AN EXPERIMENTAL STUDY OF THE DESIGN

PARAMETERS OF HYDRAULIC JET-PIPE VALVES.

by

Alan C. Curtiss

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Certified by. . . . .

_____ Thesis Supervisors

Accepted by. . . . .

Chairman, Department Committee on Graduate Students.

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ABSTRACT

The steady state characteristics of a hydraulic jet-pipe valve were experimentally studied. They were found to be strongly influenced by the precision of the alignment between the nozzle and the receiver holes. The characteristics studied included the ratio of the nozzle diameter to receiver diameter, the spacing between the nozzle and the receiver, and the distance between the two receiver holes. For most applications, the optimum values for the above parameters were found to be: the diameter ratio - 1.4; the nozzle to receiver spacing one nozzle diameter; and the receiver hole spacing - as close as possible. Fourteen pressure-flow characteristics of the valve , for various values of the above parameters, are presented in non-dimensional form.

Serious oscillations of the receiver pressure were found to exist under several conditions. The oscillations were studied but no insight into their cause or cure was found.

Thesis Supervisors: JESSE LOWEN SHEARER &
                         KARL N. REID, Jr.

Titles: ASSOCIATE PROFESSOR OF MECHANICAL
                ENGINEERING
                INSTRUCTOR, MECHANICAL ENG.
                DEPARTMENT

ACKNOWLEDGEMENTS

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I INTRODUCTION

A jet-pipe valve is a fluid power amplifier. It consists of two main elements; (1) The nozzle which converts the high-pressure low-velocity supply flow into a high-velocity low-pressure jet, and (2) The receiver plate which has two closely spaced holes drilled in it. These holes are connected to the two output ports of the valve.

The pressure difference between the two output ports and the flow passing from one to the other can be varied by directing the impinging jet more at one receiver hole than the other,   thus produce a large pressure difference for a low input force.

Valves of this type have been known and used for many years. They are currently being used by several manufacturers as the first stage of a two stage servo valve. When a jet-pipe valve is made very small, the low force level required to move the nozzle makes the valve ideal for such applications.

This type of valve has several attractive features. First, it has the ability to carry dirty fluid without tending to clog. It requires low level forces to operate. Due to it's construction it is ideally suited for a two position or "bang bang" type control system and, when used as such, is very inexpensive to manufacture. When the valve is used as a proportional control device the nozzle alignment becomes much more critical and the tolerances required in manufacturing greatly increase the cost of the valve. This type of valve is also more efficient than a flapper valve. See the thesis by Mann (1).

Among the disadvantages of this type of valve are its high leakage rate in the null position or whenever the load is blocked. It is load sensitive and the displacement required for full stroke operation is higher than a similar sized flapper valve. This means it has a lower pressure

gain for a given input displacement.

In many applications a jet-pipe valve offers the best combination of characteristics. Unfortunately there is little information available to the designer who would like to use such a valve. This work was undertaken to help remedy this situation.

2. OBJECTIVES

This work was undertaken to give a designer, interested in using a hydraulic jet-pipe valve, the necessary information to determine the various valve parameters and, after they have been fixed, be able to predict the pressure and flow outputs of the valve for any given nozzle position. This requires that he be able to pick the following variables: P_S - the supply pressure; D_N - the nozzle diameter; D_R - the receiver hole diameter; S - the spacing between the nozzle and the receiver; $2 X_{max}$ - the spacing between the two receiver holes; and know, after the above are fixed, the pressure drop ΔP_M and the flow rate Q_M the valve will produce for a given nozzle displacement (X). These variables are shown graphically in Figure (1).

The Dynamic Analysis and Control Laboratory at M.I.T. is planning to conduct a dynamic study on both hydraulic and pneumatic jet-pipe valves in the near future. A study is currently being made on the magnitude and effect of the fluid forces on the nozzle of a hydraulic jet-pipe valve. A paper has already been published by Reid (2) on the "Optimum Design Parameters of a Pneumatic Jet-pipe Valve".

It is hoped that this thesis and the above mentioned studies will help fill the need for basic information on jet-pipe valves.

JET—PIPE VALVE PARAMETERS
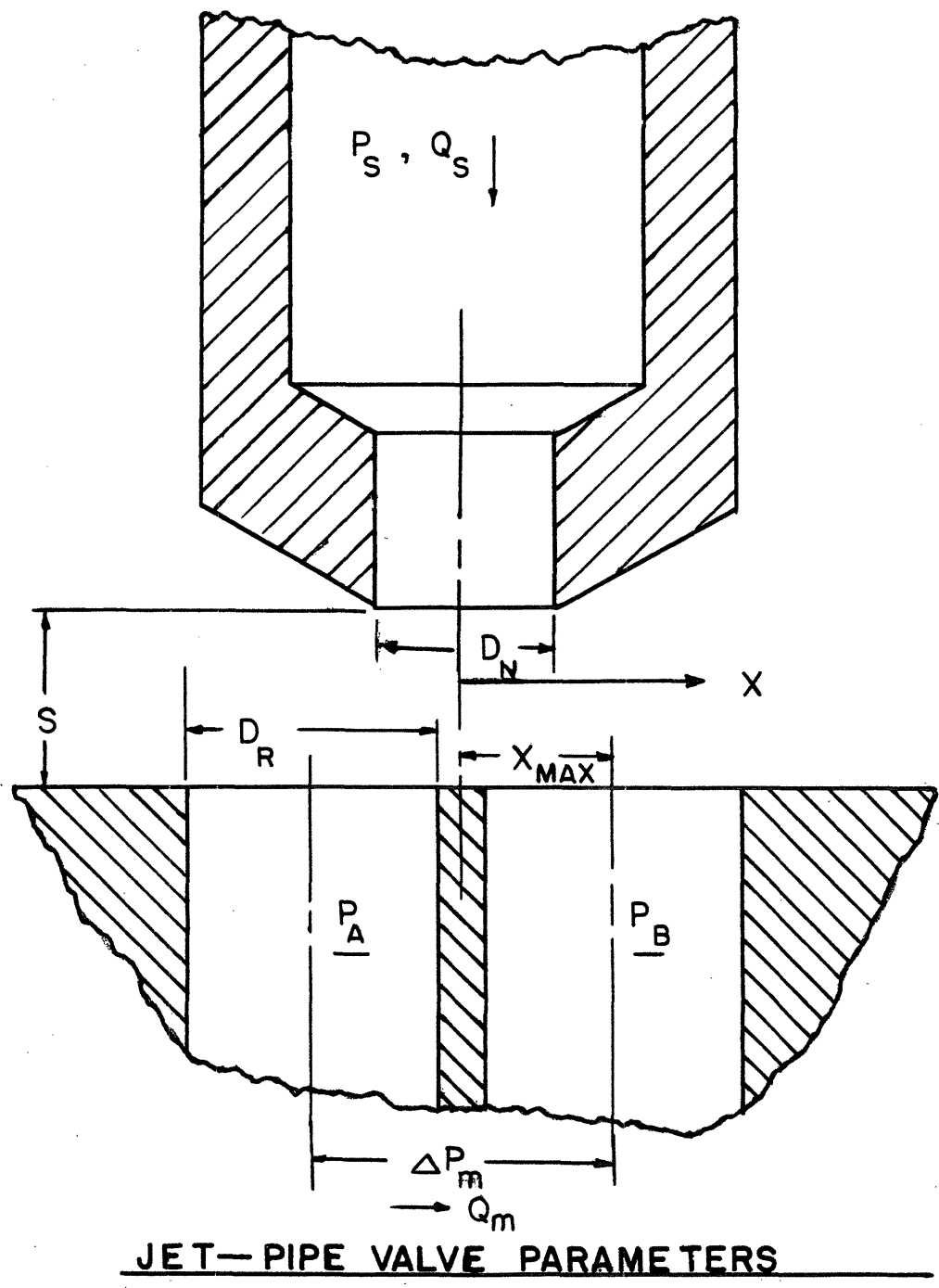
FIGURE I.

3. APPROACH

Some theoretical work has been done on J.P.V., see Dunn's thesis (3), but the return, in terms of practical information obtained for the effort put forth, has been very moderate. It was therefore decided to make an experimental study of the valves and through the use of non-dimensional analysis apply the results to the general case.

A jet-pipe valve is very sensitive to the alignment between the nozzle and receiver holes. This variable was eliminated in the first part of the test program by having the nozzle and receiver rigidly clamped in a housing that assured proper alignment. In this manner the effects of D_r, D_n, S, and P_s on the receiver flow could be made with a minimum of alignment difficulties.

A second series of tests, using a jet-pipe valve with a movable nozzle and a two hole receiver, was then made to determine the effect of the variables 2 X_M and nozzle position on the valve output.

The data from these runs were non-dimensionalized to increase their usefulness.

4. NON-DIMENSIONAL ANALYSIS

Non-dimensional analysis provides a tool for extending information obtained from one specific test to all geometrically similar cases. The method for doing this is explained in many fluid mechanics texts. A good explanation of the process is given in "Engineering Applications of Fluid Mechanics", by Hunsker and Rightmire (4).

In this case the variables considered important are those mentioned previously, plus the following: Q_S - Supply volume flow rate, P_A and P_B - Receiver pressures, ΔP_M - The difference between the two receiver pressures, and Q_M - The volume flow rate between receiver ports. The supply pressure (P_S) was assumed to be the total pressure drop across the nozzle or equal to $P_S - P_E$ if the exhaust pressure is not zero.

This gives a total of eleven variables and three units of dimensions resulting in eight non-dimensional groups which can be formed. The ones chosen were:

$$\frac{P_A}{P_S} , \frac{P_B}{P_S} , \frac{\Delta P_M}{P_S}, \frac{Q_M}{Q_S} , \frac{D_R}{D_N} , \frac{X_{MAX}}{D_N} , \frac{S}{D_N} , \text{ and } \frac{X}{X_{MAX}} .$$

If the valve is attached to a load having equal inputs and output flows then only the difference in the receiver pressures is important. In this case the following relation can be written:

$$\frac{\Delta P_M}{P_S} = f \left[\frac{Q_M}{Q_S} , \frac{D_R}{D_N} , \frac{X_{MAX}}{D_N} , \frac{S}{D_N} , \frac{X}{X_{MAX}} \right]$$

W here f stands for some unknown function. In each test four of the variables were fixed and the fifth examined as a function of the sixth. As the functional relationship was complicated it was not possible to find a mathematical statement connecting the variables.

The fluid properties of the oil were not concidered as variables in this work. A Reynolds number between 7.5×10^4 and 12.3×10^4, based on the nozzle diameter and the fluid velocity at the nozzle exit, was calculated to exist in all cases.

The test fluid was MIL 5605-SO-UNIVS-J40 hydraulic oil kept at a temperature between 60° F and 70° F.

5.   EXPERIMENTAL WORK
(SINGLE HOLE RECEIVER)

5.1   APPARATUS

The test equipment used for the clamped-nozzle single receiver test is shown in figure (2). Both the nozzle and receiver were held in a solid housing to give exact alignment and prevent nozzle movement. A total of seven receivers, providing a range of D_R from .026" to .104", and five nozzles of various geometries were used.

Both supply and receiver pressures were measured with standard, Bourdon-type, 2000 P.S.I. gauges. A glass tube rotameter was used to measure the receiver flow. A needle valve, placed upstream of the flow meter, simulated the load for the jet-pipe valve. The spacing between the nozzle and receiver was measured with a dial indicator which had divisions of .0005". During the latter part of the testing a Dynisco 0-1000 P.S.I. pressure transducer replaced the receiver pressure gauge.

5.2   TESTS PLANNED

The original plan was to measure the valve's pressure-flow characteristics at one supply pressure and one (S) spacing, but for all seven receivers. The more promising nozzle-receiver combinations were then to be used in tests where the supply pressure and (S) spacing were varied.

Soon after testing was begun it became obvious that the receiver pressure fluctuations would have to be reduced before any meaningful data could be obtained. These fluctuations were the largest when the flow through the receiver was zero but, at a ratio of $D_R/D_N = 1.5$, the oscillations remained until the receiver flow was 60% of the supply flow.
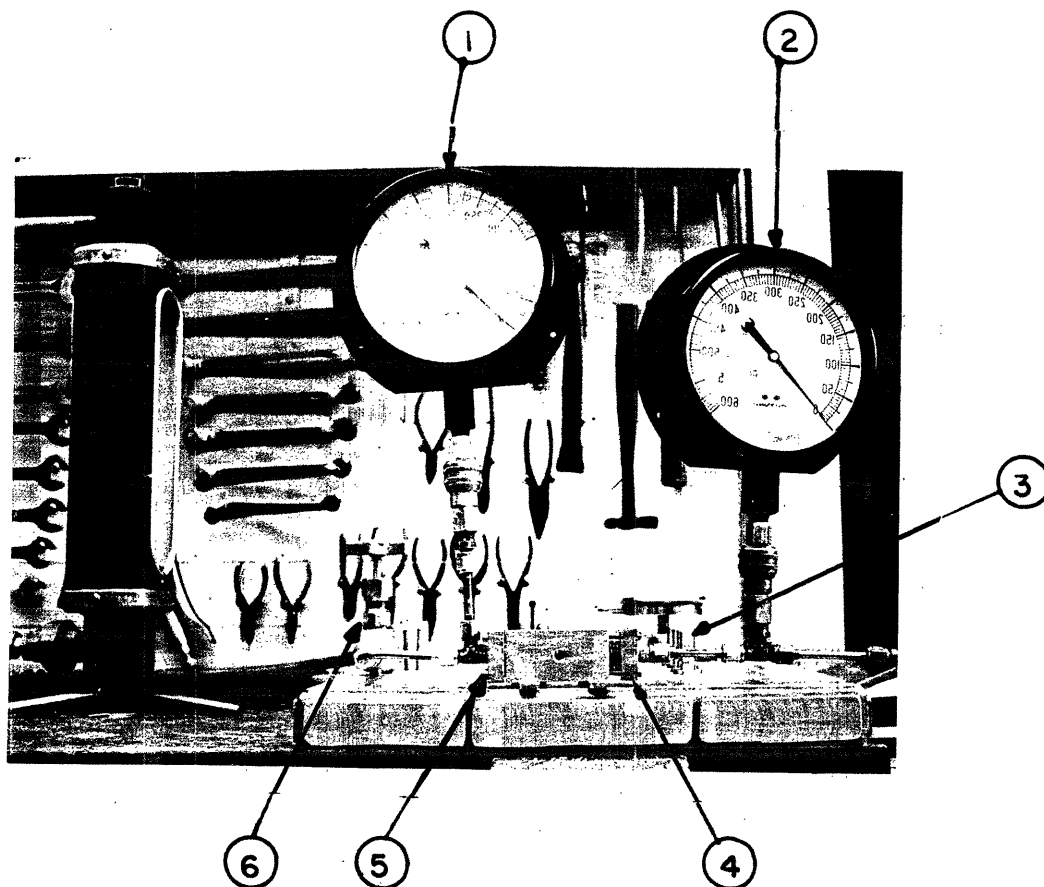
SINGLE RECEIVER TEST SET UP

1. RECEIVER PRESSURE GAUGE
2. SUPPLY PRESSURE GAUGE
3. EXTRA RECEIVERS
4. NOZZLE
5. RECEIVER
6. LOAD VALVE

FIGURE 2

Therefore the above plan was put aside so that the pressure fluctuations could be studied and hopefully eliminated. At first only a simple relative scale was used to rate the various nozzle and receiver combinations. The scale used was the magnitude the receiver pressure gauge vibrated through. It is impossible to get realistic, quantitative data in this manner but, at the time, it was felt that only relative measurements would be necessary. This was an unfortunate decision since the problem wasn't solved and actual amplitude and frequency measurements would be of greater use in indicating the direction for further study.

5.3   TEST PROCEDURE

To acquire a relative standard which the variations on the valve's geometry could be rated by, the following tests were made. A .052" I.D. nozzle, with a 120° included conical angle end, was mounted in the valve; a receiver installed; and the (S) spacing set. The load valve was closed to assure zero receiver flow. The supply pressure was then increased , in steps of 100 P.S.I., until 1000 P.S.I. was reached. The receiver pressure fluctuations were recorded at each step. Similar tests were run with all seven receivers. Any new geometry to be tried was subjected to part or all of the above test and the results compared.

At the end of the geometry study the most promising combination was used to obtain a set of pressure-flow curves.

This was done in the following manner. The valve was assembled with an (S) spacing equal to 1 D_N. The supply pressure was set at 1000 P.S.I.G. and the receiver pressure recorded. The load valve was opened slightly and receiver pressure and flow recorded.

This process was repeated until enough data was obtained to plot the receiver flow as a function of the receiver pressure.

5.4 RESULTS AND DISCUSSION

5.41 OSCILLATION STUDY

Because of the method used much of the results can only be given qualitatively.

(a) Effect of D_R/D_N

If the ratio of D_R/D_N was either 0.5 or 2.0 the oscillations were small but so was the power recovery. All ratios between 0.75 and 1.75 produced large oscillations with the 1.25 and 1.50 ratios producing the largest amplitude oscillations. When the above oscillations were measured with a pressure transducer, instead of a gauge, they were found to be less than expected. At a supply pressure of 1000 P.S.I. the maximum amplitude measured was $\pm$ 50 P.S.I. at 95-100 C.P.S. The results from one run are shown in figure (6).

(b) Effect of S/D_N

In all cases the amplitude of the fluctuations increased and the average value of the receiver pressure decreased as the distance between the nozzle and receiver increased.

(c) Effect of supply pressure.

Each nozzle-receiver combination exhibited a supply pressure below which no receiver pressure oscillations existed. This pressure was in the 300 to 400 P.S.I. range for the .052" I.D. nozzle. Ninety percent of the maximum amplitude of oscillation was reached within one hundred P.S.I. of this point. Beyond that, increasing the supply pressure only slightly increased the amplitude of the oscillations.

(d) Effect of Nozzle's exterior geometry.

Several variations were tried. First the effect of having diff-
erent diameter flats on the end of the nozzle was examined. A large di-
ameter flat forces the fluid, splashing off the receiver, out through
the thin curtain area between the nozzle and receiver. Since the (S)
spacing was only one nozzle diameter it was believed that a large flat
would prevent large eddies from forming. A zero diameter flat, or
conical nozzle would allow the splashing fluid to go more where it wish-
ed but would also allow larger eddies to form near the nozzle jet.

The diameter of this flat was varied between zero and ten times
the nozzle diameter. The results showed that by varying the flat size
it was possible to change the amplitude of the oscillations at a given
supply pressure but the high amplitude oscillations would then occur
at a different supply pressure. There was no apparent relation between
flat width and maximum pressure oscillation at a given nozzle supply
pressure.

Further experimenting along the same line involved a nozzle of
the following shape:

It was hoped that any eddies, forming around the nozzle, would flow par-
allel to the nozzle jet without interfering with it. At a supply pressure
of 300 P.S.I. the flow began to go unstable. Pressure pulses could be
felt in the valve's exhaust line and, at the same time, the receiver
pressure would drop. See figure (6) for a Brush recording of the receiv-

er pressure during this time.

By increasing the supply pressure quickly it was possible to pass through the unstable region. At 1000 P.S.I. supply pressure this nozzle showed a sizeable reduction in the receiver pressure fluctuations. It was the most promising of the steel nozzles tried.

The most satisfactory nozzle incorporated an 0.8 millimeter I.D. jewel bearing to form the jet. This type of jewel is made from an artificial ruby or sapphire. Since they are designed for use as bearings in watches the jewels are manufactured to very close tolerances, and have an excellent surface finish. A wide variety of sizes and shapes are available. The coefficient of discharge for the above size was found to be 0.79. Because of the above factors and the low cost, only 50¢ each, it was decided to use a nozzle of this type in the remaining portion of the tests.

The jewel nozzle worked very well giving 100% recovery at $D_R/D_N = .83$, only a slight pressure oscillation when $D_R/D_N = 1.2$, but at $D_R/D_N = 1.6$ the receiver pressure again began to oscillate excessively but less than was found when a steel nozzle was used. This test did show that a better formed jet can greatly reduce the pressure fluctuations.

(f) Effect of the receiver's exterior geometry.

Two variations were tried. One, the valve was run in reverse so that the nozzle with a conical end became the receiver. In this case the pressure fluctuations were reduced approximately 50%. Since this is not a type of geometry that could be produced in a two hole receiver jet-pipe valve no further study was made.

The second variation was suggested by the above. It consisted of producing a receiver from a sharp-edged thin-wall tube. Unfortunately it

did not show any reduction in the pressure oscillations.

Further work with variations on the internal geometry of both the nozzle and the receiver may prove more rewarding than the above experimenting with the exterior geometry.

(g) Effect of downstream volume changes.

No effect on the amplitude or frequency of the pressure oscillations was noticed when the volume downstream of the valve was varied, indicating the problem area is in the internal part of the valve.

(h) Effect of changing the exhaust chamber volume.

An increase in this volume by a factor of eight did not produce any noticeable changes.

