THE INVESTIGATION OF FLUID FLOW THROUGH A CONTROL VALVE OPENING

by

SIDNEY CARROL MILLER

Associate of Arts
Northeastern Oklahoma Agricultural and Mechanical College
Miami, Oklahoma
1955

Bachelor of Science
Oklahoma State University of Agriculture and Applied Science
Stillwater, Oklahoma
1957

Submitted to the faculty of the Graduate School
of the Oklahoma State University of
Agriculture and Applied Science
in partial fulfillment of the
requirements for the degree
of
MASTER OF SCIENCE
August, 1958

THE INVESTIGATION OF FLUID FLOW THROUGH
A CONTROL VALVE OPENING

Thesis Approved:

Thesis Adviser

Dean of the Graduate School

PREFACE

A significant amount of fluid-control research has been accomplished, which had for its purpose the development of adequate components for hydraulic systems. The spool valve is one such component but basic studies of this component are still needed to determine its flow characteristics in order to facilitate the design of more efficient valves. The problem considered in this study is the utilization of a two-dimensional model to determine the instability of fluid jet effluent angles for flow in a spool valve type of configuration.

The author is indebted to the Division of Engineering Research and the School of Mechanical Engineering of Oklahoma State University for providing funds to build the test setup. Further indebtedness is due Gillespie and Sons, Drilling Company, Cushing, Oklahoma, for the loan of a pump and an engine for the test apparatus. Appreciation is also extended to Professor B. S. Davenport and his staff of the Mechanical Engineering Laboratory for their constant advice.

The author expresses a sincere appreciation to Professor E. C. Fitch for his advice and suggestions toward the completion of this thesis. My expression of gratitude to my parents is unlimited and an especial appreciation is due my wife, Ladoris, for the encouragement and inspiration instilled beyond the hardships endured.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF ILLUSTRATIONS

LIST OF ABBREVIATIONS

fps	feet per second
ft	feet
gpm	gallons per minute
in.	inch
lb	pound
No.	number
psi	pounds per square inch
rpm	revolutions per minute
sec	second
spm	strokes per minute

LIST OF SYMBOLS

F	momentum force
Q	total rate of flow
u	velocity of fluid in vena contracta
w	peripheral width of orifice
x	axial clearance
y	radial clearance
θ	fluid efflux angle with the valve axis
ρ	density of flowing fluid
μ	viscosity

$<$	less than
C_d	coefficient of discharge
ΔP	difference between upstream and downstream pressure
$\cos^{-1}$	arc cosine

CHAPTER I

INTRODUCTION

The forces imposed by the velocity head of an incompressible fluid are an integral part of the design considerations for any flow control valve. An explicit knowledge of these hydrodynamic forces greatly facilitates the design of the valve because such knowledge provides a criteria by which the designer can determine the actual stresses imposed in the material which is to be used to fabricate the control mechanism. A number of scientists have investigated these forces as they exist in two-dimensional valve configurations such as the flapper valve, needle valve, orifice plate valve, gate valve, butterfly valve, and the spool valve. It has been extensively noted that the results of two-dimensional flows can, for quantitative purposes, be used to represent the flow in an axial plane of an "axi-symmetric" flow if correspondence is carried out on the basis of flow areas. Thus the method of utilizing two-dimensional models as investigation media has been used extensively.

In addition, the forces inside the valve are of interest to the engineer for use in determining the external force that would be necessary to operate the valve. In order to calculate this internal force and also to know its point of application the angle at which the flowing fluid leaves the orifice must be known for all relative positions of the flow-control element.

Because of recent widespread application and use of the spool valve type of control configuration in the hydraulic power systems of the aircraft and missile industry, study of this valve has been initiated at Oklahoma State University. Initial work by Karl Reid (1) with a two-dimensional model simulating this type of fluid flow control element has indicated that for compressible fluids there are some flow rates which produce unusual effluent angles. This work was instigated to augment work accomplished by A. H. Stenning. (2).

The work of Stenning indicated that at least two angles of efflux can be obtained in a stable condition, one which is vertical to one wall of the control element and another at the familiar position of a free jet at an angle of approximately 66 degrees with the valve axis.

From these observations the question arose as to whether or not the same phenomena might occur with an incompressible fluid. The purpose of this thesis is to make an attempt to find the answer to this question. The primary purpose is to either prove or disprove the existence of more than one stable angle of efflux for the flow of an incompressible fluid through a spool-type control valve.

An extensive library search brought very little information about this particular phenomena to light. Lee and Blackburn (3) came to the conclusion that the vertical jet would occur at very small openings and therefore would not affect greatly the design calculations since the force for this position would be relatively small. The definition of small openings was not made clear to the satisfaction of this writer so an attempt will be made to ascertain experimentally the limits of the mentioned small openings and the effect of them on the fluid-jet-effluent angles.

This work was accomplished under the direction of Professor E. C. Fitch of the School of Mechanical Engineering of the College of Engineering of Oklahoma State University. The Division of Engineering Research provided the funds to build the test apparatus as part of the initial phase of Fluid Power Project No. 144.

CHAPTER II

INITIAL INVESTIGATION

Professor A. H. Stenning (2), with the use of a glass-sided model, has accomplished an experimental study on air flow through slide-valve orifices with a pressure ratio (downstream to upstream) across the orifice in the range 0.95 to 0.20 with air velocities in the range of 100 to 1000 feet per second (fps). This gave a Reynolds number of the order of 1×10^5, as a value corresponding to air at a pressure of 150 pounds per square inch (psi) supply pressure flowing through a valve orifice with 0.010-in. opening. Initial testing with no radial clearance indicated that either of two jet efflux angles would be established when starting from zero flow. One of these angles was the familiar free jet flow making an angle of 66 degrees to the horizontal. The other was a jet with an 85 degree angle and then curling toward the vertical bounding wall. If the downstream pressure was lowered to 0.36 of the upstream pressure an initial free jet flow would become unstable and be replaced by the vertical flow which would then persist even if the downstream pressure was increased above 0.36 of the upstream pressure. (2).

Because of an interest in this work and a belief in its ever increasing value, work was initiated at Oklahoma State University to study the flow behavior in pneumatic spool valves. A model very similar to one used by Professor Stenning (2) was designed, constructed , and tested under the auspices of an honors research fellowship awarded to Karl Reid. The results of this study substantiated the presence of more than one flow regime; however, these were not the same as had been observed in the previous work cited. With no radial clearance introduced into the flow study it was not possible to establish the free-jet flow pattern. Instead, a curling free jet was observed leaving the orifice at the same angle as the free jet but curling sharply toward the vertical wall. Then, upon reduction of the downstream to the upstream static pressure ratio to about 0.30, this jet became unstable and there was a transition to the vertical jet with an efflux angle of approximately 80 degrees with the horizontal. In contrast to Stenning's (2) observations, upon increasing the pressure ratio above 0.30, the 80 degree jet shifted almost instantaneously to the curling free jet. Thus, no hysteresis effect in the flow pattern was observed. With the introduction of a small amount of radial clearance into the model flow study, two jet flow regimes were obtainable upon starting flow through the model. However, these were not the free jet and the 80 degree jet, but instead, they were the free jet and the free sharply curling jet, both of which would shift to the 80 degree jet if the pressure ratio was lowered to approximately 0.30. Thus, three different flow regines were observed instead of the two observed in the work by Stenning. (2). With the introduction of slightly more radial clearance into the flow study, the two previously mentioned flow regimes were still formed upon starting the flow through the model. If the curling free jet was formed, it would become unstable at a pressure ratio very near 0.30 and shift to the 80 degree position. Then upon increasing the pressure ratio above 0.30, transition back to the free curling jet would take place. However, if the free jet flow was established upon starting flow through the model, a transition would occur to take a position between the free jet and the 80 degree jet at some pressure ratio higher than 0.30. Further lowering of the pressure resulted in a shift from this intermediate position to the 80 degree jet. If the pressure ratio was then raised above 0.30, the 80 degree jet would then shift back to the curling free jet.

Thus, this investigation by Karl Reid (1) revealed that there were four distinct stable positions of the jet, none of which correspond to the free jet effluent angles that are commonly thought of in conjunction with fluid flow through control valve orifices.

A study by Lee and Blackburn (3) of the Dynamic Analysis and Control Laboratory, Massachusetts Institute of Technology, was referred to in determining what the flow forces were for incompressible fluid flow in a spool type valve configuration. Figure 1 is presented to show the configuration which was analyzed in this paper. This figure represents the ideal case of zero clearance and sharp land edges.

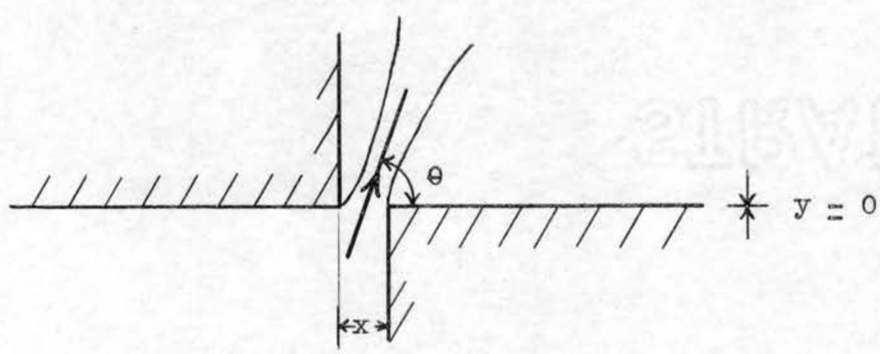


Fig. 1. Ideal Jet Configuration

The figure is skew-symmetrical about the center of the opening. It is the skewness of the symmetry that causes the axis of the jet to deviate from the vertical, and it is this deviation and the resulting X-component of momentum that give rise to the axial force on the control element. (3). Lee and Blackburn (3) indicated that for this ideal condition the jet angle is 69 degrees with the horizontal for any opening. However, F. F. Ehrich (4) tabulates angles varying from 69 degrees for zero axial clearance to 53 degrees for infinite axial opening. This indicates that there are some discrepancies occurring in the published material dealing with this type of valve configuration. Lee and Blackburn (3) conclude further that for actual valve configurations in which neither the radial clearance nor the radius of curvature of the land edges are zero, the angle of efflux approaches zero at small or negative values of axial opening. This is not particularly serious, of course, since in most cases the curve of force versus axial opening decreases and approaches that for an ideal valve before the actual value of the force is very large; that is, the phenomenon occurs essentially only for small values of axial opening. (3). The derivation of the force equations is included in Appendix A. From this derivation it is apparent that the angle of efflux has an important part in determining the forces imposed with any degree of accuracy.

Thus, from material available in the literature it cannot be determined explicitly what angle can be expected for a given axial opening, with a specific pressure ratio across the control valve orifice.

