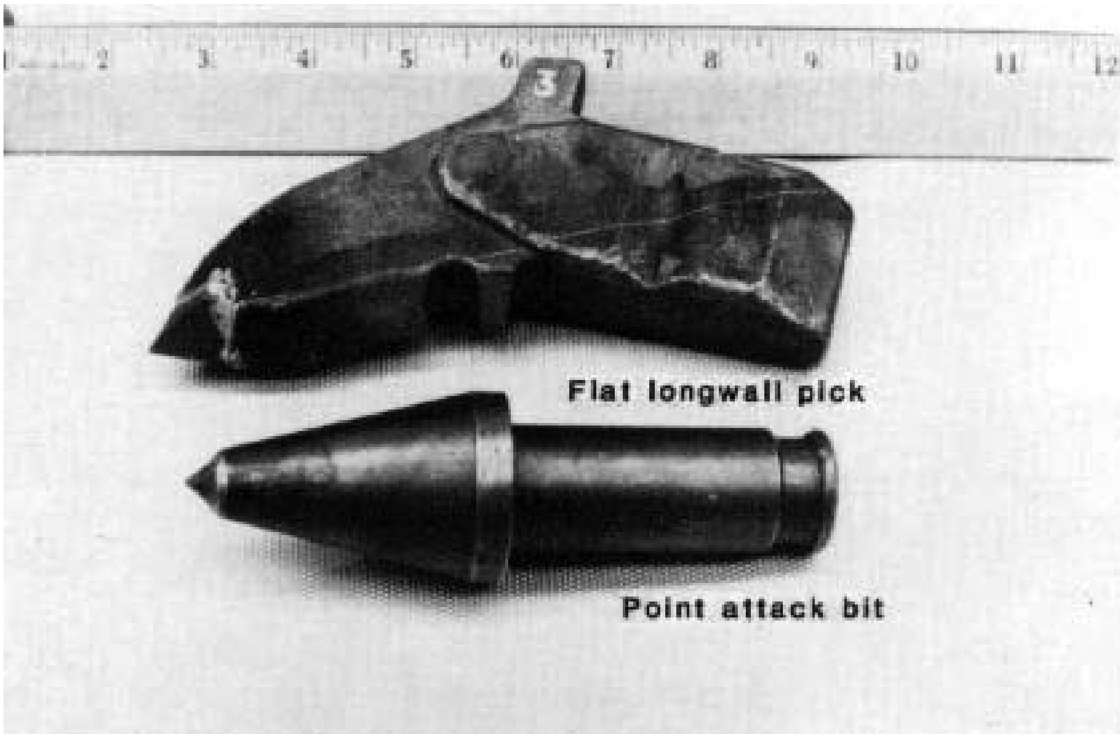


Figure 4. Drag-type rock cutting tools.

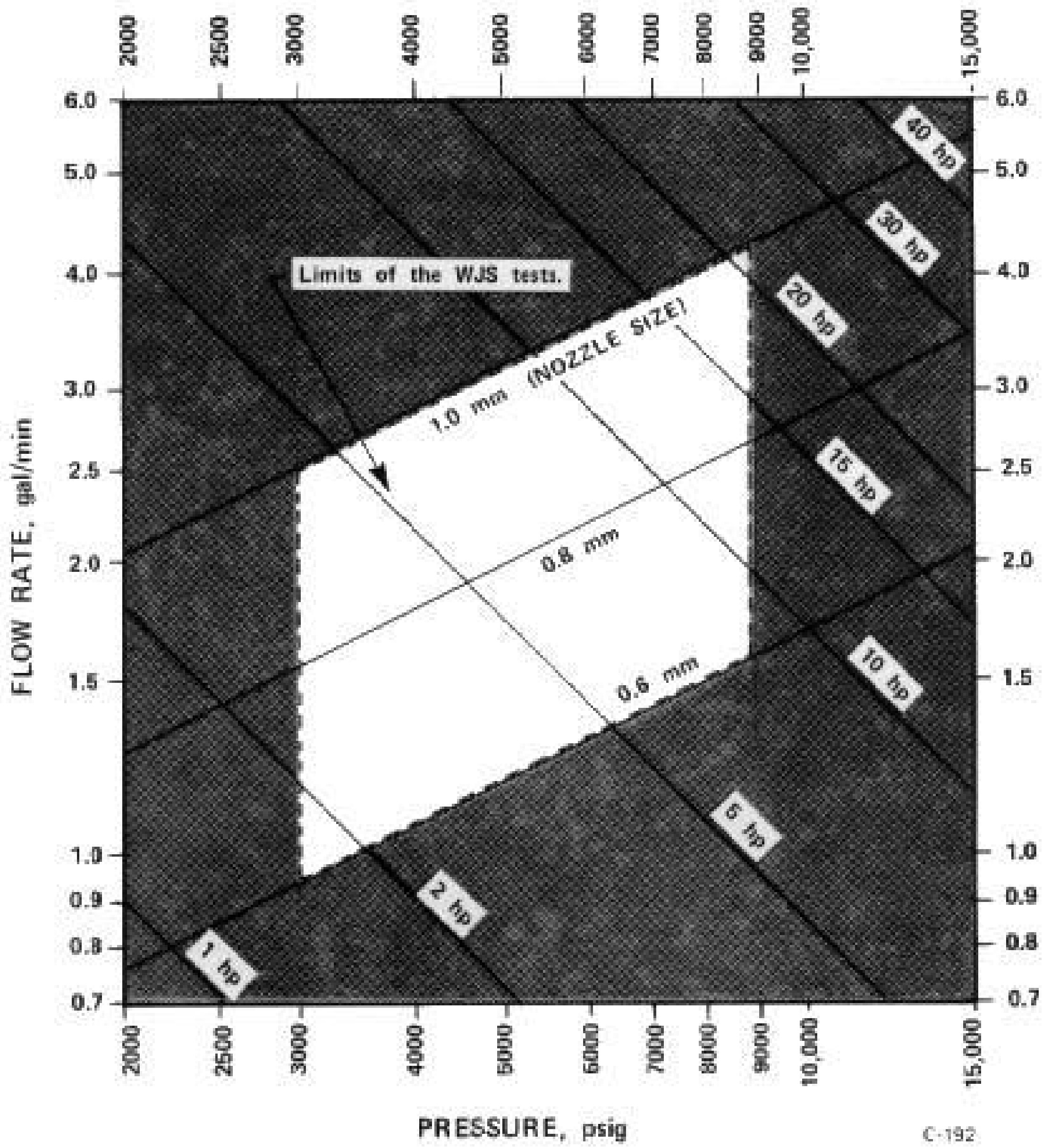
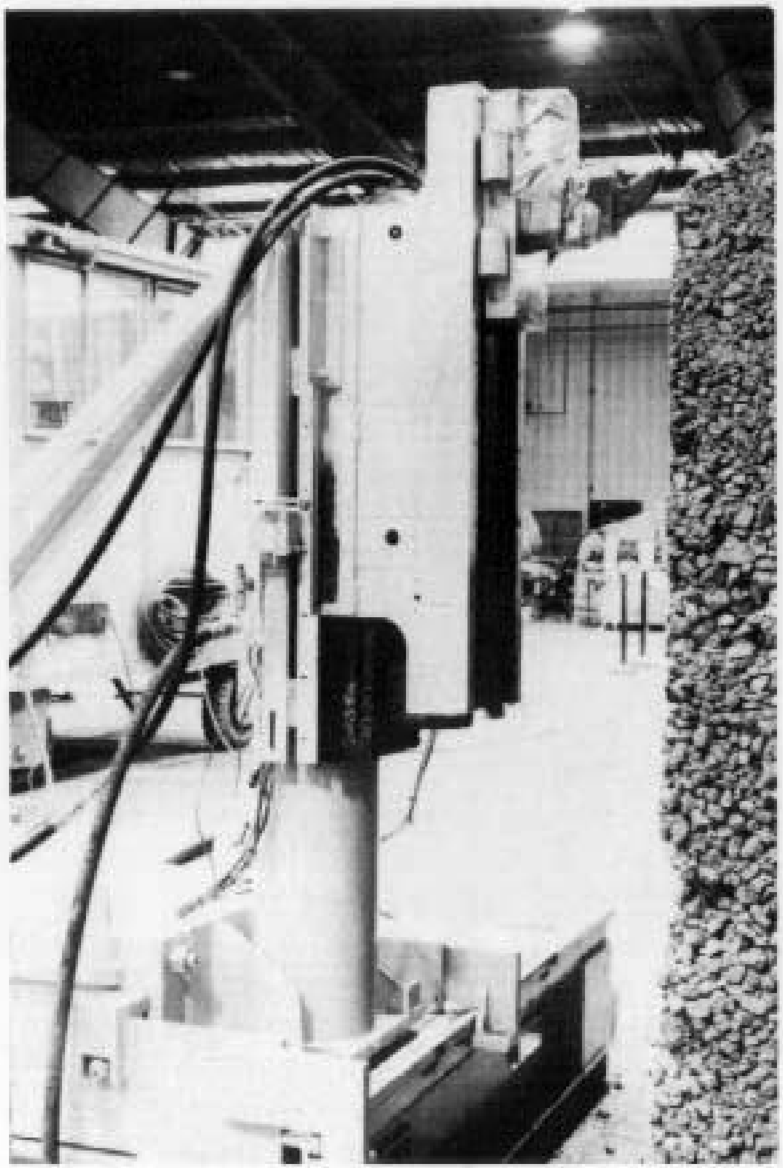


