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Paper

CHARACTERIZATION OF PHYSICAL FEATURES IMMEDIATELY UPSTREAM OF A NOZZLE INSERT AND THEIR EFFECTS ON WATERJET QUALITY

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ABSTRACT

Many waterblast applications depend on the benefits of a cohesive waterjet. Producing a cohesive waterjet within the size constraints of most applications affords a challenge which relies on many variables. The most critical of these variables affecting waterjet quality are the physical features of the passageways immediately upstream of a nozzle insert. The purpose of this research is to determine, within the size constraints of typical waterblast tooling, the critical parameters of immediate upstream passageways and their contribution to a cohesive waterjet.

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1. INTRODUCTION

Achieving a more cohesive waterjet has been the subject of much study. Without tooling space constraints, the ability to produce a cohesive, far-reaching waterjet is greatly simplified. Few cleaning applications offer such an opportunity. For the applications which are space constrained, understanding the important parameters in optimizing a waterjet are critical and may even determine success or failure.

It's worth noting that not all applications benefit from a cohesive jet. The use of fan jets is a good example of beneficial use of a poor quality, non-cohesive waterjet.

This paper details the important physical features immediately upstream of a nozzle insert to maintain a cohesive jet within the limited space typically available.

2. TEST ARRANGEMENT

Testing was completed by traversing a target across a waterjet at a constant traverse rate, 536 mm/s (21 in/s), with the waterjet normal to the surface of the target throughout the traverse. The target material chosen was blue machinable wax of size 76 mm x 76 mm x 178 mm (3 in x 3 in x 7 in).

Standoff distances from the face of the orifice to the target material surface were 1118 mm (44 in) and 813 mm (32 in). Proper choices of standoff distances to adequately describe waterjet performance are worthy of a paper. However, for this paper, these specific distances were chosen to allow for each test to provide a measurable sample for each test while detailing the difference in performance of varying jet sizes, and facilitate an expedient test program.

To record the results, depth measurements were taken in reference to the flat, undisturbed faces of the machinable wax using a drop gauge and small probe as well as a heaping teaspoon of patience. Ten measurements were taken, each approximately 4 mm apart (.15 in), and arithmetically averaged to determine the depth of cut for each result.

High-pressure water was supplied from a 242 kW (325 hp) triplex pump with pressure measured at the inlet of a 20 mm (0.787 in) supply hose of length 15.24 m (50 ft). All tests were conducted at 68.9 MPa (10,000 psi), filtered to 10 micron (0.0004 in) with carbide nozzles and holders shown in Figure 1.

When flow straighteners were used, they were each 38 mm (1.5 in) long and consisted of the designs shown in Figure 2. Placement of the flow straightener can be seen in Figure 1, with the water passing the flow straightener maintaining the flow straightener's position adjacent to the nozzle insert. Each flow straightener has a simple, geometric division, similar to a pie slice, to detail the effect of additional divisions while limiting uncontrolled variables.

No standard metric for waterjet cohesion exists and so non-space constrained parameters are reported and used for gauging the effects of the physical features upstream of the nozzle insert in space constrained parameters. Non-space constrained parameters include the feeder pipe inside diameters up to 20.7 mm (0.815 in) and feeder pipe lengths up to 914 mm (36 in).

3. FOUNDATIONS FOR TESTED PARAMETERS

Several common practices are used to improve waterjet cohesiveness. This section aims to provide a foundation for the mechanisms behind these improvements to define the important characteristics of flow upstream of a nozzle insert.

3.1 Feeder Pipe Length

Increasing the feeder pipe length has a significant and beneficial effect on waterjet quality. Shavlovsky recommended a minimum ratio of feeder pipe length to feeder pipe inside diameter of 40-50, although he described flow straightener design to be able to reduce the same ratio to 10-15 (A6-87). The goal was "removing circulatory flow arising from the flow twisting and turbulent pulsations" (Shavlovsky, A6-87). One can interpret this as achieving fully developed turbulent flow, with the ratios provided to be the requirements for the flow to stabilize.

Literature regarding the length required for fully developed turbulent flow varies widely. Nikuradse stated that the velocity distributions at length to diameter ratios of 100, 65, and 40 were equal (21). Lien, Monty, Chong and Ooi recommended values of 130 to 150 for fully developed turbulent flow (4). The latter paper details some of the discrepancies on recommended values for fully developed turbulent flow.

Feeder pipe length does not have to be established from an extension arm. A continuous passageway upstream of a nozzle insert can be formed from a tap drill and result in an improvement to waterjet quality. These improvements are typically small since the length to diameter ratio lies well below any of the previously stated values.

Regardless of the incongruities, the entry length, or the length required to establish fully developed turbulent flow is related to the hydraulic diameter and the flowrate and any of the published minimum lengths are usually significantly longer than most waterblast applications can accommodate.

3.2 Feeder Pipe Diameter

Though the Reynolds number lies well within the turbulent region within the feeder pipes of waterblast systems, reductions in Reynolds Number are correlated with the benefits of increasing feeder pipe diameter.

Reynolds Number:

$$\mathrm{Re} = \frac{uL}{\nu} = \frac{QD_h}{A\nu} \qquad \text{(Equation 1)}$$

With an increase in feeder pipe diameter, the Reynolds Number is reduced because the velocity of the fluid within the feeder pipe is reduced. Shavlovsky noted this also and recommended maximizing the ratio of nozzle diameter to feeder pipe inside diameter, with values up to 9-10 suggested (A6-88). Wright found similar benefits of this ratio with an optimal value of 5-13, depending on the length of the feeder pipe (4).