CHAPTER III

STATEMENT OF THE PROBLEM

In order to facilitate the design of control valve orifices such as those encountered in the spool type configurations, the angle of efflux created by the jet of fluid passing through the element must be known explicitly. Because of an apparent discrepancy in the previously published material for certain conditions of flow, this study was undertaken to determine experimentally the limits which dictate the position of the efflux angles for the region where these angles are other than the commonly used free jet of approximately 66 degrees with the horizontal axis.

To accomplish this study a two-dimensional simulation of the spool valve was utilized. This model, which has already been designed, constructed, and utilized in the study of compressible fluid flow by Karl Reid (1), was so constructed that visual observation of the flow pattern would be feasible. However, the jet efflux may also be determined through the use of internal pressure probes if at some flow condition it becomes impossible to observe the flow pattern.

Water at atmospheric temperature of approximately 75 degrees Fahrenheit, which means that the flowing fluid will have a dynamic viscosity of 1.95×10^{-5} lb sec/ft^2, was used as the incompressible fluid.

The following assumptions were made and are true for most practical cases and are considered to hold for this study:

1. The fluid is considered as being nonviscous.
2. The fluid is incompressible.
3. The flow in the upstream portion of the model is irrotational.

A secondary method of visualization, when the flow pattern becomes such that it is not apparent to the unaided eye, is the use of an optically bi-fringent fluid and polarized light. This technique has been used as a fluid flow investigation medium by several authorities. The method encompasses use of bentonite clay particles dispersed in water to form a bi-fringent fluid and a parallel beam light source which is polarized in order to show minute changes of density in the flowing fluid. However, a basic study of the size particles and concentration necessary for use with a given light source to produce optimum visualization would be a tremendous benefit to users of this method. Originally, such a study was the object of this author but the occurrence of the vertical jet effluent angles became evident and it was decided to explore these and leave the bi-fringence study as another phase of Fluid Power Project No. 144.

CHAPTER IV

THE TEST APPARATUS

Drawings of the two-dimensional flow model simulating the flow condition through a spool type control valve configuration are included as Appendix B. Some slight modifications were made in the original model for its use by this author. The upstream and downstream total pressure probes were removed and the positions converted to static pressure taps. To better facilitate precise positioning of the movable blocks a reference point was scribed onto each of the movable blocks with a corresponding scale of position etched into the plastic wall of the model. The original movable blocks were tipped in order to prevent erosion of the edge, with 3/16 by 3/16-in. tungsten carbide steel bonded to the aluminum with Eastman Formula 910 adhesive. With the increased force imposed on the orifice lands by the flow of fluid, these tips would not stay in place. Consequently, they were replaced with 1/4 by 9/16-in. REX-95 tool steel in an attempt to increase the bonding area for the adhesive. This proved to be sufficient for the lower block but the upper block would not stay in position for high velocities of flow. In order to hold it securely, a 17/64-in. recess was drilled into the block and a 1/4-in. stainless steel rod was silver soldered to the back side of the tip. This configuration was then inserted in place and held securely with a 3/16 by 1/8-in., 32-threads per inch, set screw.

The initial calculations to determine the flow rate necessary to produce approximately 100 fps velocity in the vena contracta of a 1/4-in. orifice opening are shown in Appendix C. Also included are calculations of Reynolds number in the 2-in. pipe to determine if the flow would be within the laminar range. These calculations indicated it would be necessary to have a fluid pump capable of producing 80 gallons per minute (gpm) at a maximum pressure of 190 psi.

A 4 by 6-in. double acting duplex, positive displacement pump was listed by manufacturers catalogs as capable of a flow rate of 94 gpm at 450 psi, utilizing 74 strokes per minute (spm). Thus, a Gaso pump with the above specifications was installed as the fluid moving medium. This pump was driven with a Buda, 4 cylinder, 49 horsepower, internal combustion, gasoline fuel engine. A photograph of this installation is included as Appendix E.

The positive displacement pump introduced excessive surges of fluid through the model at low speeds so it became necessary to incorporate surge dampeners into the test setup. A Westinghouse Desurger was installed directly adjacent to the discharge pipe of the pump. This device provided a surge volume of 190 cu.in. which did not prove to be sufficient. An air cushion, surge chamber of 1500 cu. in. surge volume was installed and this provided relatively smooth flow for a major portion of the flow range. These surge chambers were connected to a 130 psi air source in order that they might be precharged or pressurized to provide their most efficient operation for a given flow rate. However, for high flow rates the flow surges for the suction side of the pump produced excessive water hammer and a surge chamber of 164 cu. in. displacement was placed in the suction line. This chamber was so constructed that the flow of fluid had to ingress approximately 6 in. before it could change its direction of flow and proceed on toward the pump. This arrangement resulted in the suction fluid being purged of air before it reached the pump pistons.

Two, 4 X 4 X 4-ft, 478-gal capacity, storage tanks were utilized to contain the flowing fluid. These were connected to the discharge side of the flow model through two Grinnell-Saunders, 2-in. diaphragm valves so arranged that flow could be diverted to either tank (see Appendix D). The suction side of the fluid pump was connected to the tanks through a 2-in., 2-way, plug valve which provided a means by which fluid could be drawn at will from either tank.

The piping arrangement, in addition to a connection through the model, provided a fluid bypass whereby fluid could be transferred to either tank without passing through the flow model. In arranging the pipe, a 9-ft section of unrestricted run was provided upstream to the model in order to ensure uniform flow into the entrance section. A plan view, line sketch is included in Appendix B.

A bank of six Bourdon tube pressure gauges provided the means to obtain readings from the various pressure taps on the flow model (see Appendix D). A pressure gauge on the discharge side of the pump was also provided to indicate the pressure when fluid was being bypassed. A pressure indicator for the air supply of the surge chambers on the discharge side of the pump was also provided. In addition to these pressure gauges the only other instrumentation was a remote reading tachometer for the pump driving engine and a throttle linkage to the same from the table supporting the flow model.

For a complete parts list and component nomenclature see Appendix F.

CHAPTER V

OPERATIONAL PROCEDURE

In order to use the various components of the test apparatus the following procedure was followed. One of the fluid reservoir tanks was filled with water from the line connected to the laboratory water supply. The fluid pump was then started with the fluid flow being bypassed back to the reservoir tank. After the flow had purged all of the air from the pump and suction line, the bypass valve was closed sufficiently to create a back pressure of approximately 25 psi on the pump. The valve which was in the pipe circuit through the model was then partially opened and flow was established for a time sufficient to remove all of the air in this conduit. Complete closing of the bypass valve was then carried out which caused all of the flow to be transferred through the flow model.

For a given orifice setting the flow was initiated from zero flow rate by use of the friction clutch mechanism connecting the engine to the pump drive. Thus, flow was established with zero up-stream to downstream pressure difference and zero velocity through the orifice. The flow rate was increased incrementally by increasing the engine speed, with a value which would result in the recommended maximum pump speed of 100 spm. A minimum flow rate was also dictated by the idle speed of the engine being used to drive the fluid pump. The flow rate of the fluid through the model with a given orifice opening was determined by diverting the flow into the tank from which fluid was not being taken by the suction line. The time necessary to increase the level in the tank by six inches was determined with a manually operated stop watch and calculations were made to determine the flow rate from the known volume of fluid for a known time.

The engine speed was noted for each incremental setting of the pressure at which the flow rate was determined. A timed flow rate was not determined for every setting however. A final compilation of all the determined flow rates was transposed to a graph which is shown as Appendix H. The flow rate for any given engine speed setting was determined from the graph and this method was used for the calculation of fluid velocities at points where there was a change in the position of the jet efflux angle.

For each orifice setting the pressure of the transition from a vertical jet to the free jet was noted with assurance of correctness being effected by three such determinations for each setting. Flow was then started again from zero and a specific back pressure held in the downstream section of the model by manipulation of the downstream valve through which flow was being diverted at that particular time. Increase of the back pressure was effected in 5-psi increments until the limits of the test setup were approached, in all cases no higher than 15-psi.

To change the position of the internal movable blocks in order that a different orifice opening could be investigated, it was necessary to stop the flow and loosen the bolts in the model adjacent to the block which was to be moved. After the block was repositioned as desired, tightening of the bolts was effected and flow again instigated. Thus, it was impossible to move the block with flow occurring at the same time in order to determine what effect this phenomena would have on the flow pattern.

The determination of the angle of the fluid efflux was accomplished with the use of a variable protractor. This apparatus was placed against the outside of the flow model and the edge aligned by sighting along the edge and through the model. When it was deemed that this edge was parallel with the jet efflux angle, the mechanism was locked, removed, and the angle read directly from the instrument.

The accuracy of measured values of pressure was estimated to be $\pm$ 2-psi in the upstream chamber and $\pm$ 0.5-psi in the downstream section. The engine indicator was scaled in 100 rpm increments and the readability was estimated at $\pm$ 5 rpm. The accuracy of the volumetric determination of flow was approximated as 3 per cent. With this accuracy an error of 4 fps could be possible in determining the velocity of the fluid in the vena contracta.

CHAPTER VI

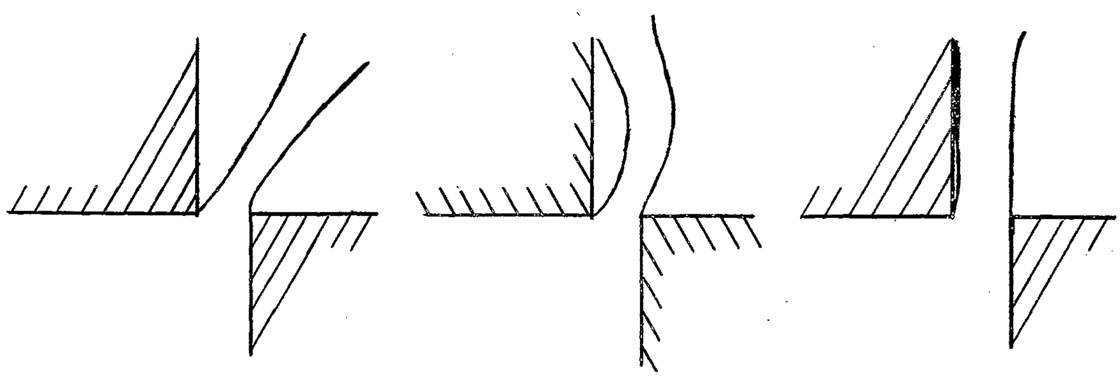
RESULTS AND CONCLUSIONS

The immediate conclusion which was fairly obvious was that the vertical jet phenomena definitely does occur in a spool type valve configuration. For this study the limits of its existence were with a maximum opening of 0.175 in. and a minimum of 0.05 in.; however, the test setup limitations were instrumental in setting the lower axial clearance limit.