Figure 5. Waterjet research parameters.



A. **Highest mast position**

Figure 6A.

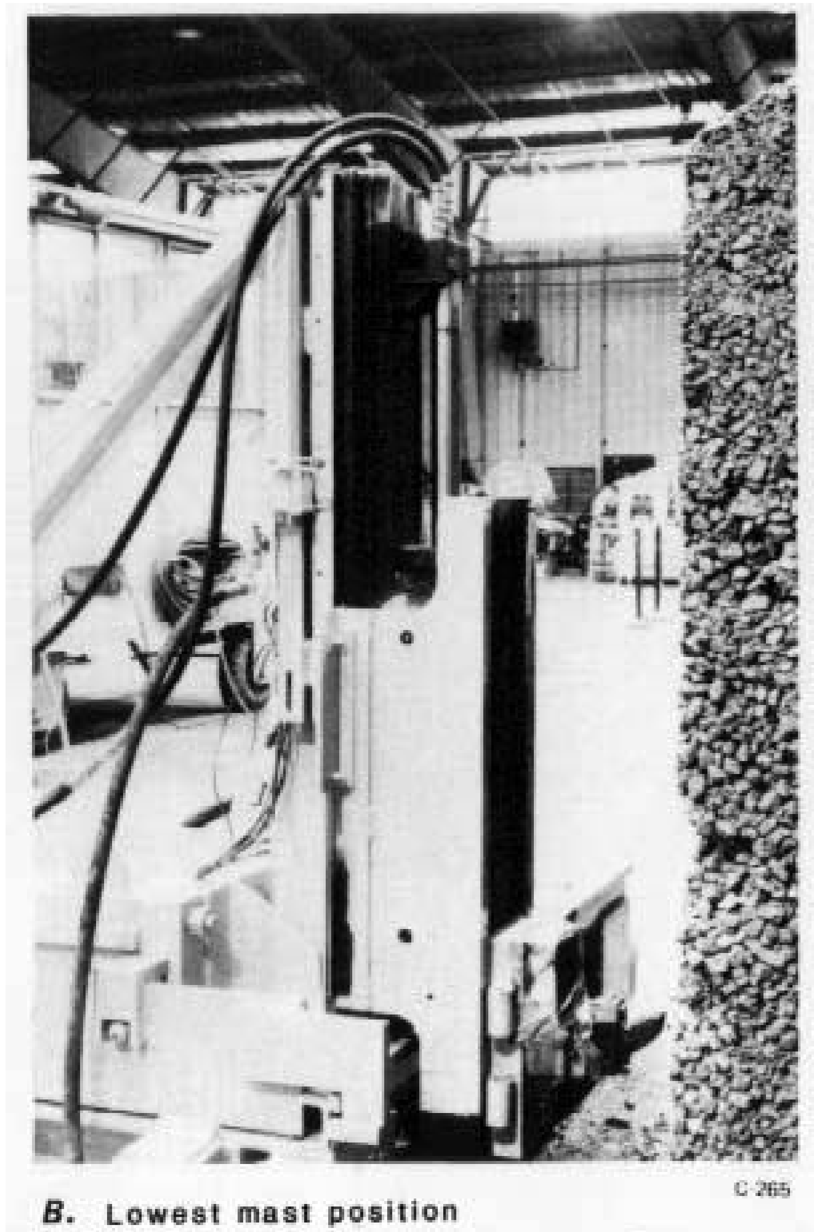


Figure 6B. In-seam tester and simulated coal block.

