A study of modified plug designs for a globe valve

Robert W. Wagner

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A STUDY OF MODIFIED
PLUG DESIGNS
FOR
A GLOBE VALVE
BY
ROBERT W. WAGNER

A
THESIS
submitted to the faculty of the
UNIVERSITY OF MISSOURI AT ROLLA
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[Signatures]
This thesis presents an empirical study of the characteristics of a cylindrical skirted disk valve used as a flow regulating device. Holes were drilled in two separate removable cylinders, which, when respectively attached to the valve plug adapter, provided control of the flow rate. These cylinders could be moved to permit exposure of more or less flow area by means of the valve stem and crank mechanism. At the closed position of the valve no holes were available for flow within the differential pressure zone of the valve, while at the full-open position all the drilled holes were within the flow zone of the valve, permitting maximum water flow rate.

The method used in adding the holes to the cylinders was "cut and try" because the desired flow rate was already determined before any holes were drilled. In other words, holes were drilled until the desired flow rate was achieved.

The flow rate through each cylinder was established at several cylinder positions by means of the weir trough measuring device. This relationship was used to determine the valve characteristic. The valve characteristic is merely a graphical comparison of the flow rate through the valve versus the percent valve opening.

A second set of curves was plotted to find the relationship between the flow rate and the exposed flow area.
for each cylinder to establish a possible design criterion.

The cylindrical skirted disk plug exhibits a new and accurate approach to fluid flow control. The results of this study indicate that the experimental procedure pursued can be used to obtain desired valve characteristics.
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# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>ABSTRACT</td>
<td>ii</td>
</tr>
<tr>
<td>ACKNOWLEDGEMENT</td>
<td>iv</td>
</tr>
<tr>
<td>LIST OF TABLES</td>
<td>vi</td>
</tr>
<tr>
<td>LIST OF FIGURES</td>
<td>vii</td>
</tr>
<tr>
<td>I. INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>II. REVIEW OF LITERATURE.</td>
<td>3</td>
</tr>
<tr>
<td>III. DISCUSSION</td>
<td>5</td>
</tr>
<tr>
<td>DESCRIPTION OF APPARATUS.</td>
<td>6</td>
</tr>
<tr>
<td>Equipment Design</td>
<td>6</td>
</tr>
<tr>
<td>Water Supply System</td>
<td>11</td>
</tr>
<tr>
<td>Measuring Devices</td>
<td>11</td>
</tr>
<tr>
<td>DESIGN ANALYSIS AND PROCEDURE</td>
<td>14</td>
</tr>
<tr>
<td>EXPERIMENTAL PROCEDURE</td>
<td>16</td>
</tr>
<tr>
<td>First Test Set-up</td>
<td>16</td>
</tr>
<tr>
<td>Second Test Set-up</td>
<td>17</td>
</tr>
<tr>
<td>Test Procedure</td>
<td>17</td>
</tr>
<tr>
<td>WEIR AND ORIFICE THEORY</td>
<td>19</td>
</tr>
<tr>
<td>RESULTS</td>
<td>23</td>
</tr>
<tr>
<td>IV. CONCLUSIONS</td>
<td>28</td>
</tr>
<tr>
<td>V. RECOMMENDATIONS</td>
<td>29</td>
</tr>
<tr>
<td>VI. BIBLIOGRAPHY</td>
<td>31</td>
</tr>
<tr>
<td>VII. VITA</td>
<td>32</td>
</tr>
<tr>
<td>VIII. APPENDIX</td>
<td>33</td>
</tr>
<tr>
<td>Table</td>
<td>Page</td>
</tr>
<tr>
<td>-------</td>
<td>------</td>
</tr>
<tr>
<td>1 WEIR CALIBRATION</td>
<td>34</td>
</tr>
<tr>
<td>2 DATA COLLECTED ON THE LINEAR FLOW PLUG TEST</td>
<td>34</td>
</tr>
<tr>
<td>3 DATA COLLECTED ON THE LINEAR HEAD PLUG TEST</td>
<td>35</td>
</tr>
</tbody>
</table>
# List of Figures