The initial indication that this phenomenon was going to occur was in a trial run with an approximate opening of 3/16 inch. For this setting and with a 40-psi pressure upstream and a 5-psi back pressure, it was noted that the jet was showing signs of a curl character. The jet was emitting at the free jet angle but curling back toward the chamber wall instead of continuing in a straight line into the discharge chamber. By increasing the flow rate through this 3/16-in. axial clearance to a point which resulted in an upstream pressure of 110-psi and a 5-psi downstream pressure, it was noted that there was a momentary increase on the 85 degree pressure probe. Just as quickly it returned to the 5-psi which was the static pressure in the chamber at that time. This phenomenon was due to the fact that the issuing jet of fluid had passed across this probe in changing from a curling position back against the vertical wall, out to the position of a free jet at approximately 66 degrees with the horizontal axis.

Further investigation substantiated this observation. Upon closing the opening to 2/16-in. the curl jet was established with zero psi back pressure which indicated that this phenomenon was a definite fact and not just a freak that might have occurred at the particular conditions of the previous opening. Thus, the purpose of this investigation was to define the limits of this phenomenon.

Figure 2 is an illustration showing the free jet, curl jet, and vertical jet configurations as the terms are used throughout this discussion.



(a) Free Jet (b) Curl Jet (c) Vertical Jet

Fig. 2. Definition of Jet Types

From preliminary observations it was determined that the vertical jet phenomenon would occur in the neighborhood of 0.20-in. so it was decided to start with an opening of 0.40-in. for axial clearance x. For this value of axial opening the data tabulated as Table I were observed. Unless specified otherwise the radial clearance y, was zero for the observed data.

TABLE I

OBSERVED DATA FOR 0.40-IN. AXIAL CLEARANCE

Upstream pressure psi	Downstream pressure psi	Fluid Velocity fps	Jet configuration
17	0	29.8	free
25	0	42.0	free
40	0	50.5	free
50	0	59.3	free
35	5	42.0	free
37	10	42.0	free
42	15	42.0	free
55	10	50.5	free
60	15	50.5	free

The tabulation of Fig. 2 indicates that the free jet was the only effluent configuration that could be obtained. The fluid effluent angle was approximated at 56 degrees with the horizontal by the method that was outlined in Chap. V. This value does not correspond to the value as indicated by Ehrich (4) of 69 degrees. A further discussion of the variation of these angles will be included later in this writing.

By taking decreasing increments of 0.025 in. in the axial opening neither the establishment of a curl nor a vertical jet could be accomplished in this area and therefore a formal tabulation of data was not included.

The first occurrence of the unusual phenomenon which is the subject of this thesis occurred with an axial opening of 0.175 in. Table II shows the data as observed for this particular setting.

TABLE II

OBSERVED DATA FOR 0.175-IN. AXIAL CLEARANCE

Upstream pressure psi	Downstream pressure psi	Fluid Velocity fps	Jet configuration
25	0	42.6	curl
35	o	46.0	curl
45	0	50.0	free
70	5	61.1	vertical
75	5	65.1	curl
76	5	65.8	free
95	10	67.7	curl
103	10	68.9	curl
106	10	70.4	free
120	15	71.6	vertical
126	15	75.8	curl
128	15	76.2	free

As indicated by Table II, it was observed that the jet always changed to a curl type before it would flip to the free jet position with this amount of axial opening. Upon decreasing the flow incrementally for any setting of back pressure no hysteresis effect was present in the jet efflux angle. Once the free jet was established it would remain stable in this position within the limits of the flow rate until flow was completely stopped. Upon starting the flow from zero, either the curl jet or the vertical jet would be established, depending on the back pressure in the discharge chamber. For high back pressure the vertical jet would be established while for low pressures the curl jet would become evident.

The data for an axial opening of 0.15 in. was tabulated as Table III.

TABLE III

OBSERVED DATA FOR 0.15-IN. AXIAL CLEARANCE

Upstream pressure psi	Downstream pressure psi	Fluid Velocity fps	Jet configuration
45	0	57.3	vertical
50	0	59.1	vertical
59	0	61.3	free
80	5	64.9	vertical
92	5	67.2	vertical
99	5	70.0	free
127	10	78.3	vertical
131	10	82.1	free
153	15	83.6	vertical
160	15	84.5	free

A significant change in the characteristics was observed for the setting of axial clearance considered in Table II. The jet effluent configuration which immediately preceded the point at which it shifted was vertical in all instances. For the axial clearance of 0.175 in. the jet would become the curl jet type of configuration before the jet shift occurred. This possibly could be attributed to the changing symmetry of the orifice configuration. For this value of axial clearance as well as for the value of 0.175 in., no hysteresis effect was observed in the jet effluent configuration as the flow was incrementally decreased after the establishment of the free jet flow.

In the determination of the point of jet shift while holding a constant back pressure it was noted that upon a change in the jet configuration there was an increase in the pressure for the downstream chamber. This same occurrence was observed in the setting of axial opening considered in Table II, but no significance was attached until the same thing happened for the setting under consideration in Table III. No definite pattern seemed to be apparent and the value of pressure ranged from 6-psi for a minimum to 14-psi as a maximum. Final results and conclusions for this phenomenon over the complete test range are discussed later in this thesis.

The next incremental setting of axial clearance was 0.125 in. and a selection from the observed data is tabulated in Table IV. The secondary part of Table IV is a tabulation of the change from a free to a vertical jet with the corresponding pressures. This was the first setting in which it was possible to induce this occurrence of a hysteresis effect to present itself. Thus it becomes apparent that for the smaller values of axial clearance the back pressure becomes a factor affecting the jet efflux angle.

TABLE IV

OBSERVED DATA FOR 0.125-IN. AXIAL CLEARANCE

Upstream pressure psi	Downstream pressure psi	Fluid Velocity fps	Jet Configuration
60	0	68.7	vertical
70	0	70.8	vertical
80	0	73.6	free
125	5	77.9	vertical
135	5	81.1	free
150	10	87.4	vertical
177	10	93.0	free
191	15	96.3	vertical
215	15	99.8	free
125	30	----	free to vertical
100	25	----	free to vertical
<100	--	----	no shift

Table V is a tabulation of the observed data taken with an axial clearance of 0.10 in. for the two-dimensional flow model. The observance of data for a back pressure of 15-psi was discontinued because it became obvious that the shift for this setting would be beyond the limits of the pressure measuring instrument which had a maximum reading of 300 psi.

TABLE V

OBSERVED DATA FOR 0.10-IN. AXIAL CLEARANCE

Upstream pressure psi	Downstream pressure psi	Fluid Velocity fps	Jet configuration
115	0	82.6	vertical
120	0	84.3	free
170	5	95.3	vertical
180	5	97.8	free
180	10	97.8	vertical
230	10	105.0	free
150	10	----	free to vertical
120	5	----	free to vertical
<120	--	----	no shift

The establishment of flow for an axial clearance value of 0.075 in. brought to the attention of this author a very vivid illustration of a phenomenon which is commonly known as valve chatter. This is the occurrence of a very rapid vibration which in an actual valve can create forces sufficient to destroy the valve in a relatively short time. However, one contributing factor to this destruction in an actual valve is the small amount of free movement which is possible in the control element. In the flow model the control element was completely rigid and the vibration was most vividly portrayed by the high frequency surges of fluid flow through the orifice.

The occurrence of a flip for this axial clearance value came to pass with an upstream pressure of 140-psi while holding 0-psi back pressure. With a 5-psi back pressure the shift from a vertical to a free jet occurred with 230-psi in the upstream chamber. This pressure was approaching the limits of the test setup so no further values of back pressure could be obtained. A hysteresis effect occurred with 0-psi pressure in the downstream chamber when the flow rate was lowered sufficiently to create a pressure in the upstream chamber of 125-psi. This was the first setting which produced a hysteresis effect while decreasing the flow rate and maintaining an unrestricted exit flow.

With an axial clearance of 0.05-in., measurement of pressure presented difficulty because the engine would not provide sufficient torque at slow speed to produce a flow rate with less than the 300-psi limit in the upstream chamber. After several unsuccessful attempts to obtain an exact pressure a good approximation was established at 210-psi with 0-psi back pressure and the reverse shift of hysteresis occurring near 175-psi.

With the introduction of radial clearance into the model study it was observed that only a small amount could be introduced before the vertical or curl jet phenomenon would no longer exist. For an axial clearance of 0.125-in. and a radial clearance of 0.010-in. the vertical jet could be induced and a tabulation of the data is included as Table VI. The introduction of radial clearance apparently only caused the point at which the shift occurred to be displaced from the values observed for the same axial opening with no radial clearance.

However, the displacement for higher values of back pressure conceivably could be more than for lower values because the slope of the curve tends to decrease for higher pressures. This may be observed from the dashed curve as it appears in Fig. 3.

TABLE VI

OBSERVED DATA FOR 0.125-IN. AXIAL CLEARANCE WITH RADIAL CLEARANCE OF 0.010-IN.

Upstream pressure psi	Downstream pressure psi	Fluid Velocity fps	Jet configuration
46	0	64.4	vertical
55	0	68.7	vertical
61	0	71.4	free
95	5	78.1	vertical
111	5	80.0	free
124	10	81.2	vertical
141	10	85.0	free

With a radial clearance of 0.013 in. or larger while holding 0.125 in. axial clearance, neither the curl nor a vertical jet could be established. This was the last value where observations were made and the remainder of this paper will deal with the overall compilation of these results.