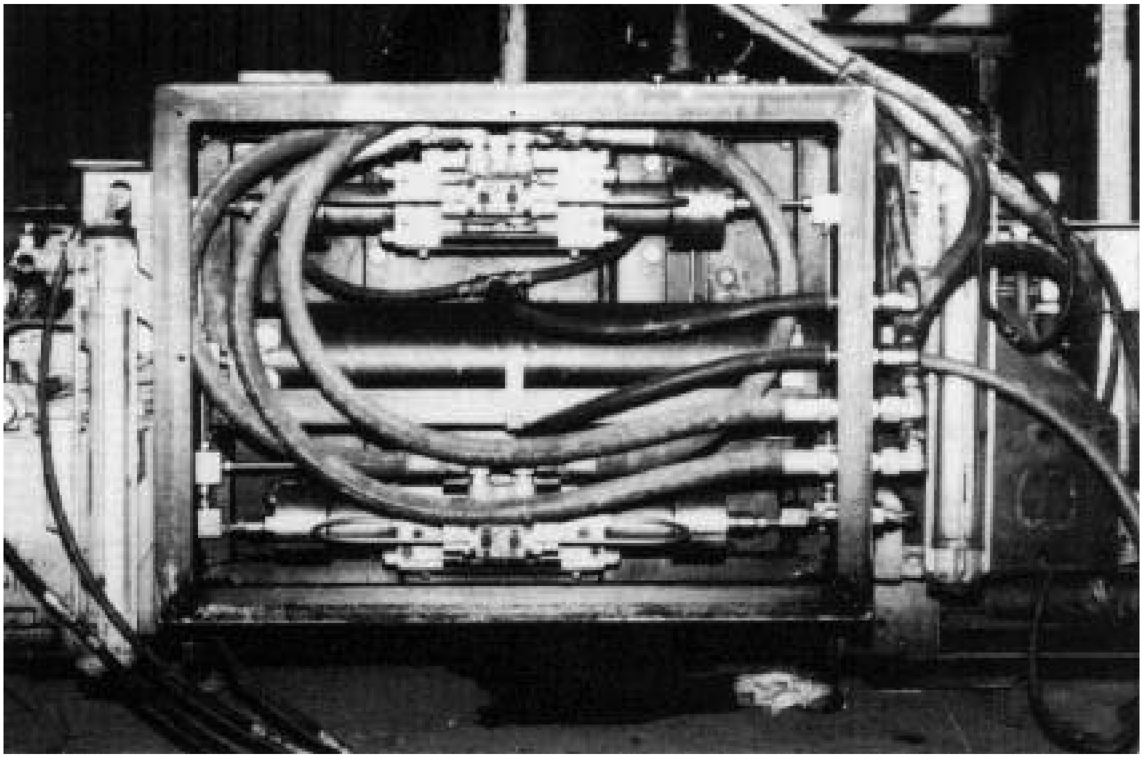
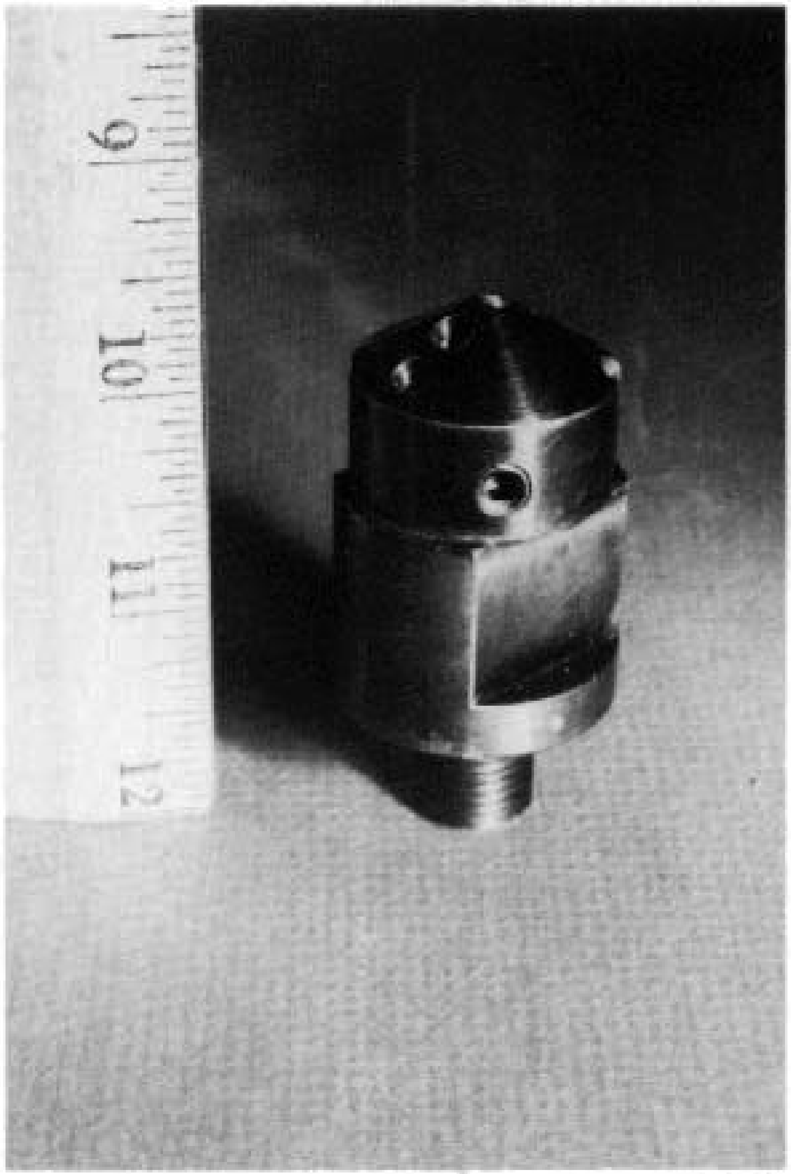


