An Analytical Method for Determining Effective Flow and Force Areas for Refrigeration Compressor Valving Systems

D. D. Schwerzler
General Motors Research Laboratories

J. F. Hamilton
Purdue University

AN ANALYTICAL METHOD FOR DETERMINING EFFECTIVE FLOW AND FORCE AREAS FOR REFRIGERATION COMPRESSOR VALVING SYSTEMS

Dennis D. Schwerzler, Associate Senior Research Engineer,
Vehicle Research Dept., General Motors Research Laboratories, Warren, Michigan

James F. Hamilton, Professor, Dept. of Mech. Engr.
Ray W. Herrick Laboratories, Purdue University, Lafayette, Indiana

An analytical method was developed for predicting the coefficients necessary for determining the mass flow rate through a refrigeration compressor valving system and the force on the valve. The method was applied to seven different types of valving systems, and the results were compared to experimental data. In general, it was found that the geometry-dependent equations obtained from the analytical method approximated the experimental data reasonably well to warrent their usage as design tools.

INTRODUCTION

Present day research in the refrigeration industry is aimed at developing an analytical model of the refrigeration compressor for purposes of improving its performance by optimization. Most of the models (1,2,3,8) that have been developed require some form of empirical data obtained from the specifically modeled compressor. Among the empirical data needed for these models are the effective flow and force areas which are used to determine the mass flow rates through the valving systems and the forces on the valves, respectively. Since the design of the valving system plays the most critical part in determining the efficiency of the compressor, it is desirable to have some means of modeling this system without the aid of empirical data.

In this work approximate methods for obtaining the effective flow and force area are developed. The method for determining the effective flow area is based on the concept of considering the restrictions in the valving system as simple orifices in series and in parallel. The assumption of incompressible flow theory is used to simplify the resulting equations governing the mass flow rate through the entire valving system. The method for determing the effective force area for calculating the drag force on the valve is highly dependent on the effective flow area for calculating the pressure in the control volume underneath the valve. The incompressible flow theory assumption also eliminates the effect of gas properties on the effective flow and force areas leaving the resulting equations as functions only of the geometry of the valving system and the valve lift.

BASIC EQUATIONS

If the valving system is treated as an equivalent orifice, the mass flow rate of the refrigerant through the valving system can be defined as

$$\dot{m}_v = (KA)\ Y\ \sqrt{2\ \rho_u\ \Delta P_v\ g_c} \tag{1}$$

where (KA) is the effective flow area and ΔP_V is the difference in the stagnation pressures upstream and downstream of the valving system. The drag force on the valve can be defined as

$$F = (FA)\ \Delta P_v \tag{2}$$

where (FA) is the effective force area.

The experimental technique that has been used most for obtaining these areas is outlined in detail by Wambsganss (1). The technique is based on steady state flow tests in which the section of the compressor encompassing the valving system is attached to the end of a flow meter. By measuring the mass flow rate through the flow meter for a given valve lift, pressure drop across the valving system, and the density of the gas upstream of the valving system; and equating it to the mass flow rate of the gas through the valving system the effective flow area (KA) of Equation 1 can be determined. By measuring the force on the valve for a given pressure drop across the valving system, the effective force area can be determined from Equation 2.

In previous work (1,3), the effective flow area for a given valve system and valve lift was found to be only a slight function of the pressure drop across the valve while there was no noticeable effect on the effective force area. The effect of the valve lift velocity was eliminated by the use of steady state flow values. This assumption is justified because the velocity of the valve is usually considerably lower than the speed of sound of the gas.

EFFECTIVE FLOW AREA

Consider the simple valving system for a ring type valve as shown in Figure 1. As the first step in analyzing the flow through the valving system, the geometric area perpendicular to the direction of the flow is plotted as a function of the flow path. In Figure 2, the major restrictions to the flow at different points can be seen. Each of these restrictions can be considered to behave as an orifice, where the mass flow rate through each orifice is governed by the difference in the pressures in the control volumes on either side of the orifice.