Another important relationship is between turbulence and power. With increasing nozzle size comes increasing flow and, since flow is a linear variable in both the Nozzle Power equation and the Reynolds Number equation, the ratio of Nozzle Power to Reynolds Number is constant for each feeder pipe diameter. In other words, the power applied to the system and the degree of turbulence in the system change at the same rate. Some testing has shown peak effective nozzle sizes for certain flow regimes that would indicate the Nozzle Power to Reynolds Number ratio could be used as an Efficiency Index. As the Efficiency Index increases, the peak effective nozzle diameter is increased, and more power may be applied for the same amount of turbulence in the system.

Nozzle Power:

$$P = \frac{Qp}{C} \qquad \text{(Equation 2)}$$

Efficiency Index:

$$E.I. = \frac{P}{Re} = \frac{pA\nu}{D_h C} \qquad \text{(Equation 3)}$$

As with feeder pipe length, the diameter of feeder pipes is typically limited by the application and the pressure which, again, typically dictate maximum sizes for tooling.

3.3 Flow Straighteners

Using flow straighteners has two beneficial effects:

1. Water flow is more fully developed at the nozzle by decreasing the hydraulic diameter. Typically done by increasing feeder pipe length.
2. Turbulence is reduced by maintaining fluid velocity while decreasing the hydraulic diameter. Typically done by increasing feeder pipe diameter.

Flow straighteners "reduce the diameter of the effective tube by dividing the channel into a series of smaller segments" (Summers, 41). Essentially, the hydraulic diameter the water passes through decreases without significantly raising the velocity of the water within the feeder pipe.

A further reduction of the Reynolds Number can be achieved by optimizing the number of flow straightener divisions. The acting area of the feeder pipe described using the number of passages and wall thickness of the flow straightener is seen in Equation 4.

Reduced Cross-Sectional Area:

$$A_r = \frac{\pi}{4}D_h^2 - \frac{D_h}{2}Nw \approx \frac{\pi}{4}ND_f^2 \qquad \text{(Equation 4)}$$

Some liberty with the equation is taken here since dividing the circle of the feeder pipe inside diameter does not result in circular regions.

Substituting the Reduced Area into Equation 1 and some simplification gives:

$$Re = \frac{Q}{\frac{\pi}{4}\nu ND_f} \qquad \text{(Equation 5)}$$

Equation 5 shows, for each feeder pipe diameter, there exists a minimum Reynolds Number formed by the inverse relationship between number of flow straightener divisions (N) and the size of the passageways formed by the divisions (D_f).

The diameter of the passageways formed is a function of the number of divisions. Further reduction in the equation allows for the following:

$$Re = \frac{Q}{\frac{\pi}{4}\nu\sqrt{ND_h^2 - \frac{2}{\pi}N^2D_hw}} \qquad \text{(Equation 6)}$$

Substitution of the hydraulic diameter allows the partial derivative of the Reynolds Number with respect to flow straightener divisions ($\partial Re/\partial N$) to be taken to produce the optimal number of passageways. The author won't spoil all the fun for the reader, though correct answers of this partial derivative may be submitted to the author in exchange for a free haircut by the author.

It should also be noted that the flow straightener walls have a displacement thickness which should be considered. The displacement thickness forces a reduction in the effective cross-sectional area, acting as a thicker wall than the physical flow straightener actually has. Much literature exists for displacement thickness of flat plates and moderately high Reynolds numbers, but the parameters typically encountered in waterblast tooling don't allow these to be used easily.

Because of their cost, size, and effectiveness, flow straighteners are ubiquitous in the waterblast industry. Typically found in the nozzle inserts themselves, the ability of a flow straightener to improve waterjet quality in a very small space makes them a unique and powerful component.

3.4 Individual Passageways

Many tools use nozzle inserts which are fed by individual passageways to improve jet quality. Similar to increasing feeder pipe length, individual passageways allow for flow to be more developed just upstream of the nozzle insert.

Packing enough individual passageways into an appropriate size for waterblast tooling can limit this technique for improving waterjet quality. Maintaining a reasonably-sized tool results in a reduction in passageway ID and a subsequent increase in turbulence upstream of the nozzle insert. Also, the overall length of the passageway may not be contributing to fully developed flow since the curves of the passageway induce turbulence within.

High-quality waterjets may be created with this method, but it requires a difficult balance of many parameters.

3.5 Surface Roughness

Having a rough surface upstream of a nozzle insert can, just like a golf ball, produce a thin boundary layer that is beneficial to waterjet quality. However, producing a feeder pipe or other tooling with a prescribed surface finish likely isn't an economically sound pursuit and isn't common in the waterblast industry.

4. TESTED PARAMETERS

4.1 Feeder Pipe Length

Results in Figure 3 and Figure 5 show that the performance gains from feeder pipe length behave logarithmically and the largest performance differences are in the shorter lengths. Each feeder pipe diameter saw the majority of performance gains, 80%, in feeder pipes shorter than 304 mm (12 in). Something important to note is that the results don't indicate any significant decrease in performance with a feeder pipe longer than the optimal length.

Where possible, feeder pipe length should be increased to reduce standoff distance and improve waterjet quality.

4.2 Feeder Pipe Diameter

Results in Figure 7 show that the performance gains from a large feeder pipe diameter are significant, especially when coupled with a long feeder pipe.

More importantly, shown in Figures 3 and 5 are the ability to benefit from more power being applied to the system because of larger Efficiency Index values. The 11.1 mm (.438 in) feeder pipe did not see better performance with the 3.2 mm (.125 in) orifice, whereas the 20.7 mm (.815 in) feeder pipe has higher Efficiency Index values and did perform better with the larger orifice.

Particularly in high-flow applications, feeder pipe diameter should be increased to improve waterjet quality.