The problem of receiver pressure oscillations was not found in the test on the movable nozzle jet-pipe valve. Pressure oscillations did occur for only two D_R/D_N ratios and then at a greatly reduced level. One important difference between the two test valves was that the clamped nozzle set-up used a larger nozzle diameter (.052" I.D. vs .039" I.D.). The larger diameter jet may have been too large to be stable at the supply pressure used. A second difference is that the cantilevered nozzle could conceivably have moved and thus have been deflected from a condition that would otherwise have produced oscillations, However, the cantilevered nozzle was supported from two directions and any deflection would necessarily have been small.

5.42   PRESSURE-FLOW STUDY

The receiver pressure versus receiver flow curves for five different receivers are shown in figure (3). These curves show that it is possible, with the proper D_R/D_N ratio, to recover a high percentage of the supply power. The flow recovery can greatly exceed 100% if large values of

D_R /D_N are used. This is due to a jet pump effect. One test showed the valve will pull 30" of mercury when a D_R /D_N ratio equal to 2.0 is used.

The receivers used in these tests had discharge coefficients*(C_D) between 0.55 and 0.65. If care is taken to make all receiver passages smooth and all area changes gradual, discharge coefficients much closer to 1.0 can be produced with a corresponding increase in the valve's efficiency.

Overall, the testing of this valve proved to be very disappointing. Only a small portion of the desired results were obtained. It did provide a tool for studying the pressure oscillations found in many jet-pipe valves. Work with the cantilevered nozzle, two receiver hole model did prove to be much more rewarding.

* **Measured** when the flow was from the receiver chamber into the nozzle exhaust chamber.

FIGURE 3

$\frac{Q_R}{Q_S}$

$P_S = 1000 \quad PSIG$
$Q_S = 0.8 \ GPM$
$D_N = .0315''$

$S/D_N = 1.0$

$D_r/D_n = 1.2$

$D_r/D_n = 1.67$

$D_r/D_n = .825$

$D_r/D_n = 2.09$

$D_r/D_n = 2.5$

P_A / P_S
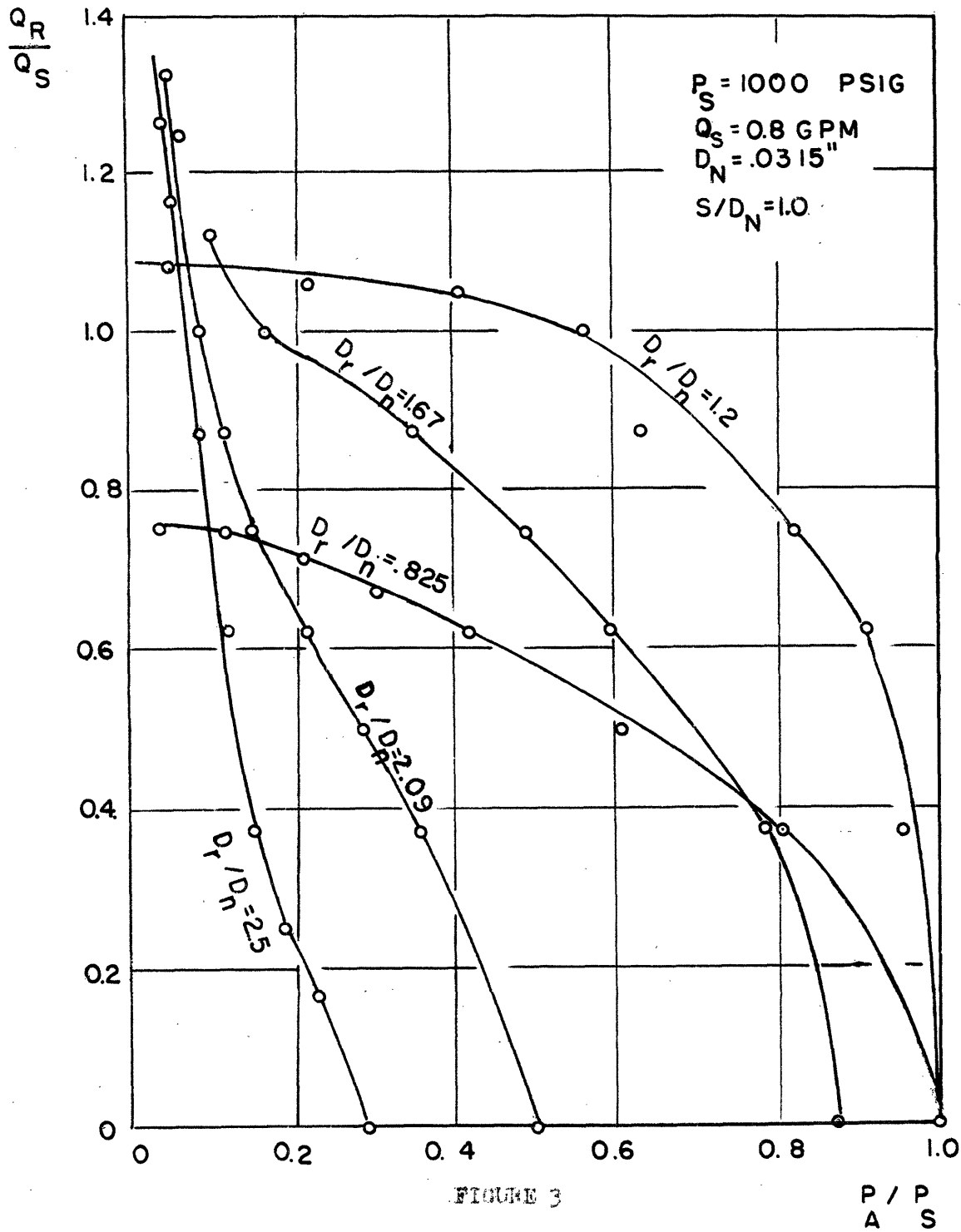
6.   EXPERIMENTAL WORK
(TWO RECEIVER HOLE VALVE)

6.1   APPARATUS

(a)   JET-PIPE VALVE

The experimental valve assembly is shown in figures (4 & 5). It has two important parts: (1) the nozzle assembly consisting of a jewel bearing mounted in the end of a cantilevered thin wall tube; and, (2) the receiver assembly consisting of a small disk, in which the receiver holes are drilled, and a barrel on which the disk is mounted.

The nozzle is positioned over the receiver holes through the use of two differential screws, mounted with their center lines 90° apart. Adjustment of one of the differential screws assures that the nozzle motion in the plus (x) and minus (x) direction is always along the line connecting the centerlines of the two receiver holes. The second provides the motion in the (x) direction.

Seven receiver disks were made giving a variation in D_R from .022" to .065" and a range of X_{max} from .0175" to .043". The barrel that these disks mount on was rotated to make sure the nozzle motion was always parallel to the line connecting the center lines of the receiving holes. It was also possible to raise or lower the barrel with a screw to provide adjustment in the spacing between the nozzle and the receiver disk.

The chamber surrounding the nozzle and receiver was exhausted directly into the splash tray of the test stand. This kept the exhaust pressure below 2 P.S.I.G.

(b)   HIGH PRESSURE FLOW METER

The flow from one side of the receiver to the other was measured

P_S , Q_S

DIAL
INDICATOR

DIFFERENTIAL
SCREW

P_E

P_A

P_B

FLOW
METER

Q_M

LOAD
VALVE
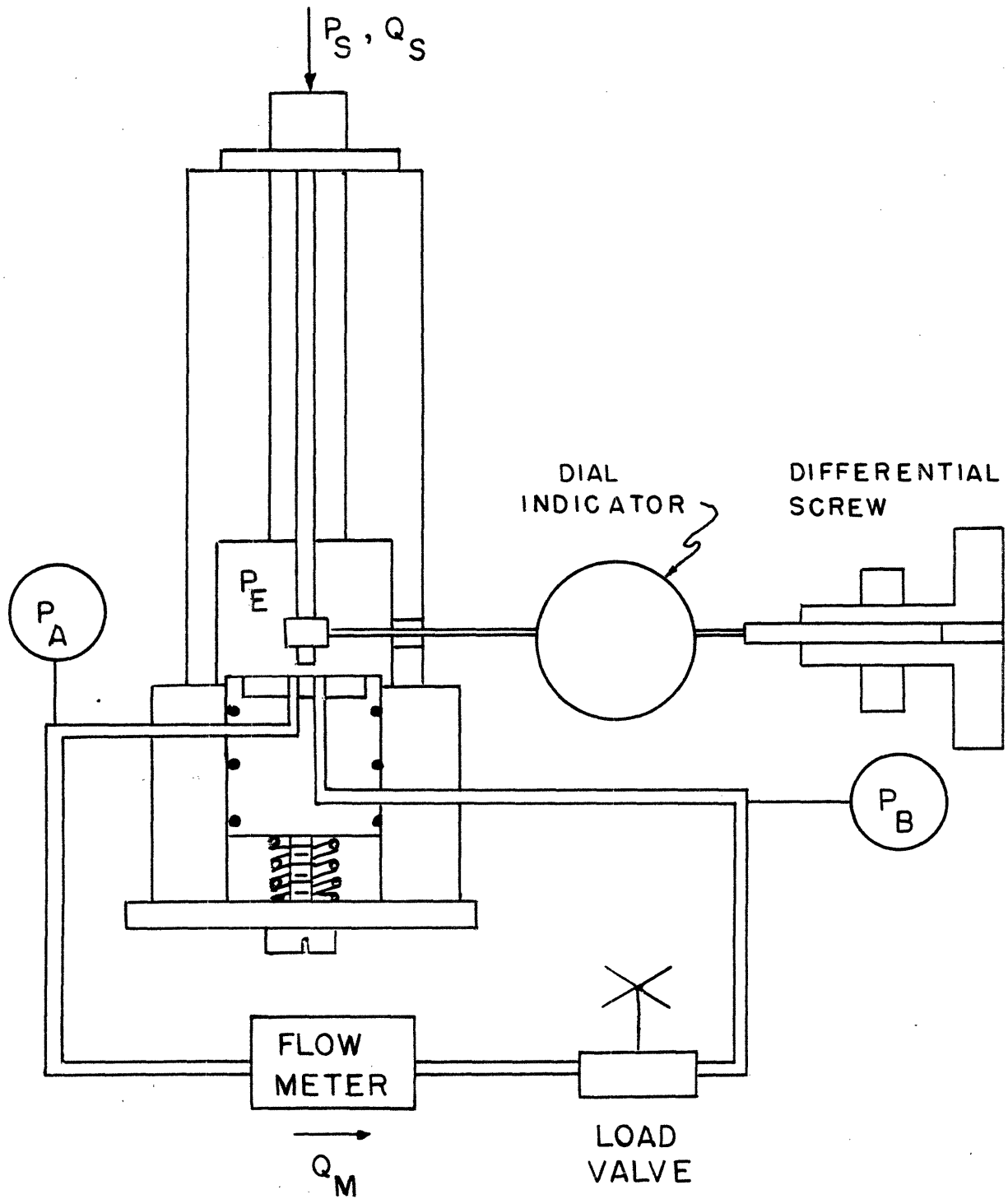
SCHEMATIC— TWO RECEIVER HOLE VALVE

FIGURE 4

TWO RECEIVER HOLE VALVE SET UP

FIGURE 5

with a small (.05 cubic inches per revolution) Vicker's piston motor driving a D.C. generator. The generator's output was calibrated with a low pressure glass tube rotameter to give the flow rate as a function of the current generated.

(c) Pressure measurements

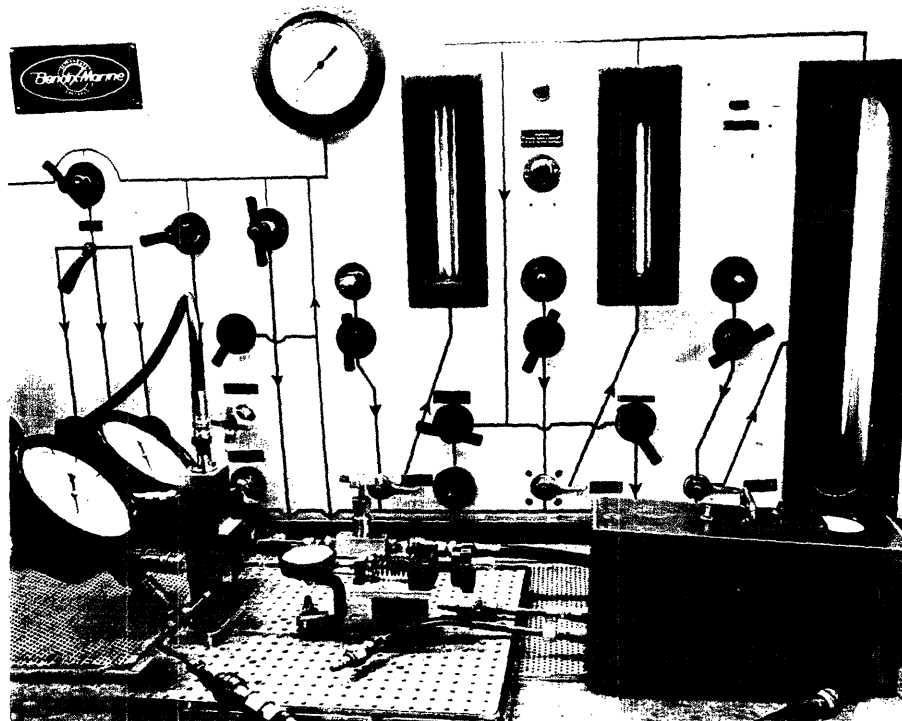All pressures were measured with 2000 P.S.I. gauges except in a few of the last runs when a Dynisco pressure transducer was substituted for one of the receiver pressure gauges.

(d) System load.

The load for the jet-pipe valve was simulated with a standard high pressure needle valve.

(e) Jet-pipe motion measurements.

The jet-pipe was stroked with a differential screw connected in series with a dial indicator having divisions of .0005".

6.2   TEST PROCEDURE

First the desired receiver disk was installed and the (S) spacing set. The valve must then be properly aligned to make sure that, when the nozzle is stroked, the jet always impinges on a line connecting the center lines of the two receiver holes.

The alignment of this valve is very critical and is therefore one of its weak points, especially when nozzle diameters in the order of .010 inches and smaller are used. The smallest one used was .031" ID so reasonable accuracy could be obtained. Even with this large a nozzle, misalignments of .001" to .002" could be detected when a plot of pressure versus nozzle displacement was made. To get optimum performance it is essential that the nozzle travel be exactly along a line drawn between the

centerlines of the two receiver holes. To assure this requires either time consuming hand adjustment or a very close tolerance on the parts that position the nozzle and the receiver.

After the valve was aligned, the nozzle was positioned mid way between the two receiver holes and the dial indicator set at zero. This position can be found readily by watching the receiver pressures. When they are equal the mid point has been reached. Since this is a very steep portion of the pressure displacement curve this method is very accurate.

To obtain the pressure-displacement curves, figures (9 to 23) the following procedure was used. The receiver ports were sure zero flow. By turning the differential screw the nozzle was positioned 10% of the way toward one of the receiver holes and the pressure at each port recorded. This procedure was repeated to give ten data points between the valves mid position and the point where the valve is aligned directly over one receiver hole. Traverses in both the plus and minus (x) direction were made.

The effect of (S/D_N) was found by a similar method, except steps of 20% of the total travel were taken. The value of (S/D_N) was varied from 0.5 to 4.0 for three different receivers. Two complete pressure-flow-displacement curves were run at two (S) spacings to determine the effect on flow recovery the value of (S) has.

To obtain the pressure-flow-displacement curves, figures (24 to 38), tests were started as above, taking 20% steps in the plus (x) direction. For each value of x, the load valve was slowly opened to obtain enough points to plot the pressure-flow curve.

6.3   NOTES ON (P_S)

The value of P_S (supply pressure) used in non-dimensionalizing the data was not the 1000 P.S.I. indicated on the gauge upstream of the jet pipe nozzle but the pressure which, under optimum conditions, would give 100% pressure recovery on one side of the receiver. This may sound quite rash at first but was made only after careful consideration of the following points and is felt to be within 1% of the actual pressure upstream of the nozzle.

A discrepancy was noticed when the pressure recovery from two jewelled nozzles, used at similar spacings and D_R/D_N ratios but with different size supply tubes, were compared. The nozzle with the .22" I.D. supply tube consistently showed 100% recovery while the one with a 4" long .078" I.D. supply tube only yielded 88% or 92% depending on the flow rate. A calculation showed the pressure difference could be accounted for by the frictional loss in the supply tube but the Reynolds number in the tube was in the region where the value of the friction factor could not be accurately determined.

An attempt to actually measure the pressure drop in the four inch long .078" I.D. tube failed because of distortion introduced at the point where the pressure tap was silver soldered on. If the measured pressure drop was correct the nozzle was giving 105% recovery.

Comparison of the nozzle-flow calibration for a 0.3 M.M. jewel mounted in a .22" ID. tube with one in the .078" I.D tube accounted for the missing 80 P.S.I. if the flow rate was 0.765 G.P.M. instead of 0.76 G.P.M. as originally recorded. This small an error is entirely within the accuracy of the system.

After the work above no doubt was left that the actual pressure

just upstream of the 0.8 M.M. nozzle was 920 P.S.I. and upstream of the 1.0 M.M. nozzle was 880 P.S.I.G., which were the maximum recovery pressures for the respective nozzles.

This points out the fact that it is important to have well designed nozzles if a high performance valve is to be built.

6.4 RESULTS AND DISCUSSION

6.41 PRESSURE-DISPLACEMENT-FLOW DATA

The primary information this work was designed to produce is presented in the pressure-flow-displacement curves (figures 24 to 38). The pressure recovery is given in terms of the pressure difference between the receiver ports. Data in this form is required if the system load **is** a ram with equal areas on each side, a piston motor, or any load that has equal flows entering and leaving. If the jet-pipe valve is used to drive a ram with unequal areas, so that flows entering and leaving the load are not equal, it is necessary to know the pressure-flow characteristics for each side of the valve.

An approximate method for obtaining this from the data presented is given below. It should give the upstream pressure within a few percentages but the downstream or low pressure side may only be accurate within 20-25%.

EXPANSION OF DATA TO GIVE $\dfrac{P_A}{P_S}$, $\dfrac{P_B}{P_S}$ vs. $\dfrac{Q_M}{Q_S}$.

---

1) From the equation $Q = .6A\sqrt{\dfrac{2\Delta P}{\rho}}$ calculate the pressure flow curve for the low pressure side of the receiver.

2) From the P_B/P_S vs X/X_{max} curves, with the desired value of the D_R/D_N and X_{max}/D_N ratios, determine the zero-flow pressure

at the values of X/X_{max} of interest.

3) Plot a series of curves, similar to those calculated in step 1), starting at the points found in step 2).

4) Add the values of P_B/P_S found in step 3) to the value of

$$\frac{P_A - P_B}{P_S} = \triangle \frac{P_M}{P_S}$$ given in the pressure-flow curves,

figures (24 to 38) to get the value of P_A/P_S as a function of

$$Q_M/Q_S = Q_A/Q_S = Q_B/Q_S.$$

This method will give a fair approximation of the pressure flow characteristics of each side of the valve. The slope of the actual P_B/P_S versus Q_M/Q_S curves decreases for increasing X/X_{max} instead of being exactly similar as assumed above.

Returning to an examination of the pressure-flow curves we find some expected and some unexpected results.

For $D_R/D_N = 0.5$ the curves look very much like those for a flow compensated valve. This would be a simple way to produce such a valve although it is very inefficient in terms of power recovery.

When D_R/D_N is in the 0.8 to 1.0 range the pressure-flow curves are very linear over most of the flow range but curve sharply and approach the zero flow line almost vertically. The results near the zero pressure drop line could only be estimated. A flow meter with a very low pressure drop was not available.

The spacing between the curves decreases as X/X_{max} approaches I. This is seen more clearly in the zero-flow pressure-displacement curves, figures (9 to 23). They have linear regions that last only for X/X_{max} up to 0.2 or 0.4 depending on other parameters. After this point the curves bend sharply so that near $X/X_{max} = .9$ very little pressure change is noticed for a 20% change in X/X_{max}.

When D_R/D_N 1.0 two points of interest are noticed. First, it is possible to have a jet pump effect existing that produces a receiver flow larger than the supply flow.

The second point of interest is the very undesirable dips right in the middle of the pressure flow curve . These dips are more pronounced when X/X_{max} = 0.4 to 0.8 than when it equals 1.0. This is one good point if this valve is to be used in a "bang-bang" type control system.

Considerable time has been spent trying to eliminate or at least explain these dips but without success. However the following facts are known:

1) The dips are only seen in the pressure-flow curve of the side of the receiver the nozzle is closest to, ie, Q_A/Q_S vs P_A/P_S.

2) The dips are not caused by an interference between the returning flow and the nozzle jet. A test run with the return flow disconnected exhibited the same pressure-flow curves.

3) In the region of the dips small pressure oscillations exist in the receiver pressure. These are on the order of 20 to 30 P.S.I. at 20 to 30 C.P.S. See figures (7 & 8).

4) If the nozzle to receiver spacing is increased to three diameters the dips are reduced substantially but so is the flow recovery (down 17%). See figures (33 & 34).

