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Estimating Valve Losses When Dynamic Effects Are Important

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ESTIMATING VALVE LOSSES WHEN DYNAMIC EFFECTS ARE IMPORTANT

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ABSTRACT

A new method has been developed for calculating valve losses in reciprocating compressors. Previously available methods have either 1) made major assumptions such as neglecting valve dynamics and the compressibility of the gas in the cylinder and as a result have been accurate for only a narrow range of conditions, or 2) used a lengthy numerical integration scheme to solve for the valve lift and cylinder pressure. The new method is designed to fill the void between these two methods and provide a reasonably accurate method applicable to the complete range of operating conditions that nevertheless is fast enough for routine use in compressor application and valve selection calculation.

The technical basis for the new method is described here and comparison of its accuracy with results of a numerical integration method are given. It is shown that adequate accuracy has been achieved over a wide range of operating conditions and that an understanding of this theory leads to a greatly improved feel for valve design and selection and for the errors in previously used methods.

INTRODUCTION

The methods traditionally used for compressor sizing and valve selection (Ref. 1, 2) make two major assumptions:

1) The valve is completely open for the complete valve event and,

2) The gas in the cylinder behaves as an incompressible fluid during the valve event.

These assumptions give a reasonable prediction of the valve losses, and hence the compressor power if, and only if, the valve design is such that the valve dynamics are good and the losses low. It gives the designer or application engineer no information on how to select valves to achieve this result or even if such a valve is possible.

To overcome these limitations many programs have been written to calculate the valve losses using numerical integration of the equation of motion for the valve and equations governing the cylinder pressure. The method we use (Ref. 3) reduces the time required for this calculation compared to earlier methods, but is still too time consuming and expensive for routine use in compressor sizing and application.

NOMENCLATURE

A	Constant $\left(\frac{n_v}{2V_1} \sin \theta_1\right)$
B	Constant $\left(\frac{1}{0{\cdot}006E}^{7{\cdot}8} \left(\frac{V_1 P_{Fo}}{n_v\ C}\right)^2\right)$
C	Dimensionless valve area
C_E	Effective dimensionless valve area
$\overline{C_E}$	Effective dimensionless valve area corrected for flutter
C_{op}	Dimensionless valve area during valve opening
CL	Dimensionless clearance (V_{cl}/V_{sw})
E	Dimensionless valve natural frequency ($60f/N$)
f	Valve natural frequency
F	Valve equivalent area at lift L

NOMENCLATURE

Feq	Valve equivalent area at full lift
K	Constant ($P_{F0}^2\ \Delta P_\lambda$)
$\overline{K}$	Constant ($P_{F0}^2\ \overline{\Delta P_\lambda}$)
L	Valve lift
$\dot{m}$	Mass flow rate
MW	Molecular Weight
n	Constant ($n_v^2/(n_v-1)$)
n_V	Isentropic volume exponent
N	Compressor speed (rpm)
p	Pressure
P	Dimensionless pressure (p/ps; p/pd)
P_{FO}	Dimensionless pressure drop to fully open valve
Δp	Valve pressure drop
ΔP	Dimensionless valve pressure drop (Δp/ps; Δp/pd)
$\overline{\Delta P}$	Volume average value of ΔP
ΔP_{op}	Dimensionless valve pressure drop during valve opening
$\Delta P_{\lambda,\alpha,\rho,\gamma,\delta}$	Dimensionless valve pressure drop after different corrections
$\overline{\Delta P}_{\lambda,\alpha,\rho,\gamma,\delta}$	Volume average value of $\Delta P_{\lambda,\alpha,\rho,\gamma,\delta}$
R	Gas constant
t	Time
T	Temperature
v	Cylinder volume
v_1	Cylinder volume at crank angle θ_1
v_{CL}	Cylinder clearance volume
v_{SW}	Cylinder swept volume
X	Constant (Equn 5)
$\overline{X}$	Constant (Equn 6, 7)