From the data presented it can be concluded that there are fluid jets with angles other than the common free jet angle. The occurrence of this phenomenon has definitely been established for values of axial

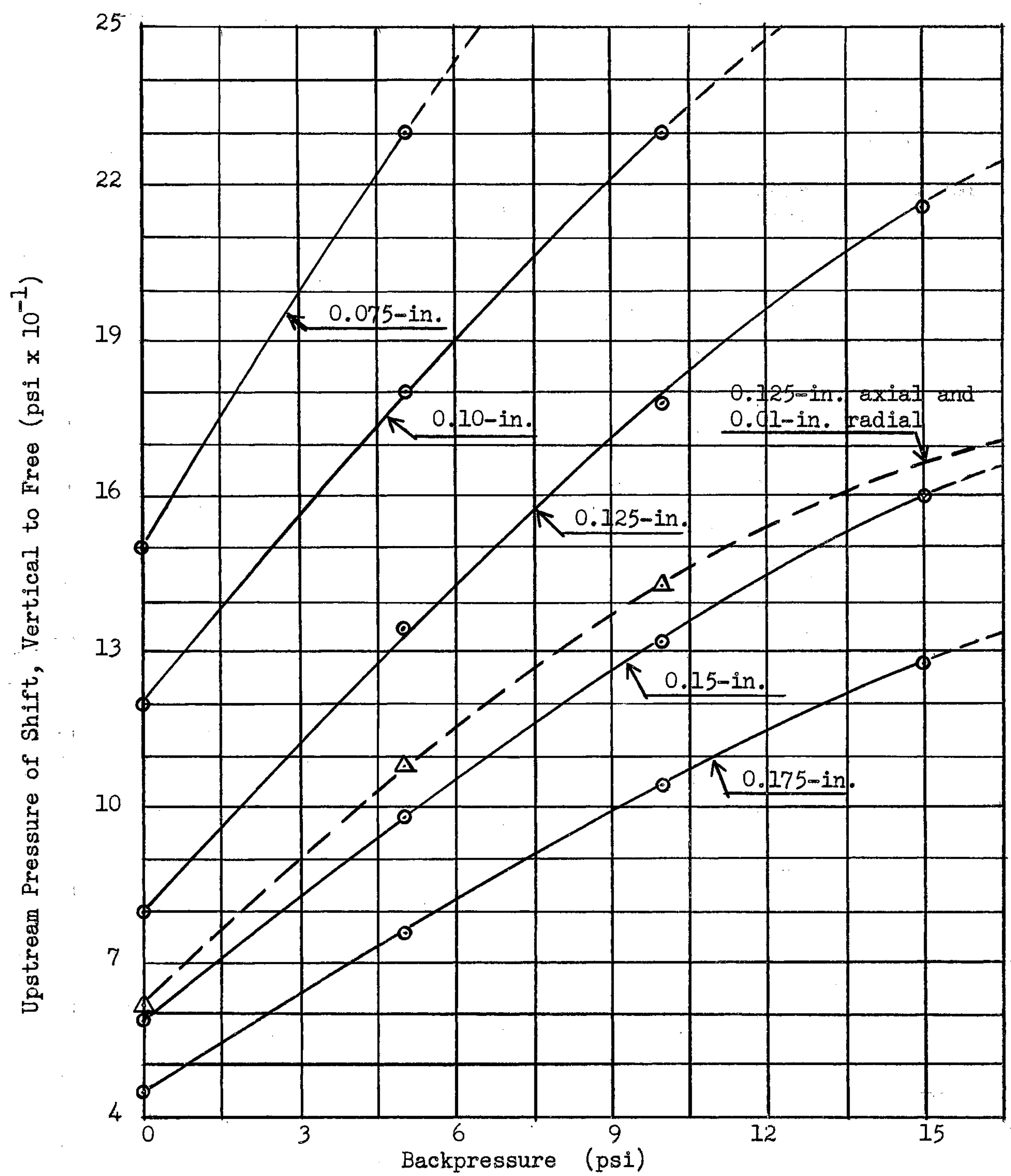


Fig. 3. Jet Shift from Vertical to Free Efflux

openings of as large as 0.175 in. With the instigation of a vertical or a curl jet upon initiation of flow, the value of upstream pressure at which the jet shifts to a free jet is shown versus back pressure on Fig. 3.

It was noted from the data of Table II that the pressure difference lessened between the shift points as the back pressure increased. The difference between 0-psi and 5-psi was 31-psi, while from 10-psi to 15-psi back pressure the difference was only 22-psi. When these values were transposed into a curve on Fig. 3, this observation was noted also, in the decreasing slope of the lines of constant axial clearance. As the back pressure increased, this phenomenon was prevalent for all values of axial clearance except the openings which were small enough to prevent the determination of high back pressure shift points. It is felt by this author that a determination of these points would follow the pattern which has been established within the limits of the present test setup.

Essentially, the same occurrence appears in Fig. 4 when lines of constant back pressure are plotted with the pressure of shift versus orifice opening as coordinates. The extrapolation of these curves, which are shown as dashed curves, further brings to focus the change in the slope as the back pressure increased for a given orifice opening.

When the observed data are transposed into the velocities in the vena contracta and velocity versus axial clearance is used as the coordinate the lines of constant back pressure are straight. From

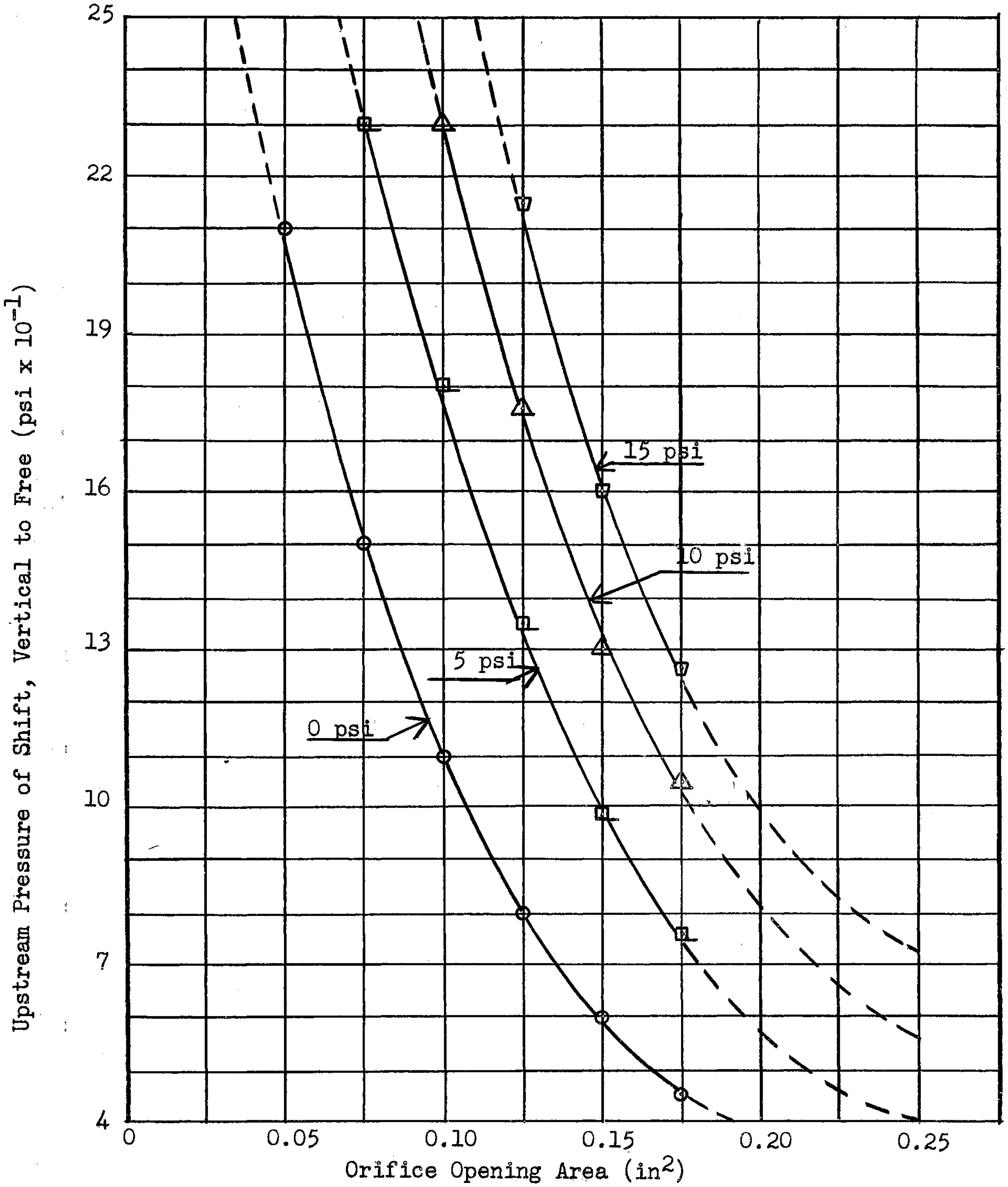


Fig. 4. Jet Shift from Vertical to Free Efflux

Fig. 5, direct correspondence between the velocities can be observed for an increasing back pressure. As the back pressure was increased the velocities of the fluid at the condition of the flip from either a curl jet or a vertical jet to a free position, decrease. This could have been predicted since the less the pressure ratio from one side of the orifice to the other, the smaller will be the velocity through that orifice. For a tabulation of the velocities from which Fig. 5 was transcribed, see Appendix G.

The occurrence of an increase in upstream pressure upon the change from a vertical or a curl jet to a free jet was observed frequently. As stated before, there seems to be no established pattern nor a predictable value for any given axial clearance setting. The maximum value noted was 14-psi and a minimum value of 6-psi (see Appendix G). This increase in the upstream pressure can undoubtedly be attributed to a change in the coefficient of discharge, because all values upon which this factor depend are constant except the orifice configuration with relation to the issuing jet of fluid. A geometric analysis of both types of flow indicated that the free jet efflux has drag on both lands of the orifice while for the vertical jet, only one land is acting on the jet discharge. Thus, for the free jet the coefficient decreases and more energy is expended in forcing a given amount of fluid through the orifice than for a vertical jet type of efflux.

It is quite possible also that the free jet instigates phenomena known as separation and this is the contributing factor which