5) Varying the nozzle flat from zero (sharpedged) to .200" diameter did not effect the dips. If turbulence was causing the nozzle to vibrate it would appear that a larger area presented to the turbulence would radically effect the results.

6) A bleed between the supply pressure and the receiver pressure will remove the dip but only if the bleed exceeds 40% to 50% of the nozzle flow. This was not a very practical solution to the problem.

7) Doubling the volume of the line connecting the two receiver ports does not effect the amplitude or frequency of the oscillations or the size of the dips. This indicates the problem is in the region of nozzle or at the entrance to the receiver.

8) The dips were much more pronounced in the run made with a supply pressure of 100 P.S.I.G. Since the jet was moving at a speed $1/\sqrt{10}$ times that in the other runs the turbulence, although it must be reduced, has a correspondingly longer time to act on the fluid jet.

9) The dips could be explained by a .002" to .003" deflection of the nozzle but the nozzle was known to be stiff enough to prevent motions of that magnitude. The possibility of the fluid jet moving, however, does exist.

The effects the above dips have on the dynamic response of the valve was not examined.

As mentioned in the single receiver test the discharge coefficients of the receiver holes were quite low, between .58 and .65. This represents an internal parasidic loss that is not available to drive the load. Since this is undesirable, care should be taken to keep all such losses to a minimum.

A valve similar to the one tested will show greater pressure and flow recovery than indicated on the graphs if it has better formed flow passages and thus higher C_D's. Therefore a correction factor must be applied to the curves if a valve with a receiver hole discharge coefficient greatly different from 0.6 is used.

6.42   PRESSURE-DISPLACEMENT DATA

The pressure-displacement characteristics of the valve are shown in figures (9 to 23). The main point of interest is the pressure gain for the valve when it is in the null position. Since the gain must be in terms of the actual nozzle displacement and not the non-dimensionalized displacement, no value of maximum gain can be found until a value of X_{max} is know. A range of X_{max} can be assumed and the corresponding gains calculated to find what value gives the highest gain.

6.43   PARAMETER OPTIMIZATION

a) D_R / D_N This ratio effects the maximum power transfer, pressure recovery, and flow recovery. Using maximum power transfer as a criterion figure (39) shows the optimum value to be 1.4. This is the same value found by Reid (2) when air was used as a working fluid. It may actually be the $\sqrt{2}$ but there is no known reason for this at the present.

From figure (40) it can be seen that the efficiency curves for the valve are reasonably broad and that the peak is reached at a pressure recovery of 60% to 70%, which is desirable.

There are two factors that may make it necessary to operate at some ratio other than 1.4. First, most systems require a high system stiffness in the null position. From figure (41) it can be seen that to achieve a null pressure to supply pressure ratio of 0.6 the two receivers would have to touch which is not practical. For ratios of D_R / D_N greater than 1.0 most applications will require the holes to be as close as possible. Aside

from the low mid-position pressure, large receiver holes require large nozzle strokes. The dynamic requirements of a system may make it advisable to sacrifice some of the valve's efficiency to permit faster response.

The second factor is the very embarrassing dips found in the pressure-flow curves when D_R/D_N is greater than 1.0. The effect of this on the valve's dynamic response is not known.

b) X_{max}/D_N This ratio determines the mid-position pressure and effects the valves pressure gain. This variable normally should be kept as small as possible. The effect of this variable on the mid-position pressure is shown in figures (41 & 42).

c) S/D_N This ratio defines how far the nozzle is from the receiver. A considerable spread was found in the data concerning this variable. Only one case is plotted in figure (43). The important results from the other test are:

1) Pressure recovery drops only 2-3% when S/D_N increases from 1.0 to 3.0 if D_R/D_N is equal to 1.0, but drops 10% when D_R/D_N is increased to 2.0.

2) The above change in S/D_N produces a 18% drop in the maximum flow recovery.

See figures (33 & 34).

3) The zero-flow pressure fluctuations become annoyingly large when S/D_N is increased much above 3.5 or 4.0.

Under the conditions used in this work a value of S/D_N equal to 1.0 was considered to give the best results.

It is possible to replot the data obtained in many different ways, depending on what factors are the most important in any specific application. As the number of figures is already getting out of hand other possibilities are not shown.

SHARP EDGE STEEL NOZZLE

$D_N = .052"$ $P_S = 900$ P.S.I.

$D_R / D_N = 1.0$ $S/D_N = 1.0$

|←—.2 SEC.—→| ←—— TIME INCREASING

860 PSIG   80 PSIG
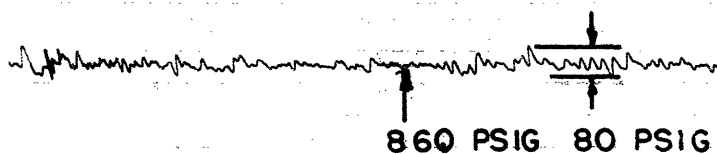
$\omega \gtrless 75$ CPS.

FIGURE 6-A

NOZZLE —→

$D_N = .052"$ $P_S = 300$ P.S.I

$D_R / D_N = 1.25$ $S/D_N = 1.0$

—| |— 1.0 SEC.   TIME INCREASING ←——

$\omega = 1.0 - 1.5$ C.P.S.

300 PSI   50 PSI
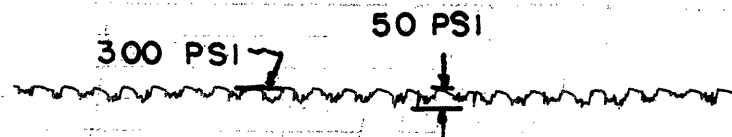
FIGURE 6 -B

RECEIVER PRESSURE OSCILLATIONS

SINGLE RECEIVER VALVE

CANTILEVERED JEWEL BEARING NOZZLE.
LOAD VALVE OPENED AND CLOSED.

$D_N = .031"$ $P_S = 1000$ P.S.I.

$X/X_{MAX} = 0.56$ $S/D_N = 1.0$

$D_R/D_N = 1.51$ $X_{MAX} = .034"$

I. SEC.

650 PSIG

TIME INCREASING

475 PSIG

40 P.S.I.   68 P.S.I.
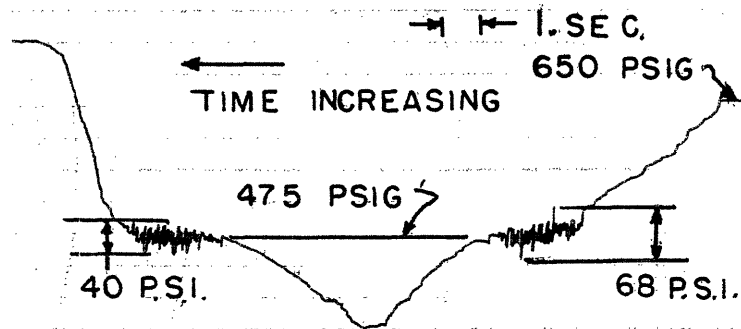
FIGURE - 7

UPSTREAM RECEIVER PRESSURE AS LOAD
VALVE WAS OPENED AND CLOSED AT
APPROXIMATELY CONSTANT SPEED. FIGURE 8
SHOWS AN EXPANSION OF THE OSCILLATORY
REIGON.

TWO-HOLE-RECEIVER VALVE

CANTILEVERED JEWEL BEARING NOZZLE

$D_N = .031"$ $P_S = 1000$ P.S.I.

$D_R/D_N = 1.5.1$ $S/D_N = 1.0$

$X/X_{MAX.} = 0.56$ $X_{MAX.} = .034"$

TIME INCREASING

0.2 SEC. $\omega = 20\text{-}30$ C.P.S.

480 PSIG

38 P.S.I.
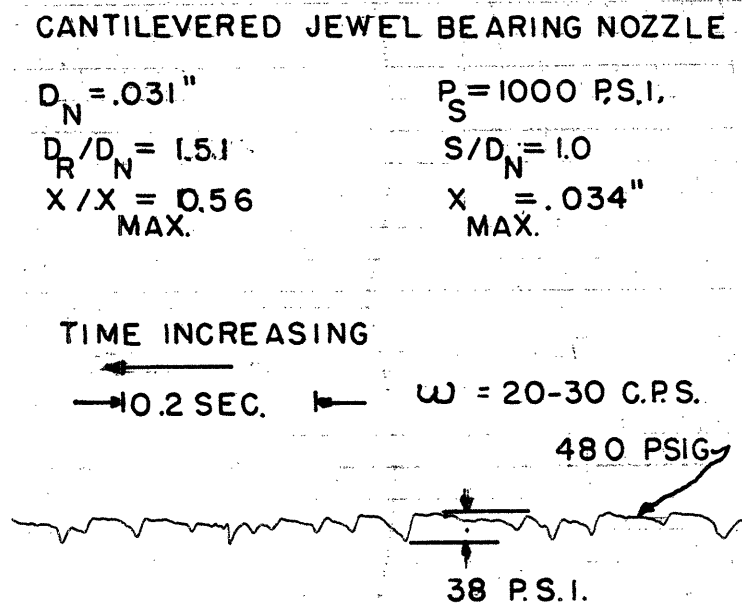
FIGURE 8

RECEIVER PRESSURE OSCILLATIONS DURING THE

DIP IN THE PRESSURE — FLOW CURVE.