The main assumption is made that incompressible flow theory can be applied (i.e. $Y = 1$ and ρ = constant). At pressure ratios across the valve above 0.90, the flow of the gas through the valve can be considered incompressible. Making this assumption, the influence of the pressure ratio on the effective flow area is eliminated and a greatly simplified equation for the effective flow area (only a function of the geometry of the valving system) is obtained. Without this assumption, the flow rate through the valving system can only be calculated by knowing the density and the pressure in each of the small control volumes inside the valving system.

A flow coefficient for each of the orifices is determined from published values (4) on incompressible flow through general configurations similar to that given restriction. The equation for the total flow rate through the valving system is then determined in a manner similar to that for finding the current through a system of resistances in series and in parallel.

Any valving system can be considered to be made up of a combination of orifices in series and in parallel. For the case of two orifices in series, as is shown in Figure 3, the expression for the mass flow rate can be determined as follows:

$$\dot{m} = K_1A_1Y_1 \sqrt{2\ \rho_1\ \Delta P_1 g_c} = K_2A_2Y_2 \sqrt{2\ \rho_2\ \Delta P_2 g_c} \quad (3)$$

where the pressure drop across each orifice will be assumed to be the difference in the stagnation pressure between the control volumes. The total pressure drop across the two orifices equals the sum of the pressure drops across each orifice, i.e.,

$$\Delta P = \Delta P_1 + \Delta P_2 \quad (4)$$

or

$$\Delta P_1 = \Delta P - \Delta P_2 \quad (5)$$

Substituting Equation 5 into the first part of Equation 3 yields

$$\dot{m} = K_1A_1Y_1 \sqrt{2\ \rho_1\ (\Delta P - \Delta P_2) g_c} \quad (6)$$

However, from the second part of Equation 3,

$$\Delta P_2 = \frac{\dot{m}^2}{2K_2^{\,2}A_2^{\,2}Y_2^{\,2}\rho_2 g_c} \quad (7)$$

Substituting this equation into Equation 6 yields

$$\dot{m} = \frac{K_1A_1Y_1}{\left[1+\left(\frac{K_1A_1Y_1}{K_2A_2Y_2}\right)^2 \frac{\rho_1}{\rho_2}\right]^{1/2}} \sqrt{2\ \rho_1\ \Delta P\ g_c} \quad (8)$$

Applying the assumption of incompressible flow theory, the density is assumed to be constant and the velocity of approach factor is set equal to one. Therefore, an effective flow area for the two orifices in series is of the form

$$(KA) = \frac{K_1A_1}{\left[1 + \left(\frac{K_1A_1}{K_2A_2}\right)^2\right]^{1/2}} \quad (9)$$

For orifices in parallel, a similar approach is used. For Figure 4, the total mass flow rate is

$$\dot{m} = \dot{m}_1 + \dot{m}_2 \quad (10)$$

where

$$\dot{m}_1 = K_1A_1 \sqrt{2\ \rho\ \Delta P\ g_c} \quad (11)$$

$$\dot{m}_2 = K_2A_2 \sqrt{2\ \rho\ \Delta P\ g_c} \quad (12)$$

Substituting Equations 11 and 12 into Equation 15 yields

$$\dot{m} = (K_1A_1 + K_2A_2) \sqrt{2\ \rho\ \Delta P\ g_c} \quad (13)$$

and the effective flow area for orifices in parallel is

$$(KA) = K_1A_1 + K_2A_2 \quad (14)$$

Equations 9 and 14 can now be used to determine the effective flow area for the valve configuration of Figure 1.

The restrictions underneath the valve can be combined together by Equation 14 to obtain

$$(KA)_1 = K_oA_o + K_iA_i \quad (15)$$

where $A_o = 2\pi R_o y$ and $A_i = 2\pi R_i y$. Then Equation 9 is used to combine $(KA)_1$ with the restriction at the entrance to the port (K_eA_e) to obtain the effective flow area for the entire valving system

$$(KA) = \frac{(KA)_1}{\left[1 + \left(\frac{(KA)_1}{K_eA_e}\right)^2\right]^{1/2}} \quad (16)$$

In Equation 16, if the port area leading to the valve is large compared to the areas between the valve and its seats $(KA)_1$, the effective flow area is approximately equal to $(KA)_1$. As the valve lift increases and $(KA)_1$ grows larger, the port area starts limiting the mass flow rate through the valve. As $(KA)_1$ approaches infinity, the effective flow area approaches the area of the port $(K_e A_e)$.