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Assembled Valve</td>
<td>5</td>
</tr>
<tr>
<td>2</td>
<td>Valve Test Section</td>
<td>6</td>
</tr>
<tr>
<td>3</td>
<td>Cylinder and Stud Mounted on the Valve Crank Mechanism</td>
<td>8</td>
</tr>
<tr>
<td>4</td>
<td>Valve Seal Components</td>
<td>10</td>
</tr>
<tr>
<td>5</td>
<td>Overall View of Test Equipment</td>
<td>12</td>
</tr>
<tr>
<td>6</td>
<td>V-Notch Weir</td>
<td>19</td>
</tr>
<tr>
<td>7</td>
<td>Orifice</td>
<td>22</td>
</tr>
<tr>
<td></td>
<td>(Note: The remaining figures are in Appendix)</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Valve Characteristic for the Linear Flow Valve Plug</td>
<td>36</td>
</tr>
<tr>
<td>9</td>
<td>Head versus Percent Valve Opening for the Linear Head Valve Plug</td>
<td>37</td>
</tr>
<tr>
<td>10</td>
<td>Relationship between Area Change and Valve Plug Position Change for the Linear Flow Plug</td>
<td>38</td>
</tr>
<tr>
<td>11</td>
<td>Relationship between Area Change and Valve Plug Position Change for the Linear Head Plug</td>
<td>39</td>
</tr>
<tr>
<td>12</td>
<td>Relationship between Area Change and Flow Rate Change for the Linear Flow Valve Plug</td>
<td>40</td>
</tr>
<tr>
<td>13</td>
<td>Relationship between Area Change and Flow Rate Change for the Linear Head Valve Plug</td>
<td>41</td>
</tr>
<tr>
<td>14</td>
<td>Comparison of Yarnall's Flow Equation and the Orifice Equation for the Linear Flow Plug</td>
<td>42</td>
</tr>
<tr>
<td>15</td>
<td>Comparison of Yarnall's Flow Equation and the Orifice Equation for the Linear Head Plug</td>
<td>42</td>
</tr>
<tr>
<td>16</td>
<td>Weir Flow Equation Flow Rate versus Actual Flow Rate (Weighed)</td>
<td>43</td>
</tr>
</tbody>
</table>
I. INTRODUCTION

There are two basic functions of a valve in a fluid flow system. These are: regulation of flow rate or regulation of pressure. The type of regulation studied in this investigation was the flow rate through a modified globe valve. Specifically, the flow rate of water was evaluated both experimentally and analytically.

There are countless uses of flow control or throttling valves. Hydraulic systems, oil transmission lines, and city water supply systems represent a partial list of applications. The use of throttling valves is as common as regulating your shower or controlling the sprinkler for your lawn.

A continuing study of plug designs for globe valves has been made in order to produce desired valve characteristics. Again, the characteristic of a valve is a plot of the flow rate versus percent opening of the valve. A basic problem encountered in designs using cylindrical plugs is that of sealing the area between the cylinder and the seat in order to constrain the flow to the configurations machined in the wall of the cylinder.

The primary objective of this investigation, therefore, was to design a cylindrical skirt plug for a globe valve with a positive seal at the valve seat. Two valve plug configurations were tested in an attempt to achieve the predetermined valve characteristics. A secondary objective
of this investigation was to try to establish a more deliberate design procedure for programming the valve characteristic.
II. REVIEW OF LITERATURE

The study of control valves dates back over one hundred years. Since then butterfly valves, plug valves, gate valves, globe valves, and others too numerous to mention have been investigated and modified to fit almost every flow requirement. Therefore, to maintain conciseness, this review will be limited to the globe valve and its derivatives.

Before 1900, piston type globe valves were already in use in the steam engines of railroads and ships. Seely and Talbot (1) conducted experiments with globe valves in 1918, for the purpose of correlating the head loss within small globe valves with various valve openings. The principle contribution of Seely and Talbot's study was the fact that the head loss depended primarily upon the shape of the exit passage of the valve. Their conclusion that head loss in a valve varied directly with the square of the flow velocity has proved to be a very useful tool in hydraulic studies. Lansford (2), in a later study of globe valves, verified these results.

In his treatise on hydraulics, Addison (3) disclosed that although losses within globe valves were rather large, such valves made excellent flow regulators. Rhodes (4), in 1941, suggested several types of ported plug designs to achieve desired valve characteristics. One was solid disk with a symmetrical parabolic cross section designed to produce a parabolic flow relationship with percent of valve
lift. He also mentioned the V-ported plug valve which produced a similar parabolic flow characteristic because of the mathematical relationship between the area and the altitude of a triangle. However, Rhodes' most significant statement in relation to this study was: "Where greater accuracy is required than will result from the use of the simple triangular shaped orifice, the plug is made with a series of small orifices of different shapes so designed as to give a total flow at any time exactly equivalent to that needed for perfect control."

Beard (5) completed a thorough investigation of V-port and parabolic plug designs in 1957. His emphasis of valve rangeability, "the ratio of maximum to minimum flow over which the usable flow characteristic exists," is something every valve designer must bear in mind.
III. DISCUSSION

The discussion is divided into two primary sections: the first subdivision is concerned with the equipment requirements and experimental procedure, and the second subdivision is concerned with analysis of the experimental results. In addition, a section on experimental apparatus discusses the purpose and capability of each piece of equipment.