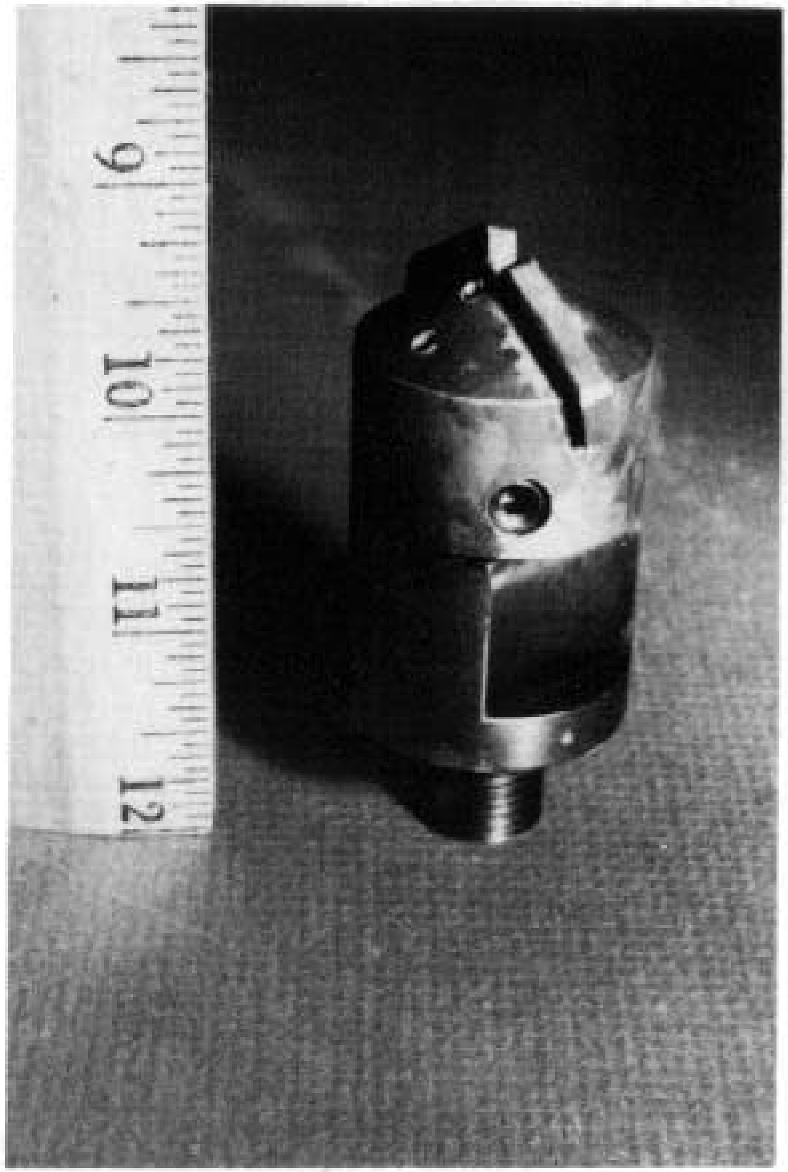
Figure 7. 40,000 psig water intensifier.

Figure 8. 40,000 psig handheld rotary drill provided by Jarvis Clark Co.



A. High-pressure nozzle assembly C-347

Figure 9A.



C-348

B. High-pressure bit assembly with tungsten carbide cap.

Figure 9. 40,000 psig drill bit (cont'd).

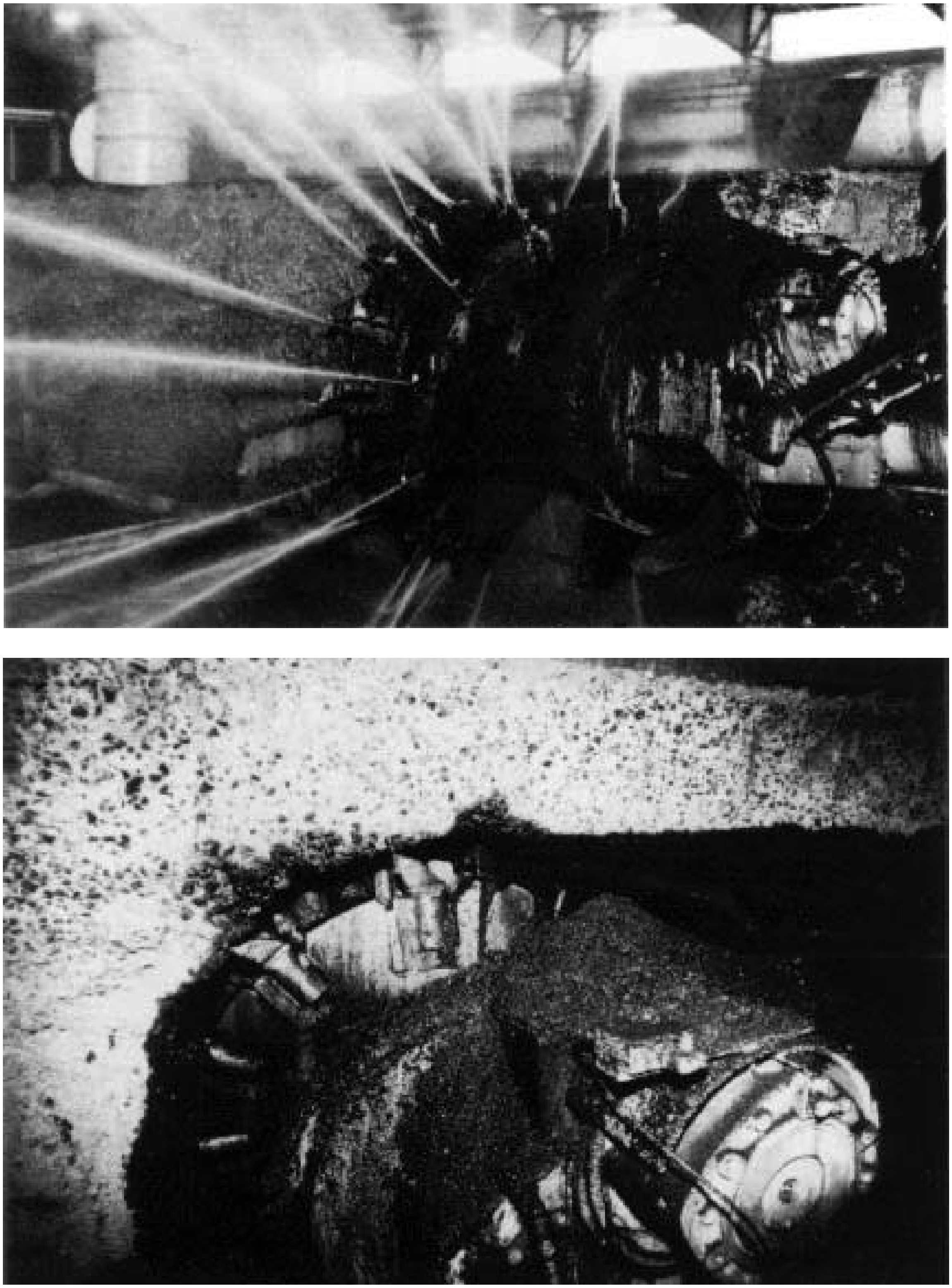


Figure 10. Waterjet assist cutter drum on longwall coal shearer.