4.3 Flow Straighteners

Flow straighteners with simple geometric divisions are shown in Figure 2 and were shown to have a significant benefit to waterjet quality as seen in Figure 3, raising the performance of the feeder pipe by 37%. The effects of increasing the number of flow straightener divisions is shown in Figures 8 and 9 indicating that, depending on orifice size, the optimal number of divisions may be greater than six for a feeder pipe of diameter 11.1 mm (.438 in).

Similar to the elevated values of the Efficiency Index from a large feeder pipe diameter correlating to better performance with larger nozzles, more flow straightener divisions raised the Efficiency Index and the trend was improved performance with larger nozzles. This point is especially important for high-flow applications; by adding the appropriate flow straightener, a cohesive jet can be made with a small diameter feeder tube and a large diameter orifice.

4.4 Individual Passageways

Limited testing was completed on components with an individual passageway feeding a nozzle insert. Benefits of individual passageways were seen in some, but not all, cases. The individual passage is not sufficient to improve waterjet quality.

The magnitude of benefits to waterjet quality were comparable to a feeder pipe with a similar straight length directly upstream of the nozzle insert. Also, smooth transition features between the high-pressure supply and the start of the individual passageways had no measurable effect on waterjet quality.

4.6 Surface Roughness

Limited testing was completed on the surface roughness of parts immediately upstream of the nozzle insert. The results showed a slight benefit when there was some degree of roughness to the wall of the feeder pipe.

5. CONCLUSIONS

Many features immediately upstream of a nozzle insert influence waterjet quality, though there are three that have a significant effect:

1) Feeder pipe length – through aiding in flow development within the pipe.
2) Feeder pipe inside diameter – through reduction in turbulence within the pipe.
3) Flow straighteners – through reduced distance required for developed flow and reductions in turbulence within the feeder.

While it is not always practical to use large feeder pipes, there is no significant penalty for making the feeder pipe as long as the application allows. Also, feeder pipe length should be considered first, as the benefits to additional length are greater than the benefits from a slightly larger diameter.

For space constrained applications, flow straighteners provide the most effective method to improve waterjet quality, especially when high flow causes extreme turbulence upstream of the orifice. Performance can be recovered in a small space and large orifices can be used more effectively.

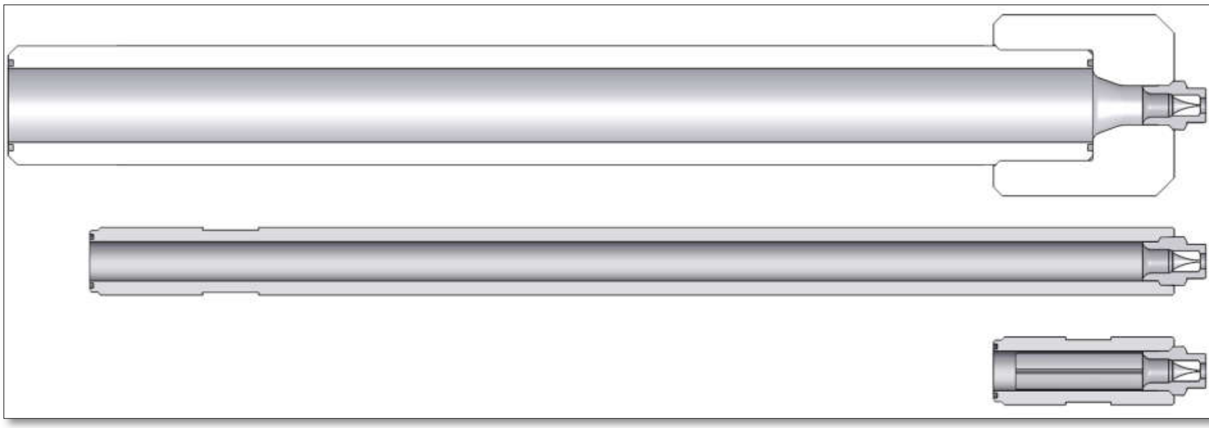

Figure 1. Top: Ø20.7 mm by 305 mm (Ø.815 in x 12 in) feeder pipe with carbide nozzle holder. Middle: Ø11.1 mm by 305 mm (Ø.438 in x 12 in) feeder pipe with carbide nozzle holder. Bottom: Ø11.1 mm by 51 mm (Ø.438 in x 2 in) feeder pipe with carbide nozzle holder and flow straightener installed.

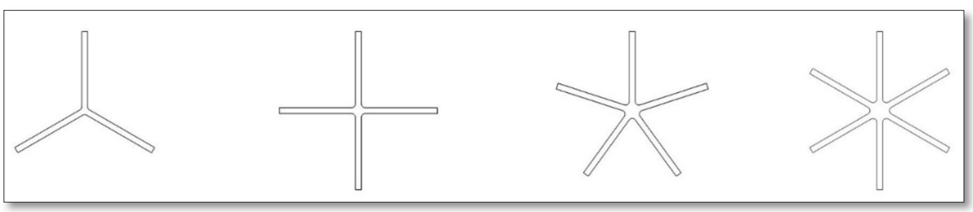

Figure 2. Flow straightener cross sections showing styles with 3, 4, 5, and 6 divisions.