A section covering design analysis and experimental procedure dwells on the methods used in designing the flow

![Diagram of assembled valve](image)

**FIGURE 1**
ASSEMBLED VALVE
controlling device, and the procedure used in conducting the tests on the complete valve. The remainder of the discussion tabulates and evaluates the experimental results with known weir and orifice theory in order to establish a mathematical relationship that can be used in future design procedures.

DESCRIPTION OF APPARATUS

Equipment Design - A major portion of this research was concerned with the design of a valve with a cylindrical skirt plug. For convenience, it was decided that the valve plug
would be fabricated to fit in an existing 3-inch diameter Crane globe valve, since this commercial valve was already installed in a pump test loop in the Mechanical Engineering Laboratory. The Crane valve stem, plug assembly, and seat were removed from the body and the plug was modified. The cylindrical skirt plug illustrated in Figures 1 and 3 was then installed in place of the plug and seat of the former valve. The seat of the new valve was machined to the same thread pitch and pitch diameter of the original Crane seat since it had to fit the threaded holder within the valve body.

The modified valve seal contained three component parts; the seat, the plug adapter, and the cylindrical skirt plug. All three were machined from aluminum round stock. The purpose of the seat was to provide a machined surface upon which the valve would seal in the closed position. In order to prevent water flow by the housing, a rubber seal, which will be described later in the discussion, was installed. The plug adapter serves as a connection between the cranking mechanism, the stem, and the actual flow controlling device— the cylindrical skirt plug. The purpose of using detachable cylinders was to reduce the machining time in case a cylinder had to be scrapped during the test period.

The main difficulty associated with modification of the valve was concerned with finding a method to seal the
valve from leaks while under fluid pressure. Close attention was given to the means of sealing the gap between the outside diameter of the cylindrical skirt plug and the inside diameter of the seat. This task was accomplished by the use of a butyl O-Ring which was flexible enough to allow relative motion between the cylinder and the seat.

In order to keep the diameter of the valve as close to 3 inches as possible, the outside diameter of the cylinder was set at 3.000 ± .005 inches and a suitable O-Ring was chosen to dynamically seal the cylinder. A # 337 Precision O-Ring was used. To insure a positive seal on the inside diameter.
of the O-Ring, a .025 inch diametral squeeze was recommended by the manufacturer. Since the inside diameter of the O-Ring was 2.975 inches, an adequate seal was obtained for pressure differentials up to 500 psi, according to the manufacturer's specifications.

After the outside diameter of the cylinder was determined, the inside diameter of the seat was bored to allow the cylinder to pass through with no interference or scarring. At that time the slot for containing the O-Ring was bored as close to the leading edge of the seat as possible to prevent appreciable flow interference by the seat itself upon opening the valve. A 45° chamfer was then turned on the leading edge of the seat to mate with the 45° bevel on the plug adapter at the full-closed position of the valve.

One of the principal machining problems was that of maintaining the concentricity of the moving parts to insure that the cylinder would not bind or cock as the valve was opened or reclosed. To prevent this from occurring, the cylinders were all initially bored to 2.750 inches, faced and threaded at one end to fit the plug assembly. The cylinders were then screwed on the plug and turned down to the specified 3.000 inch outside diameter. This operation kept the cylinder concentric with the drive stem. As a final means to prevent binding and cocking, the drive stem was connected to the plug in such a way as
to allow only a "push" or "pull" force to be transferred to the plug. Hence, the cylinder and plug assembly was constrained to move only in horizontal translation removing the possibility that the cylinder would unscrew from the plug during stem movement. Figure 3 shows the component parts of the valve seal; the seat, the detachable cylinders, and the plug adapter.

The actual mechanics of the valve is simple. The controlled variable, water, passes through the inside of the cylinder from the high pressure side of the valve to the
low pressure side-through whatever opening it "sees" in the cylinder wall. For water to flow, however, the openings must appear beyond the O-Ring on the low pressure side of the seat. It is apparent here that the O-Ring divides the high and low pressure sides of the valve. The openings mentioned above are discussed later in this thesis.

**Water Supply System** - The test set-up shown in Figure 5 was used in all the tests performed on the valve. The pump used was an Aurora centrifugal pump mechanically driven by a General Electric d-c motor. The pump capacity was 200 gallons per minute and it was already connected in the test set-up.

The suction side of the pump is connected to a sump below the lab floor, and the discharge side is connected to the valve where it empties into the weir trough. Upon leaving the tank, the water flows back into the sump to complete the circulation process. The piping from the pump to the valve has a nominal 3-inch inside diameter which is the same diameter as the pipe at the valve outlet. The pipe and valve extending above the test valve seen in the over-all test set-up figure is used for another laboratory experiment and was shut off during these tests.