FIGURE 9

$P_s = 880$ PSIG
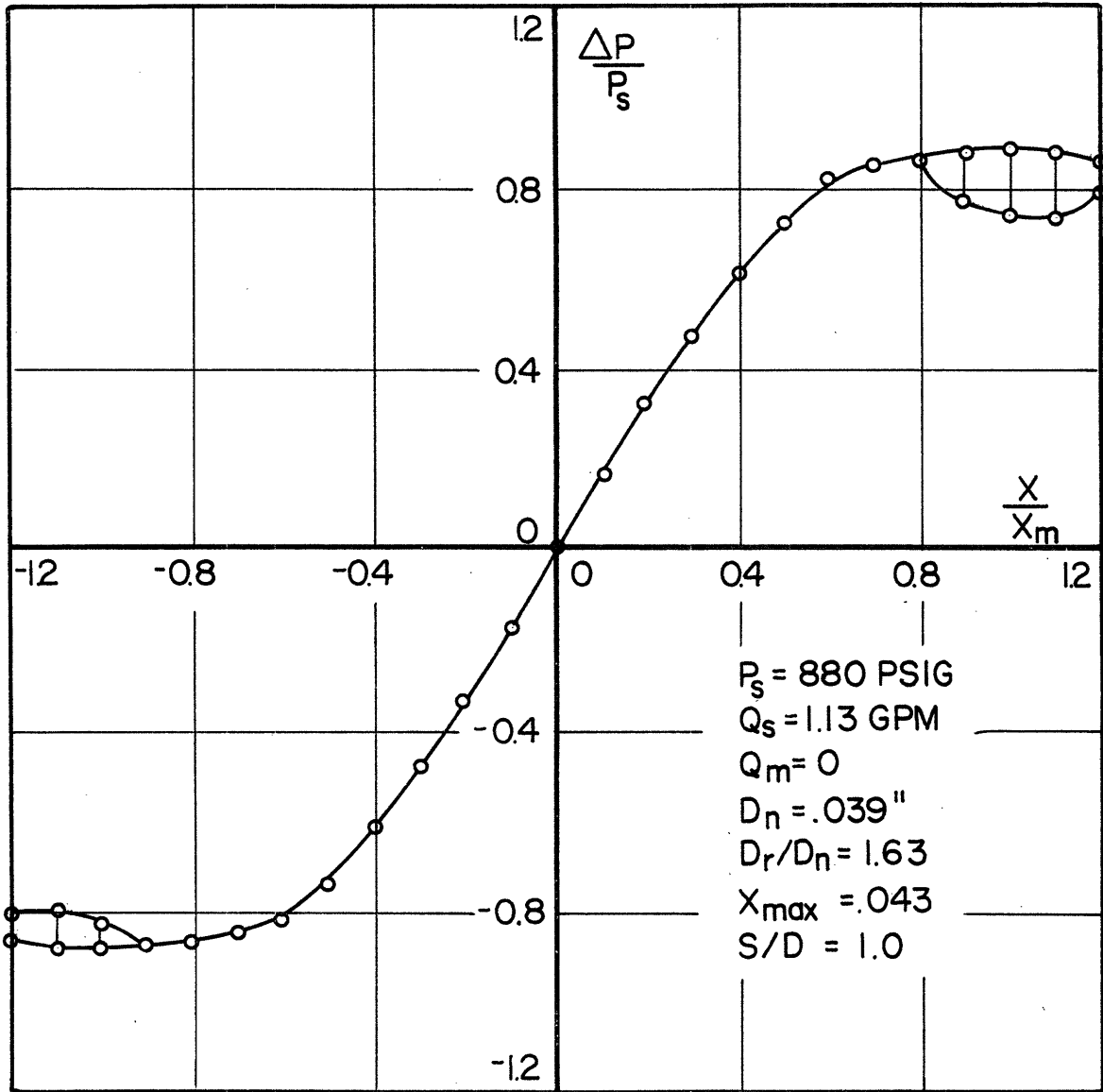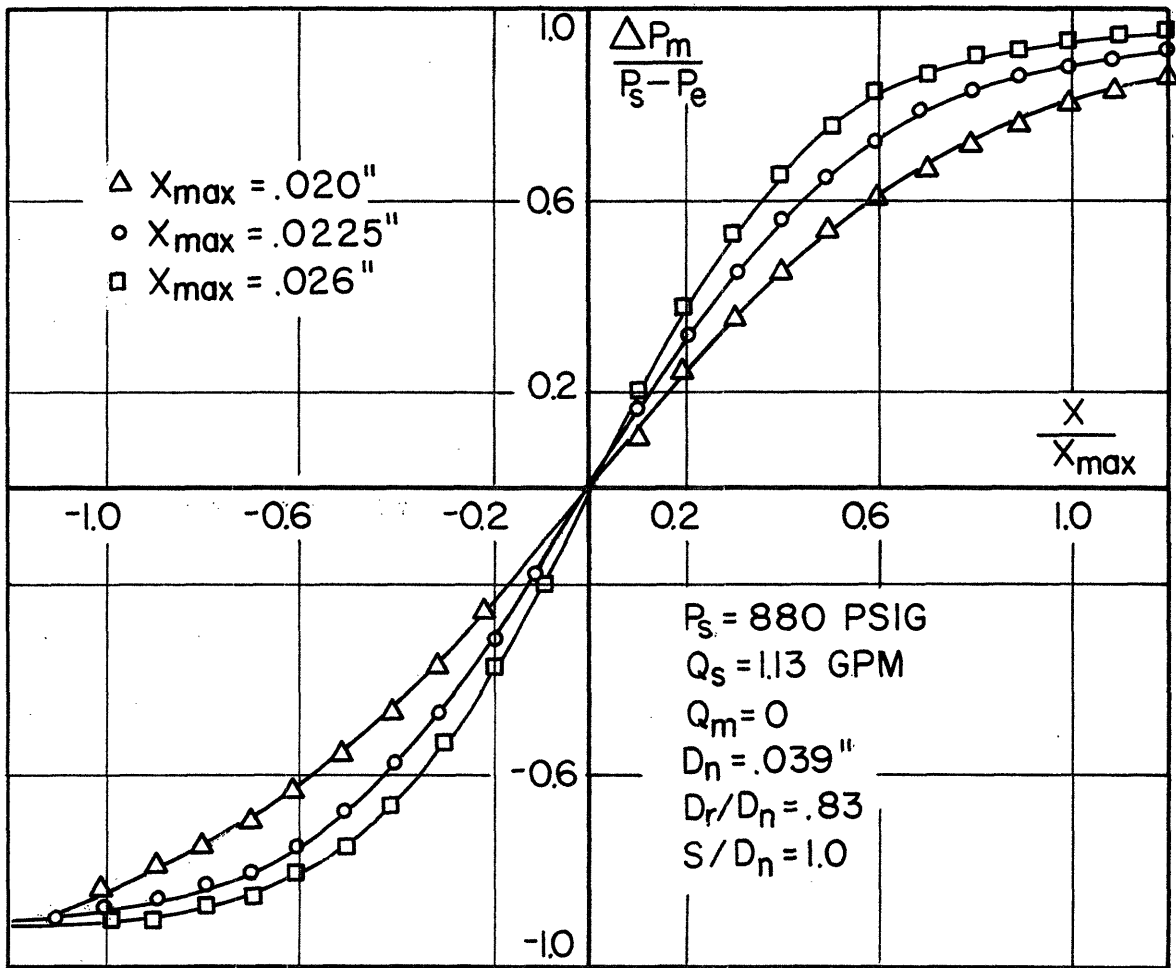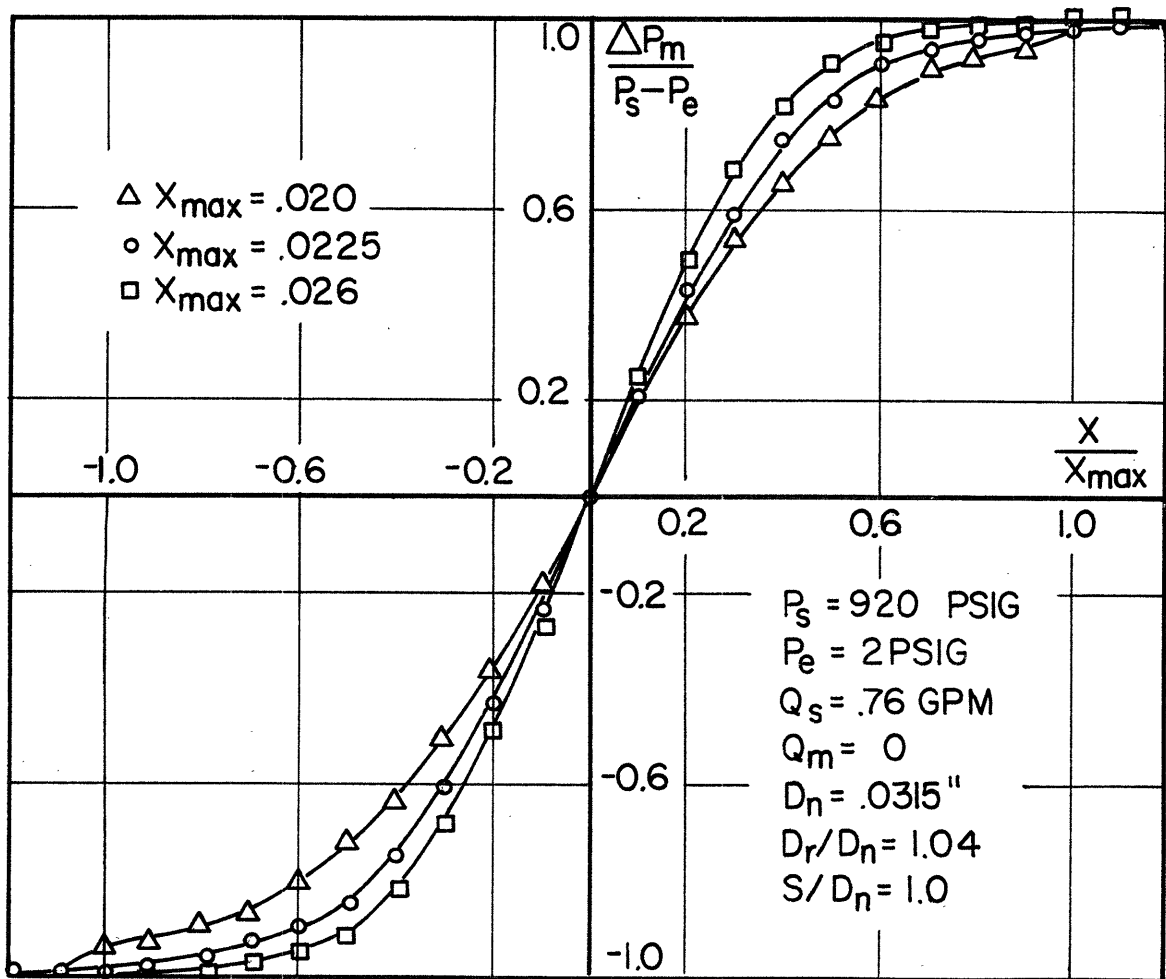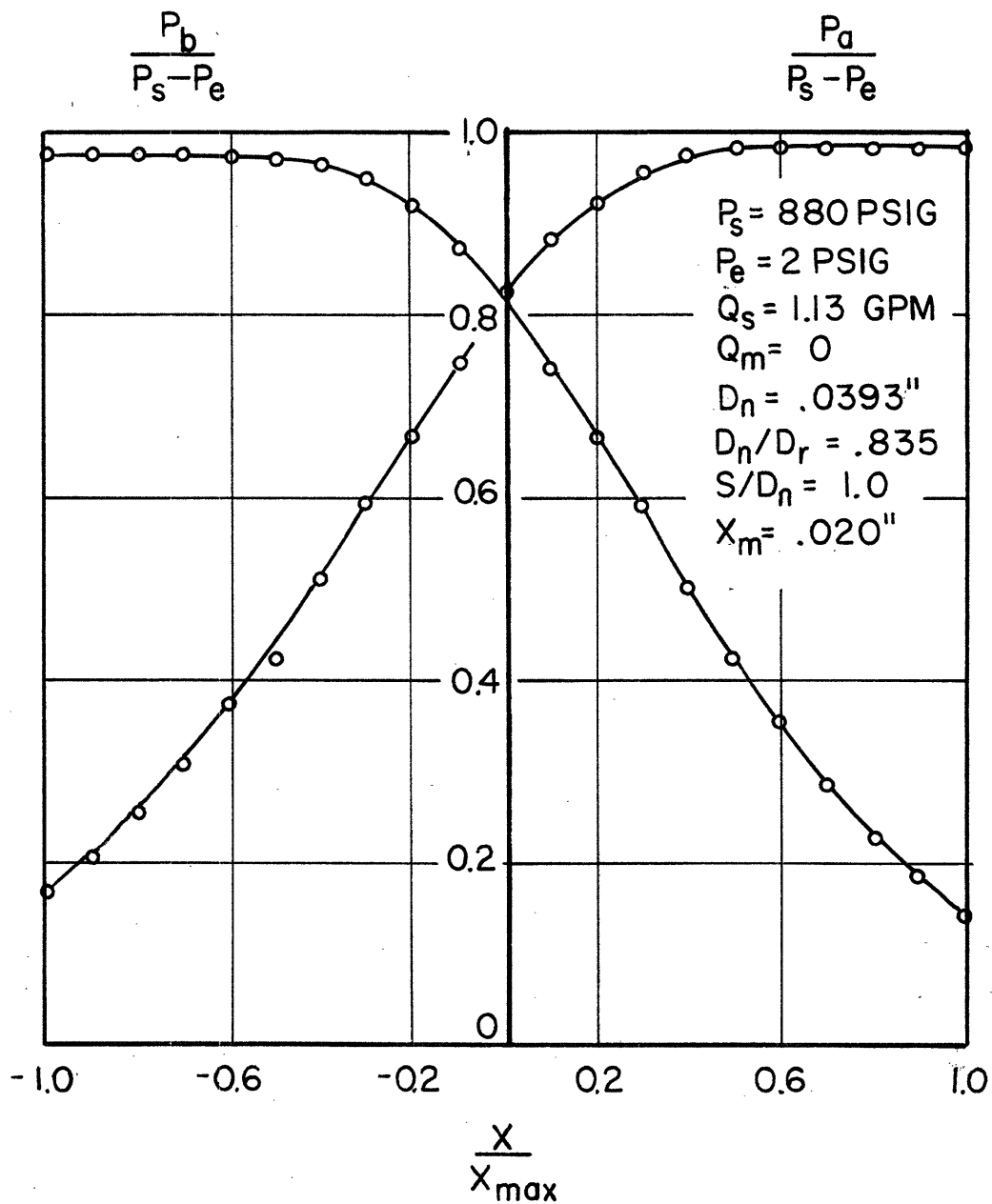$Q_s = 1.13$ GPM
$Q_m = 0$
$D_n = .039''$
$D_r / D_n = .835$
$X_{max} = .020$
$S/D_n = 1.0$
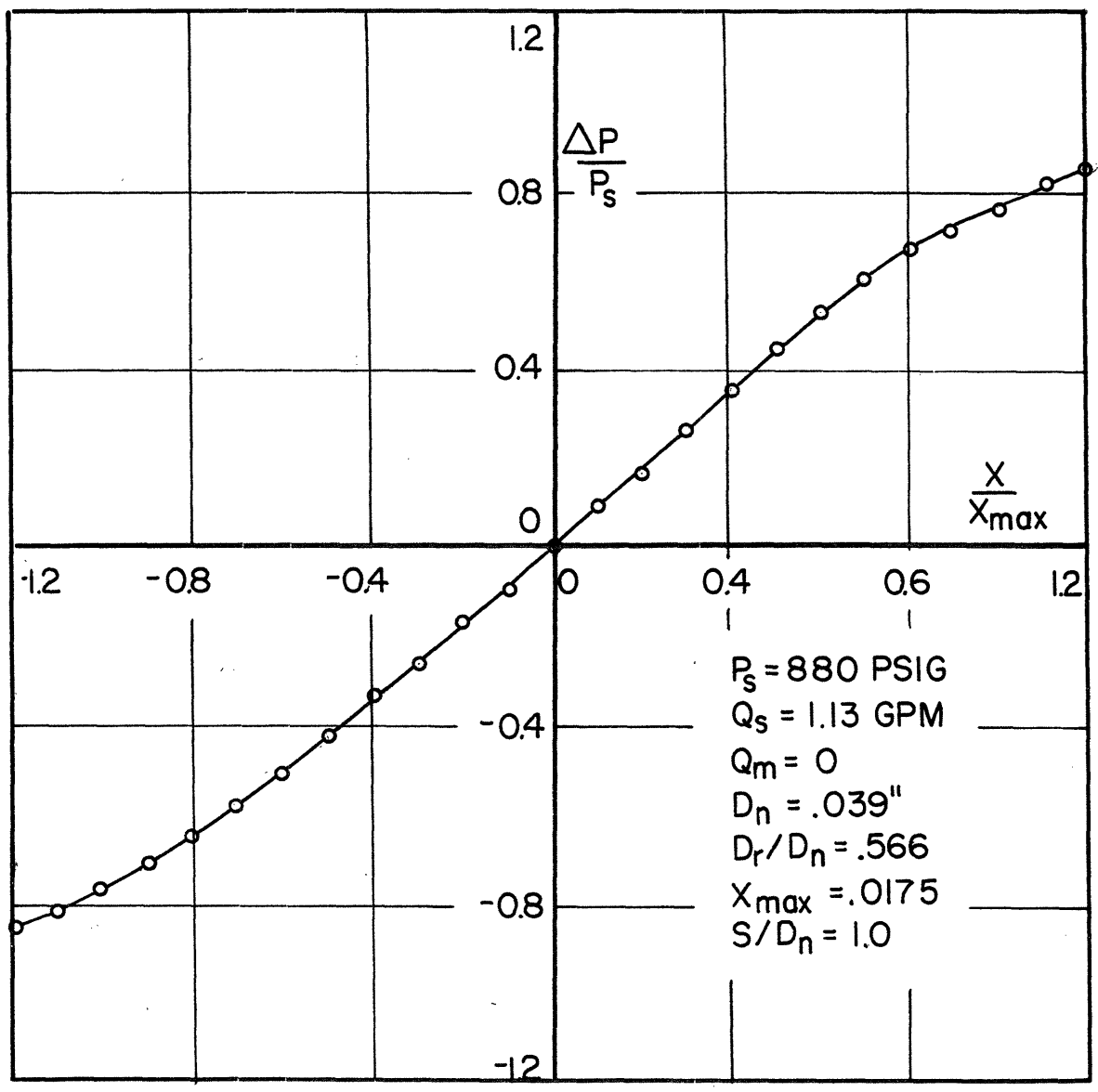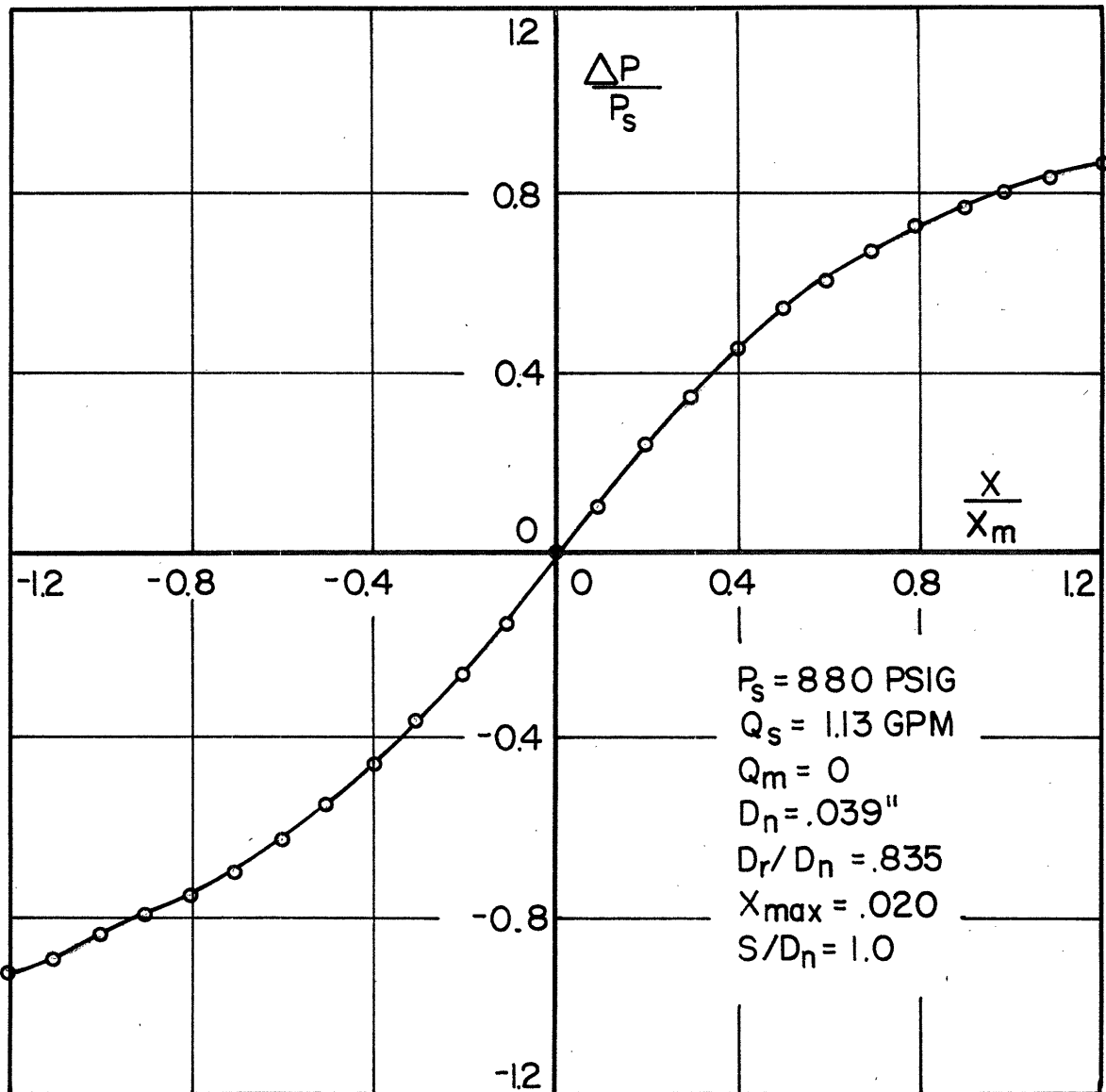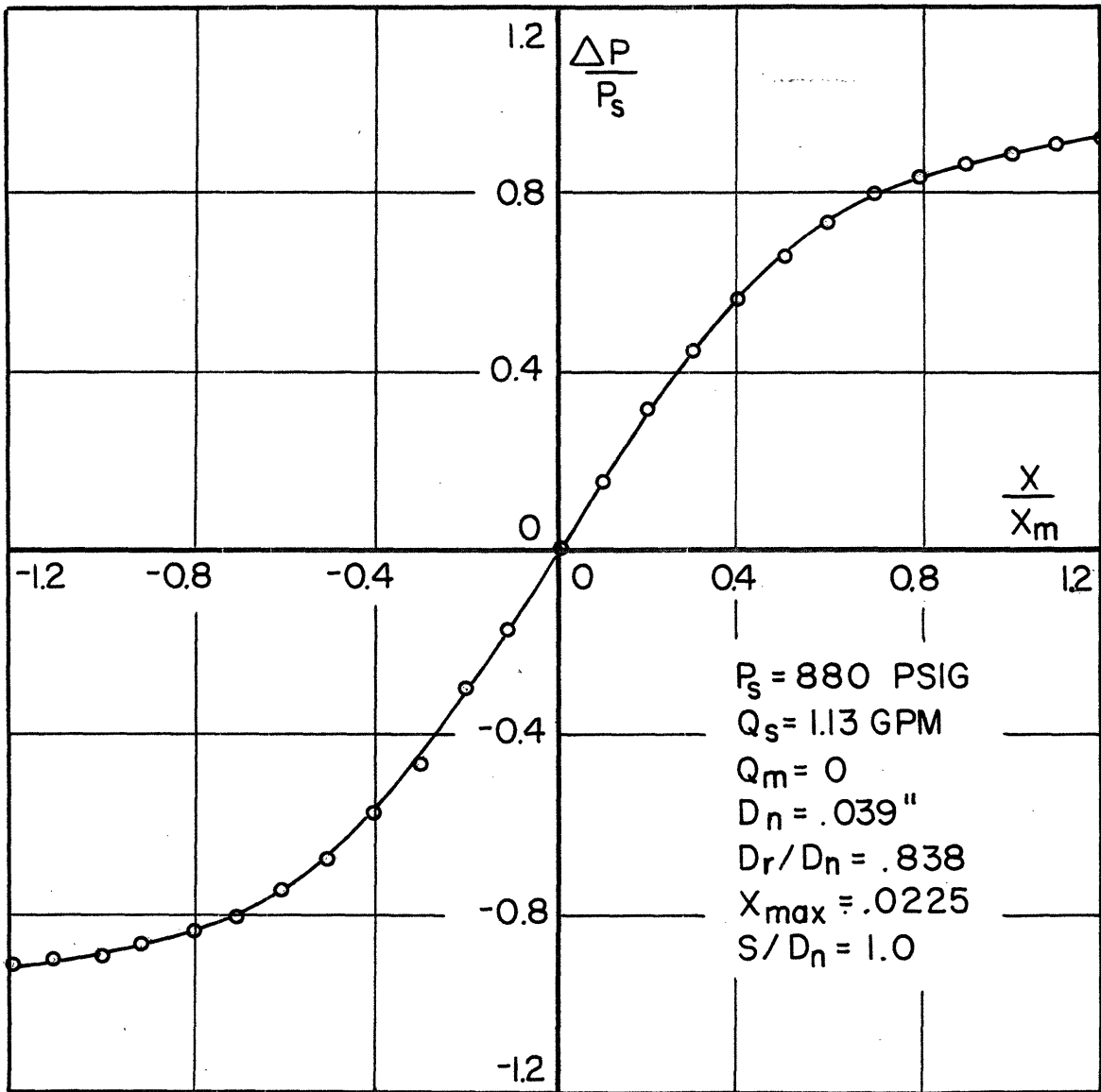
FIGURE 10