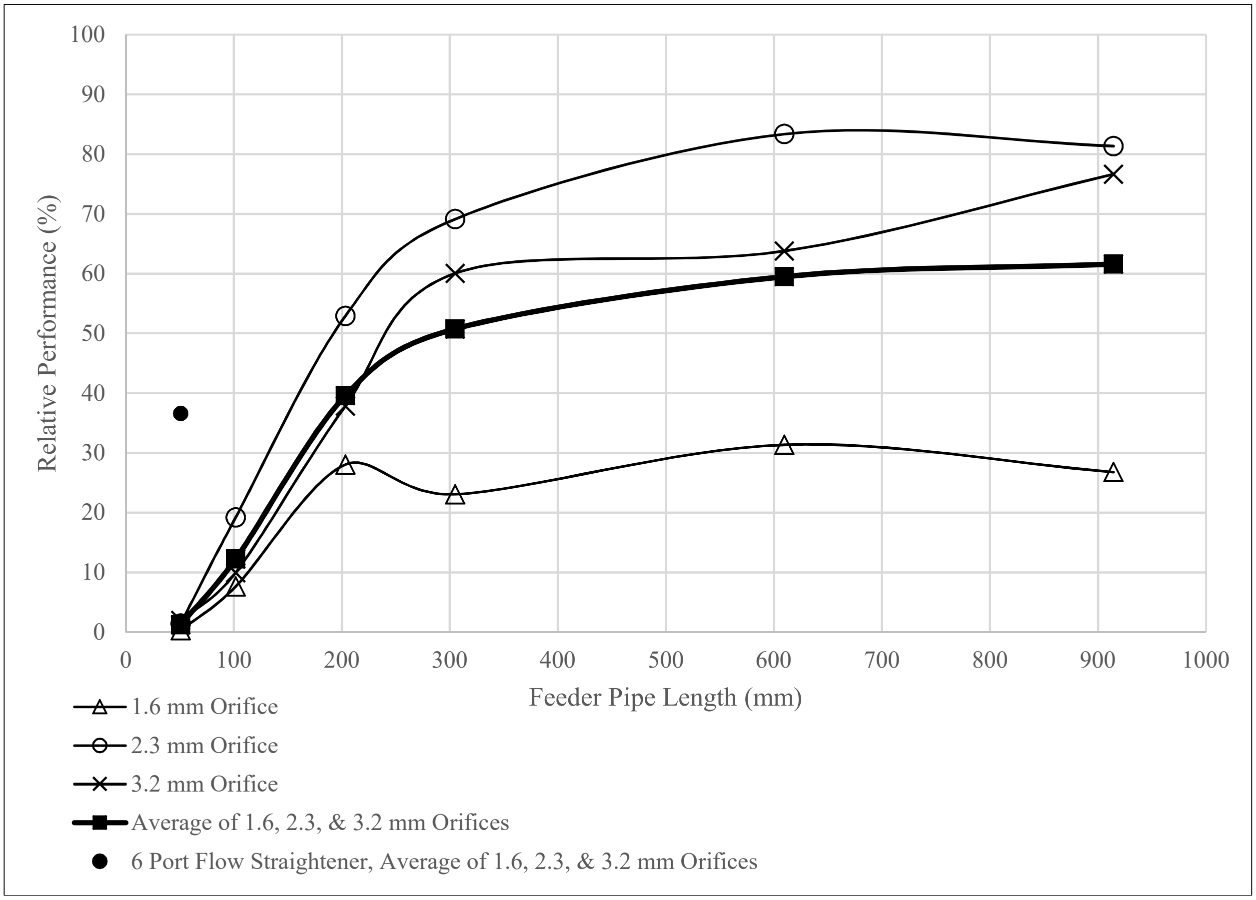

Figure 3. Relative performance of increasing feeder tube length for various orifice sizes, Ø11.1 mm ID feeder pipe. Performance relative to 3.2 mm (.125 in) orifice with Ø20.7 mm by 914 mm (Ø.815 in x 36 in) feeder pipe. Note stabilization of performance with increasing length as well as increased performance with a flow straightener.