This technique was applied to the seven different valving systems shown in Figure 5. The valves varied from one inch to over four inches in outer diameter. Each valve has its own unique port section or configuration of restrictions past the valve. For one of the valves the gas had to undergo a 90° change in direction before it reached the valve; in another, 180° change. For some of the valves the major restrictions to the flow occurred after the gas had passed the restriction between the valve and the seats.

The valves at the top left hand side of the picture (PS and PD) are respectively the suction and the discharge valves used by Payne (3) in the 3 H.P. R-22 compressor he modeled. The valves at the bottom (AS and AD) are the suction and discharge valves from a 5 H.P. R-12 compressor. Experimental flow and force coefficients for these valves were determined by Adams (7). The experimental data for these four valves were obtained by using flat plate displacements of the valves and by using a modified isentropic flow equation in place of Equation 1. Since the theoretical values were to be compared to the experimental data obtained by Payne and Adams, the experimental effective flow area data had to be modified by a conversion factor to take into account the different equations used to obtain the effective flow areas.

The three valves on the right side of the picture (SS, SOD and SID) are respectively the suction and the two discharge valves from a 50 H.P. ammonia vapor compressor. Experimental data for these three valves were taken by Schwerzler (6) using Equation 1 to solve for the effective flow area.

Equations 15 and 16 are applicable to the valving systems PD and AD. Figure 6 shows the comparisons of the experimental effective flow areas to those obtained by the above equations. The experimental values shown for PD are for a constant pressure drop across the valve of 2 psi. A range of values are given at each valve lift for the AD valving system to correspond to a range of pressure drops across the valve at that lift.

This range of experimental effective flow area values at each given valve lift for various pressure drops across the valves was also obtained by Wambsganss (1), Payne (3) and Schwerzler (6) in their data. This range can be attributed to both compressible flow effects and inaccuracies in the experimental techniques.

Figure 6 demonstrates the usefulness of this technique. As was pointed out before, as the area between the valve and its seats increases with valve lift, the port area starts limiting the flow rate. Both curves of Figure 6 show this effect. Therefore, increasing the port area of these valves would probably have more of an effect on increasing the mass flow rate through the valve than would increasing the stop depth of the valve.

The most complex valving system that was modeled is the suction valve, SS. Figure 7 shows a side view of the valving system and Figure 8 its equivalent orifice system. The casing on the back of the valve and the narrow restrictions along the sides of the valve are included in the model. Figure 9 shows the comparison between the experimental and analytical results.

From the analytical model, it was discovered that at around 0.030 inch valve lift, the area between the outer edge of the valve and the valve case limited the flow around the outer edge of the valve. At around 0.040 of an inch, the last restriction, A_p, started limiting the total flow rate.

For the four other valves, the same type of correlation was obtained between the experimental data and analytical model. The same flow coefficients were used for each of the valves where they were appropriate. The flow coefficient for the restriction between the valve and its seat was 0.70 for all seven valves.

EFFECTIVE FORCE AREA MODEL

For the valving system of Figure 10, the force on the valve is a function of the three pressures P_d, P_p, and P_r. The pressure P_d is the stagnation pressure on the downstream side of the valve. The pressure P_p is the pressure the flow exerts on the valve due to its momentum change. The pressure P_r is the static pressure of the gas as it flows through the two narrow channels between the valve and the seats.

The pressure P_p can be determined by summing the forces vertically on the rectangular section shown in Figure 10

$$P_p A_p - P_e A_e - P_{st}(A_p - A_e) = \frac{\dot{m}_v V_e}{g_c} \tag{17}$$

where P_{st} is the stagnation pressure of the internal control volume enclosed by the restrictions. See Figure 1. Solving for P_p yields

$$P_p = P_e \frac{A_e}{A_p} + P_{st} \frac{A_p - A_e}{A_p} + \frac{\dot{m}_v V_e}{g_c A_p} \tag{18}$$