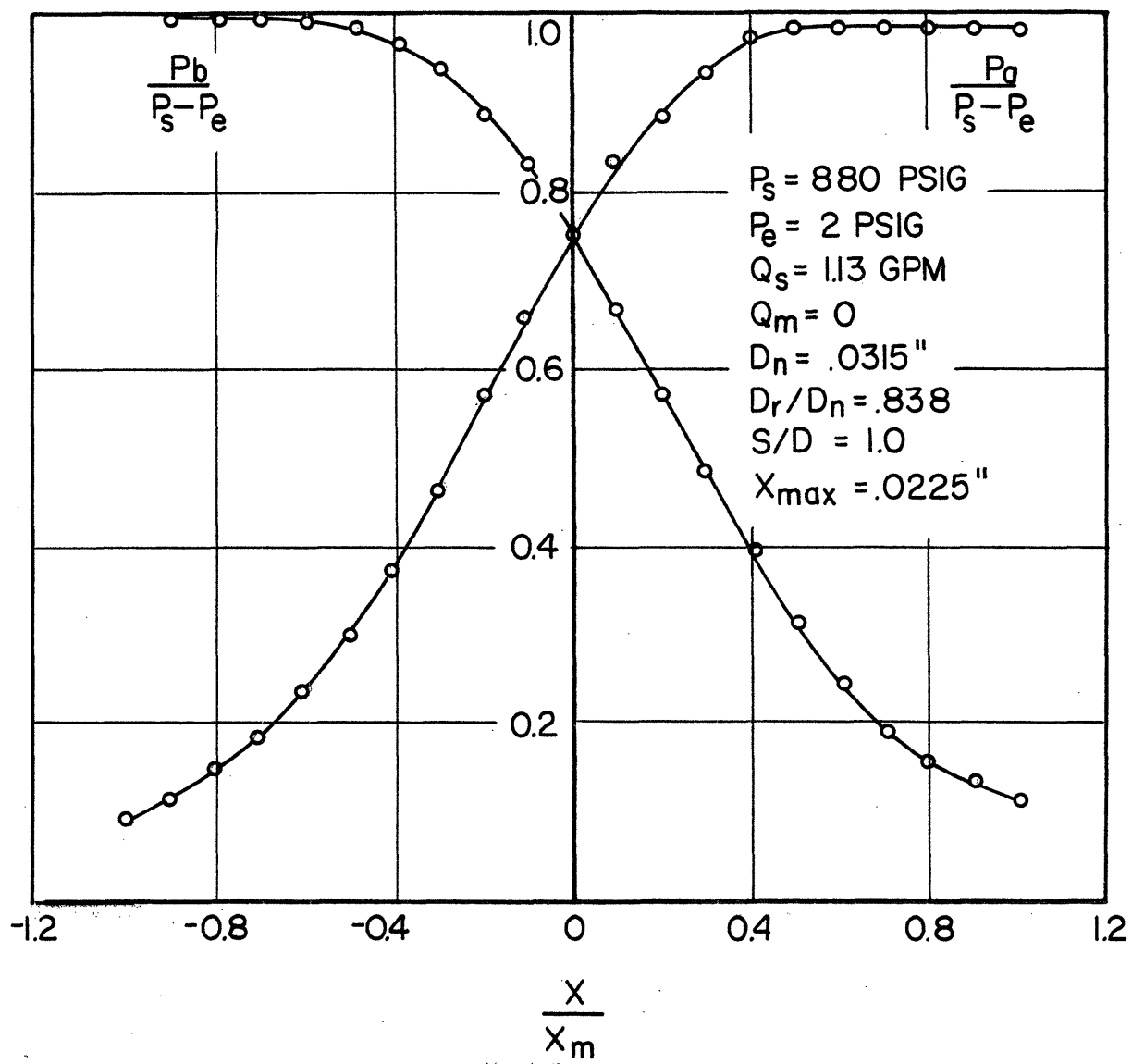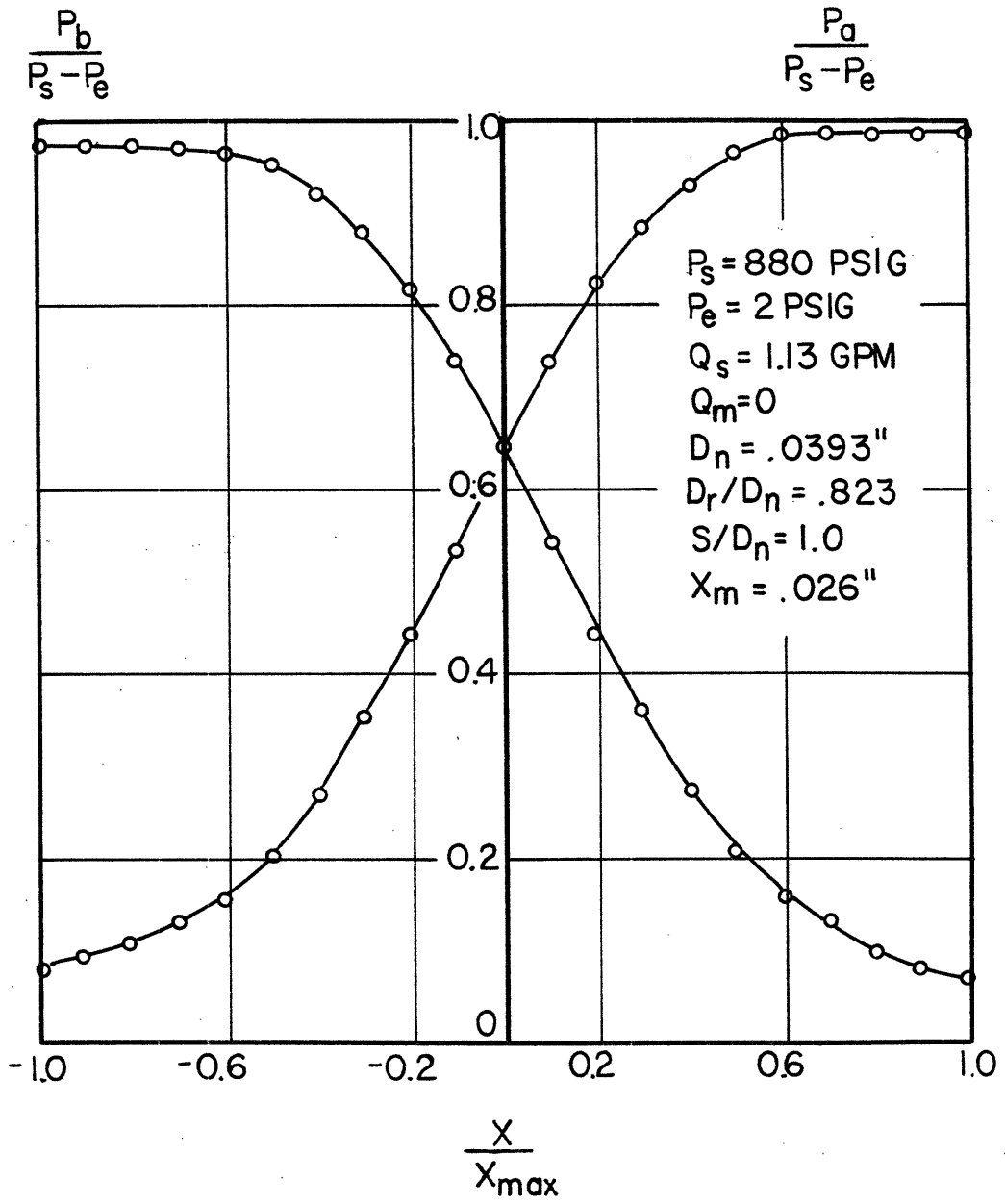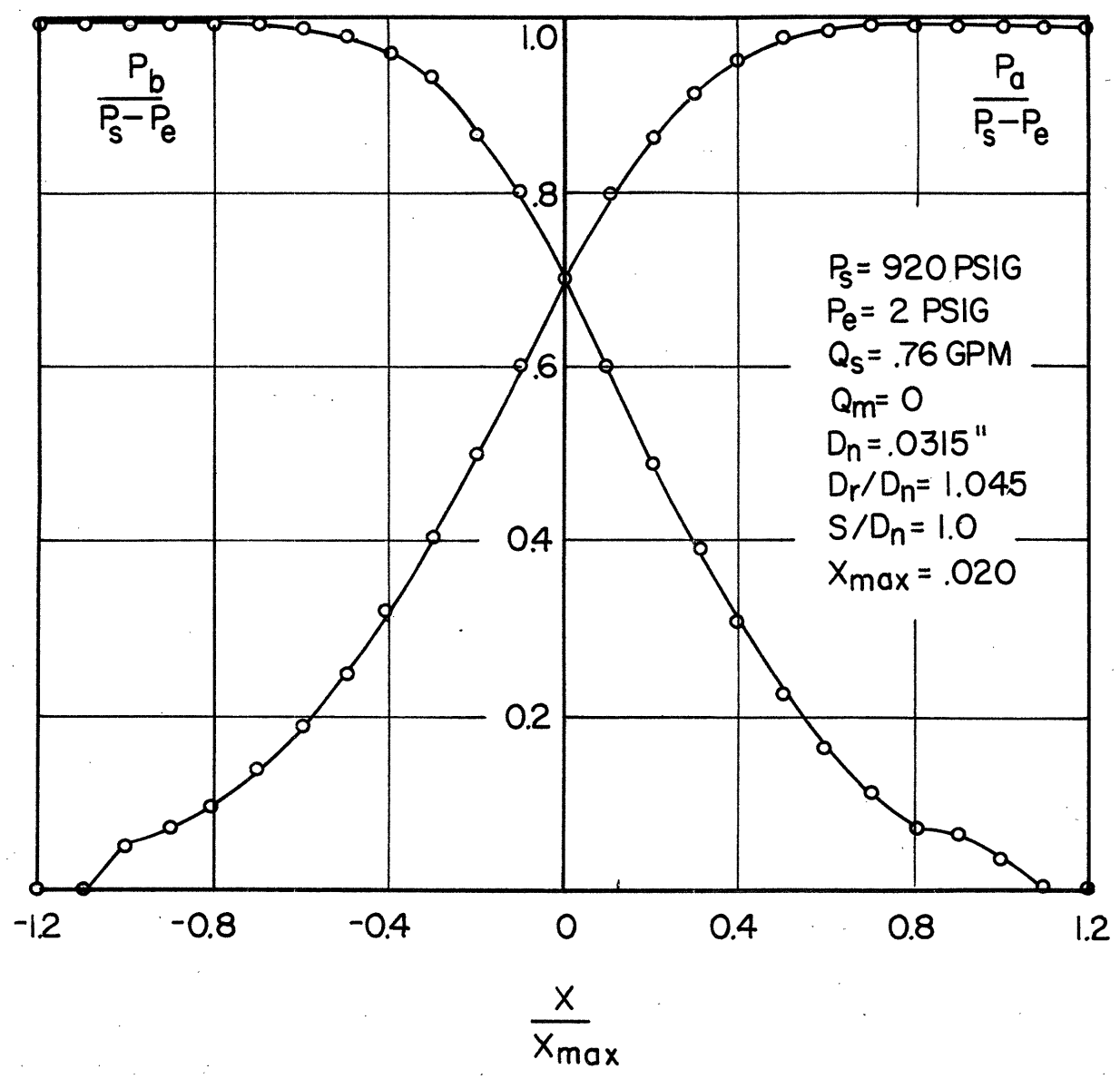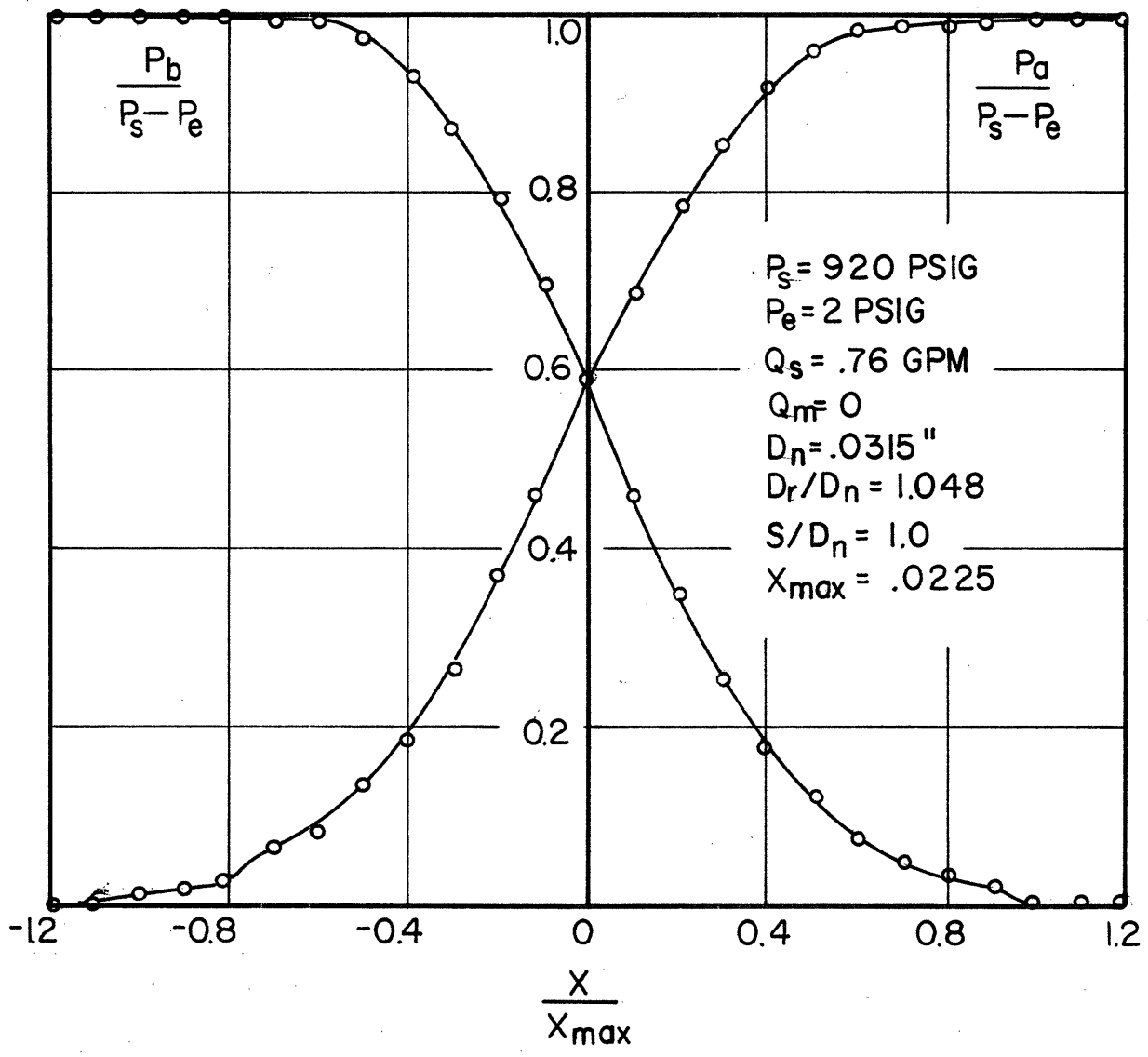FIGURE 11

FIGURE 12

FIGURE 13

$P_s = 880$ PSIG
$Q_s = 1.13$ GPM
$Q_m = 0$
$D_n = .039''$
$D_r / D_n = 1.208$
$X_{max} = .034''$
$S / D_n = 1.0$

FIGURE 14

P_s = 880 PSIG
Q_s = 1.13 GPM
Q_m = 0
D_n = .039"
D_r/D_n = 1.63
X_{max} = .043
S/D = 1.0
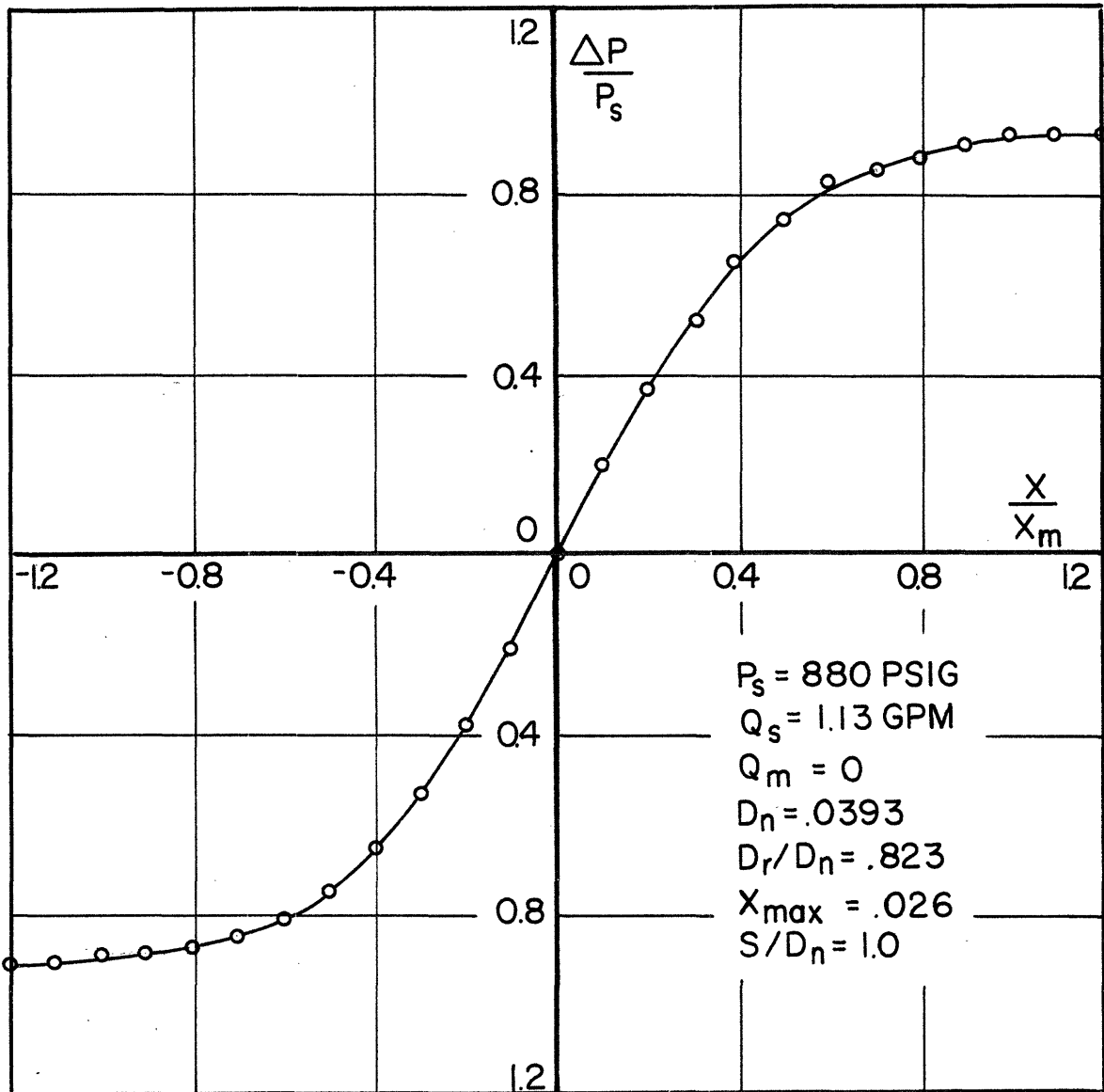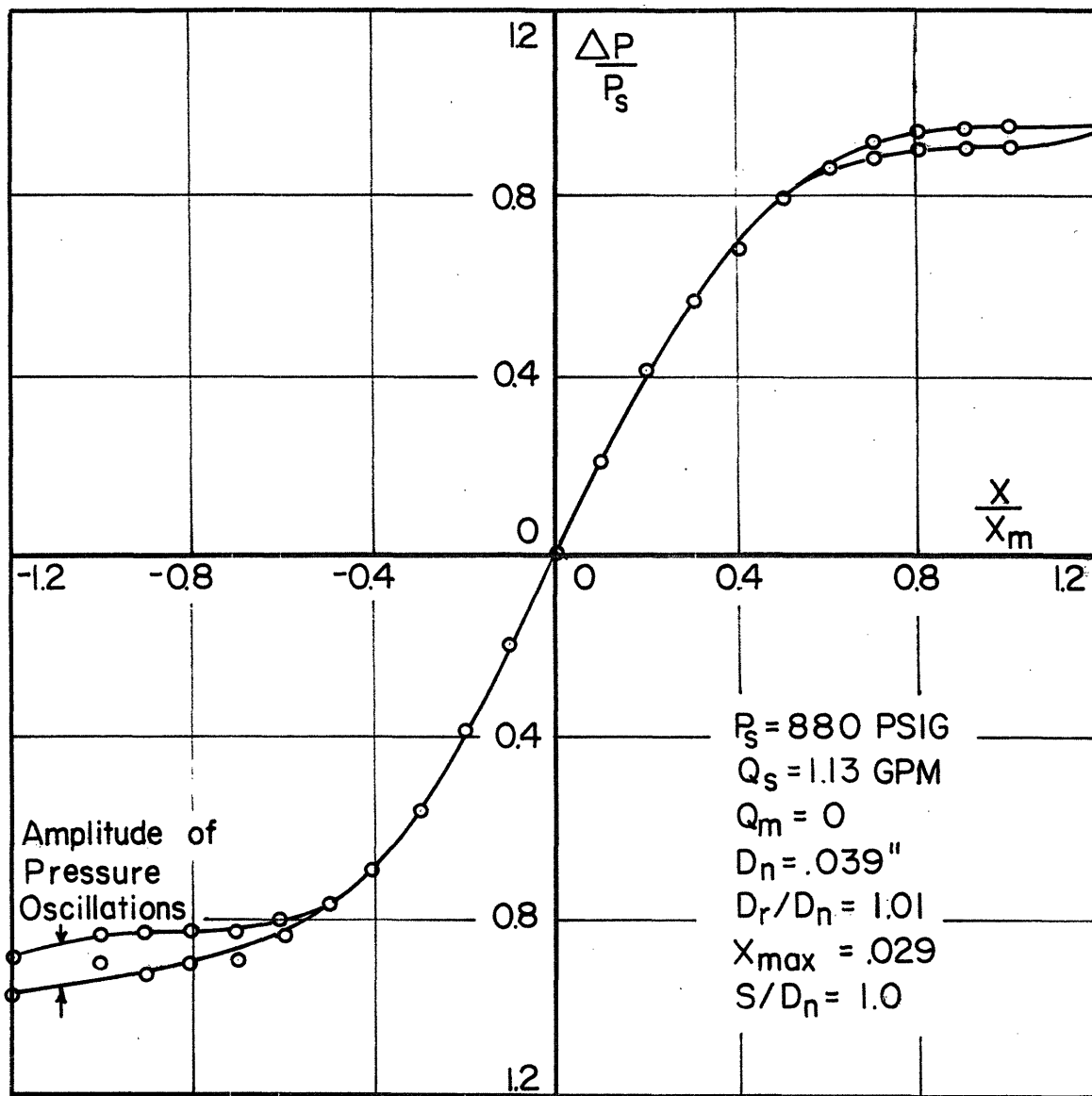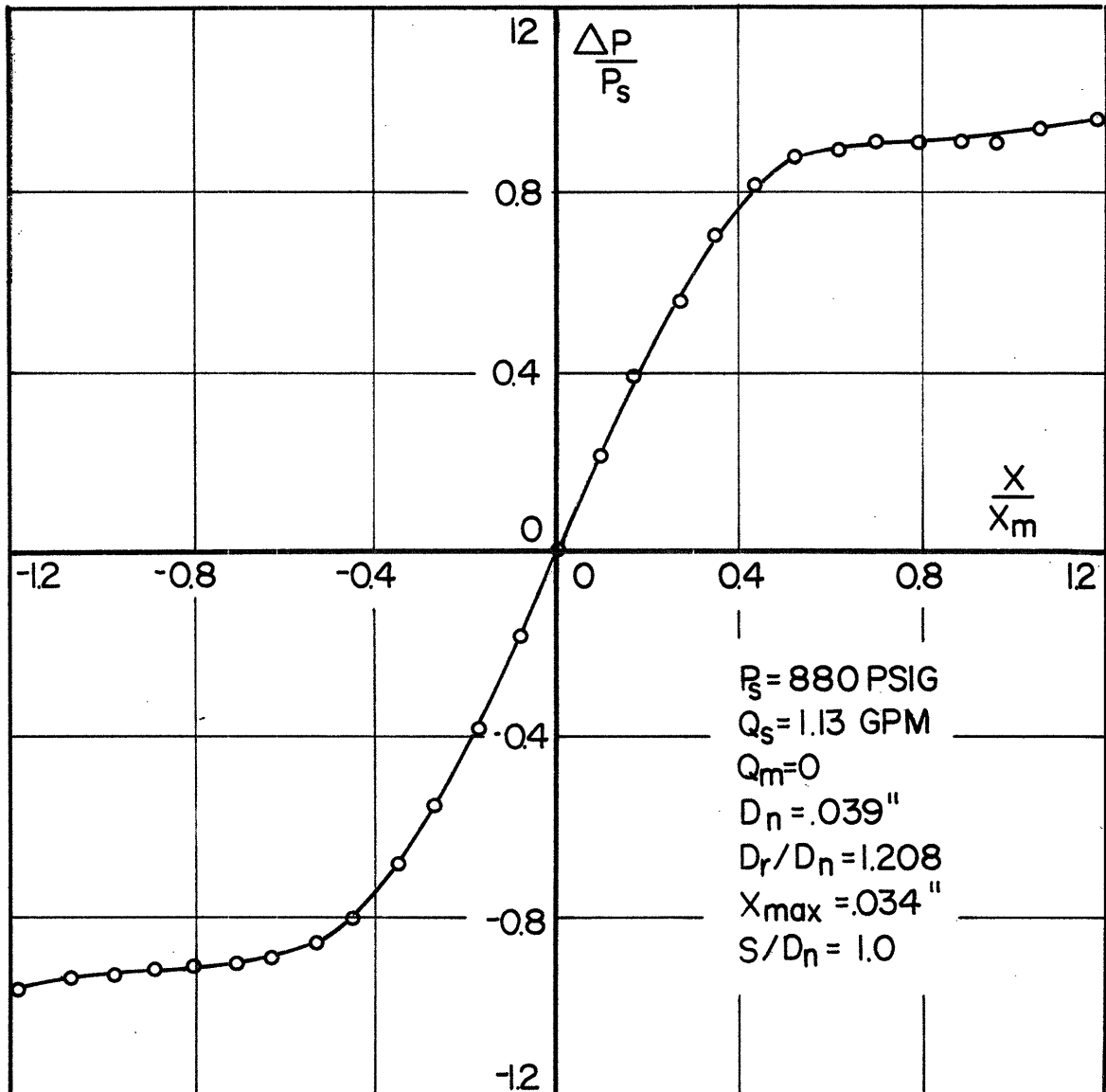
FIGURE 15

FIGURE 16

FIGURE 17

$$\frac{P_b}{P_s - P_e}$$

$$\frac{P_a}{P_s - P_e}$$

$P_s = 880 \text{ PSIG}$

$P_e = 2 \text{ PSIG}$

$Q_s = 1.13 \text{ GPM}$

$Q_m = 0$

$D_n = .0393''$

$D_n/D_r = .835$

$S/D_n = 1.0$

$X_m = .020''$

$$\frac{X}{X_{max}}$$

FIGURE 18

$$\frac{P_b}{P_s - P_e}$$

$$\frac{P_a}{P_s - P_e}$$

$P_s = 880$ PSIG
$P_e = 2$ PSIG
$Q_s = 1.13$ GPM
$Q_m = 0$
$D_n = .0315''$
$D_r/D_n = .838$
$S/D = 1.0$
$X_{max} = .0225''$

$$\frac{X}{X_m}$$

FIGURE 19

FIGURE 20

$$\frac{P_b}{P_s - P_e} \qquad \frac{P_a}{P_s - P_e}$$

$P_s = 920$ PSIG
$P_e = 2$ PSIG
$Q_s = .76$ GPM
$Q_m = 0$
$D_n = .0315"$
$D_r/D_n = 1.045$
$S/D_n = 1.0$
$X_{max} = .020$

$$\frac{X}{X_{max}}$$

FIGURE 21

$$\frac{P_b}{P_s - P_e}$$

$$\frac{P_a}{P_s - P_e}$$

$P_s = 920$ PSIG
$P_e = 2$ PSIG
$Q_s = .76$ GPM
$Q_m = 0$
$D_n = .0315''$
$D_r/D_n = 1.048$
$S/D_n = 1.0$
$X_{max} = .0225$

$$\frac{X}{X_{max}}$$

FIGURE 22

$$\frac{P_b}{P_s-P_e} \qquad \frac{P_a}{P_s-P_e}$$

$P_s = 920 \ PSIG$

$P_e = 2 \ PSIG$

$Q_s = .76 \ GPM$

$Q_m = 0$

$D_n = .0315$

$D_r/D_n = 1.03$

$S/D_n = 1.0$

$X_{max} = .026"$

FIGURE 23 $\quad \dfrac{X}{X_{max}}$

$P_s = 880$ PSIG
$Q_s = 1.13$ GPM
$D_n = .0393''$
$D_r/D_n = .566$
$S/D_n = 1.0$
$X/X_{max} = .2, .4, .6, .8, 1.0$
$X_{max} = .0175''$
$P_e = 2$ PSIG
$C_d = 0.55$

$\dfrac{Q_m}{Q_s}$

$\dfrac{\triangle P_m}{P_s - P_e}$

FIGURE 24

FIGURE 25

$P_s = 880$ PSIG
$Q_s = 1.13$ GPM
$D_n = .0393''$
$D_r/D_n = .835$
$X/X_m = .2, .4, .6, .8, 1.0$
$X_m = .020''$
$P = 2$ PSIG
$S/D_n = 1.0$
$C_d = 0.56$

FIGURE 26

FIGURE 27

FIGURE 28

$$P_s = 880 \text{ PSIG}$$
$$Q_s = 1.13 \text{ GPM}$$
$$D_n = .0393"$$
$$D_r/D_n = 1.01$$
$$S/D_n = 1.0$$
$$X_{max} = .029"$$
$$P_e = 2 \text{ PSIG}$$
$$C_d = 0.54$$

$X/X_m = 1.0$

0.8

0.6

0.4

0.2

$\dfrac{Q_m}{Q_s}$

$\dfrac{\triangle P_m}{P_s - P_e}$

FIGURE 29

FIGURE 30

The graph shows $\frac{Q_m}{Q_s}$ on the vertical axis (ranging from 0 to 1.0) versus $\frac{\Delta P_m}{P_s - P_e}$ on the horizontal axis (ranging from 0 to 1.0).