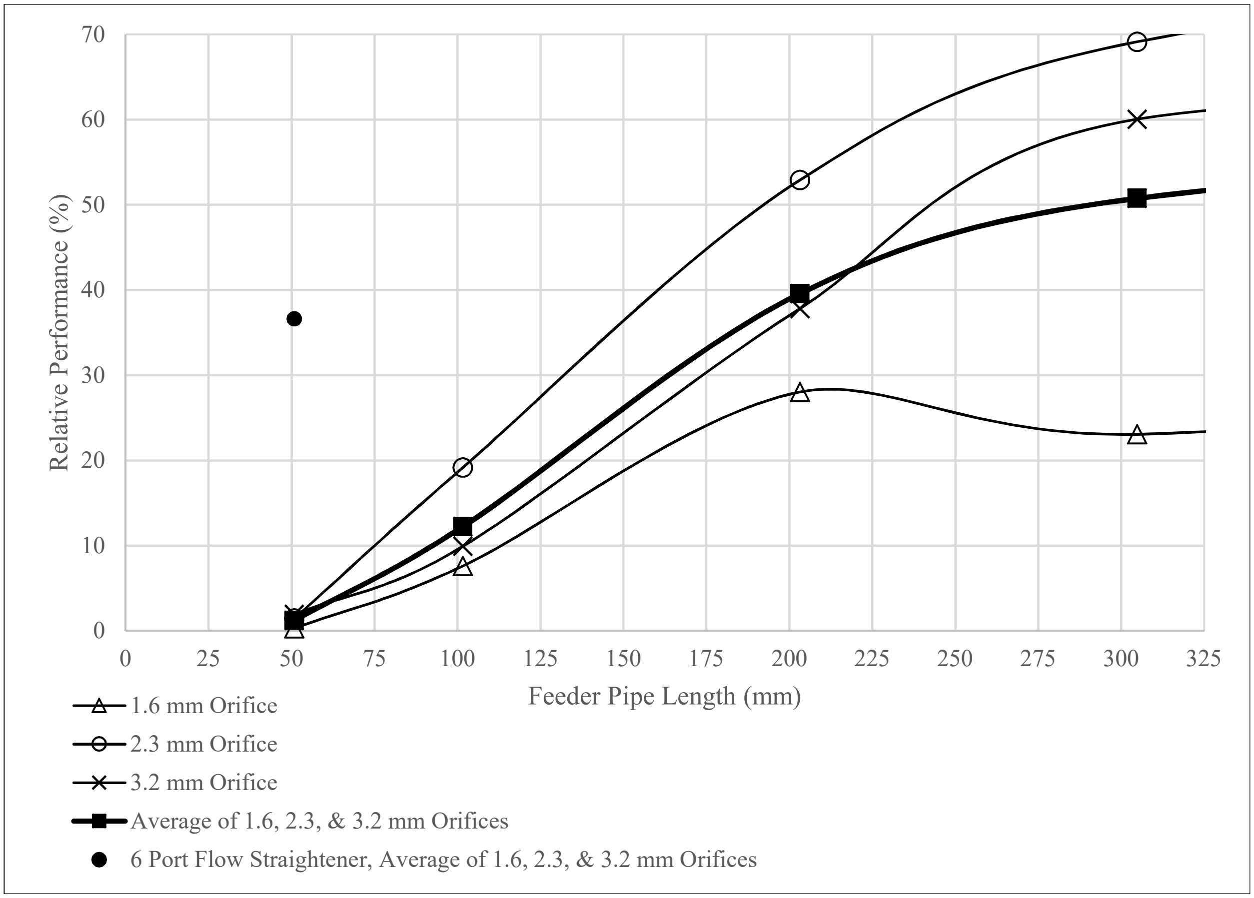

Figure 4. Curtailed view of Figure 3. Relative performance of increasing feeder tube length for various orifice sizes, Ø11.1 mm ID feeder pipe. Performance relative to 3.2 mm (.125 in) orifice with Ø20.7 mm by 914 mm (Ø.815 in x 36 in) feeder pipe. Note stabilization of performance with increasing length as well as increased performance with a flow straightener.

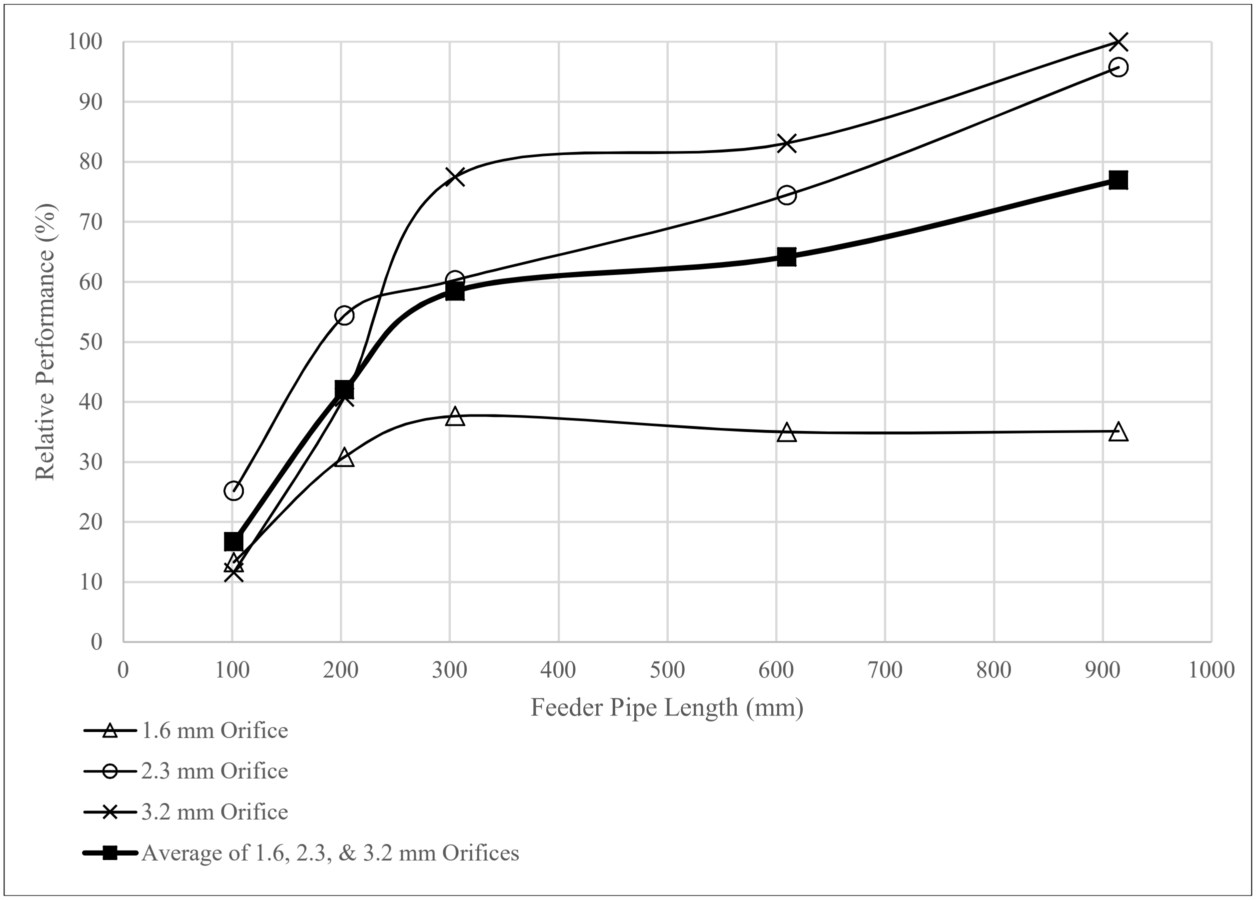

Figure 5. Relative performance of increasing feeder tube length for various orifice sizes, Ø20.7 mm ID feeder pipe. Performance relative to 3.2 mm (.125 in) orifice with Ø20.7 mm by 914 mm (Ø.815 in x 36 in) feeder pipe. Note stabilization of performance with increasing length.