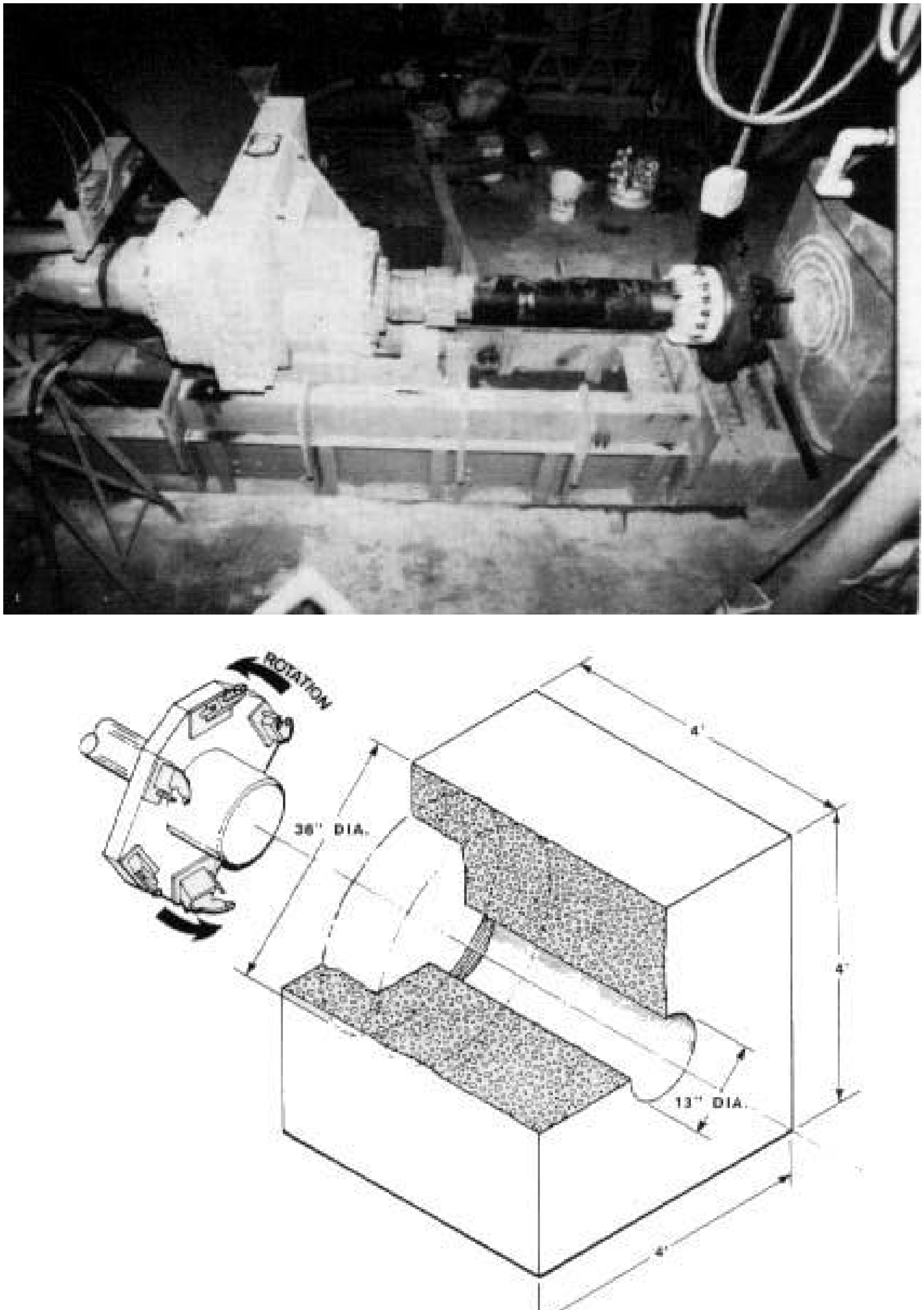


Figure 11. Sandstone block and traverse speed cutter test setup for interactive array testing.

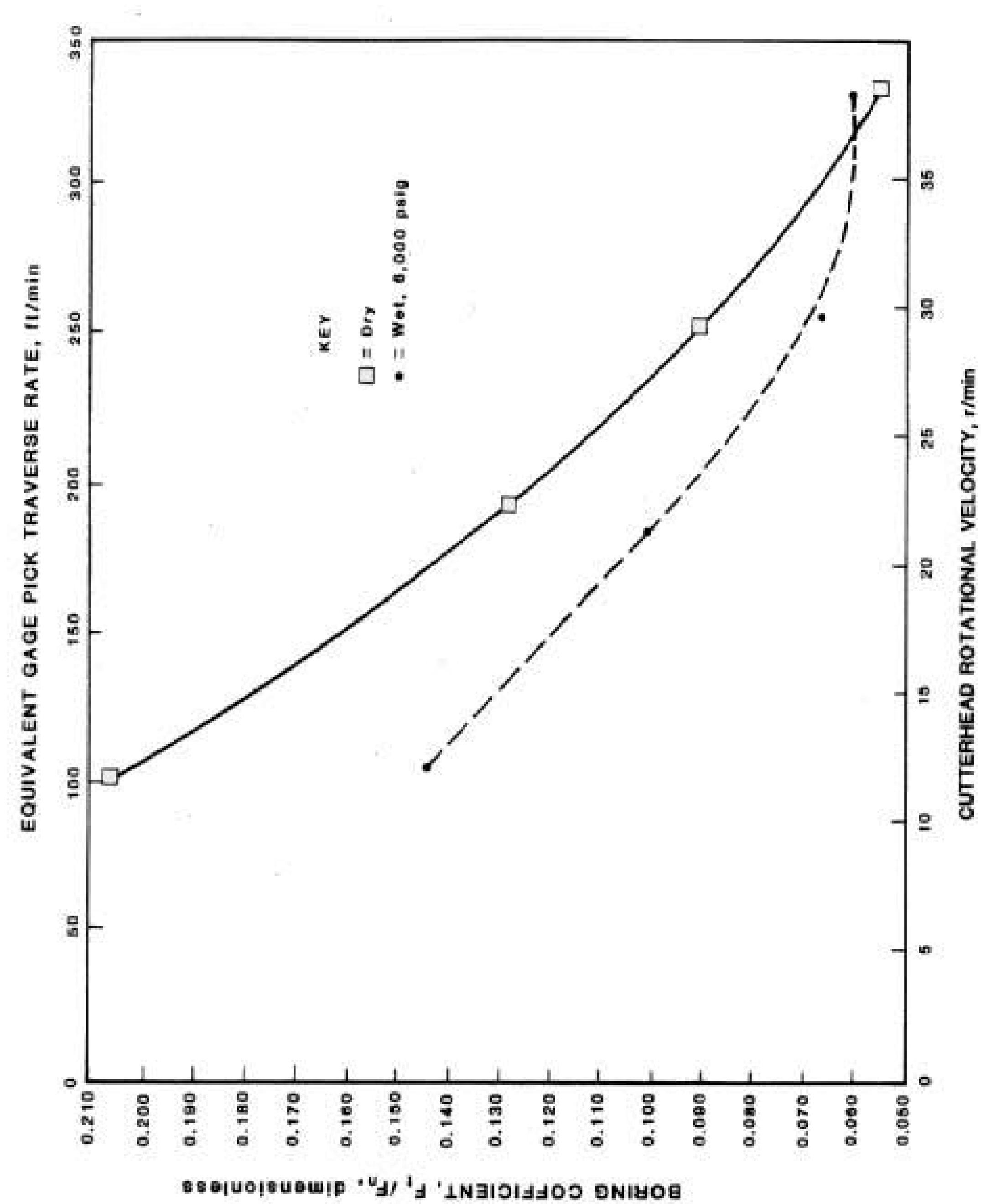


Figure 12. Boring coefficient as a function of cutter head rotational velocity with constant thrust for tests conducted in Dakota sandstone.