Curves labeled:
- $X/X_m = 1.0$
- 0.8
- 0.6
- 0.4
- 0.2

Parameters:
$$P_s = 920 \ \text{PSIG}$$
$$Q_s = .76 \ \text{GPM}$$
$$D_n = .0315''$$
$$D_r/D_n = 1.048$$
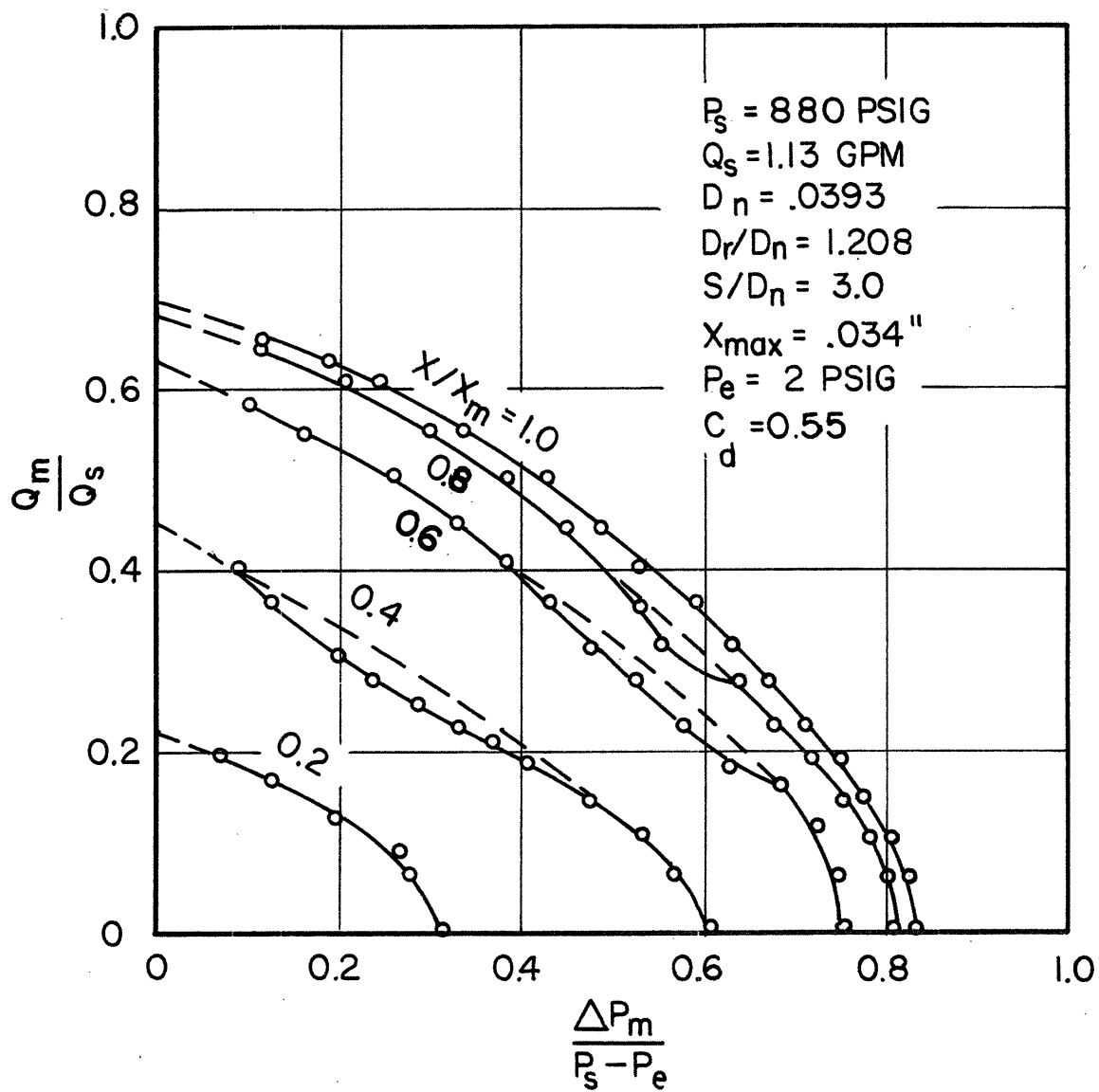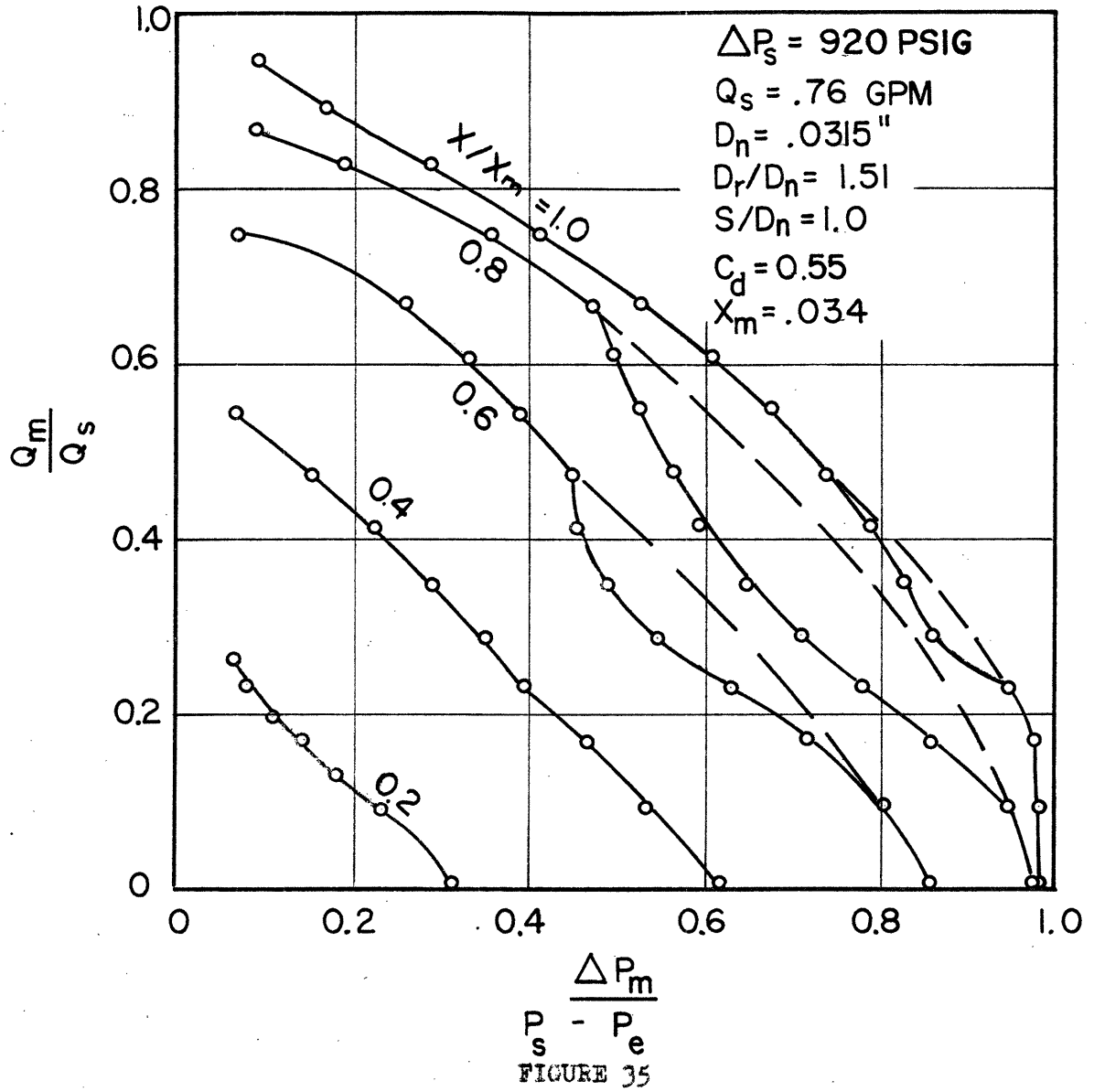$$S/D_n = 1.0$$
$$X_{max} = .0225$$
$$P_e = 2 \ \text{PSIG}$$
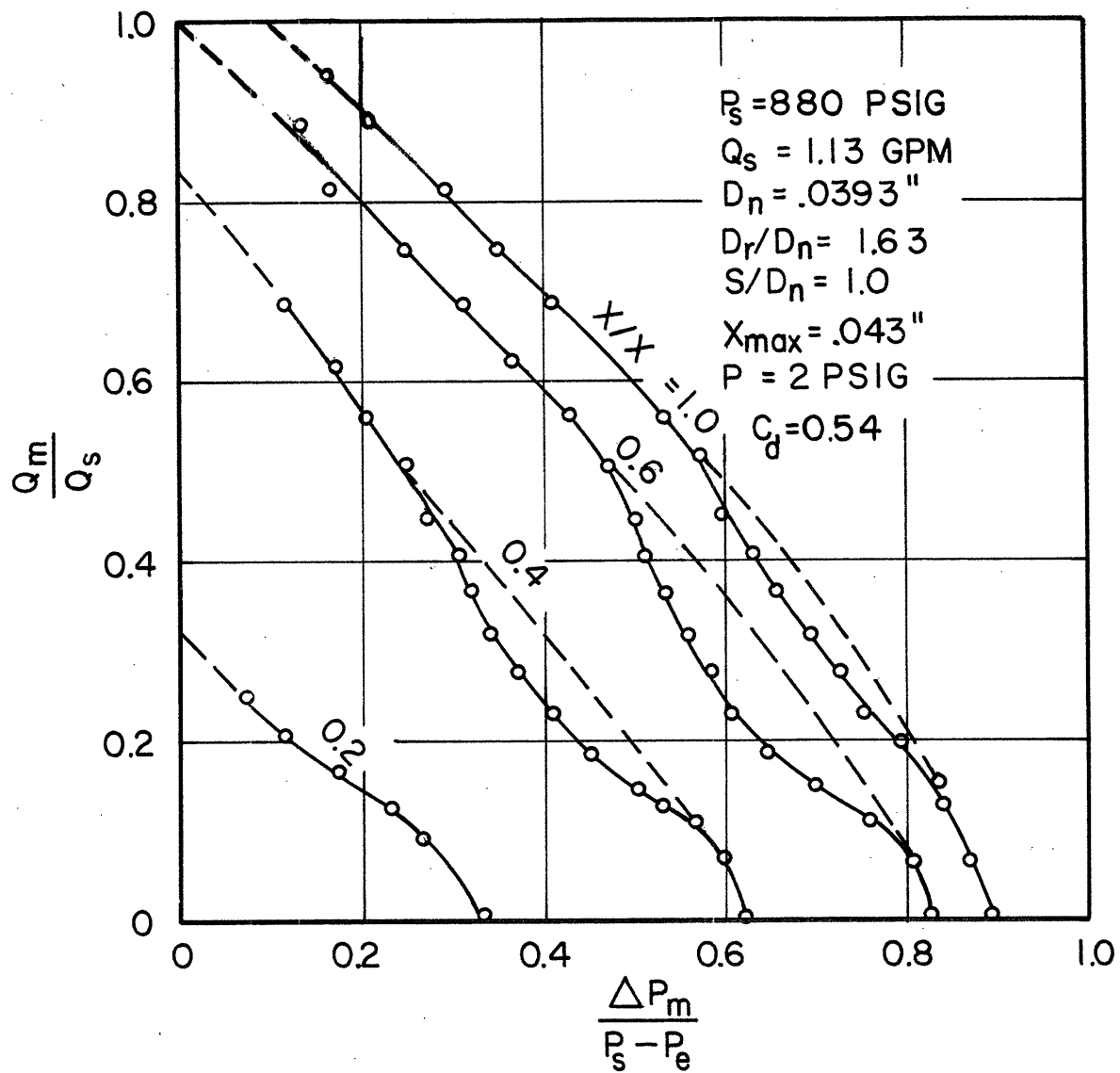$$C_d = 0.55$$
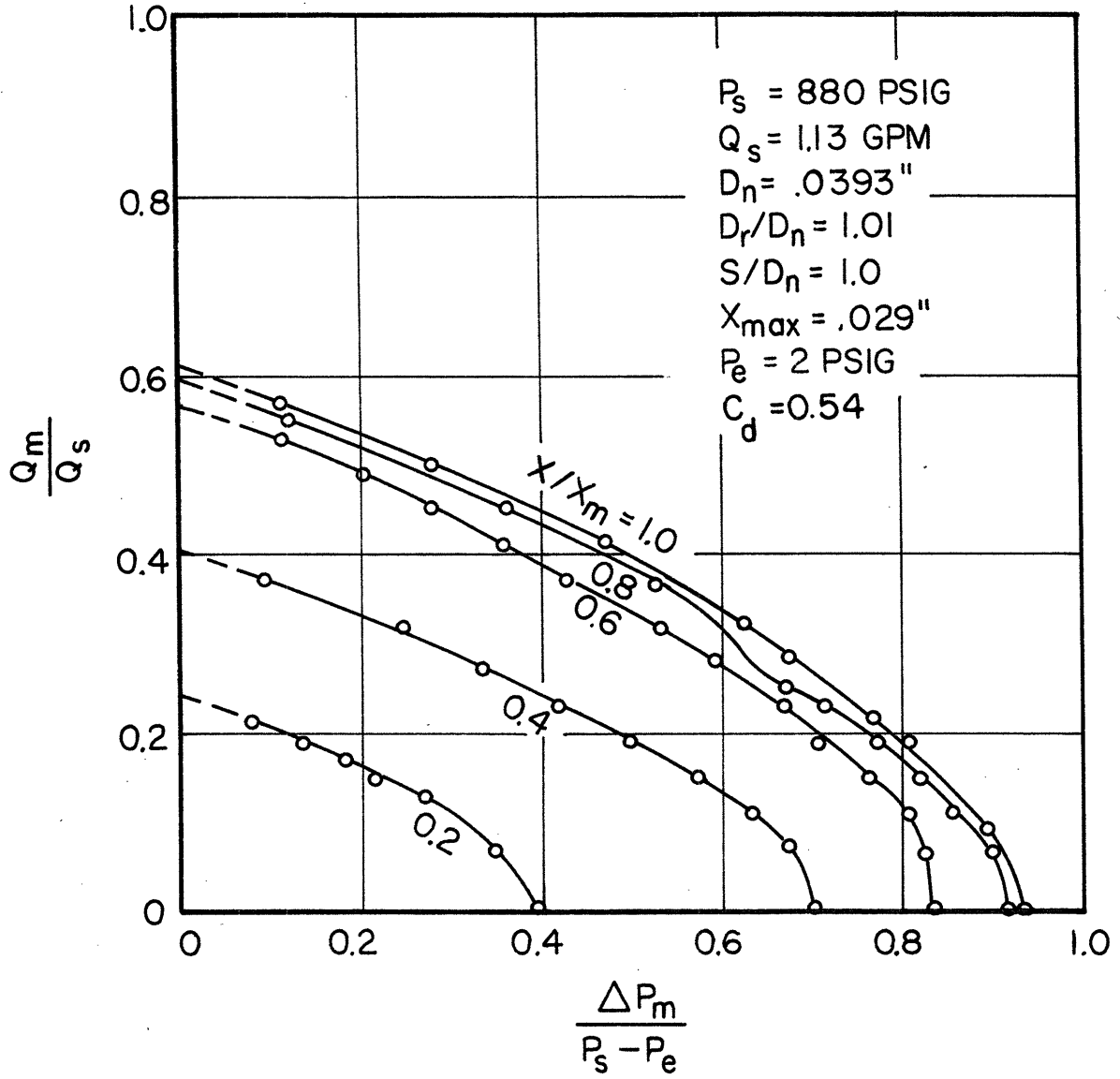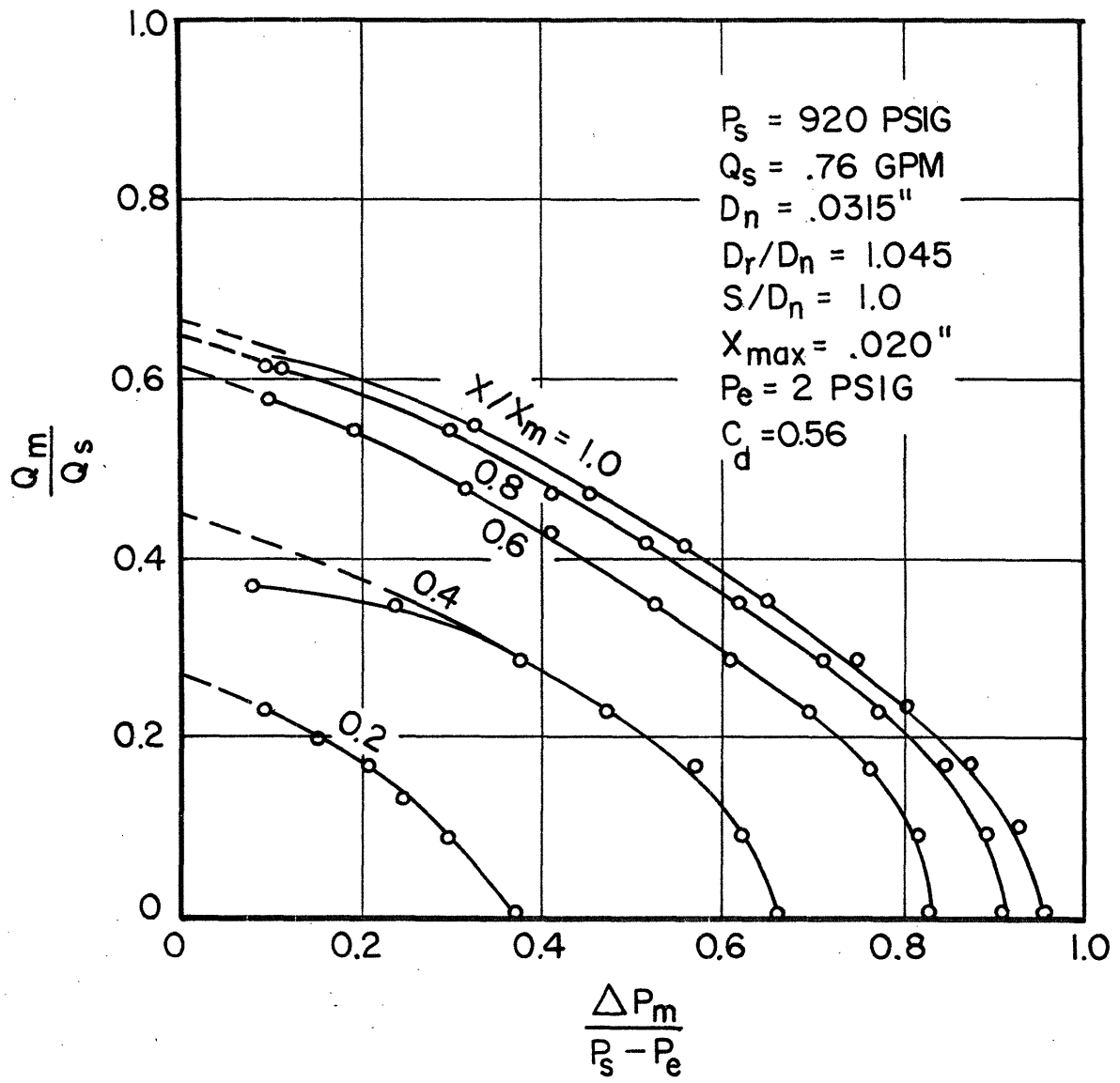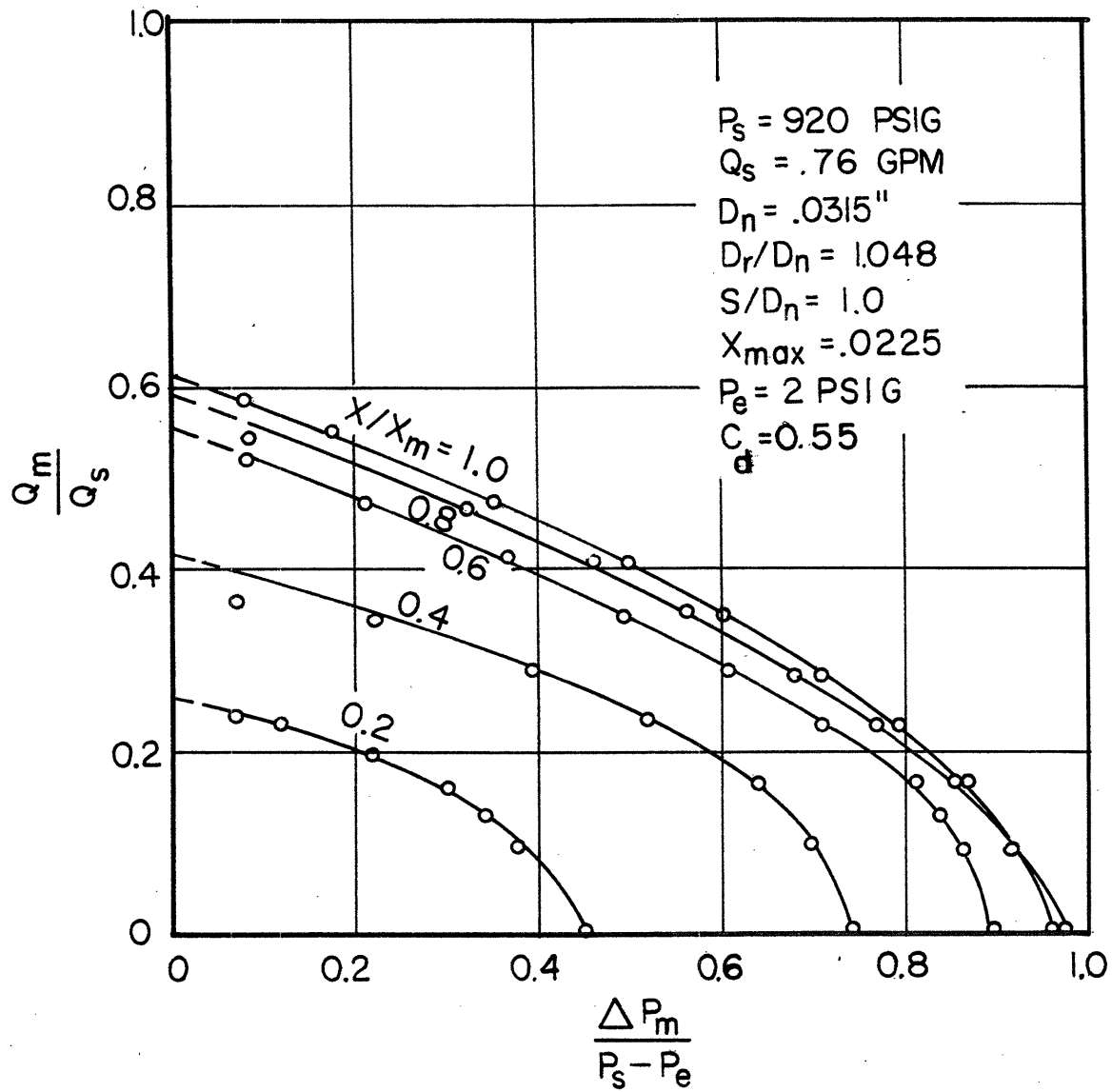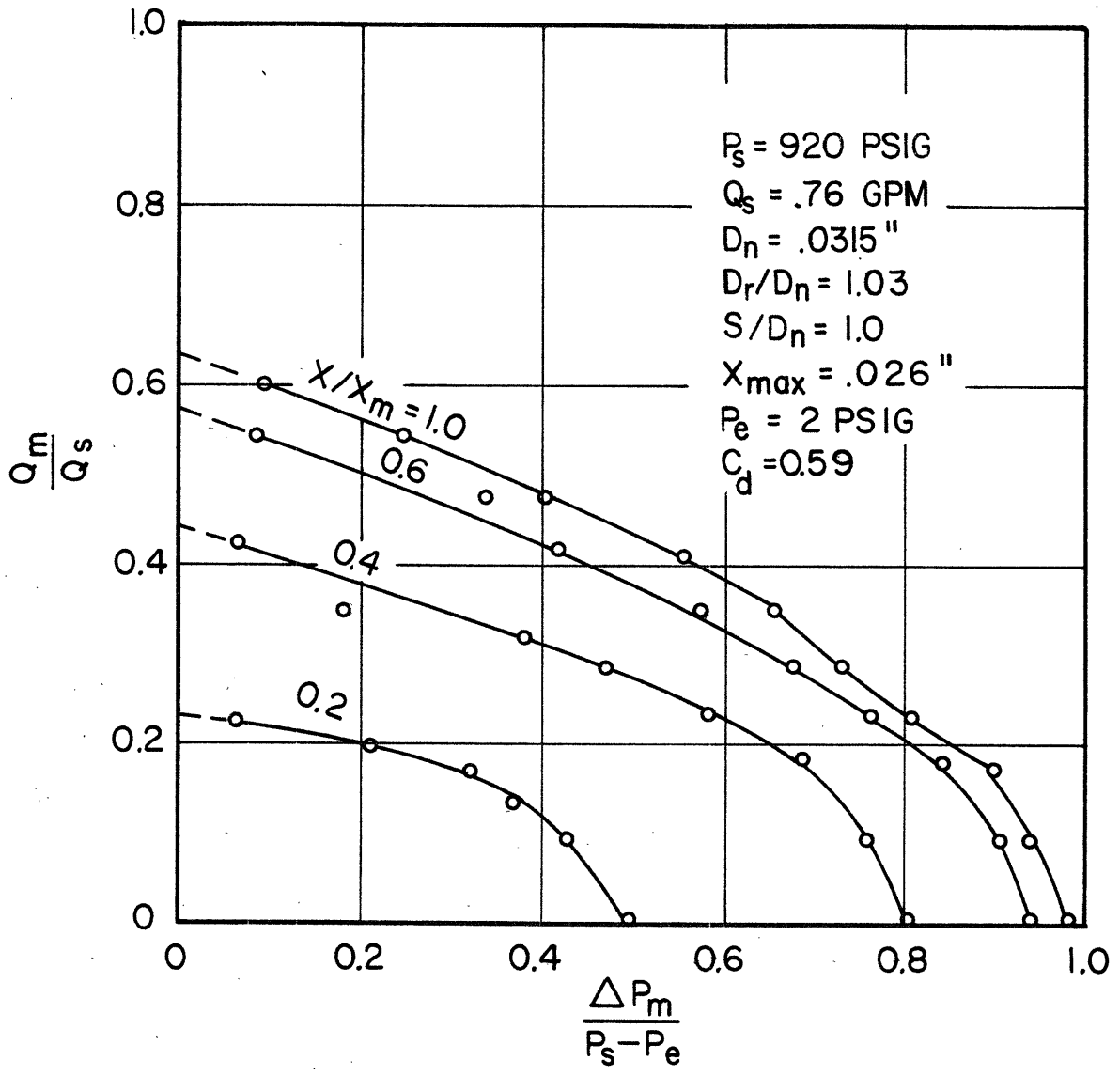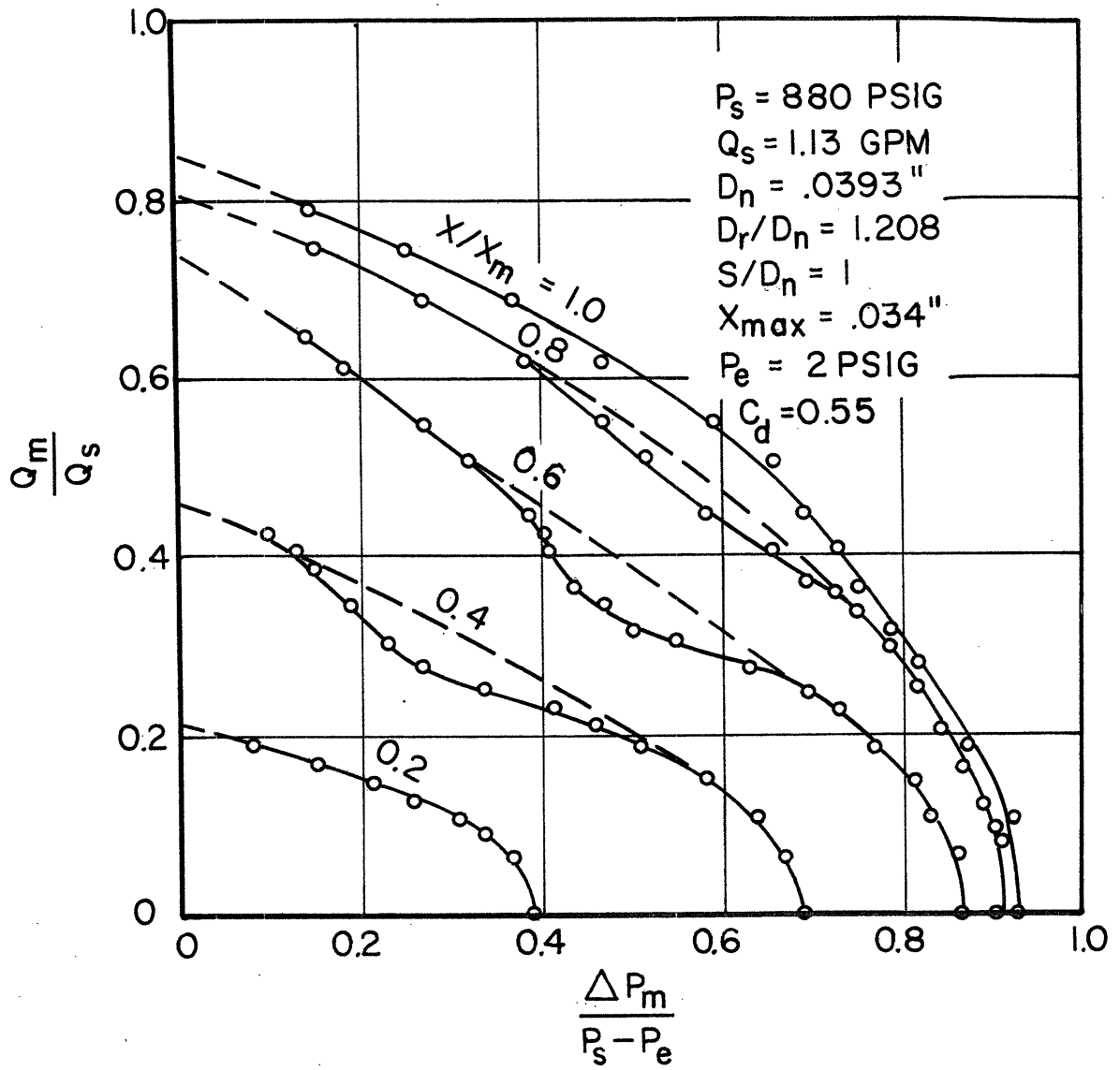
FIGURE 31