**Measuring Devices** - Only two pressure gauges were used in the experiment and both were used in conjunction with the pump to indicate the pump suction and discharge pressures. The gauge on the suction side was a U.S. Gauge with a range
of -30 inches of mercury to +60 inches of mercury. The
gauge had increments of one inch mercury. The gauge on
the discharge side was a Williams and Hussey with a range
of zero to +60 psig in increments of 1 psi.

The principle measurement required was that of the wa-
ter flow rate. The weir trough shown in Figure 5 is equip-
ped to measure flow indirectly by measuring the head with
the hook gauge attached to the weir and using the head value
in the weir equation to calculate the flow. This equation
is derived in the weir and orifice theory section. The
weir was actually calibrated for several flow rates by collecting the water for a period of one to two minutes, depending upon the magnitude of the head, and weighing the amount collected. A comparison of the calculated flow rate obtained from the weir equation and the actual weighed flow per second is illustrated in Figure 16. The curve indicates that the weir equation flow rate was nearly equal to the actual flow rate—the maximum error being about 5%.

The hook gauge mentioned in the previous paragraph was calibrated in inches. The vernier fixed to the frame of the gauge allowed the head to be measured to 0.01 inch with good accuracy. To enhance the accuracy of reading the head, flow straighteners were placed ahead of the hook gauge in the flow stream, thereby reducing oscillations within the measuring tube. Also, to remove the effects of approach from the head measurement, the gauge was placed several feet ahead of the weir. The phenomenon of adhesion of the water to the side of the glass measuring tube was compensated for by placing the point of the hook one-half the distance from the center of the cross section to the wall; i.e., an average reading was taken. During the actual test of the valves, the hook gauge was found to produce inconsistent results because of a vacuum above the water level in the hook gauge. The inconsistency was removed by drilling a hole through the top of the instrument to provide venting.
The actual procedure for locating the water level in the hook gauge was simple. The viewer had to merely watch the mirror image of the hook, while stationed below the water level, until the actual hook and the mirror image touched each other. All head readings on the weir had to be corrected to zero because the actual zero head reading was 8.57 inches on the hook gauge. This zero reading was determined by means of a hydraulic level.

**DESIGN ANALYSIS AND PROCEDURE**

Valve design is generally concerned with achieving a desired valve characteristic which is important from the standpoint of control of fluid flow. This characteristic is the variation of flow with the change of valve opening. The valves designed for this study were developed using two predetermined characteristics, which will be referred to as control graphs. They were: A linear relationship between the flow through the valve and the valve opening, and a linear relationship between head on the weir (flow measuring means) and the valve opening.

Before the first configuration was machined, the desired graph of flow versus percent valve opening was made and a straight line relationship was chosen. The chosen characteristic passed through the origin of the plot to the known flow at the full flow condition for the previous 3-inch Crane globe valve at maximum opening. This was done to obtain maximum flow at the full-open position of the
valve. The method employed in achieving a desired flow was to drill holes in the detachable cylinder which allowed the water to flow from the high pressure inside to the atmospheric pressure downstream from the valve. The actual pressure drop across the valve will be discussed in a later discussion.

A special procedure was used to drill the holes in the cylinder in order to expedite the experimental procedure. A random pattern of holes would involve too much experimenting time. Therefore, a more expedient way was developed.

Since the thread pitch of the drive screw was found to be 4 threads per inch, each one-half revolution resulted in a 1/8 inch cylinder movement. Therefore, the cylinder was divided in 1/8 inch circular bands, each of which must contain enough holes to meet the flow requirement for the corresponding valve position. In this way each successive one-half revolution would add a new band of holes—no single hole of which would be divided by the sealing mechanism (the O-Ring). To keep the holes within the respective bands, a 3/32 inch diameter drill was used together with a rigid clamping arrangement that insured that the hole centers were in the same plane even though the cylinder had to be removed and rotated each time a hole was drilled.

To determine the number of holes within each band, reference was made to the two control graphs depicting the desired flow characteristics. Both cylinders were designed
with this procedure. For each 1/8 inch advance of the cylinder from the full-closed position to the full-open position a corresponding flow rate existed on the respective control graph. This was the criterion for the design procedure. Hence, the method was "cut and try" in that a series of holes was drilled to prevent overrunning the specified flow rate for the position. When the flow rate was accomplished, the cylinder was advanced to the next position and the procedure was repeated.

EXPERIMENTAL PROCEDURE

First Test Set-up - The initial set-up was made to test the feasibility of the valve itself, since the experimental set-up including the pump, piping, and the flow measuring device had been tested repeatedly by a Mechanical Engineering Systems Laboratory class. The first configuration machined in the cylindrical shell was a triangular shaped hole. The opening was symmetrical on opposite sides of the cylinder. The sum of the areas of the two configurations was equated to the area of the entrance pipe in order to obtain the maximum flow rate at the full-open position of the valve. The triangle was isosceles and was positioned so that the side opposite the apex was parallel and directly under the periphery of the O-Ring at the full-open position of the valve.