Using the assumption of incompressible flow,

$$P_e = P_{st} - 1/2 \frac{\rho V_e^2}{g_c} \tag{19}$$

and

$$V_e = \frac{\dot{m}_v}{\rho A_e} \tag{20}$$

Substituting Equation 19 and 20 into Equation 18 and rearranging yields

$$P_p = P_{st} + 1/2 \frac{\dot{m}_v^2}{\rho A_e A_p g_c} \tag{21}$$

Summing the forces on the valve gives

$$F = - P_d A_v + P_r (A_v - A_p) + P_p A_p \tag{22}$$

where

$$P_r = P_{st} - 1/2 \frac{\rho V_r^2}{g_c} \tag{23}$$

and

$$V_r = \frac{\dot{m}_v}{\rho (KA)_1} \tag{24}$$

Substituting Equations 21, 23, and 24 into Equation 22 and rearranging

$$F = - P_d A_v + P_{st} A_v + \frac{\dot{m}_v^2}{2 \rho g_c} \left[\frac{A_p}{(KA)_1^2} + \frac{1}{A_e} - \frac{A_v}{(KA)_1^2} \right] \tag{25}$$

Now,

$$P_{st} = P_d + \Delta P \tag{26}$$

where ΔP is the pressure drop across the effective flow area $(KA)_1$ and is therefore given by

$$\Delta P = \frac{\dot{m}_v^2}{2\rho (KA)_1 g_c} \tag{27}$$

The mass flow rate is obtained through the use of the effective flow area model as is given by the expression

$$\dot{m}_v = (KA) \sqrt{2 \rho \Delta P_v g_c} \tag{28}$$

where (KA) is the effective flow area coefficient for the given valve lift. If Equations 26 to 28 are substituted into Equation 25 and the effective force area coefficient $(F/\Delta P_v)$ is solved for, the following equation is obtained,

$$(FA) = F/\Delta P_v = (KA)^2 \left[\frac{A_p}{(KA)_1^2} + \frac{1}{A_e} \right] \tag{29}$$

If A_p is assumed to remain a constant value, this equation is only valid for small valve displacements since it was implicitly assumed in the derivation that the valve always changes the direction of the gas a complete ninety degrees within this area as the gas flows out from under the valve. As the value approaches large displacements, this assumption will no longer hold true as more area of the valve is needed to turn the flow. In order to extend this equation to higher valve lifts, the area A_p was allowed to increase beyond the inside edges of the seats so that it was equal to the diameter of the largest semicircle that could be placed inside the port as is shown in Figure 11. A semicircle was chosen because it best defined the stagnation region enclosed by the middle streamlines which can be considered to approach the shape of equilateral hyperbolas. Using Equation 21 for P_p becomes somewhat questionable with this approach for A_p. However, for the maximum valve lift considered for these seven valves (0.055 of an inch), the maximum percentage increase encountered in the diameter of the semicircle was less than 19%. Therefore, Equation 21 remained a good approximation for these valves.

In the above expression, both (KA) and $(KA)_1$ are a function of the valve lift. For zero valve lift, both values are zero. Therefore, the lower limit on the effective force area is A_p; the area between the valve seats.

Equation 29 was applied to the valving systems PD and AD. Figure 12 shows the comparison between the experimental values and the results predicted by Equation 29. Without allowing A_p to increase, the correlation would have been good only up to 0.020 of an inch valve lift.

In Equation 29 there are no negative terms although in reality a Bernouli effect could create a negative force on the valve. This factor is the result of assuming that the control volume is of the form shown in Figure 1. The stagnation pressure of the gas between the valve and its seats is therefore the same as that inside the port. The stagnation pressure does not change until the gas reaches the edge of the valve. If the change in stagnation pressure is assumed to occur at the inside edge of the valve seats a negative term will appear in Equation 29.

This same technique was applied to the other types of valving systems. The resulting equation for the effective force area differed for the various valves. The reason for the variations was due to the different types of restrictions the flow encountered past the valve which affected the pressure drop, ΔP, across the valve. The most complicated valving system for determining the effective force area equation was for the SS valve. As can be seen in Figure 7, due to the velocity of the gas through the control volume (3L), the pressure p_d behind the valve is no longer the downstream stagnation pressure but is the static pressure of the gas in this control volume.

Figure 13 shows the comparison between the analytical equation and the experimental data for this valve.