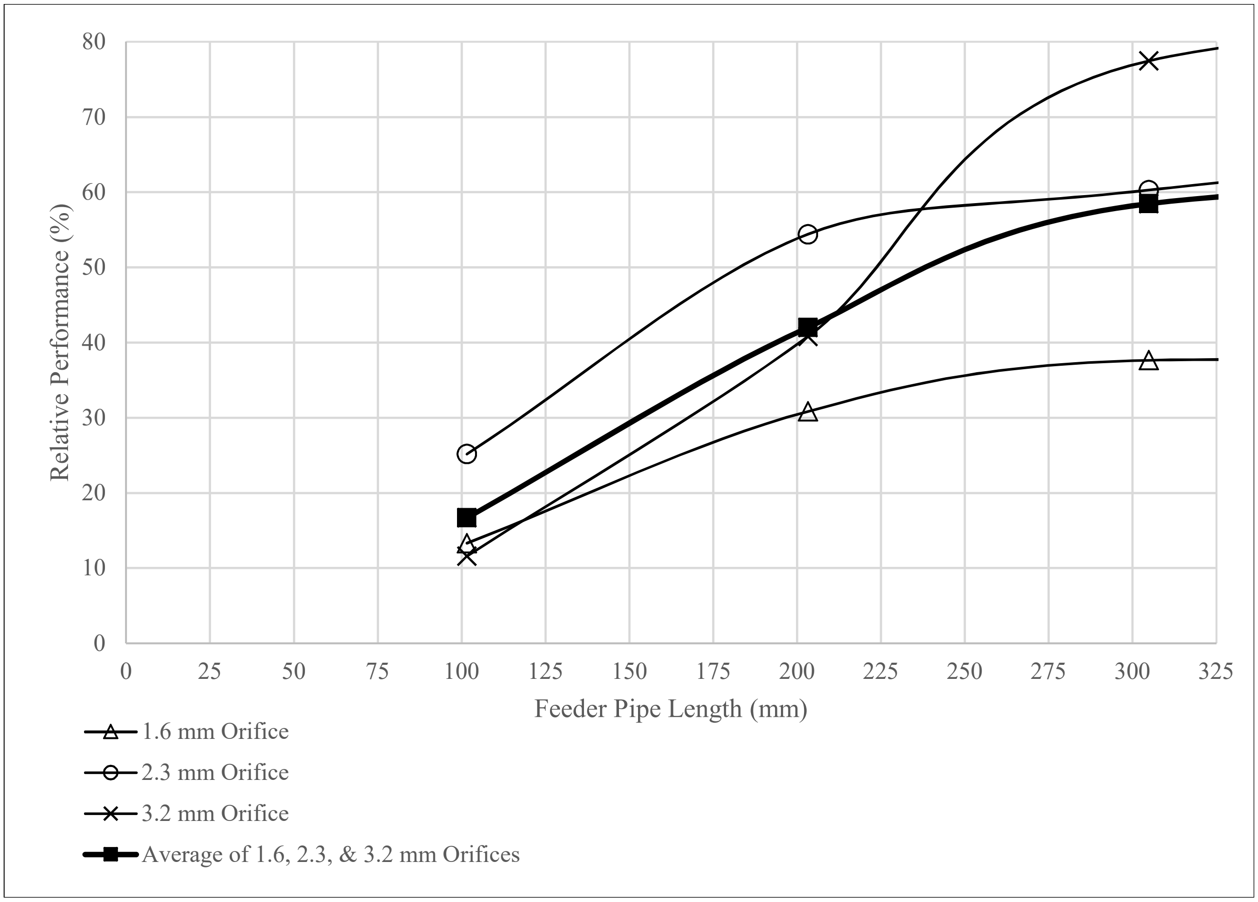

Figure 6. Curtailed view of Figure 5. Relative performance of increasing feeder tube length for various orifice sizes, Ø20.7 mm ID feeder pipe. Note stabilization of performance with increasing length.