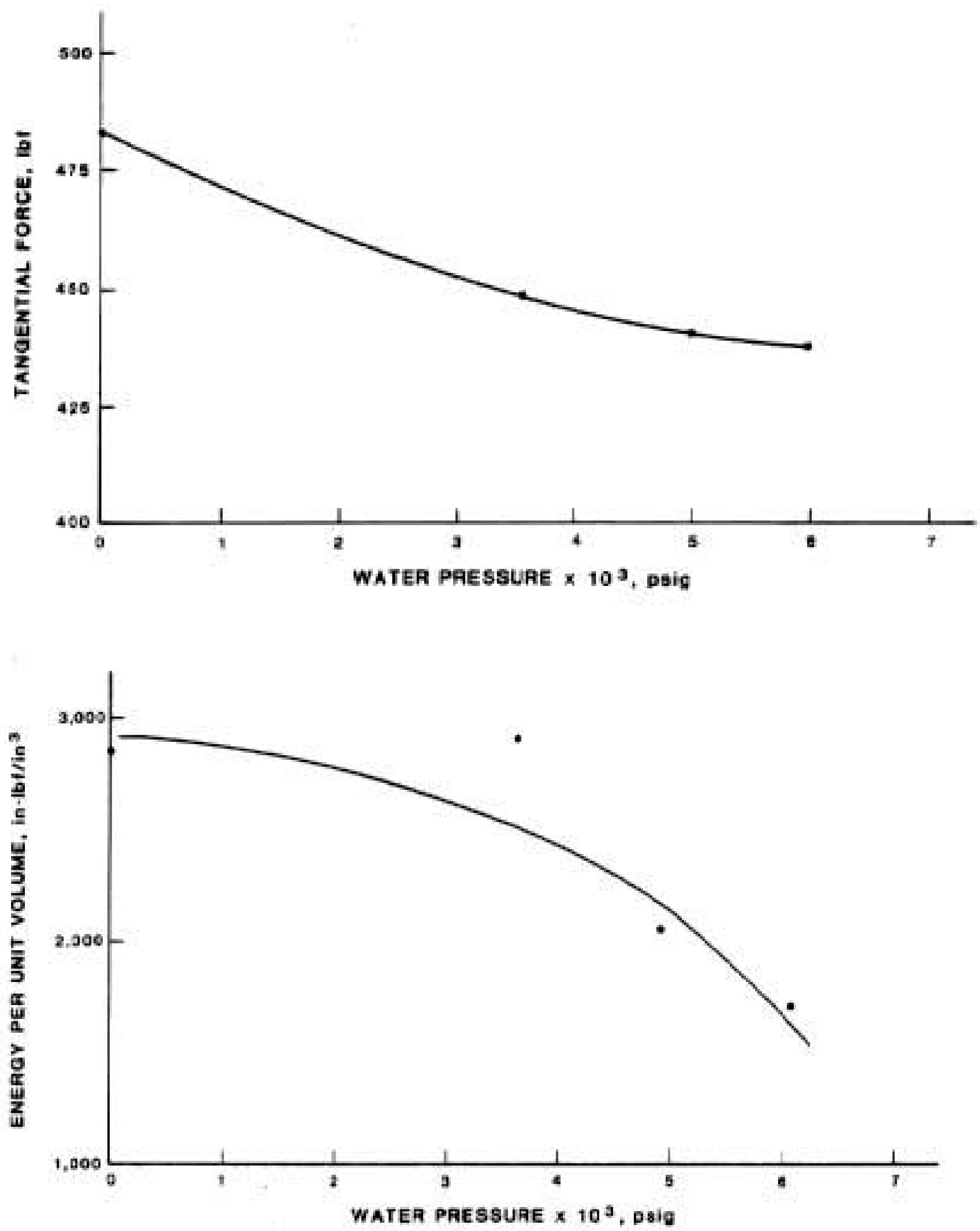


Figure 13A. Tangential force and energy per unit volume as a function of water pressure with constant thrust for tests conducted in Dakota sandstone.

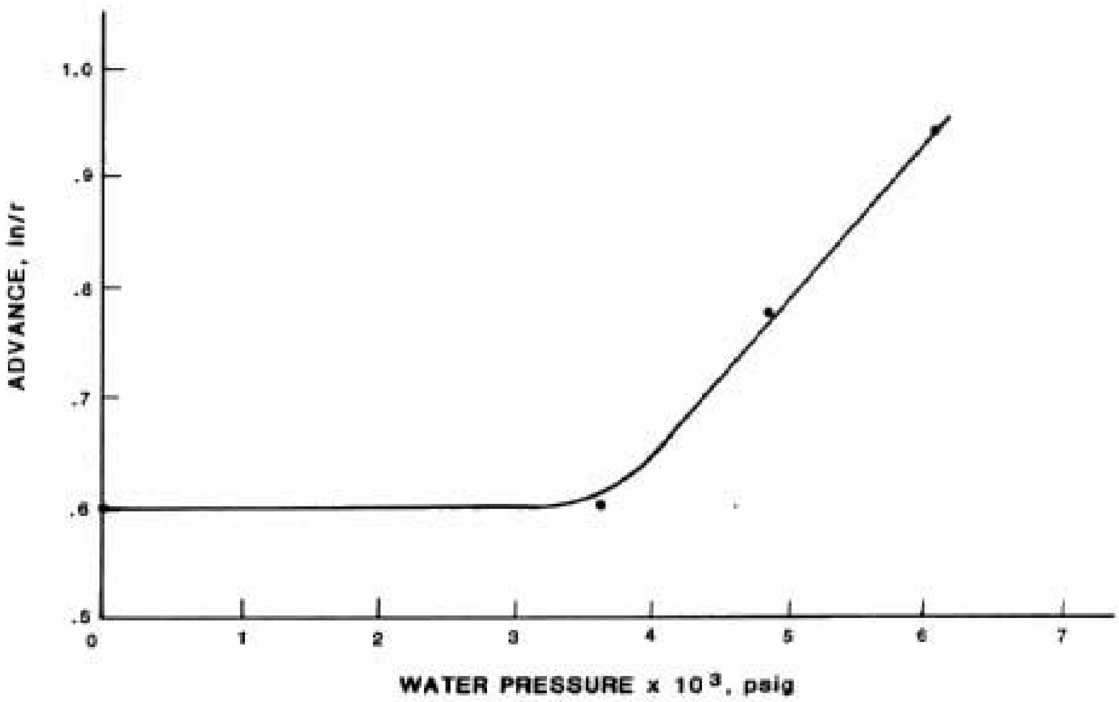


Figure 13B. Power and advance as a function of water pressure with constant thrust for tests conducted in Dakota sandstone.