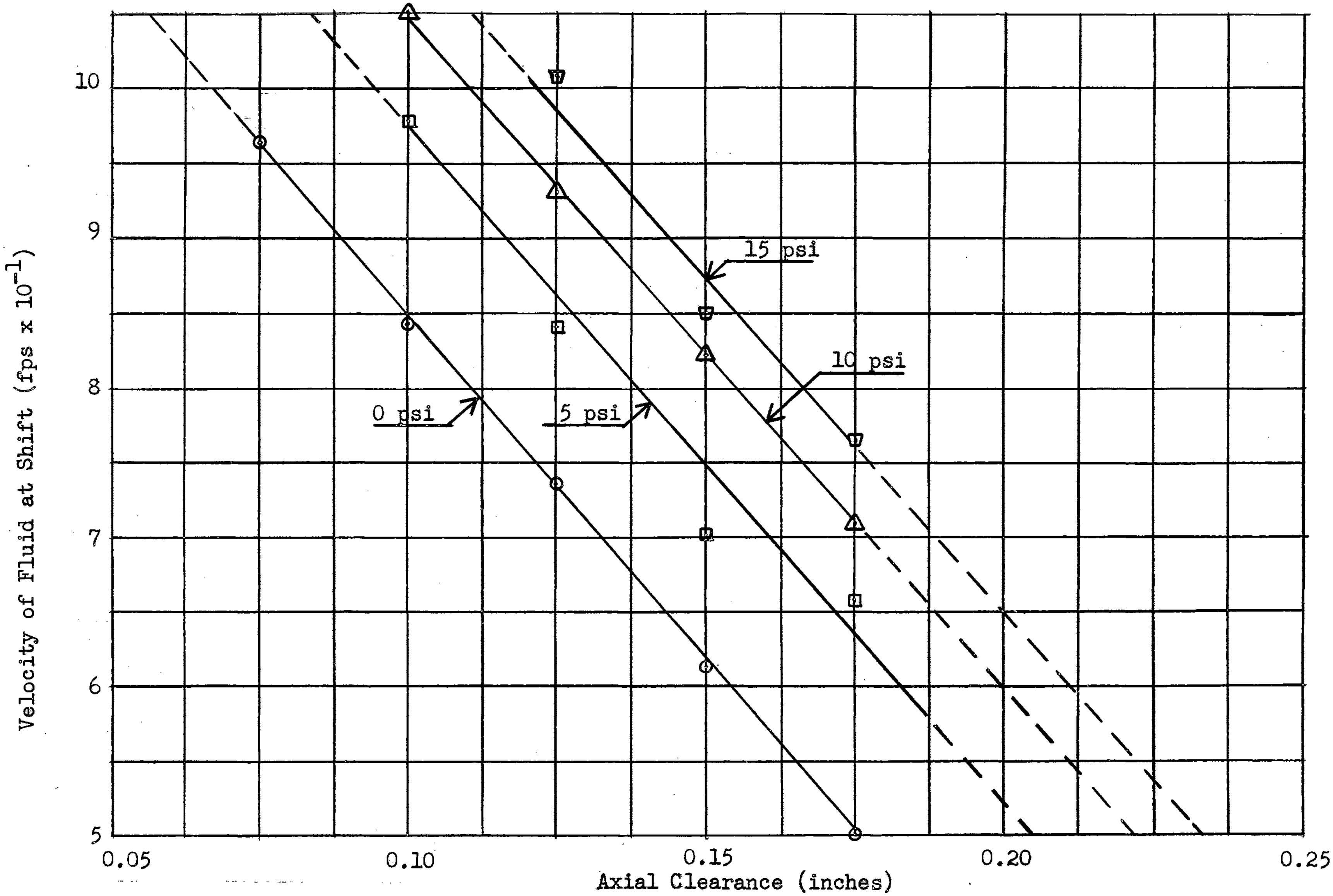


Fig. 5. Velocity at Shift from Vertical to Free Efflux

causes the coefficient of discharge to decrease. Separation is an occurrence that is responsible for many of the difficulties encountered in the design of hydraulic devices and is frequently the cause of differences between theory and experiment, according to N. M. Sverdrup. (5). Sverdrup indicates that separation is a source of instability because of back-flow around the boundary, and the formation of eddies with consequent increase in turbulence. Separation in a fluid passage depends upon the shape of the solid boundaries, the velocity of the fluid, and the pressure gradient. As fluid approaches a sharp edge orifice with high velocity, the fluid separates on the downstream side and the formation of eddies and increased turbulence result in dissipation of energy. Kinetic energy thus dissipated into thermal energy can never be recovered in useful form and manifests itself as an excessive pressure loss through the hydraulic device. (5).

The efflux angles which were measured by the method outlined in Chap. V did not correspond to the values as tabulated by Ehrich. (4). The values that were measured are tabulated here as Table VII. The corresponding radial and axial clearances as well as the pressures are also included. Ehrich (4) indicated that for small values of axial clearance, less than the width of the entrance flow section, the angle of efflux would be 69 degrees. From Table VII it is seen that the efflux angle varies from 57 degrees for an axial clearance of 0.30 in. to 64 degrees corresponding to 0.15 in.. According to Ehrich (4) the value should be 69 degrees for all values of axial opening up to 0.90 in..

TABLE VII

FREE-JET EFFLUX ANGLES

Axial clearance in.	Radial clearance in.	Upstream pressure psi	Downstream pressure psi	Jet efflux angle deg.
0.30	0	90	0	57
0.30	0	140	0	58
0.20	0	50	0	60
0.20	0	107	0	61
0.20	0	225	0	61
0.15	0	60	0	61
0.15	0	70	0	62
0.15	0	80	0	64
0.15	0.025	50	0	53
0.15	0.025	75	0	54
0.15	0.025	100	0	56
0.125	0.025	48	0	53.5
0.125	0.025	75	0	55
0.125	0.025	100	0	57
0.125	0.15	75	0	48
0.125	0.15	100	0	50.5

With the introduction of radial clearance into the model the efflux angles decreased with the evident relation of the greater the radial clearance the smaller the angle. This relationship can be predicted from the geometry of the flow jet efflux configuration.

CHAPTER VII

SUMMARY

For axial clearances between the limits of 0.175 in. and 0.05 in. for a two-dimensional simulation of a spool valve there occur fluid jet effluent angles which do not correspond with the accepted opinion of authorities on this subject. Within these limits three stable jets were observed, the free jet, the curl jet, and the vertical jet. The curl jet and the vertical jet are types which are not referred to in any literature which has been observed by this author. The existence of a vertical jet with velocities greater than 100 fps while holding a back pressure of only 15-psi or less indicates that this phenomenon could be an important factor in the design of control valves.

Either a curl jet or a vertical jet was obtained with no radial clearance upon increasing the flow rate from zero if the axial clearance was between 0.05 in. and 0.20 in. and the back pressure was between 0-psi and 15-psi. With an increase in the flow rate, there is a value at which the jet efflux angle will suddenly change to a position which will be equal to or less than 69 degrees with the axis of the valve. Only after this shift has occurred was the jet efflux in a position which has been predicted by authorities on this subject.

With the introduction of radial clearance into the model study the phenomenon of obtaining an efflux angle other than the generally supposed angle of less than 69 degrees was also observed. However, this radial clearance must be kept small; less than 0.013 in. This value is greater than the radial clearance which would occur in an actual valve and it is thus to be considered as a potential design consideration.

The occurrence of an increase in the upstream chamber pressure when the jet changes from a curl jet or a vertical jet to a free jet has been attributed to a decrease in the coefficient of discharge. This indicates that if a design could be accomplished which would keep the jet in the vertical position, a decrease in the energy expended by flow through the configuration would be accomplished.

The occurrence of a hysteresis effect in the position of the jet can be induced with back pressure for values of axial clearance equal to or less than 0.125 in. For values of 0.075 in. or less, there is a change from the free jet to the vertical jet upon decreasing the flow sufficiently, even though there is negligible back pressure. Thus hysteresis is self induced.

One of the principal methods used to move the control element of a servo type spool valve is by the use of an external torque motor and lever-arm mechanism. This arrangement can adequately compensate for the change in force as the fluid jet efflux varies slowly within a range of approximately 15 degrees. However, an instantaneous change from an angle very near 90 degrees to a position of approximately 66 degrees would create a force for which the control mechanism ordinarily could not compensate. Thus, precise operation of the valve, in an area where this shift phenomenon occurs, would be very difficult. A knowledge of the exact conditions where the jet angle shifts would be used effectively in designing a stable control system which would result in precise positioning of the actuating element. This thesis has attempted to lay the groundwork for such a study wherein the precision and accuracy necessary for the use of the results in actual valve design could be found.

CHAPTER VIII

RECOMMENDATIONS FOR FUTURE ANALYSIS

With a test setup of more extensive nature it would be possible to define the occurrence of fluid jet efflux angles with sufficient precision that the experimental data could be incorporated into design calculations. This could possibly mean using higher back pressure values where visual observation of the jet position could not be accomplished by utilizing the air entrained in the fluid, as was used in this thesis. Another method could be one which would use an optical bi-fringent fluid and crossed polarized light to show changes in fluid flow characteristics. This type of fluid can be made by introducing bentonite clay particles into the water. A selected bibliography for preliminary study of this method is included as Appendix I. However, this material does not contain any information on the proper amount of clay nor the size of particles that would result in optimization of the visualization.

In order to define the limits of the vertical or the curl jet phenomenon, a steady-flow fluid supply would be necessary. The use of an axial-flow pump with a bypass arrangement, such that all or a very small portion of the pump discharge could be utilized through the flow model could accomplish this steady-flow condition. When this method is used some means should be incorporated by which the properties necessary to the determination of the coefficient of discharge could be measured over short periods of time. This would allow the determination of the coefficient at the time just before and after the jet shift to indicate which type is the best.

For utilizing the same flow model with higher pressures, it is suggested that all matching surfaces be polished to a high degree of smoothness in order to prevent excessive leakage of fluid. The addition of at least two more bolts along the top edge in the vicinity of the 69 degree pressure probe should be included to prevent leakage of fluid in this area.

BIBLIOGRAPHY

(1). Reid, Karl N., "Static Flow Characteristics in Pneumatic Spool Type Valves," (unpub. report on Fluid Power Project 144, Oklahoma State University, 1958).

(2). Stenning, A. H., "An Experimental Study of Two-Dimensional Gas Flow Through Valve Type Orifices," ASME Paper No. 54-A-45.

(3). Lee, Shih-Ying and Blackburn, J. F., "Steady State Axial Forces on Control Valve Pistons," Transactions ASME, 74, 1952, pp 1005-1011.

(4). Ehrich, F. F., "Some Hydrodynamic Aspects of Valves," ASME Paper No. 55-A-114, 1956.

(5). Sverdrup, N. M., "Theory of Hydraulic Flow Control," Product Engineer, V-26, n 4, April, 1955, pp 161-76.

(6). Mises, R. von, Calculation of Flow Coefficients for Nozzles and Orifices and for Weirs, tr. H. Hug and J. Dunn (Cambridge 39, Mass.), Sept. 1956.

(7). Binder, R. C., Fluid Mechanics, (New York, 1955), pp 47-48.

APPENDIX A

Following is the derivation of the axial force on the control element as given by Lee and Blackburn. (2).

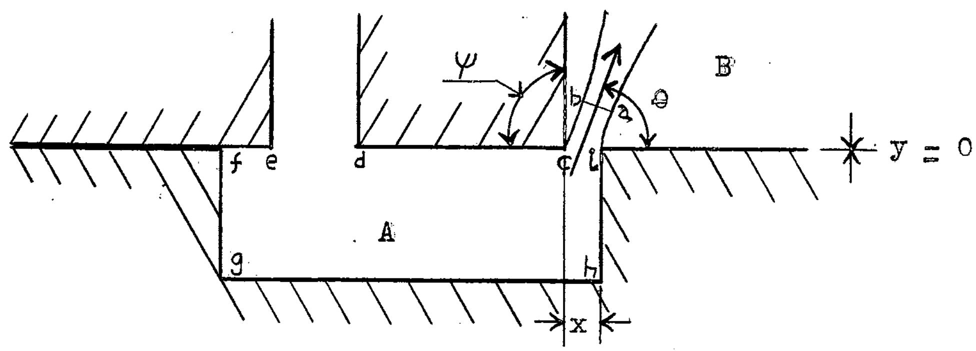


Fig. 6. Square-land Chamber Configuration

With the assumption that the flow is two-dimensional, irrotational, nonviscous, and incompressible, the solution of the flow pattern in the region upstream from the orifice becomes a solution of Laplace's equation for the configuration shown in Fig. 6. This solution has been obtained by von Mises (4) and it was found that for a square-land valve when the axial opening is small compared with the other dimensions of the upstream chamber, the angle θ which the axis of the stream makes with the piston axis is $\cos^{-1} 0.36$, or 69 degrees. For lands with angles other than 90 degrees the value of θ can be found either experimentally or theoretically. (2).