The correlation between the analytical equation and the experimental data for the other four valves was of the same form.

CONCLUSION AND SUMMARY

The simple approach of considering the restrictions in a valving system to behave as a system of orifices in series and in parallel resulted in an analytical model that showed excellent correlation to experimental data to warrant its use as a design tool. The assumption of incompressible flow theory in the derivation of the techniques for the effective flow and force areas does limit the use of the results. However, the benefits in obtaining equations that are only dependent on the geometry of valving system and the valve height are useful. Parameter studies on the geometry of the valving system can be made in only a matter of minutes by these techniques compared to the weeks needed to do the studies experimentally. An example of a parameter study which used these techniques can be found in reference 6.

NOMENCLATURE

A = Area
F = Drag force on the valve
FA = Effective force area
g_c = Constant of Proportionality in Newton's Second Law
K = Flow coefficient
KA = Effective flow area
$\dot{m}$ = Mass flow rate
P = Pressure
R = Radius of the valve
V = Velocity
Y = Velocity of approach factor
ρ = Density

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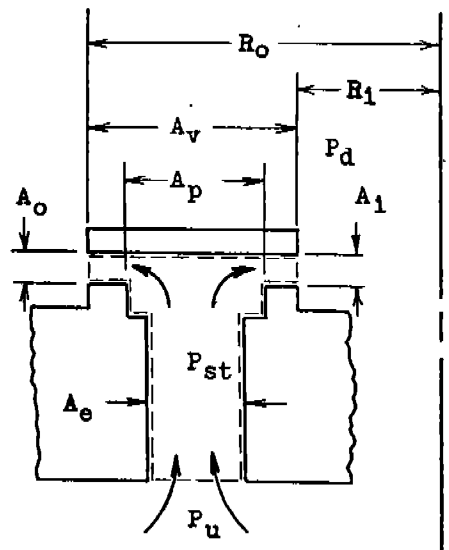


Figure 1 - Schematic of a Simple Ring Type Valving System.

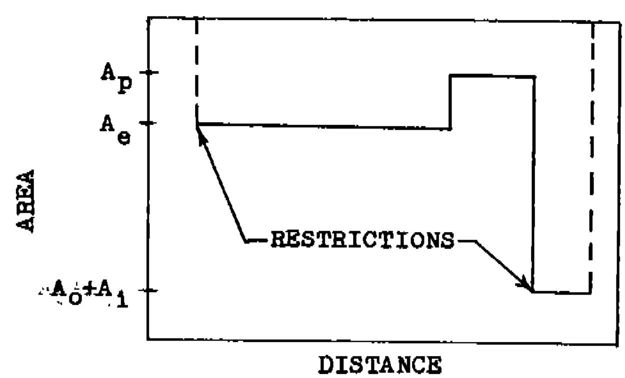


Figure 2 - Flow Area Perpendicular to Flow Path.

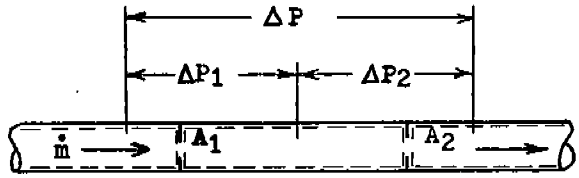


Figure 3 - Orifices in Series.

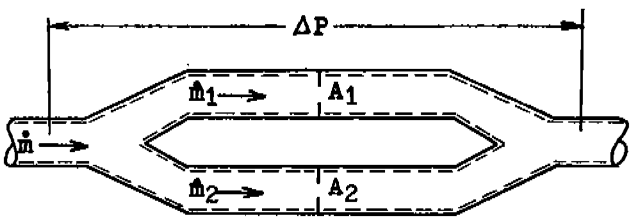


Figure 4 - Orifices in Parallel.