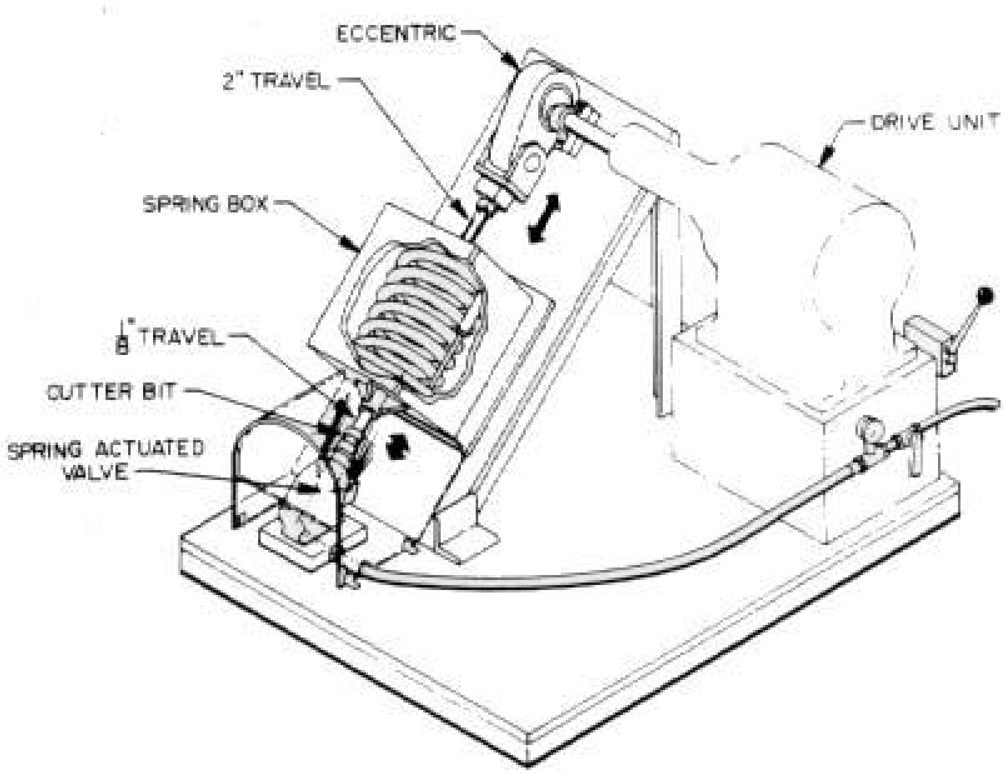


Figure 14. Self activated bit test fixture.