When θ is known, the axial force on the piston can be determined in the following manner: The axial force on the piston equals the axial component of the difference in fluid momentum as it changes within the boundary a-b-c-d-e-f-g-h-i-a in Fig. 6. The a-b portion of this boundary is the vena contracta of the jet. Since the area of a-b is small compared to d-e and the relative velocities are inversely proportional to the areas, the momentum through d-e is negligibly small. Considering the area of the chamber A to be large, the flow rate in this area is small enough to consider the flow velocity as zero; then the basic relation, force equals mass rate of flow times the velocity change becomes a product of rate of flow, density of the fluid, and the velocity at the vena contracta. (5) Thus:

$$F = Qu\rho \qquad \text{A-1}$$

in which F = momentum force,
Q = total rate of flow,
u = velocity at the vena contracta, and
ρ = density of the flowing fluid.

The velocity u is given by the Bernoulli's equation as:

$$u = \sqrt{\frac{2\Delta P}{\rho}} \qquad \text{A-2}$$

where ΔP is the pressure difference from upstream to downstream chambers.

The net axial force is:

$$F_{ab} = F_{fg} - F_{hi} = Qu\rho \cos \theta \qquad \text{A-3}$$

This can be transformed into the more useful form:

$$F_{ab} = 2\, C_d w x \Delta P \cos \theta \qquad \text{A-4}$$

in which C_d is the coefficient of discharge defined as:

$$C_d = \frac{Q}{wx\sqrt{\frac{2\triangle P}{\rho}}} \qquad \text{A-5}$$

where w = peripheral width of the orifice,
x = axial length of the orifice.

The above equations apply only to the ideal valve with perfectly sharp lands and zero radial clearance. For an orifice with radial clearance y, Eq. A-4 becomes:

$$F_{ab} = 2\, C_d\, w \triangle P \sqrt{x^2 + y^2} \cos \theta \qquad \text{A-6}$$

This is the force equation in its most useful form.