FIGURE 32

FIGURE 33.

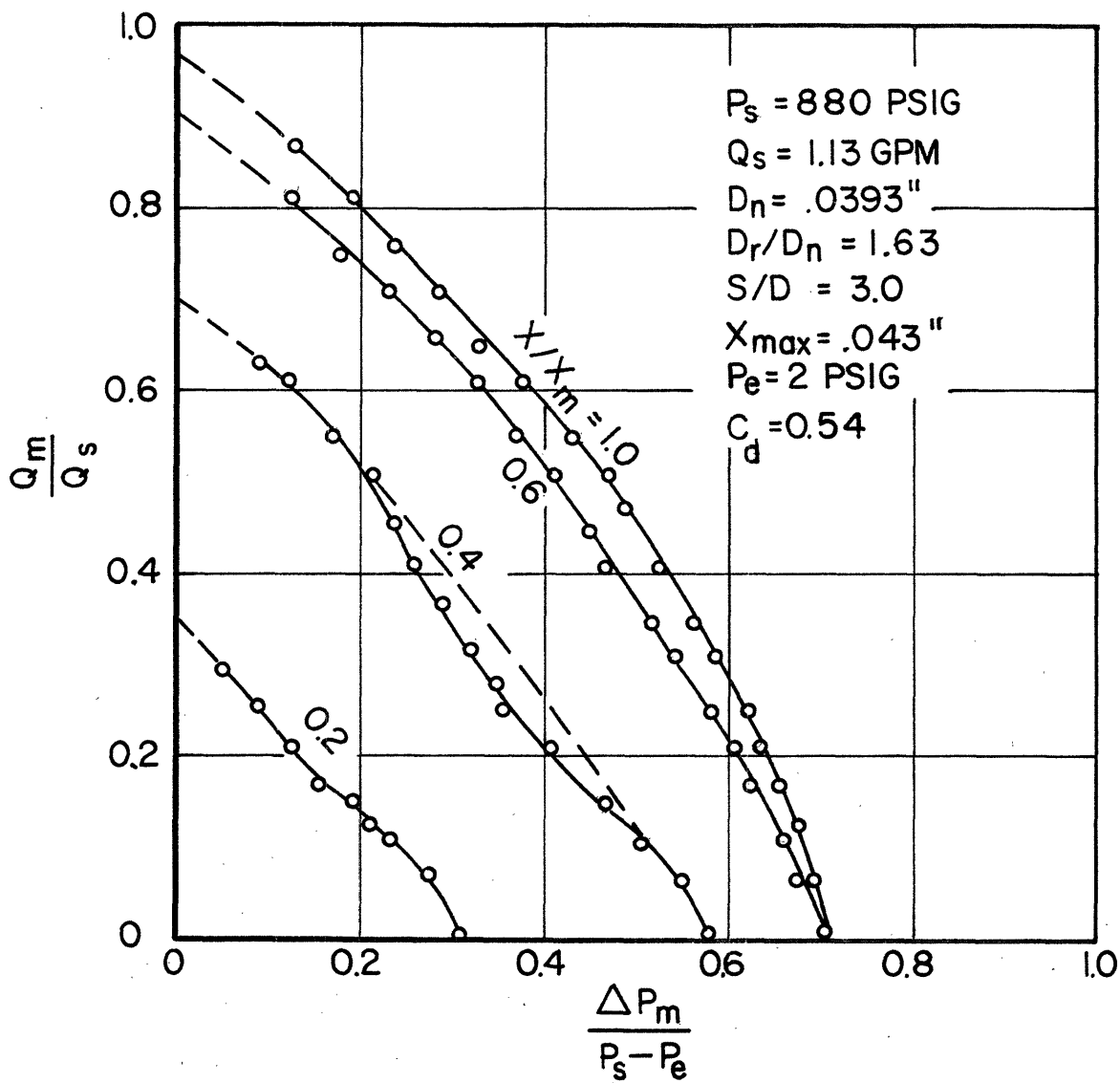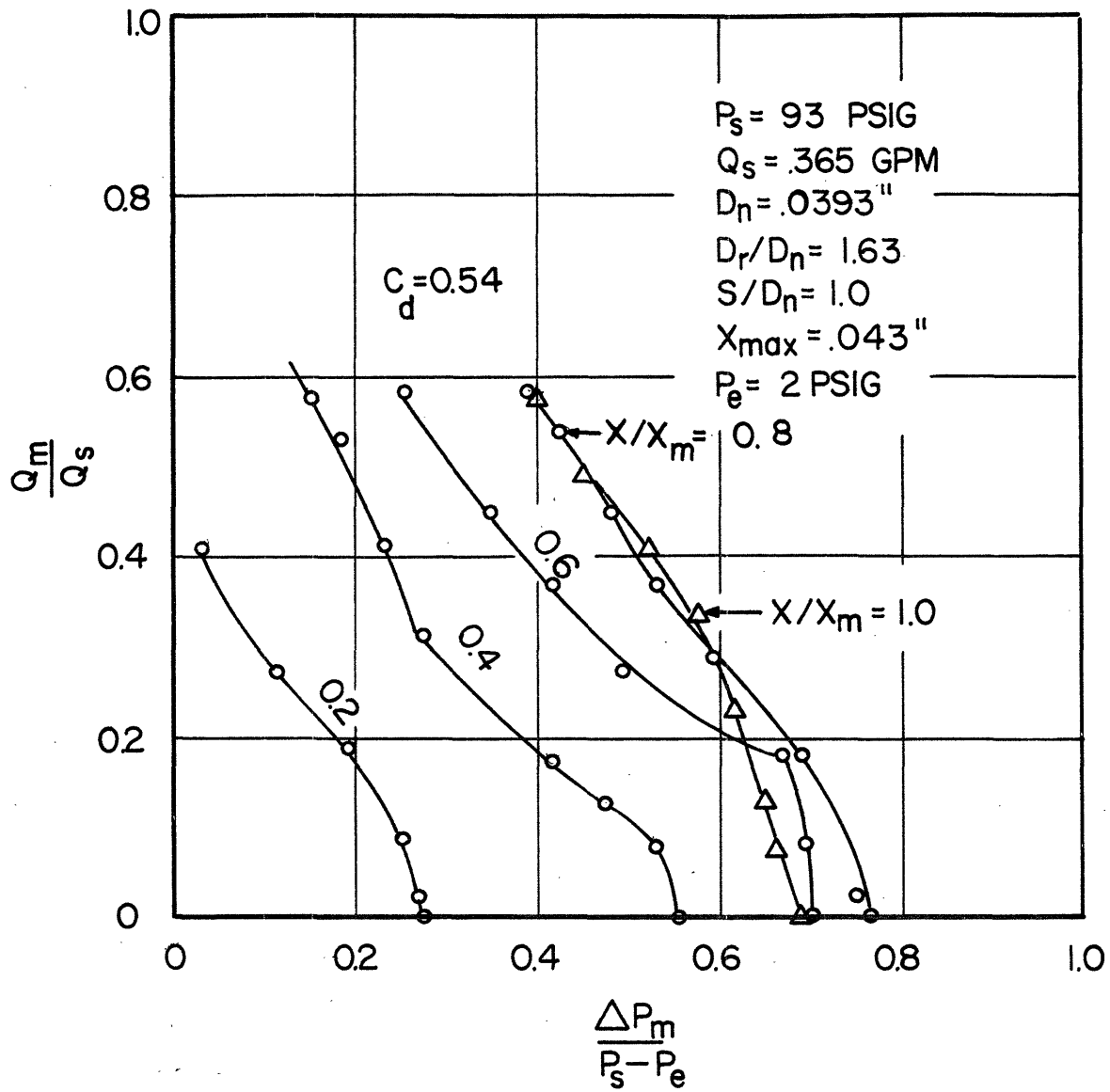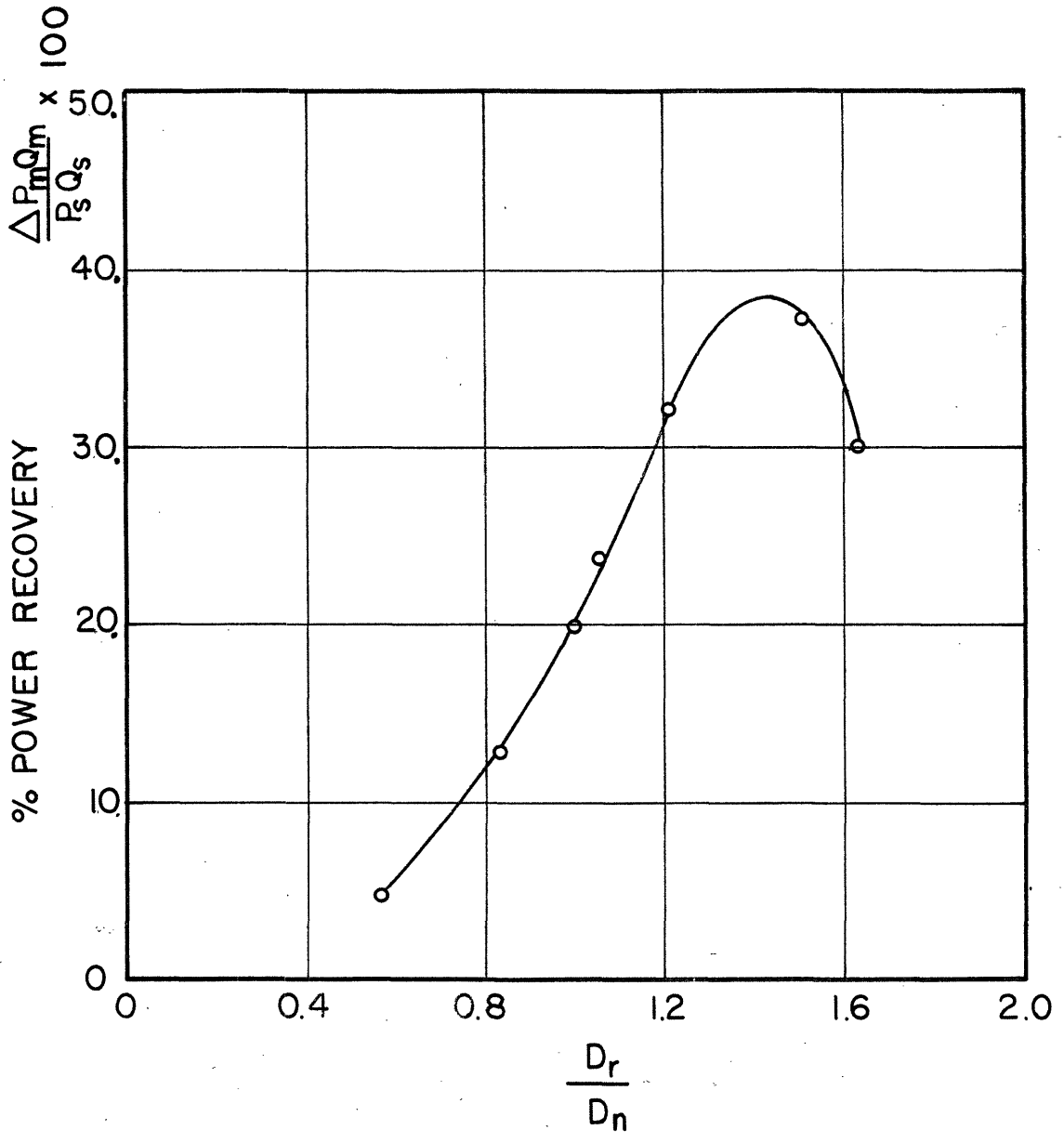FIGURE 34