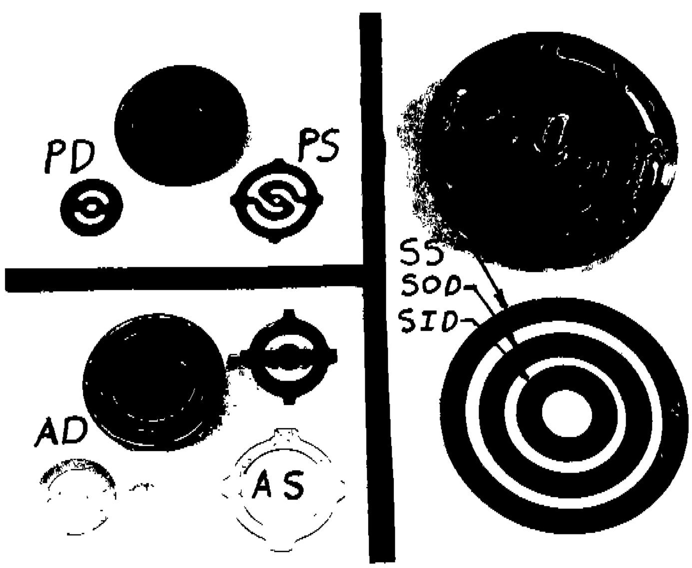


Figure 5 - The Valving Systems to which the Theoretical Effective Flow and Force Area Techniques were Applied.

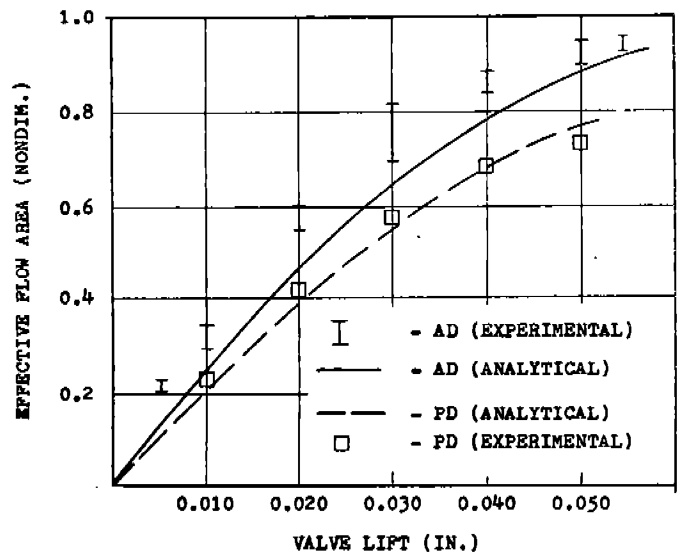


Figure 6 - Comparison of the Analytical and Experimental Effective Flow Areas for the PD and AD Valving Systems.

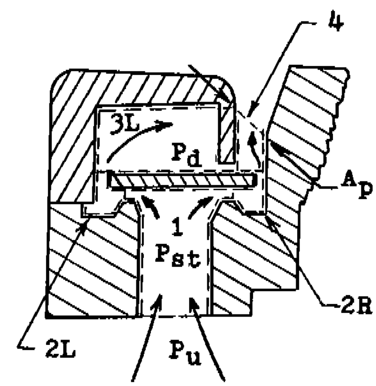


Figure 7 - Schematic of the SS Valving System.

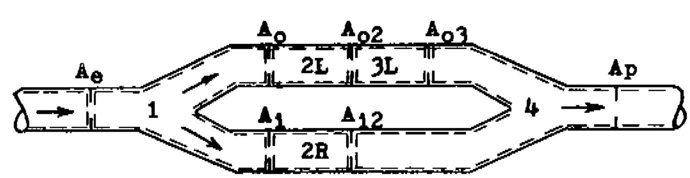


Figure 8 - Equivalent System for the SS Valving System.

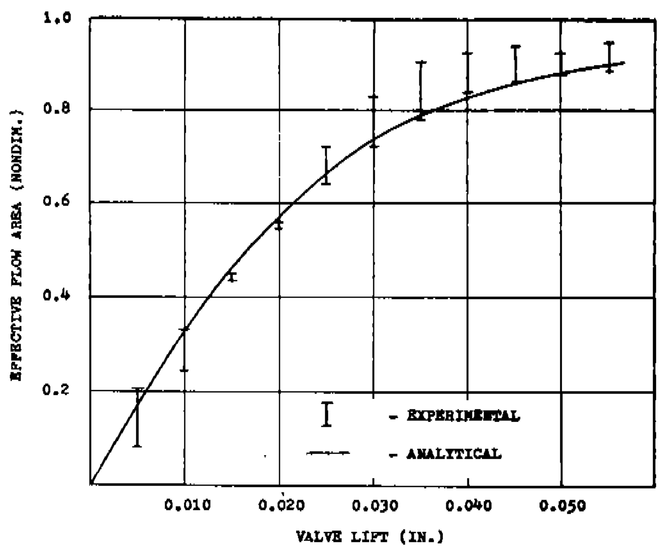


Figure 9 - Comparison of the Analytical and Experimental Effective Flow Areas for the SS Valving System.