APPENDIX B

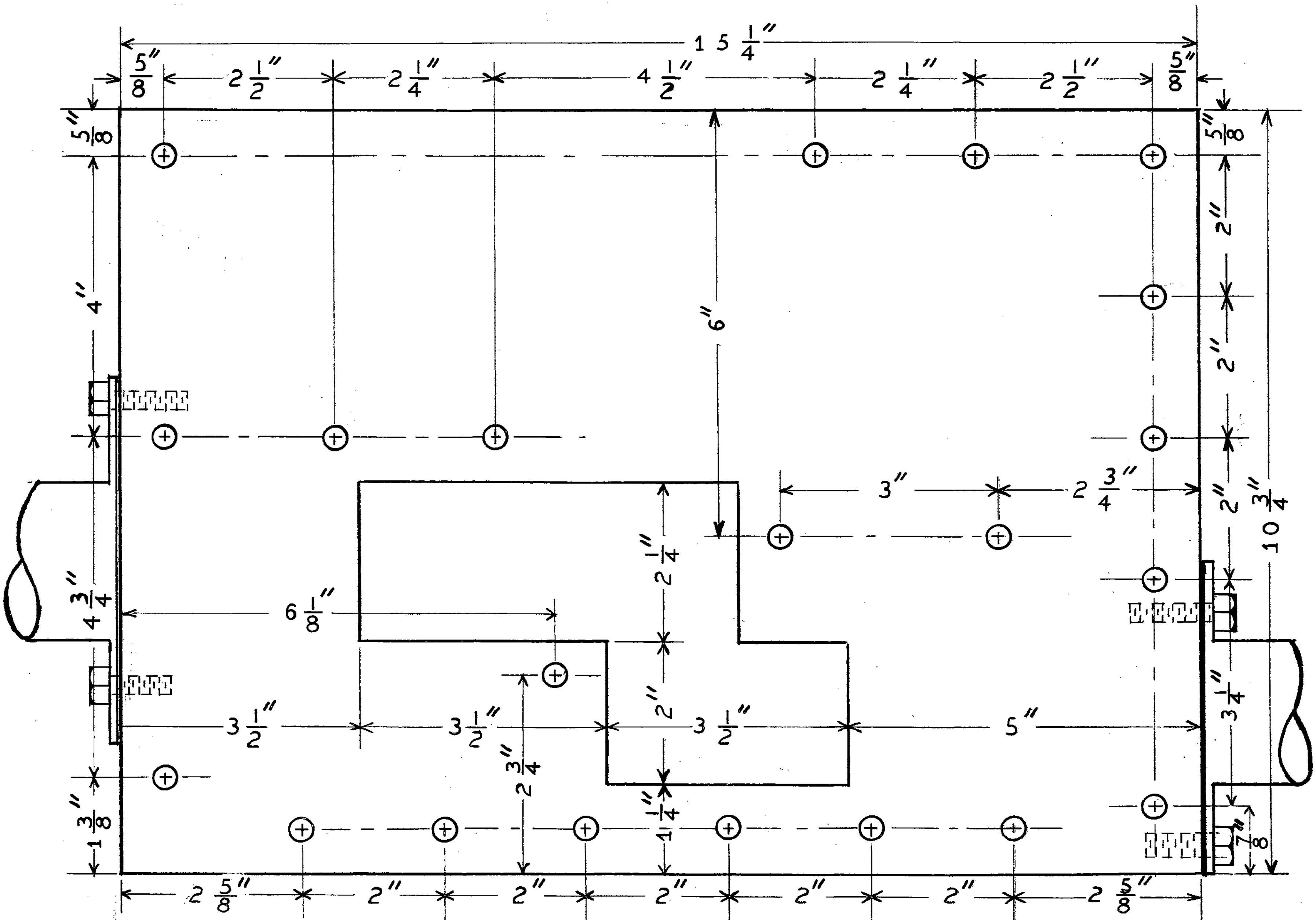


Fig. 7. Flow Model, External Dimensions

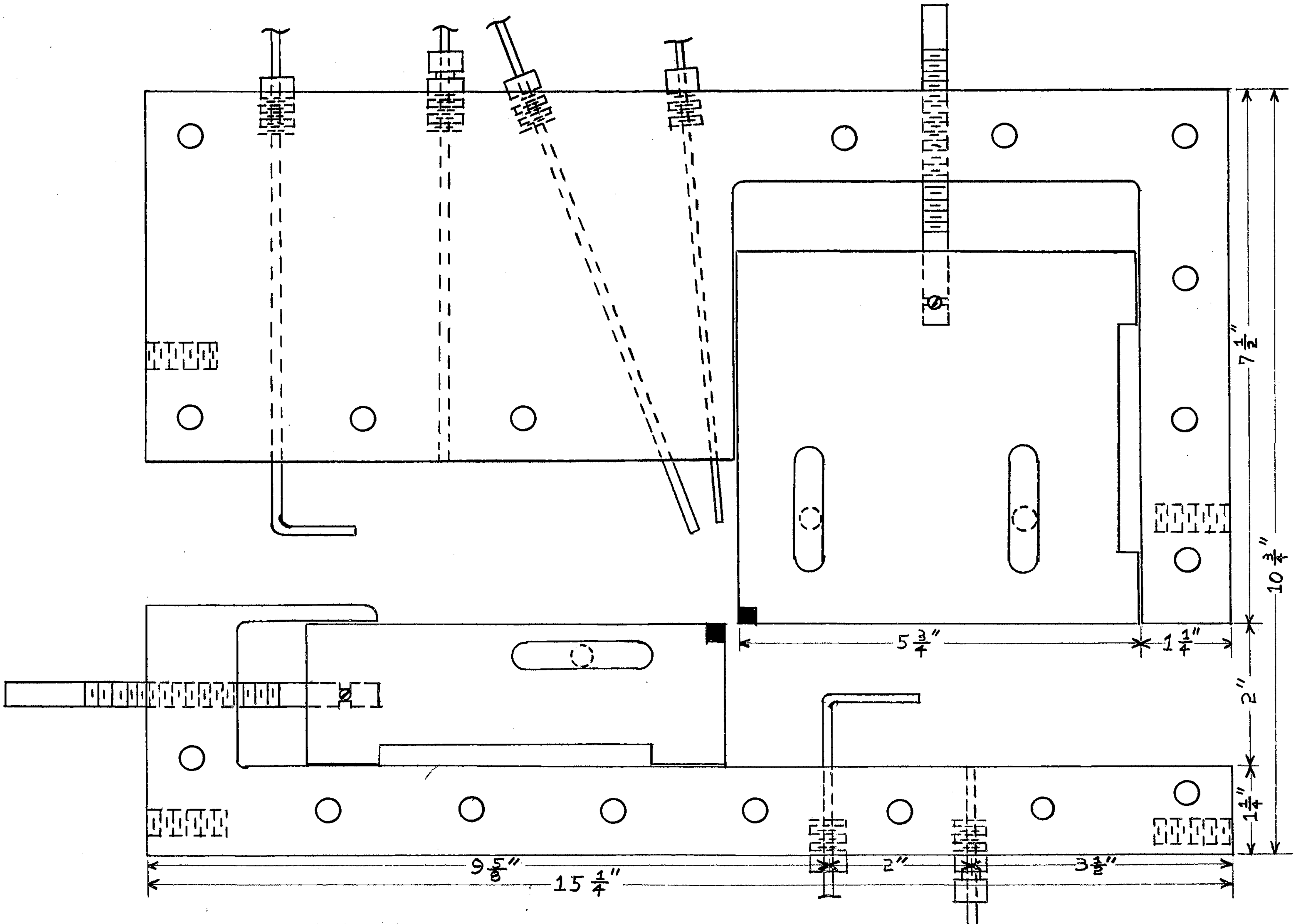


Fig. 8. Flow Model, Internal Detail

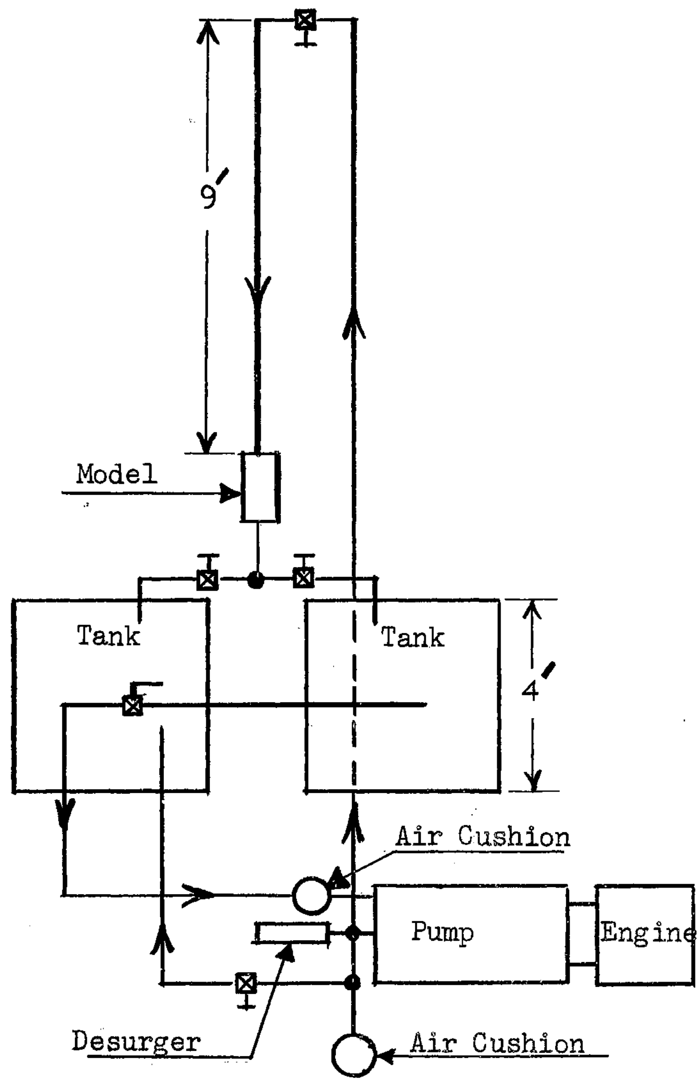


Fig. 9. Plan View, Line Sketch of Test Setup

APPENDIX C

For any given area of orifice and flow rate, the velocity of the fluid going through the orifice can be determined by the use of the continuity equation, such as:

$$Q = AV \qquad \text{C-1}$$

in which, Q = flow rate,
A = area of the orifice, and
V = velocity.

For a velocity of 100 fps and a 1/4 in. orifice, we have:

$$A = \frac{0.25 \text{ in.}^2}{144 \text{ in.}^2/\text{ft}^2} = 0.00173 \text{ ft}^2$$

$$Q = \frac{0.00173 \text{ ft}^2 \times 100 \text{ ft/sec} \times 60 \text{ sec/min} \times 1728 \text{ in}^3/\text{ft}^3}{231 \text{ in}^3/\text{gal}}$$

$$Q = 79 \text{ gpm}$$

This indicates it will be necessary to obtain a flow rate of approximately 80 gpm in order to achieve 100 fps velocity through the orifice.

This flow rate will produce a velocity in the 2 in. pipe of:

$$V = \frac{231 \text{ in.}^3/\text{gal} \times 79 \text{ gal/min}}{0.7854 \times 2^2 \text{ in.}^2} = 5720 \text{ in. per min.}$$

or 7.9 ft. per sec.

With 15-psi back pressure, the upstream pressure would be:

$$P_1V_1 = P_2V_2 \qquad \text{C-2}$$

in which, P_1 = upstream pressure,
P_2 = downstream pressure,
V_1 = upstream velocity, and
V_2 = downstream velocity.

$$P_1 = \frac{15 \text{ lb/in.}^2 \times 100 \text{ ft/sec}}{7.9 \text{ ft/sec}}$$

$$P_1 = 190 \text{ psi.}$$

The Reynolds number for the 2 in. pipe would be:

$$N = \frac{VD\rho}{\mu} \qquad \text{C-3}$$

in which, N = Reynolds number,
V = velocity
D = diameter of the pipe
ρ = density of the flowing fluid, and
μ = viscosity.

$$N = \frac{7.9 \text{ ft/sec} \times 2 \text{ in.} \times 62.4 \text{ lb/ft}^3}{1.5 \times 10^{-5} \text{ lb-sec/ft}^2 \times 32.2 \text{ ft/sec}^2 \times 12 \text{ in./ft}}$$

$N = 1.75 \times 10^4$ or 17,500 which is a value outside the laminar range.

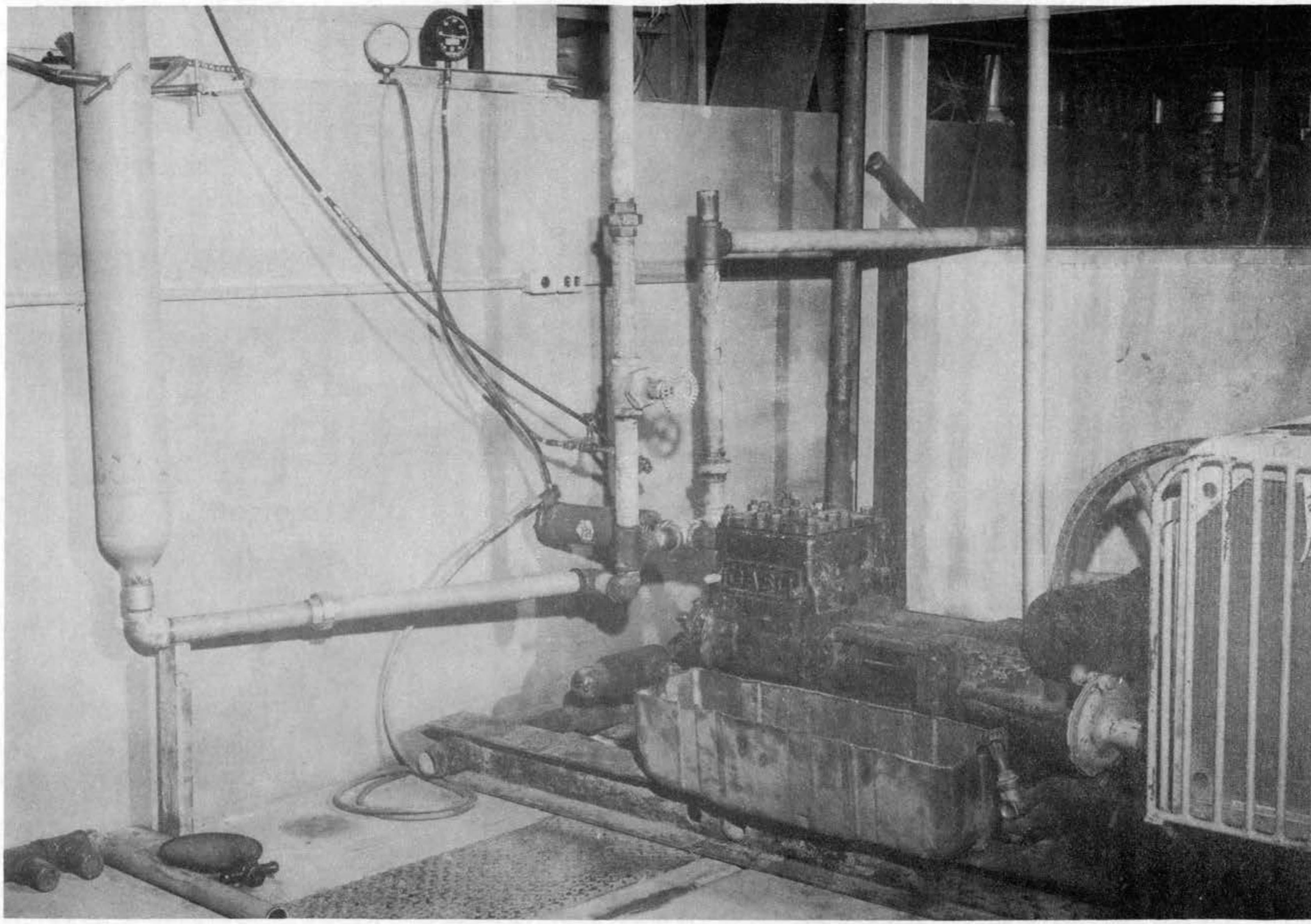


Fig. 10. Photograph of Pump Installation.

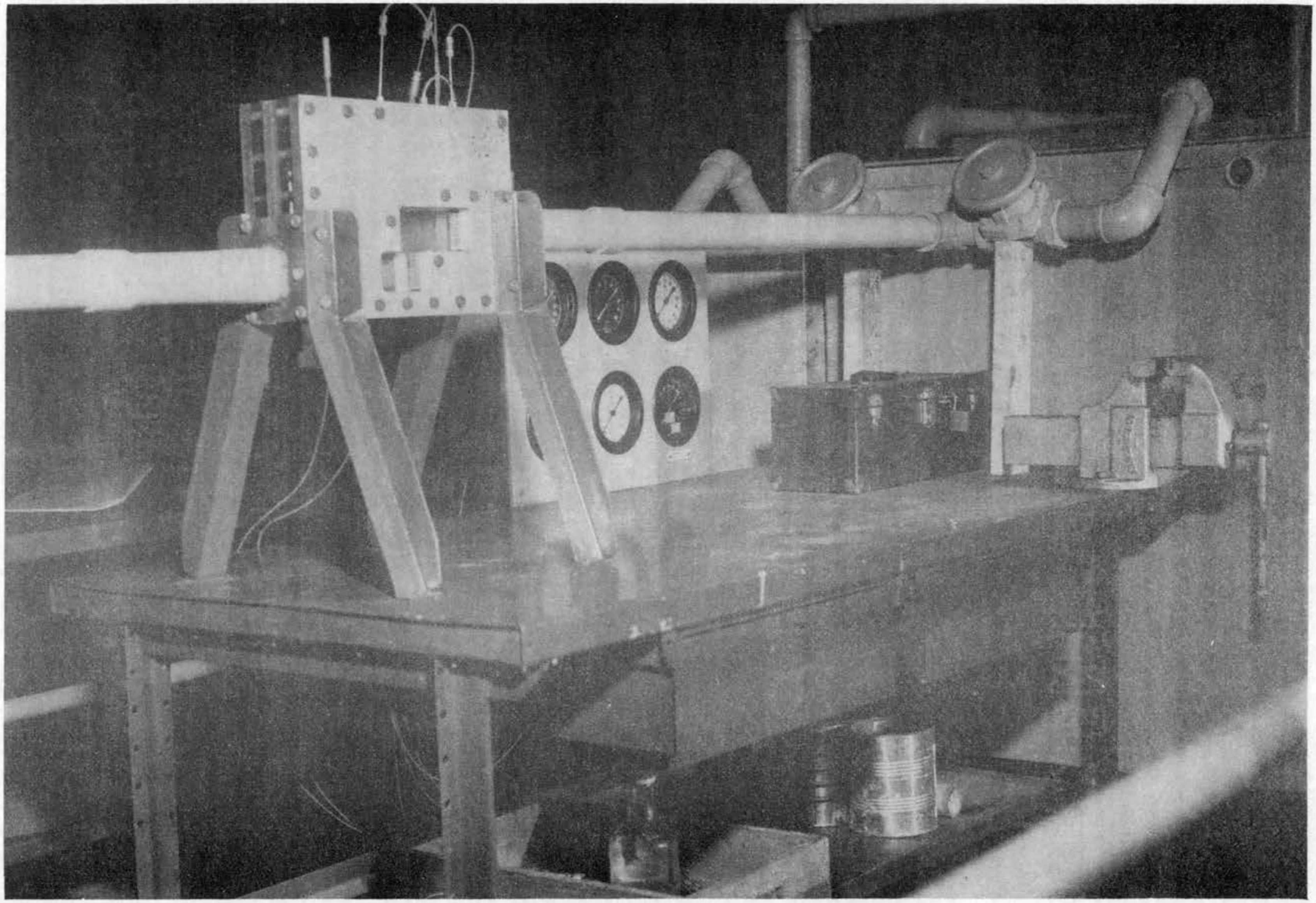


Fig. 11. Photograph of Flow Model and Instrumentation.