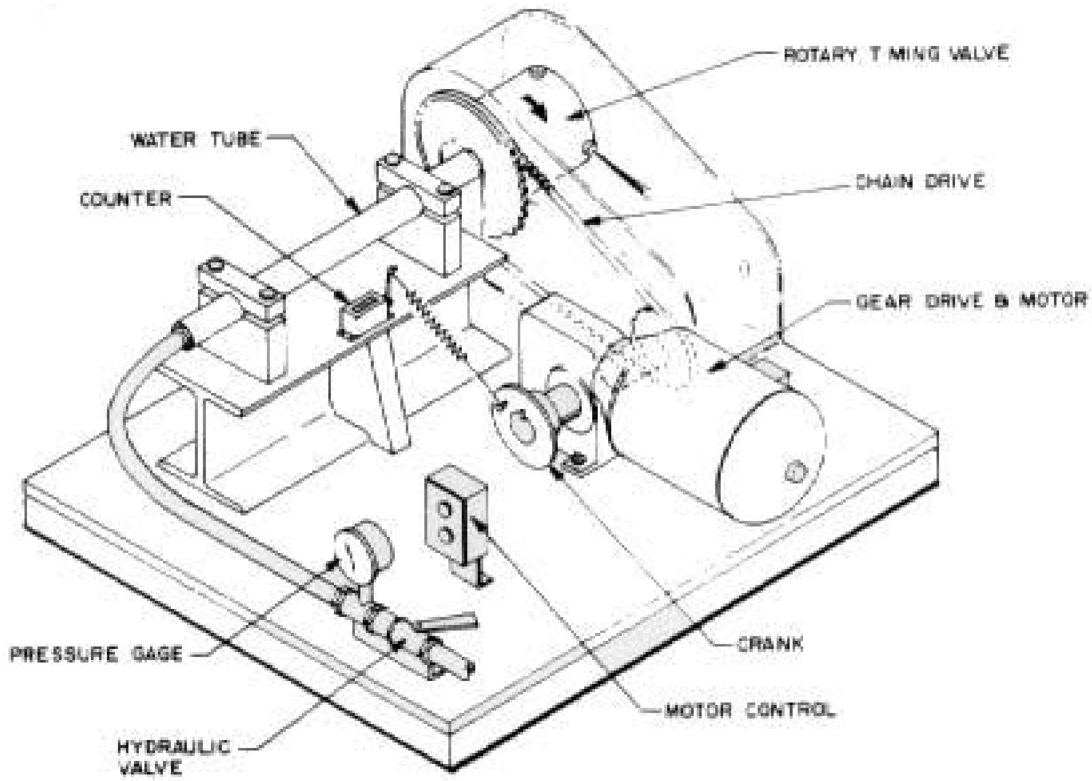


Figure 15. Rotary timing valve test fixture

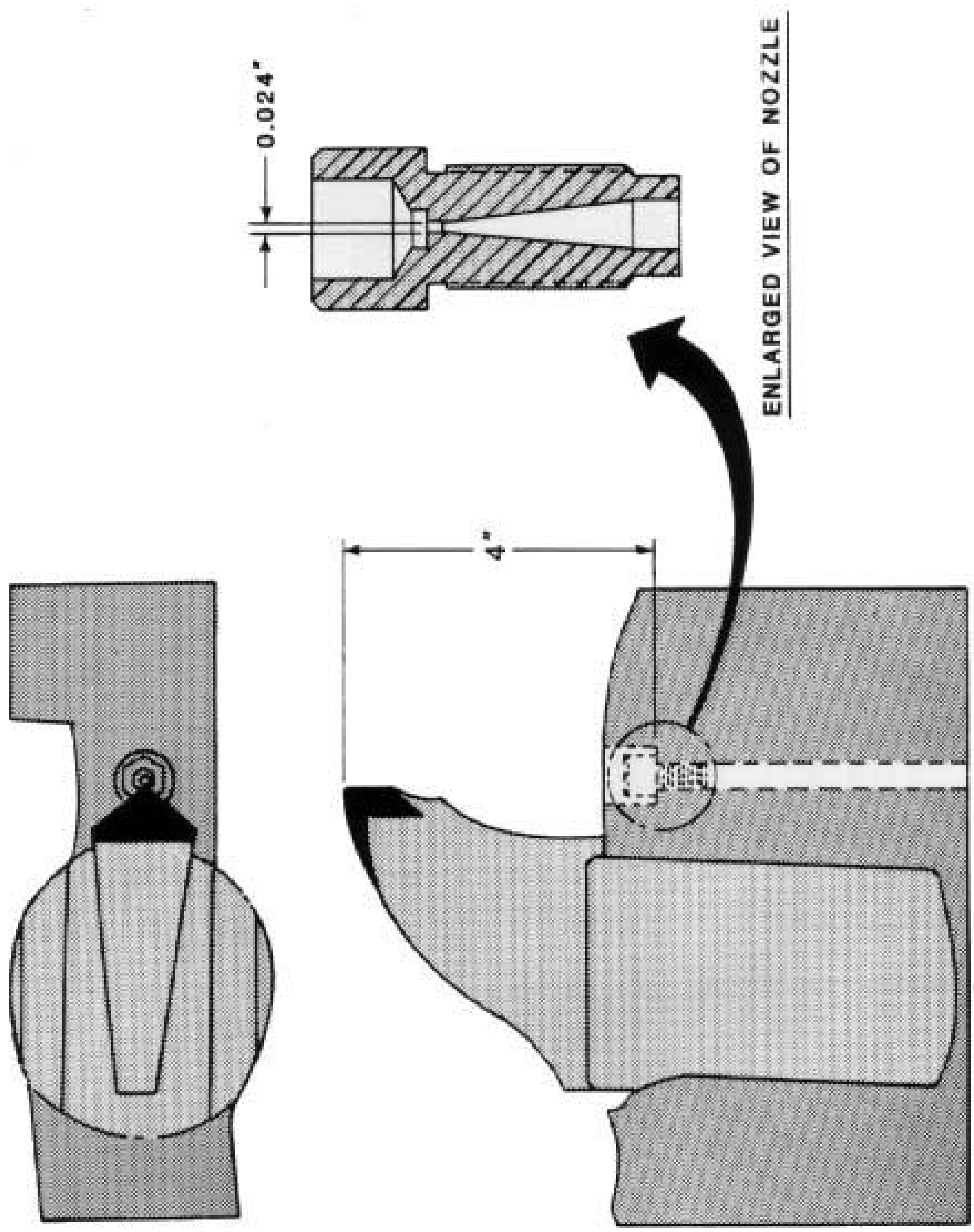


Figure 16. Bit mounting and jet nozzle used on the longwall coal shearer.

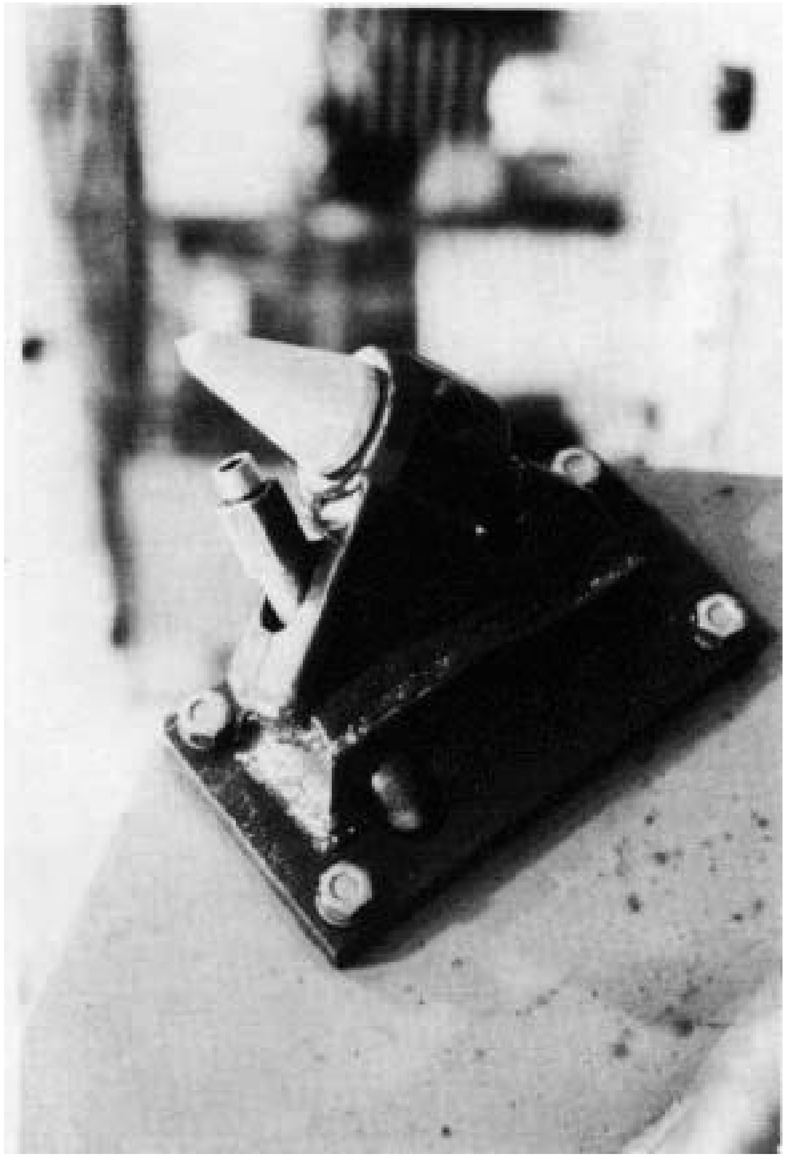


Figure 17. Conical bit mounted on traverse speed test fixture.