NOMENCLATURE

Y	Constant (Equn 5)
$\overline{Y}$	Constant (Equn 6, 7)
Z	Gas compressibility
α	Angle
θ_1	Crank angle at valve opening
θ_2	Crank angle at end of constant rate of pressure rise (Fig. 4)
θ_3	Crank angle at end of valve opening period (Fig. 4)
ρ	Gas density

Subscripts

c	Cylinder
d	Discharge
s	Suction

ASSUMPTIONS

For simplicity in the analysis, the following assumptions are made:

1) There is no spring preload. That is, the spring force is zero when the valve is closed (Fig. 1).
2) The curve of equivalent area against pressure drop across the valve under steady state conditions (Fig. 1) is linear up to the pressure drop that causes the valve to open fully.
3) The connecting rod is long compared to the stroke.
4) The natural frequency of the valve element on its springs is constant (i.e. independent of lift).
5) Flow through the valve is incompressible.
6) All valve elements are identical (or average values are used).

The theory can be modified to avoid these restrictions at the cost of some extra complexity. For purposes of developing the technique it is convenient to make these assumptions.

In this paper, the theory is developed for suction valves. The theory for discharge valves is almost identical and the differences are pointed out where they occur.

APPROACH

As a first step in understanding the effects of valve design on compressor valve losses a dimensional analysis was done. This showed that a dimensionless form of the average valve pressure drop ($\overline{\Delta P} = \overline{\Delta p}/p$) is a function of the following dimensionless parameters.

1) Dimensionless Equivalent Area *

$$c = \frac{F_{eq} \sqrt{2ZRT}}{V_{sw}\, \pi\, N/60}$$

or if Feq is in in.^{2}; Vsw in in.^{3}; T in ^oR;

R in ft.lb./lb. ^oR and

N in RPM then

$$c = 72{,}280 \frac{F_{eq}}{NV_{sw}} \sqrt{\frac{ZT}{MW}}$$

2) Dimensionless pressure drop to fully open the valve

$$P_{FO} = p_{FO}/p$$

3) Dimensionless valve element natural frequency

$$E = 60f/N$$

where f is in Hz and N in rpm.

4) Compressor volumetric efficiency (suction or discharge, depending on which valve is considered).

5) Compressor dimensionless clearance

6) Isentropic volume exponent for gas

Charts (e.g. Fig. 2) can be prepared using a numerical integration scheme (Ref. 3) to show the variations of average valve pressure drop with the above independent parameters and compare this to results of the conventional method. It is found that the isentropic volume exponent is not significant, but even so, it is not practical to plot sufficient curves to cover the range of five independent parameters and so it is necessary to take the analysis one step further.

* c = Equivalent valve area/Valve area that would give pressure drop equal to line pressure with a gas velocity given by the maximum piston velocity.

The approach used is to start with the conventional assumptions that the valve is fully open for the complete valve event and that the gas in the cylinder is incompressible. This simple theory is then modified to account for the actual effects until adequate agreement with results of the numerical integration is obtained. At each stage, the simplest theory that will do the job is used. Care should be taken in improving any of the assumptions made, even if the new assumption is obviously more accurate than the one given here, as in some cases this will result in poorer overall accuracy. To simplify and speed up the calculation, assumptions that lead to equations that require an iterative solution or numerical integration have been avoided.

Each correction is described separately below. They allow for errors in the conventional method caused by:

1) Change in gas density. The conventional method assumes that the density of the gas in the cylinder is the same as that in the passage. A correction is made to allow for the fact that the density in the cylinder, and for discharge valves, that of the gas flowing through the valve, changes as the pressure changes.

2) Valves that don't open fully. The conventional method assumes that the valve is fully open for the complete valve event. The new method recognizes that the valve will not open fully if the spring is such that the average pressure drop across the valve is less than that required to hold the valve fully open.

3) Valves that flutter. If the valve does not open fully it will usually flutter. This correction allows for the fact that the pressure drop is a non-linear function of the lift and so flutter requires a correction of the average valve area.