Upon testing this cylinder, difficulties were encountered with severing of the O-Ring during the valve advance-
ment period. This was caused by the expansion of the O-Ring in triangular non-seal areas of the cylinder, resulting in a pinching action on the O-Ring by the edges surrounding the triangle at the outside surface of the cylinder. Hence, a different configuration had to be used which would be compatible with the sealing device or a new sealing device would be required.

**Second Test Set-up** - With the small 3/32 inch diameter holes as the new configuration, a new cylinder was tested and found to perform very smoothly with only a slight vibration at the initial position of the valve after it was unseated.

A preliminary test indicated that the head level in the hook gauge was too oscillatory to be measured accurately, so a short flexible hose was connected to the hook gauge inlet and extended to the center of the tank—keeping the opening perpendicular to the flow so as not to add the velocity head of the flowing water. This remedied the oscillation problem in the hook gauge, thus permitting experimental evaluation of the valve characteristic.

**Test Procedure** - The actual test procedure was simple and could be performed by one man. After the holes had been drilled in the wall of the cylindrical skirt in accordance with the design procedure, the cylinder was mounted on the plug adapter and the complete valve was installed in the housing. The crank mechanism was advanced one-half revolution from the seated or full-closed position to allow the
first \( \frac{1}{8} \) inch band to appear between the high and low pressure zone of the valve. It is in this first band that the holes were placed to allow just enough flow rate for the initial valve position to correspond to the desired flow rate from the control curves.

The d-c motor driving the pump was started and the water in the weir trough was allowed to come to steady-state for the specific valve position. Since the capacitance of the weir trough was large, time was required to reach steady-state conditions. For small flow rates the steady-state condition required approximately one-half hour.

For each test position (one-half revolution of the crank) the design procedure of "cut and try" was carried out until the desired flow rate corresponding to the respective control curve was realized. The procedure was repeated until the full-open position was reached. At this time the overall test of the newly designed cylindrical skirt plug was completed. The procedure for the final test closely paralleled the previous procedure except that readings of the pump suction pressure, pump discharge pressure, and the hook gauge were recorded for each valve position. This operating procedure was repeated for the second cylinder and the same readings were recorded for the valve positions studied.
Since the flow rate could not be measured directly during the experimental procedure, it had to be calculated using Yarnall's equation. The following is the derivation of the weir equation for a 90° V-notch weir. Figure 6 applies to this derivation, where:

\[ Q = \text{total weir discharge in ft}^3/\text{sec.} \]
\[ H = \text{static head on weir in ft.} \]
\[ h = \text{static head at any distance below the water surface in ft.} \]
\[ \theta = \text{included angle of V-notch, 90°} \]
\[ C_d = \text{coefficient of discharge for the weir} \]
\[ V_m = \text{velocity of approach of upstream channel in ft./sec.} \]
\[ dh = \text{differential element of head.} \]
Assumptions:

(1) Torricelli's equation for discharge prevails for weir discharge or \( V_{avg} = (2gh)^{1/2} \).

(2) The velocity of approach is negligible, hence the velocity head at the plate location is negligible.

Therefore,

\[
Q = C_d A V
\]

and,

\[
dQ = C_d V dA \\
= C_d [2(H - h) \tan \frac{1}{2} \theta] (2gh)^{1/2} dh,
\]

where,

\[
dA = 2(H - h) \tan \frac{1}{2} \theta dh
\]

\[
V = (2gh)^{1/2}. \quad \text{(Torricelli's equation)}
\]

Hence,

\[
Q = C_d \int_0^H [2H \tan \frac{1}{2} \theta (2gh)^{1/2} - 2h \tan \frac{1}{2} \theta (2gh)^{1/2}] dh,
\]

which reduces to

\[
Q = C_d \frac{8}{15}(2g)^{1/2} \tan \frac{\theta H^5}{2}.
\]

Since for V-notch weirs \( C_d \) has been experimentally found to be 0.58, and \( \theta \) is 90°, the above equation reduces to Yarnall's equation which will be referred to as Equation 1 in the following discussion of experimental results,

\[
Q = 2.48H^{5/2}. \quad (1)
\]
In the following section, the discussion of experimental results, an analogy using orifice theory will be used in an attempt to correlate the test results. A brief derivation will be given at this time concerning water flow through orifices and will serve as a reference for that article. Figure 7 shows the location of the reference points 1 and 2 with the corresponding nomenclature:

- $V_1 =$ upstream velocity in ft./sec.
- $V_2 =$ velocity at the vena-contracata in ft./sec.
- $p_1 =$ pressure at the upstream position in lb./ft.$^2$
- $p_2 =$ pressure at the downstream position in lb./ft.$^2$
- $\gamma =$ specific weight in lb./ft.$^2$
- $g =$ acceleration due to gravity in ft./sec.$^2$
- $z_1 =$ elevation of flow at upstream position in ft.
- $z_2 =$ elevation of flow at downstream position in ft.
- $Q =$ volume flow rate through the orifice in ft.$^3$/sec.
- $A_1 =$ area of inlet pipe in ft.$^2$
- $A_2 =$ area of cross section of flow at vena-contracata in ft.$^2$
- $A =$ area of orifice in ft.$^2$
- $C_v =$ discharge coefficient of the orifice
- $C_c =$ contraction coefficient of the orifice

Bernoulli's equation for steady flow is:

$$\frac{V_1^2}{2g} + \frac{p_1}{\gamma} + z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\gamma} + z_2,$$
but because the elevation change for a very small orifice even for vertical flow is negligible,

\[ Z_1 = Z_2. \]

According to the continuity equation,

\[ Q = AV = A_1V_1 = A_2V_2, \]

hence,

\[ V_2 = V_1A_1/A_2; \]

which, when combined with Bernoulli's equation along with the discharge and contraction coefficients of the orifice yields Equation 2,

\[ Q = CA\sqrt{2g(p_1 - p_2)/\gamma}, \]  

(2)
and Equation 3,

\[ C = C_v C_e / \sqrt{1 - C_e^2 (A/A_1)^2} \]  \hspace{1cm} (3)

RESULTS

The resulting curves for the two cylinders tested proves rather conclusively that nearly any desired valve characteristic may be produced with the experimental procedure used in this investigation. Figures 8 and 9 are the basic experimental relationships that were produced from the two cylinders tested. Figure 8 shows that a linear relationship between flow rate and percent valve opening was accomplished. Figure 9 reveals that the second requirement of the investigation - a linear relationship between weir head and percent valve opening - was also realized. Although both curves were tested with the same procedure at one-eighth inch advancements a check was made at smaller intervals to see if the flow rates at the un-tabulated positions justified the plotted straight line between the tabulated positions. This investigation revealed that the error in assuming the straight line relationship between tabulated positions was negligible.

Another important aspect of this investigation was the relationship between area change and cylinder advance, depicted by Figures 10 and 11. For the linear head cylindrical skirt plug the curve was parabolic in nature, while the linear flow plug had a very distinct linear curve
especially at smaller flow rates. A further and more significant analysis produced from these curves is shown, however, in the flow rate versus percent available area curves, Figures 12 and 13. These graphs indicate that for both cylinders the curves did not differ appreciably from a straight line; hence, the configuration area (the holes) appeared to be directly proportional to the flow rate. In orifice theory the flow rate was also found to vary directly with the area permitting flow, namely,

\[ Q = CA\sqrt{2g\Delta p/\gamma}; \]

hence, the analogy between the configuration flow through the cylinders and simple orifice flow may now be supported. First, however, in order to validate the use of this orifice equation, the following three assumptions were made:

1. The cylindrical plug is an orifice plate and the holes in the cylinder wall are considered as individual orifices.

2. The roundness of the orifice plate is negligible for each orifice because the 3/32 inch diameter is small compared to the 3-inch diameter of the cylinder.

3. The high pressure side of the valve will be considered to be an infinite reservoir with a constant pressure of 20 psig.
The first two assumptions are self-explanatory due to the geometry of the cylinder and the flow configurations. The third, however, needs some explanation. The pump discharge pressure, recorded for each test position, indicated a variable pressure, first increasing, then decreasing with increasing flow rates for both cylinders. Since the pressure did not change appreciably throughout the tests and the head losses within the piping system amounted to approximately 5 psi, the average pump discharge pressure minus the head losses within the piping system resulted in the 20 psig pressure at the valve inlet. It must be kept in mind here that a 3 psi pressure drop from the pump to the valve is immediately accounted for because of the elevation change of the water.

With the assumptions stated, the coefficient $C$, of the orifice flow equation may be evaluated. Since by the second assumption, the area of the individual orifices is small in comparison with the reservoir area, the pipe cross section, the $A/A_1$ term of Equation 3 is approximately zero, hence,

$$C = C_v C_c.$$

Vennard (6) states that the product of $C_c$ and $C_v$ is 0.61 for a sharp edged orifice and 0.98 for a rounded edge orifice. An average of these two coefficients yielded 0.80 for the discharge coefficient since the sharpness or roundness of the inside edge of the holes could not be
determined. Therefore, Equation 2 can be reduced to Equation 4,

$$Q = 0.80A\sqrt{2g\Delta p/\gamma}.$$  (4)