FIGURE 35

FIGURE 36

FIGURE 37

FIGURE 38

FIGURE 39

$$\frac{Q_m \, \Delta P_m}{Q_s(P_s-P_e)} \times 100$$

% POWER TRANSFER

$\frac{D_r}{D_n} = 1.208$

$\frac{D_r}{D_n} = 1.51$

$\frac{D_r}{D_n} = 1.63 \rightarrow$

$\frac{D_r}{D_n} = 1.01 \rightarrow$

$\frac{D_r}{D_n} = .838 \rightarrow$

$\frac{D_r}{D_n} = .566 \rightarrow$
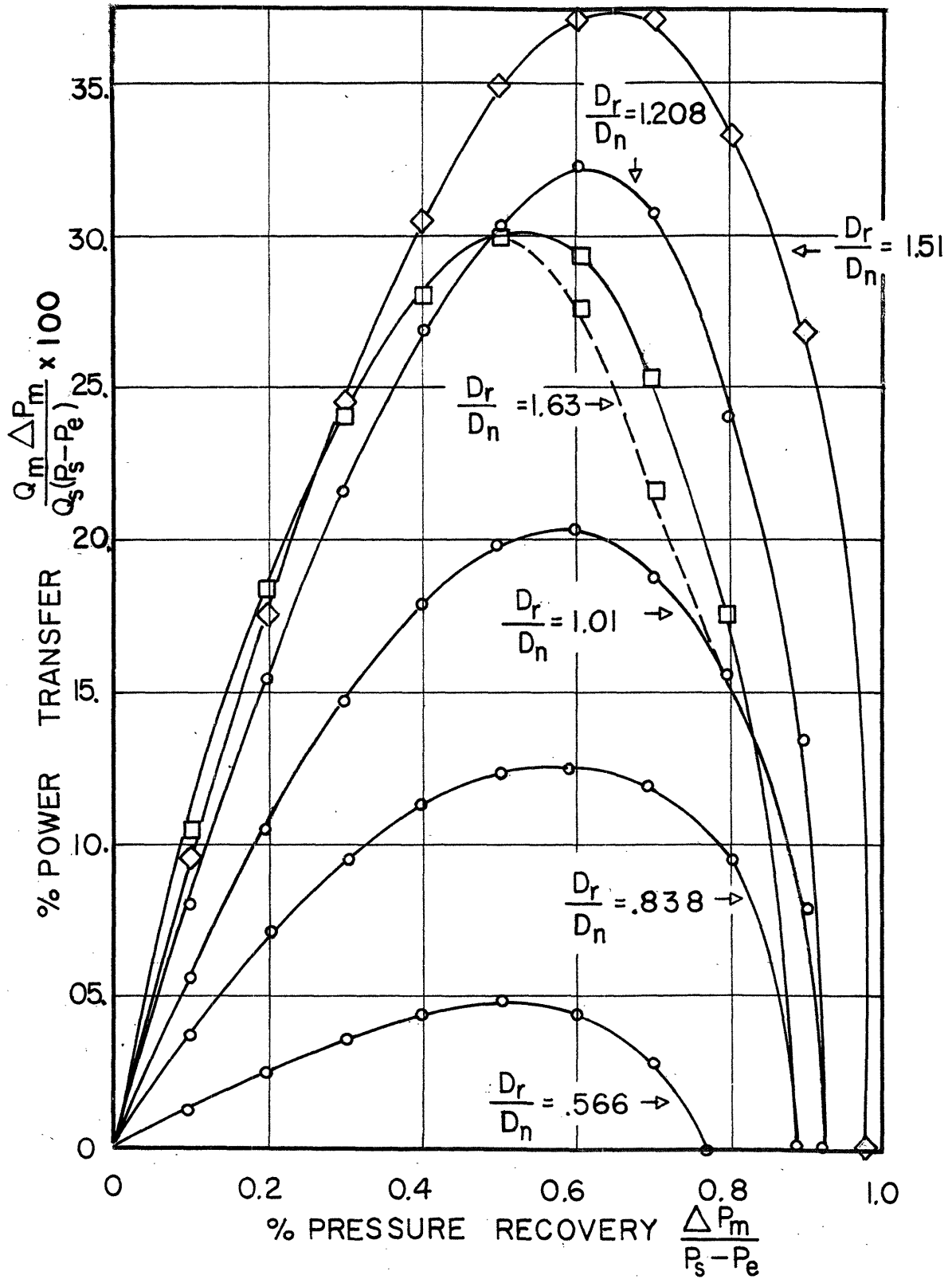
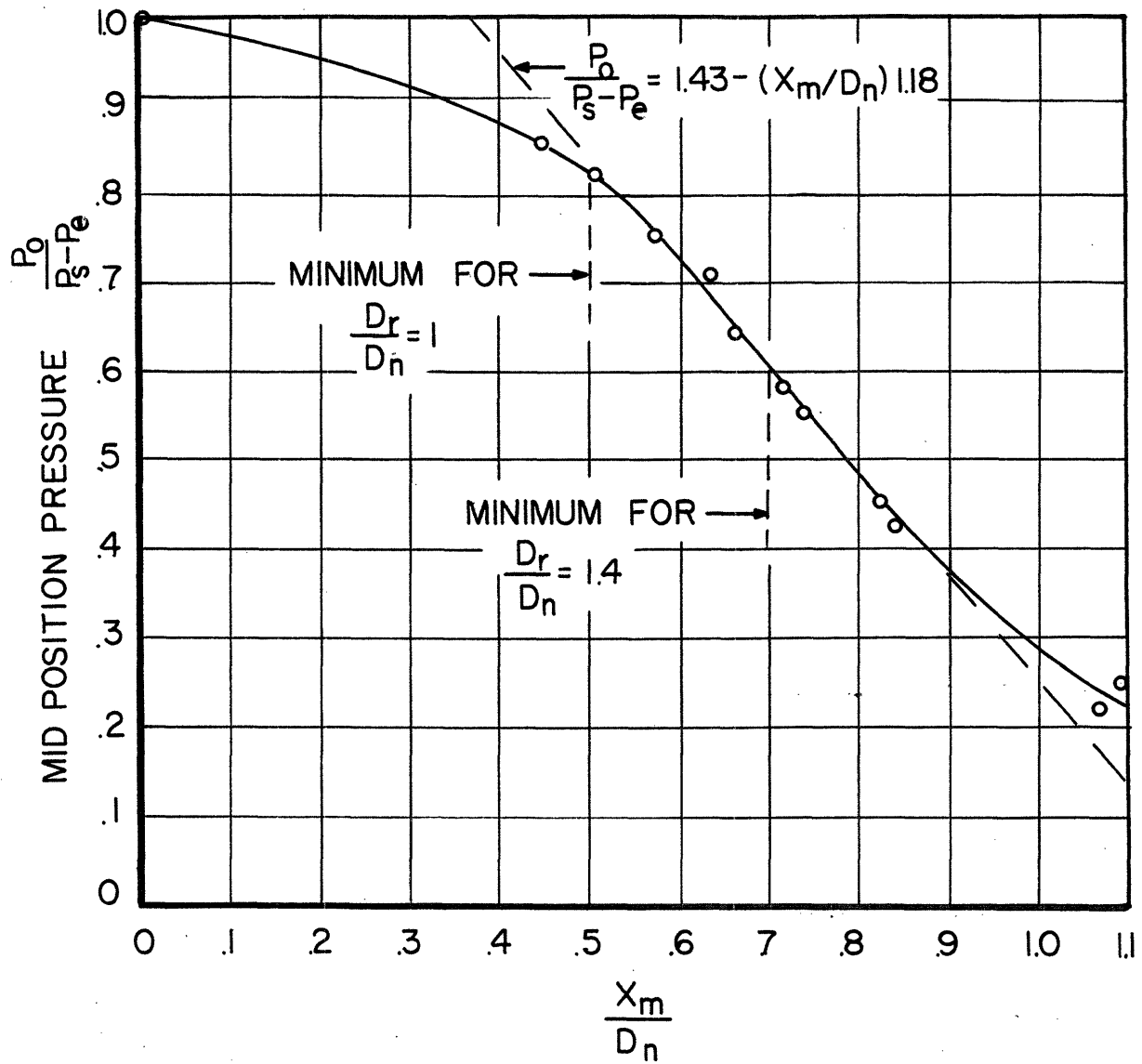% PRESSURE RECOVERY $\frac{\Delta P_m}{P_s-P_e}$
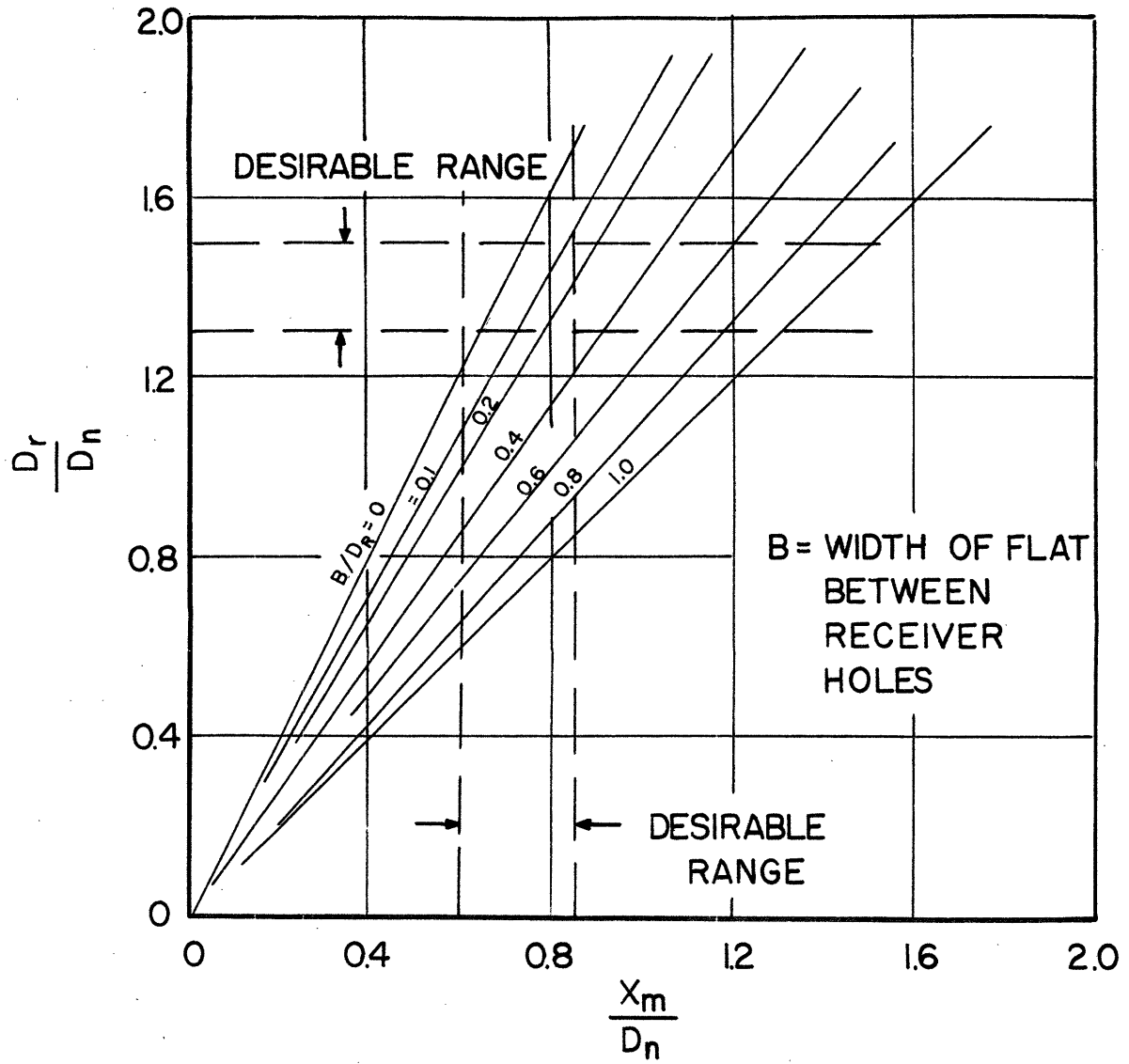
FIGURE 40
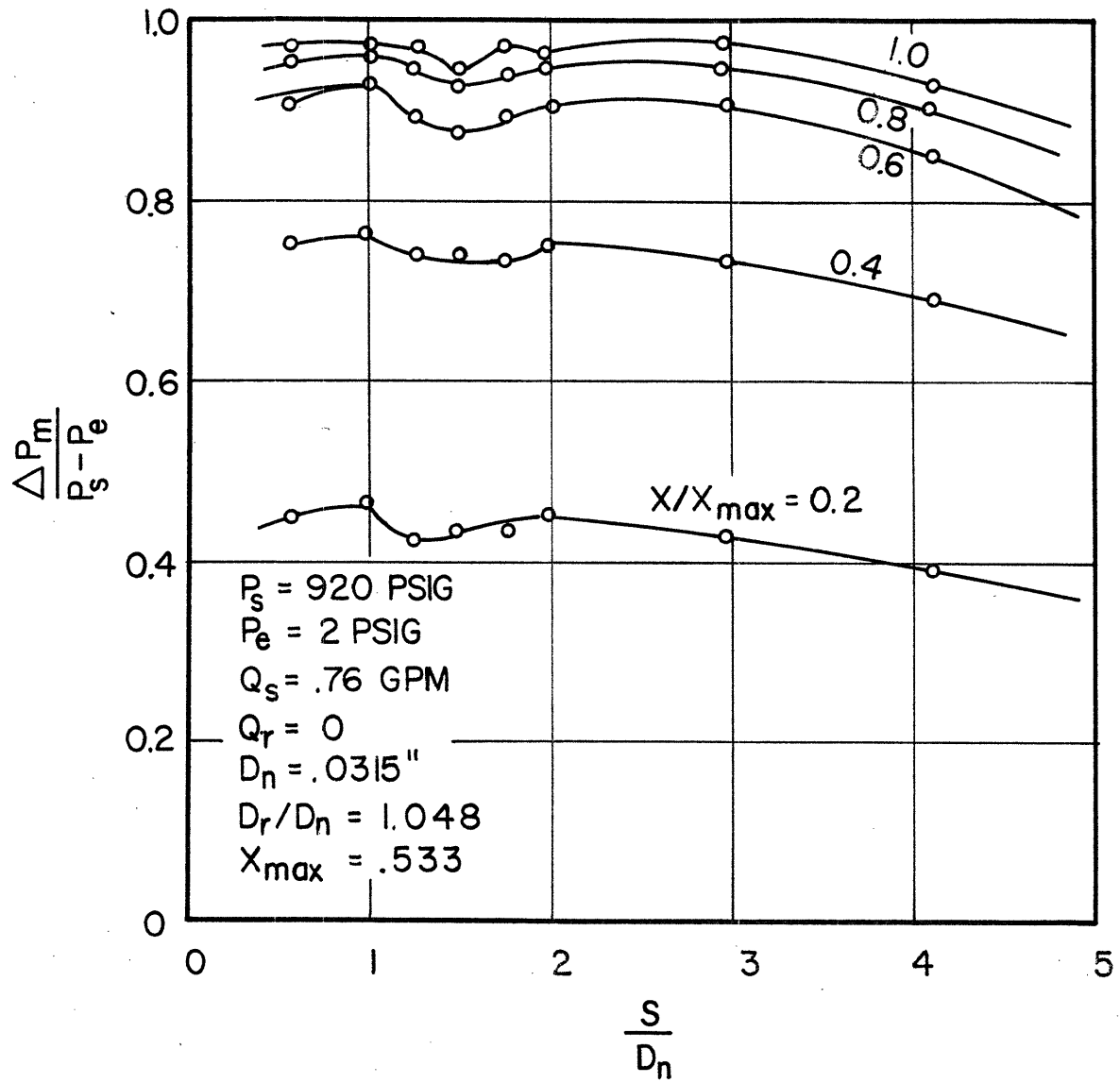
FIGURE 41

FIGURE 42

FIGURE 43

7. CONCLUSIONS

A jet-pipe valve can be designed to give very good performance. This requires that two very important factors be met. First, the align-ment between the jet-pipe nozzle and the receiver holes must be very care-fully set. Secondly the parasidic internal losses in the valve, i.e., pressure loss in the nozzle and receiver flow passages, must be kept at a minimum.

The optimum values for the non-dimensionalized parameters were found to be $D_R/D_N = 1.4$, $X_{max}/D_N = 0.7$ and $S/D_N = 1.0$. These are not absolute rules but must be examined in the overall outlook of the specific applications.

Fluctuations in the receiver pressure can occur under various op-erating conditions. The magnitude may be such that the valve will be un-usable under these conditions.

The pressure-flow characteristics for the jet-pipe valve have very non-linear dips whenever the value of D_R/D_N is greater than 1.0 and X/X_{max} is between 0.4 and 0.8.

8.   SUGGESTIONS FOR FURTHER WORK

Two big problem areas still exist with the jet-pipe valve. One is the cause of the oscillations in the receiver pressure and the second is the cause of the dips in the pressure-flow curves. A large, well-instrumentated model of the valve that would allow visual study of the fluid flow patterns would be of great use in solving the above problems.

NOMENCLATURE

C_D . . . . Discharge Coefficient

D_n . . . . Jet-pipe Nozzle Diameter

D_r . . . . Receiver Hole Diameter

P_A . . . . Pressure in Receiver (A)

P_B . . . . Pressure in Receiver (B)

P_e . . . . Nozzle Exhaust Pressure

ΔP_M . . . . $P_A - P_B$ = Pressure Drop Across the Load

P_S . . . . Nozzle Supply Pressure

Q_M . . . . Volume Flow Rate Through the Load

Q_S . . . . Nozzle Volume Flow Rate

S . . . . Jet-Pipe Nozzle to Receiver Spacing

X . . . . Jet-Pipe Nozzle Stroke Measured from
the Mid-Position

X_{max} . . . . . . Distance from Mid-Position to the Center
Line of a Receiver Hole

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