4) Effects during valve opening. The conventional method assumes that the pressure drop increases to the value given by the piston velocity and the valve equivalent area as soon as the valve is supposed to open. This correction allows for two actual effects: a) The pressure in the cylinder cannot possibly fall below (for suction valves) the pressure that would occur if the valve did not open and b) that due to its inertia, the valve takes a finite time to open.

CONVENTIONAL ASSUMPTIONS

Assuming that the compressor connecting rod is long compared to the stroke, the cylinder volume is given by:

$$V = CL.\ Vsw + \frac{Vsw}{2}(1 - \cos\theta)$$

$$\text{and } dv/d\theta = \frac{Vsw}{2}\sin\theta$$

Assuming that the gas in the cylinder is incompressible,

$$\dot{m} = \rho\frac{dv}{d\theta}\cdot\frac{d\theta}{dt} = Feq\sqrt{2\rho\,\Delta p}$$

$$\Delta P_\lambda = \Delta p/p = \left(\frac{\sin\theta}{c}\right)^2 \quad \ldots\ldots (1)$$

For the calculation of compressor horsepower loss we require a valve pressure drop averaged with respect to the cylinder volume:

$$\text{i.e. } \overline{\Delta P_\lambda} = \int_{V_1}^{V_{sw}+V_{cl}}\Delta P.dv \Big/ \int_{V_1}^{V_{sw}+V_{cl}} dv$$

using (1)

$$\overline{\Delta P_\lambda} = \frac{6VE - 4VE^2}{3c^2} \quad \ldots\ldots (2)$$

Thus using the conventional assumptions the average valve loss is a function of only the volumetric efficiency and the dimensionless valve area.

CORRECTION FOR CHANGE OF GAS DENSITY

The above analysis assumes that the gas in the cylinder has the same density as that in the passage. For valves with a large pressure drop, this can lead to significant errors. For suction valves it is reasonable to assume that the temperature of the gas in the cylinder is the same as that in the passage and so the density is proportional to the pressure.

$$\rho c/\rho s = 1 - \Delta P$$

Then

$$\dot{m} = \rho c\frac{dv}{d\theta}\frac{d\theta}{dt} = Feq\sqrt{2\rho_s\,\Delta p}$$

Or

$$\Delta P_\alpha = \frac{1}{2}\left(\frac{c}{\sin\theta}\right)^2 + 1 - \sqrt{\frac{1}{4}\left(\frac{c}{\sin\theta}\right)^4 + \left(\frac{c}{\sin\theta}\right)^2}$$

Applying this correction to $\overline{\Delta P_\lambda}$ gives a better approximation to $\overline{\Delta P}$, namely:

$$\overline{\Delta P_\alpha} = \frac{1}{2\,\overline{\Delta P_\lambda}} + 1 - \sqrt{\frac{1}{4}\left(\frac{1}{\overline{\Delta P_\lambda}}\right)^2 + \frac{1}{\overline{\Delta P_\lambda}}}$$

For discharge valves, it is assumed that change of state in the cylinder is isentropic and the density of gas flowing through the valve is the density of the gas in the cylinder. Then

$$\overline{\Delta P_\alpha} = \frac{n}{n_v} - \frac{n}{\overline{\Delta P_\lambda}} + \sqrt{\left\{\frac{n^2}{\overline{\Delta P_\lambda}^2} - \frac{2n^2}{n_v\,\overline{\Delta P_\lambda}} + \frac{n^2}{n_v^2} + 2n\right\}}$$

$$\text{where } n = \frac{n_v^2}{n_v - 1} \quad \ldots\ldots. (4)$$

or with the assumption that

$$\frac{\overline{\Delta P_\lambda}^2}{2n} \ll 1 + \frac{\overline{\Delta P_\lambda}}{n_v}$$

$$\overline{\Delta P_\alpha} = \frac{\overline{\Delta P_\lambda}}{1 + \overline{\Delta P_\lambda}/n_v} \quad \ldots\ldots (4a)$$