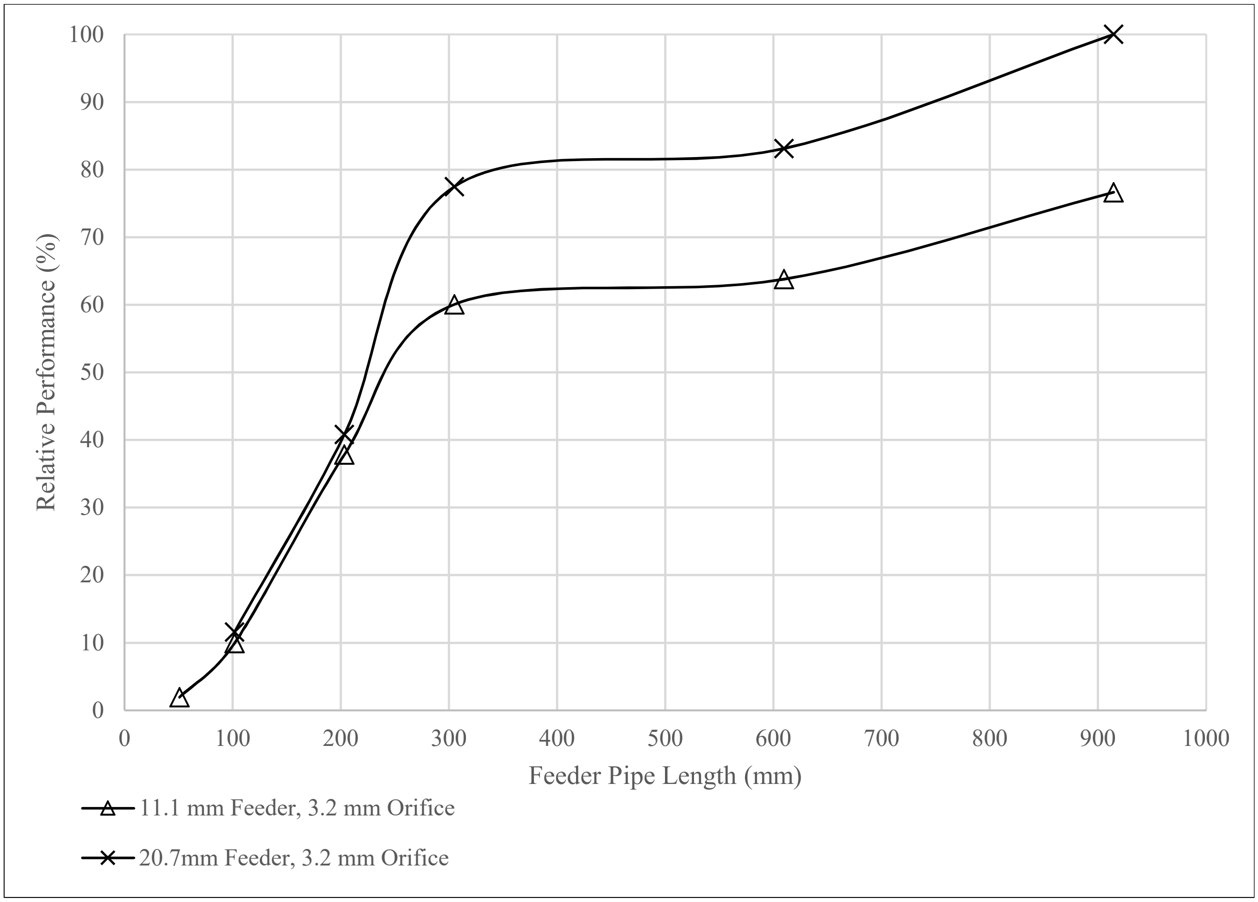

Figure 7. Relative performance of 3.2 mm (.125 in) orifices in 11.1 mm (.438 in) and 20.7 mm (.815 in) feeder pipes of varying length. Performance relative to 3.2 mm (.125 in) orifice with Ø20.7 mm by 914 mm (Ø.815 in x 36 in) feeder pipe. Note similar performance with short feeder pipes and improved performance with a larger diameter feeder pipe as the length increases.