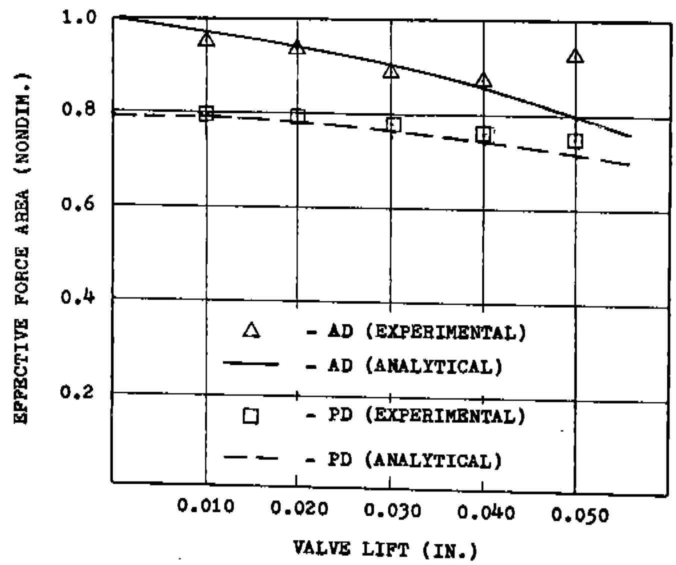


Figure 12 - Comparison of the Analytical and Experimental Effective Force Areas for the PD and AD Valving Systems.

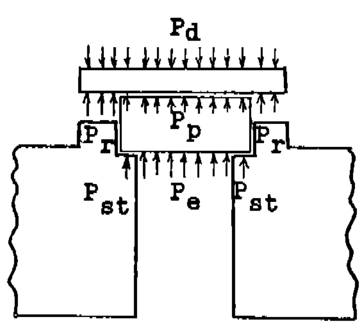
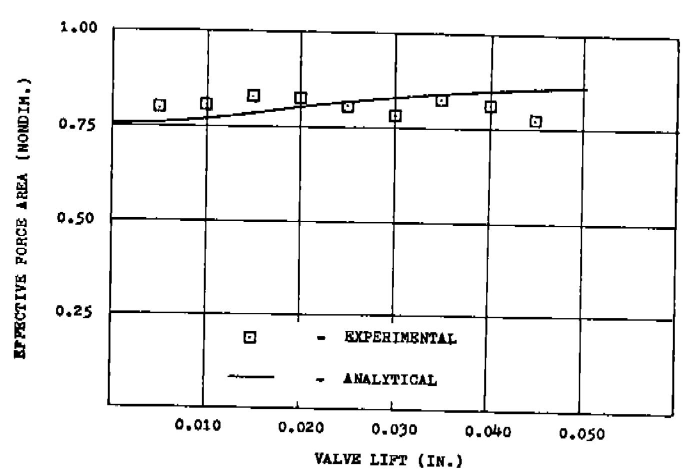


Figure 10 - Location of the Internal Pressures.



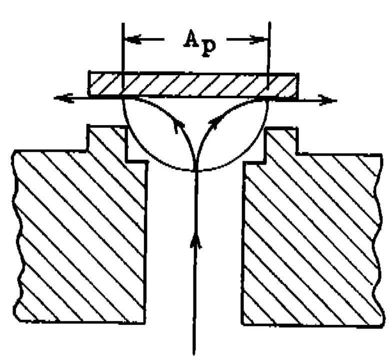


Figure 11 - Technique for Defining the Stagnation Area.

Figure 13 - Comparison of the Analytical and Experimental Effective Force Areas for the SS Valving System.