APPENDIX F

PARTS LIST

1. Model, Two-dimensional simulation of a spool valve, designed by Karl Reid (1), constructed by the Research and Development Laboratory of Oklahoma State University.

2. Pump, Gaso Pump Manufacturing Company, 4 x 6 in., double acting duplex. Gear ratio, 5.2:1, piston rod 1 1/8 in. dia, No. 19585.

3. Engine, Buda Division of Allis Chalmers Corporation, 4 cylinder, 153 cu. in. displacement, 49 hp. Model 4-B-153.

4. Desurger, Industrial Products Division, Westinghouse Air Brake Company, 190 cu. in. surge volume with a neoprene sleeve arrangement.

5. Air Chamber, discharge, 6 in. OD, 55 in. long, bolted blank pipe flange for top, 6 x 2 in. bell reducer at bottom.

6. Air Chamber, suction, 8 1/2 in. OD, 33 in. long, reverse flow inlet to outlet.

7. Valves, Grinnell-Saunders Manufacturing Company, 2 in. diaphragm type, 150-psi working pressure.

8. Gauges, U. S. Gauge Company, New York.

9. Pipe, 2 in. galvanized line pipe, sundry fittings.

APPENDIX G

VELOCITY AND UPSTREAM PRESSURE INCREASE FOR CHANGE FROM VERTICAL TO FREE JET

Axial clearance in.	Radial clearance in.	Upstream pressure psi	Pressure increase psi	Velocity at flip fps
0.175	0.00	45	6	50.0
		76	14	65.8
		106	6	70.4
		128	12	76.2
0.15	0.00	59	--	61.3
		99	13	70.0
		131	6	82.1
		160	13	84.5
0.125	0.00	80	5	73.6
		135	13	84.1
		177	11	93.0
		215	5	101.4
0.10	0.00	120	10	84.3
		180	--	97.8
		230	--	105.0
0.075	0.00	150	--	96.4
		230	--	110.8
0.125	0.01	61	13	71.4
		111	7	80.0
		143	9	85.0

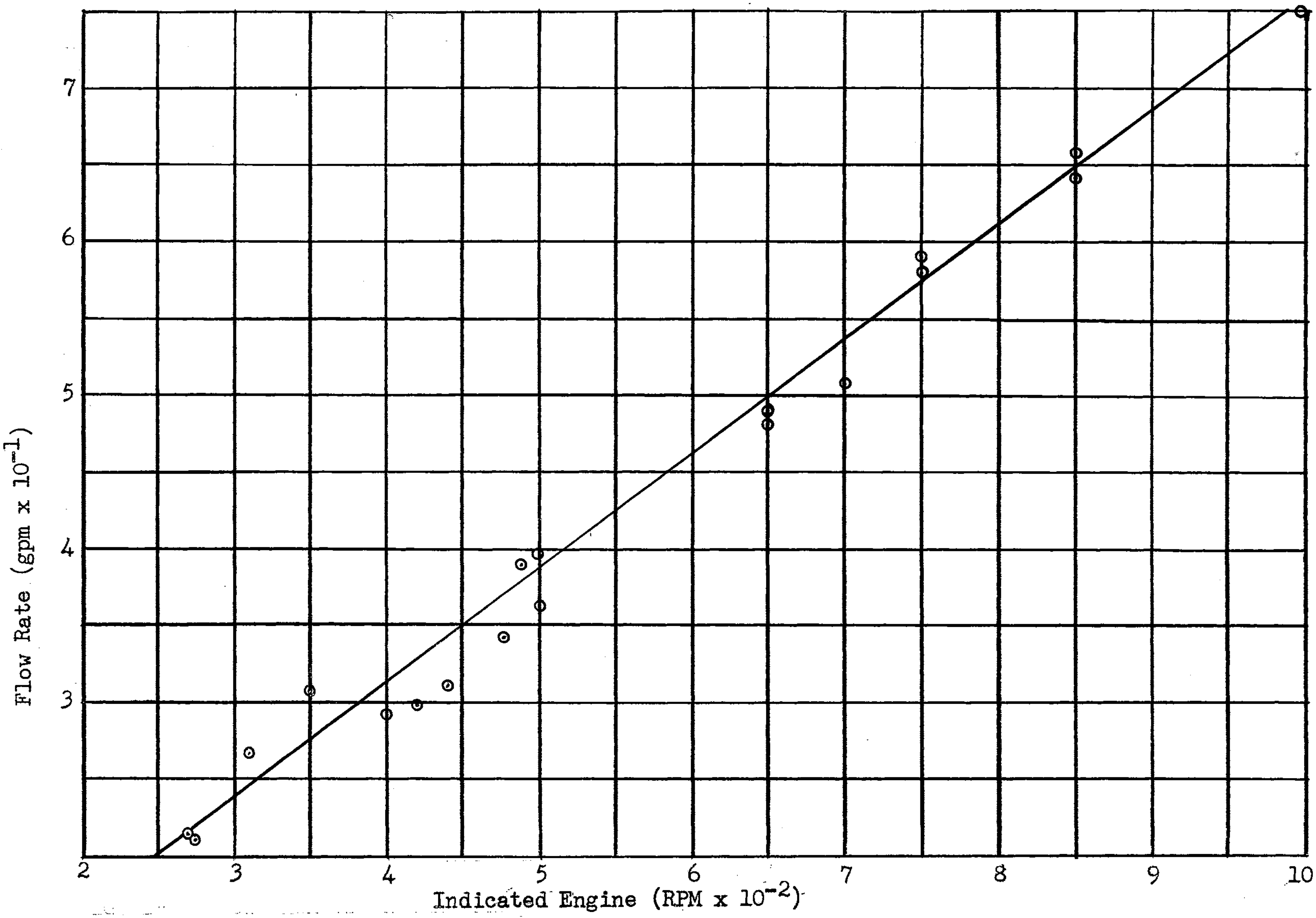


Fig. 12. Flow Rate vs Engine Speed

APPENDIX I

SELECTED BIBLIOGRAPHY

Alcock, E.D. and Sadron, C.D., "An Optical Method for Measuring the Distribution of Velocity Gradients in Two-Dimensional Flow," Physics, 6, 1935, pp 92-95.

Atherton, E. and Peters, R. H., "Some Aspects of Light Scattering from Polydisperse Systems," Brit. J. Applied Physics, v 4, n 11, Nov. 1953, pp 344-49.

Balint, E., "Flow Visualization - Tool of Fluid Research," Civ. Eng. (Lond) v 51, n 603-604, Sept. 1956, pp 991-93.

Dewey, D. R., "Visual Studies of Fluid Flow Patterns Resulting from Streaming Double Refraction," (Unpub. Ph.D. Dissertation, Massachusetts Institute of Technology, 1941).

Hargrove, L. E. Jr. and Thurston, G. B., "Photographic Method for Analysis of Fluid Motion," J. Acaust. Soc. Am., v 29, 179A, 1957.

Hauser, E. A. and Dewey, D. R. II, "Study of Liquid Flow," Ind. Eng. and Chem., 31, 786, 1939.

Maxwell, J. C., "On Double Refraction in a Viscous Fluid in Motion," Proc. Roy. Soc., v 22, 1873, pp 46-47.

Peebles, F. N., Garber, J. H. and Jury, S. H., "Preliminary Studies of Flow Phenomena Utilizing a Double Refracting Liquid," Proc. Third Midwestern Conference on Fluid Mechanics, Minneapolis., The University of Minnesota Press, 1953.

Peebles, F. N. et. al., "A Study of Laminar Flow Phenomena Utilizing a Double Refracting Liquid," Progress Report No. 1, under Contract No. NONR - 811(04), Eng. Exp. Sta. and Dept. of Chem. Engr. of University of Tennessee, 1953.

Perry, C. C., "Visual Flow Analysis," Product Engineer, v 26, n 8, Aug. 1955, pp 154-61.

Rosenberg, B., "The Use of Double Refracting Solutions in Investigation of Fluid Flow Phenomena," Navy Dept. David W. Taylor Model Basin. Washington 7, D. C., Report No. 617, 1952.

VITA

Sidney Carrol Miller

Candidate for the Degree of

Master of Science

Thesis: THE INVESTIGATION OF FLUID FLOW THROUGH A CONTROL VALVE OPENING

Major Field: Mechanical Engineering

Biographical:

Personal data: Born near Claremore, Oklahoma, February 5, 1931, the son of Sidney A. and Florence S. Miller.

Education: Attended Sequoyah grade school and high school near Claremore, Oklahoma; graduated as class salutatorian in 1949; entered Northeastern Oklahoma A. and M. College and received the Associate of Arts degree in 1955; received the Bachelor of Science degree from Oklahoma State University with a major in Mechanical Engineering in January, 1957; completed the requirements for the Master of Science degree in August, 1958.

Experience: Two years as communications chief, U. S. Army, Korea; sundry odd jobs during undergraduate career; roustabout with The Carter Oil Company, summer, 1955; Junior Petroleum Engineer for Pan American Oil Corporation, summer, 1957; instructor of Mechanical Engineering, 1957-58, Oklahoma State University.

Professional organizations: Engineer-in-Training No. 471, Oklahoma; American Institute of Mining, Metallurgical, and Petroleum Engineers, Inc.

Honorary organizations: Phi Theta Kappa, 1953; Sigma Tau, 1955; Pi Tau Sigma, 1955.

Honors and Awards: Commendation Award, U.S. Army; Achievement Award, N.E.O. A. and M.; Dean's Honor Roll, 1955; OSE Salutes Award, 1957.