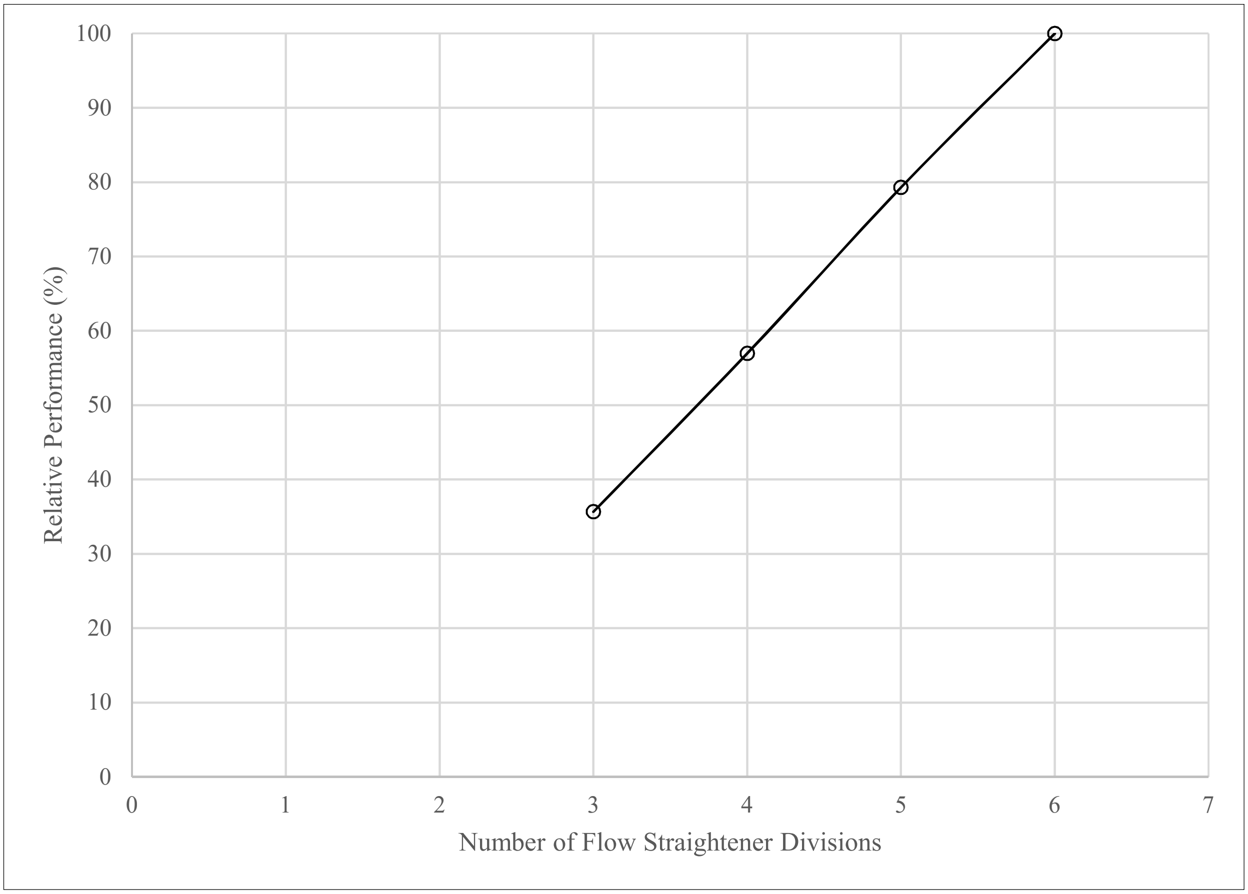

Figure 8. Relative performance of number of flow straightener divisions. Performance relative to average of 1.6, 2.3, and 3.2 mm (.063, .090, .125 in) orifices with Ø11.1 mm by 51 mm (Ø.438 in x 2 in) feeder pipe. Note the trend of increasing performance with increasing divisions as well as the slope of the line suggesting that the trend may continue past six divisions.

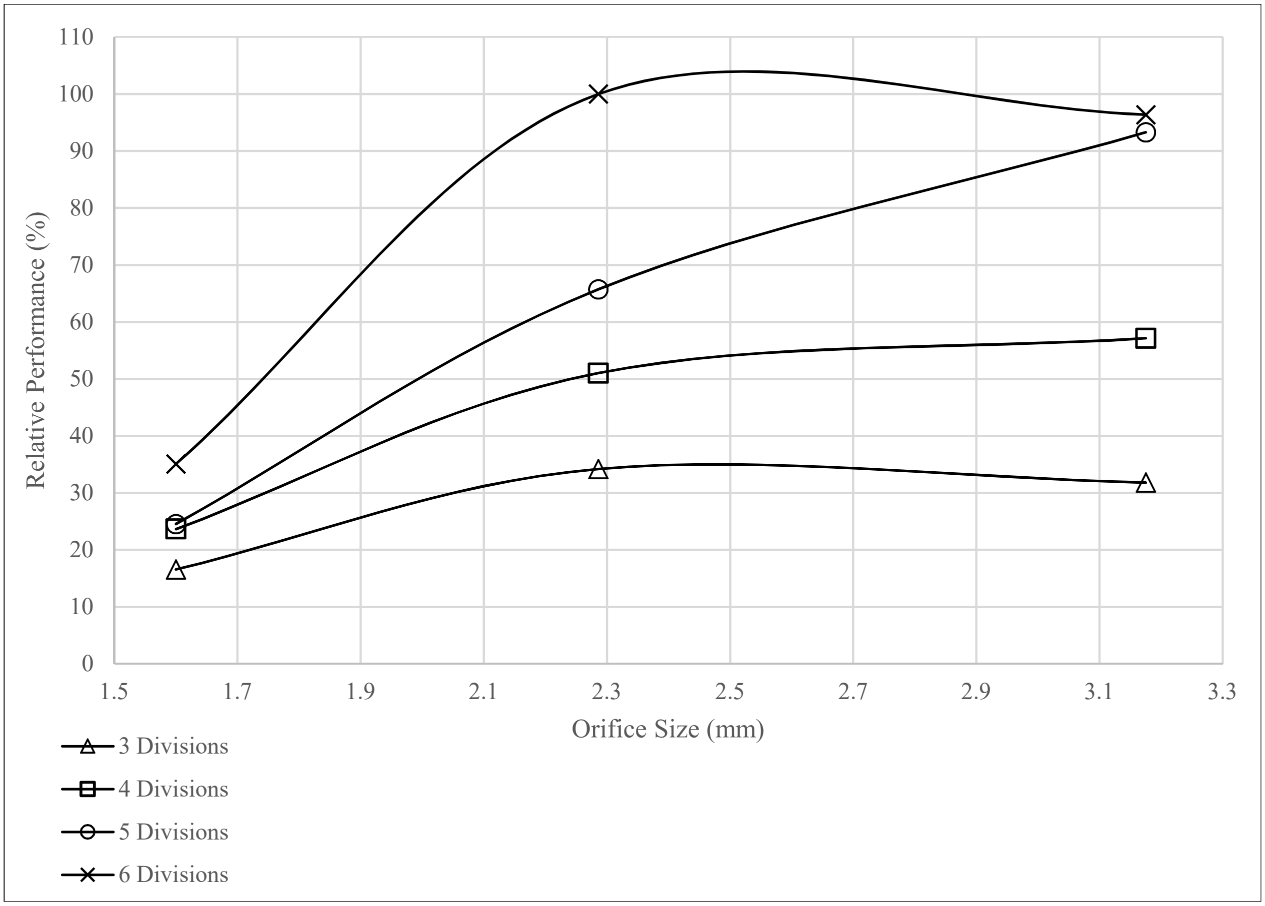

Figure 9. Relative performance of increasing the number of flow straightener divisions vs nozzle size. Performance relative to a 2.3 mm (.090 in) orifice with a six-division flow straightener installed in a Ø11.1 mm by 51 mm (Ø.438 in x 2 in) feeder pipe. Note that the optimal orifice size may be larger than 3.2 mm (.125 in) for the flow straighteners with four and five divisions.

6. REFERENCES

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7. NOMENCLATURE

Re = Reynolds number
u = velocity of fluid
L = characteristic length
ν = kinematic viscosity
Q = volumetric flowrate
A = cross-sectional area of flow pathway
A_r = reduced cross-sectional area of flow pathway. Reduced by wall thickness of flow straightener
P = power
p = pressure
C = constant
E.I. = efficiency index
D_h = hydraulic diameter of feeder pipe
D_f = hydraulic diameter formed by flow straightener
N = number of flow straightener divisions
w = wall thickness of flow straightener