This equation evaluates the flow through a single hole in the cylinder, where again, $A$, is the area of the hole and $\Delta p$ is the pressure drop across the orifice - 20 psi - since the gauge pressure of the low pressure side of the valve was atmospheric pressure. To find the total flow rate for each valve position, Equation 4 was multiplied by the total number of holes permitting flow. The theoretical flow rate at each valve position, Equation 4, was then plotted versus the actual flow rate found by the weir equation. Figures 14 and 15 indicate an equality between the actual weir flow rate and orifice flow rate for the low flow rate positions of the valve. The deviation of the curves from the linear relationship at high flow rates was expected, however, since the assumption of a constant pressure reservoir was no longer valid due to a finite upstream velocity in the pipe at the highest flow positions of the valve. The fact that close spacing of orifices on a plate reduces the flow capability of each orifice should not contribute a significant error since only the inlet flow stream is affected by the close spacing of holes. This type of error would combine directly with the coefficient of discharge, a term that was already approximated. As stated above, the orifice flow equation was indeed a good
approximation of the flow rate and offers a mathematical approach to the problem of flow control.
IV. CONCLUSIONS

The results of this investigation have led to the following conclusions:

1. Almost any flow characteristic can be achieved by use of the cylindrical skirt plug - depending only on the cylinder size and stroke length.

2. There exists a linear relationship between configuration area change and percent opening for the valve plug with a linear characteristic.

3. For small flow rates the orifice flow equation offers a reliable mathematical approximation of the flow rate, thus providing a means of predicting the number of holes required to achieve a given flow rate.

4. The flow rate at all but the full-open position of the valve varied directly with the percent available flow area.

5. The head loss of the cylindrical skirt plug is greater than the head loss of the original poppet plug in the Crane valve.

6. The type of plug designed offers an accurate flow control capability from the 0% to 60% open positions.
V. RECOMMENDATIONS

In order to benefit from what has already been done, the author suggests that the following minor changes be made to improve the valve design, as well as system performance. The primary suggestion is that more instrumentation, particularly within the valve test section, be added because the pressure drop across the cylinder portion of the valve is a critical factor in the valve's design. A pressure tap above the valve, as well as below the valve, is suggested, since at low flow rates a small but definite suction condition existed due to the vertical drop in the pipe at the valve exit. This condition could be heard by obstructing the air flow into the flange with the hands.

A second centrifugal pump, which is shown in the overall test set-up, might be coupled to stabilize and increase the flow capability of the water supply system. This would also increase the stability of the upstream pressure in the valve test section, a very desirable condition.

There are two basic changes within the valve itself that would improve flow stability through the configuration. One change would be to increase the lift of the valve by redesigning the stem connection to allow more room for the stroke. Another possibility would be to remove the volume on the stud beyond the outside diameter of the cylinder since the valve actually doesn't have to seat at the "off" position.
Thus, the cylinder length could be increased to the constrained distance from the O-Ring to the rear body of the valve. Both of these suggestions would increase the control stroke of the valve, the axial distance of the cylinder within which all the holes must be located.

Aside from the design changes, there are several areas within the scope of this thesis that merit further investigation. A study of higher flow phenomenon should be undertaken to see if the control aspects of the valves studied in this thesis are valid for higher flow rates. A similar study could be made of the pressure regulation capabilities of the cylindrical skirt plug at small and large flow rates. The effect of reducing the hole size should be investigated to improve control of both pressure and flow rate. An area that also suggests further investigation is that of a solid plug using the percent area available for flow versus flow rate curves of this thesis as a design criterion.

One of the primary disadvantages of the cylindrical skirt plug designed in this thesis is the large head loss incurred by the directional change of the flow stream. The head loss of the modified plug was larger than the original poppet plug for all valve openings. A second disadvantage of the modified plug is concerned with contamination due to solid matter existing in the flow medium. Because of the size of the holes clogging could become an acute problem.
VI. BIBLIOGRAPHY

1. Seely, F.B. and Talbot, A.N. (1918), University of Illinois Experimental Station, Bulletin Number 105.

2. Lansford, W.M. (1943), University of Illinois Experimental Station, Bulletin Number 340.


VII. VITA

The author was born January 20, 1943, in St. Louis, Missouri. He received his primary and secondary education in Webster Groves, Missouri, a suburb of St. Louis. In September, 1960, he entered Colorado University, Boulder, Colorado, then transferred to the University of Missouri School of Mines and Metallurgy in February, 1961, where he received his Bachelor of Science degree in Mechanical Engineering in May, 1964.

He has been enrolled in the Graduate School of the University of Missouri at Rolla since September, 1964, and has held a Graduate Assistantship in Mechanical Engineering during his enrollment in the Graduate School.