CORRECTION FOR VALVES THAT DON'T OPEN FULLY

With the assumption that the equivalent area is proportional to the pressure drop across the valve up to the pressure drop required to fully open the valve (Fig. 1), the equivalent area F at pressure drop ΔP is given by:

$$F/Feq = \Delta P/P_{FO}$$

During the time when the valve is not fully open

$$\dot{m} = \rho c\frac{dv}{d\theta}\frac{d\theta}{dt} = F\sqrt{2\rho_s\,\Delta p}$$

The corresponding instantaneous valve pressure drop is

$$\Delta P_\beta = \left(\frac{K}{6}\right)^{1/3}\left\{(X+Y)^{1/3} - (X-Y)^{1/3}\right\} + \frac{K}{3}$$

$$\text{where } K = \left(\frac{P_{Fo}\sin\theta}{c}\right)^2 = P_{Fo}^2\,\Delta P_\lambda \quad \ldots\ldots (5)$$

$$X = \sqrt{9 - \frac{4}{3}K}$$

$$Y = \frac{2}{9}K^2 - 2K + 3$$

To obtain the average value of ΔP_β the average value of ΔP_λ is used in the definition of K.

$$\overline{\Delta P_\beta} = \left(\frac{\bar{K}}{6}\right)^{1/3}\left\{(\bar{X}+\bar{Y})^{1/3} - (\bar{X}-\bar{Y})^{1/3}\right\} + \frac{\bar{K}}{3} \quad \ldots\ldots (6)$$

where $\bar{K} = P_{F0}^2\ \overline{\Delta P_\lambda}$

$$X = \sqrt{9 - \tfrac{4}{3}\bar{K}}$$

$$Y = \tfrac{2}{9}\bar{K}^2 - 2\bar{K} + 3$$

For discharge valves, the correction gives:

$$\overline{\Delta P_\beta} = \left(\frac{\bar{K}}{6}\right)^{1/3}\left\{(\bar{Y}+\bar{X})^{1/3} + (\bar{Y}-\bar{X})^{1/3}\right\} - \frac{\bar{K}}{6n} \quad \ldots\ldots (7)$$

$$\text{where } \bar{X} = \sqrt{\left\{9 - \frac{(9n_v - 5)}{3n_v^3}\bar{K} + \frac{3-2n_v}{12n^2 n_v^2}\bar{K}^2\right\}}$$

$$\bar{Y} = 3 - \frac{\bar{K}}{2nn_v} - \frac{\bar{K}^2}{36n^3}$$

$\overline{\Delta P_\beta}$ is used to calculate the average valve equivalent area C_E from:

$C_E = \dfrac{\overline{\Delta P_\beta}}{P_{F0}} c$ if $\overline{\Delta P_\beta}$ is less than P_{FO}

CORRECTION FOR VALVES THAT FLUTTER

If the valve does not open fully, it will usually flutter. If the average valve area C_E is greater than half the full lift area (Fig. 3A), we assume that

$$c^1 = C_E + (c - C_E)\ \sin\alpha$$

To calculate the pressure drop, the average value of c^1 is required:

$$\overline{C_E}^2 = \frac{\int_0^{2\pi} c'^2\, d\alpha}{2\pi}$$

$$= (3C_E^2 + C^2 - 2CC_E)/2 \quad \ldots\ldots (8A)$$

If the average value area is less than half the full lift area (Fig. 3B), we assume

$$c^1 = C_E\ (1 + \sin\alpha)$$

$$\text{or } \overline{C_E}^2 = 1.5\ C_E^2 \quad \ldots\ldots (8B)$$

The pressure corrected for this effect is:

$$\overline{\Delta P_\gamma} = \frac{C_E^2}{\overline{C_E}^2}\ \overline{\Delta P_\beta}$$