The author is a student member of the American Society of Mechanical Engineers and the American Institute of Aeronautics and Astronautics. He is also a member of Tau Beta Pi and Pi Tau Sigma engineering honor societies.
VIII. APPENDIX

The following appendix contains Tables 1 to 3, tabulated data taken during the test period. Table 1 is the weir calibration curve, while Tables 2 and 3 represent recorded and calculated data pertinent to the test of each cylinder. Figures 8 through 16 are a series of plotted curves representing the tabulated data in Tables 1, 2, and 3.
### TABLE 1
WEIR CALIBRATION

<table>
<thead>
<tr>
<th>Run Number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Head (in.)</td>
<td>1.36</td>
<td>2.36</td>
<td>3.08</td>
<td>3.91</td>
</tr>
<tr>
<td>Time (sec.)</td>
<td>60</td>
<td>120</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>Weight (lbs.)</td>
<td>45</td>
<td>330</td>
<td>285</td>
<td>604</td>
</tr>
<tr>
<td>Density (lbs./ft.)</td>
<td>62.4</td>
<td>62.4</td>
<td>62.4</td>
<td>62.4</td>
</tr>
<tr>
<td>Qa (ft./sec.)</td>
<td>0.012</td>
<td>0.044</td>
<td>0.076</td>
<td>0.161</td>
</tr>
<tr>
<td>Qth (ft./sec.)</td>
<td>0.011</td>
<td>0.045</td>
<td>0.077</td>
<td>0.151</td>
</tr>
</tbody>
</table>

### TABLE 2
DATA COLLECTED ON THE LINEAR FLOW PLUG TEST

<table>
<thead>
<tr>
<th>% Valve Opening</th>
<th>Pump Discharge Pressure (psig)</th>
<th>Flow Rate (cfs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>14.3</td>
<td>27</td>
<td>0.035</td>
</tr>
<tr>
<td>28.6</td>
<td>30</td>
<td>0.104</td>
</tr>
<tr>
<td>42.9</td>
<td>30</td>
<td>0.156</td>
</tr>
<tr>
<td>57.2</td>
<td>28</td>
<td>0.196</td>
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<tr>
<td>71.5</td>
<td>27</td>
<td>0.241</td>
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<tr>
<td>85.8</td>
<td>24</td>
<td>0.298</td>
</tr>
<tr>
<td>100.0</td>
<td>23</td>
<td>0.348</td>
</tr>
</tbody>
</table>
### TABLE 3
**DATA COLLECTED ON THE LINEAR HEAD PLUG TEST**

<table>
<thead>
<tr>
<th>% VALVE OPENING</th>
<th>PUMP DISCHARGE PRESSURE (psig)</th>
<th>FLOW RATE (cfs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>16.6</td>
<td>27</td>
<td>0.055</td>
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<tr>
<td>33.2</td>
<td>29</td>
<td>0.027</td>
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<tr>
<td>49.8</td>
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<td>0.083</td>
</tr>
<tr>
<td>66.4</td>
<td>28</td>
<td>0.163</td>
</tr>
<tr>
<td>83.3</td>
<td>24</td>
<td>0.280</td>
</tr>
<tr>
<td>100.0</td>
<td>23</td>
<td>0.375</td>
</tr>
</tbody>
</table>
FIGURE 8 - VALVE CHARACTERISTIC FOR THE LINEAR FLOW VALVE PLUG
FIGURE 9 - HEAD VERSUS PERCENT VALVE OPENING FOR THE LINEAR HEAD VALVE FLUG
FIGURE 10 - RELATIONSHIP BETWEEN AREA CHANGE AND VALVE PLUG POSITION CHANGE FOR THE LINEAR FLOW PLUG
FIGURE 11 - RELATIONSHIP BETWEEN AREA CHANGE AND VALVE PLUG POSITION CHANGE FOR THE LINEAR HEAD PLUG
FIGURE 12 - RELATIONSHIP BETWEEN AREA CHANGE AND FLOW RATE CHANGE FOR THE LINEAR FLOW VALVE PLUG
Figure 13 - Relationship between area change and flow rate change for the linear head valve plug.

Percent of total area available for flow vs. flow rate (cubic feet per second).
FIGURE 14 - COMPARISON
OF YARNALL'S FLOW EQUATION
AND THE ORIFICE EQUATION
FOR THE LINEAR FLOW PLUG

YARNALL'S EQUATION
FLOW RATE (cubic feet per second)

FIGURE 15 - COMPARISON
OF YARNALL'S FLOW EQUATION
AND THE ORIFICE EQUATION
FOR THE LINEAR HEAD PLUG

YARNALL'S EQUATION
FLOW RATE (cubic feet per second)
FIGURE 16 - WEIR FLOW
EQUATION FLOW RATE VERSUS
ACTUAL FLOW RATE (WEIGHED)

ACTUAL FLOW RATE (cubic feet per second)