If $C_E > 0.5c$ $\left(\overline{\Delta P_\beta} > 0.5 P_{F0}\right)$

$$\overline{\Delta P_\gamma} = \frac{2\overline{\Delta P_\beta}}{3 + \left(\frac{c}{C_E}\right)^2 - 2\frac{c}{C_E}}$$

$$= \frac{2\overline{\Delta P_\beta}}{3 + \left(\frac{P_{F0}}{\overline{\Delta P_\beta}}\right)^2 - 2\frac{P_{F0}}{\overline{\Delta P_\beta}}} \quad \ldots\ldots (9A)$$

If $C_E < 0.5c$

$$\overline{\Delta P_\gamma} = \frac{\overline{\Delta P_\beta}}{1.5} \quad \ldots\ldots (9B)$$

CORRECTION FOR EFFECT OF VALVE OPENING

At the crank angle θ_1, when the valve opens, the rate of change of cylinder volume:

$$\left(\frac{dv}{d\theta}\right)_1 = \frac{V_{sw}}{2}\ \sin\theta_1$$

and the rate of change of cylinder pressure is:

$$\left(\frac{dp}{d\theta}\right)_1 = \frac{n_v p_1}{v_1}\ \frac{V_{sw}}{2}\ \sin\theta_1$$

$$\left(\frac{d\Delta p}{d\theta}\right)_1 = \frac{n_v}{2v_1}\ \sin\theta_1$$

Using the methods of Ref. 3 to calculate the valve lift with initial conditions that the valve lift and velocity are zero and that the pressure drop is as assumed above gives:

$$L = \frac{1}{P_{F0}}\left\{\frac{\sin(\varepsilon(\theta-\theta_1))}{\varepsilon} - (\theta-\theta_1)\right\}\frac{n_v \sin\theta_1}{2v_1}$$

or using the approximation
$u - \sin u = 0.155\ u^{2.9}$

$$L = \frac{n_v \sin\theta_1}{2v_1 P_{F0}\varepsilon}\left\{0.155(\varepsilon\theta - \varepsilon\theta_1)^{2.9}\right\}$$

The dimensionless valve area at angle θ_1, is given by:

$$C_{op} = \frac{\Lambda_v \sin\theta_1 \, c}{2 v_1 P_{F0} E}\left\{0.155\,(E\theta - E\theta_1)^{2.9}\right\}$$

Using the conventional assumptions (Equn 1)

$$\Delta P_{op} = \left(\frac{\sin\theta_1}{C_{op}}\right)^2$$

$$= \frac{B}{(\theta-\theta_1)^{5.8}} \qquad \ldots\ldots (10)$$

$$\text{where } B = \frac{1}{0.006\,E^{3.8}}\left(\frac{v_1 P_{F0}}{\Lambda_v c}\right)^2$$

Equn (7) because it is based on the conventional assumptions gives a pressure drop greater than that that would occur if the valve did not open. Indeed it will give infinite pressure drop at θ_1, when the valve area is zero. This is impossible and we assume that the pressure drop rises at a constant rate of $(d\Delta P/d\theta)$, until it reaches the pressure drop given by equation 10 at crank angle θ_2 (Fig. 4).

Thus from θ_1 to θ_2, $\Delta P = A\,(\theta - \theta_1)$

$$\text{where } A = \frac{\Lambda_v}{2 v_1}\sin\theta_1$$

θ_2 is given by $A\,(\theta_2 - \theta_1) = B/\,(\theta_2 - \theta_1)^{5.8}$

$$\therefore\quad \theta_2 - \theta_1 = (B/A)^{1/6.8}$$

We assume that the pressure drop is given by equation 10 from θ_2 to the angle θ_3 at which this pressure drop equals the average pressure drop calculated without this correction $\overline{\Delta P_\alpha}$

i.e. θ_3 is given by:

$$\frac{B}{(\theta_3-\theta_1)^{5.8}} = \overline{\Delta P_\alpha}$$

$$\text{or}\quad \theta_3 - \theta_1 = \left(\frac{B}{\overline{\Delta P_\alpha}}\right)^{1/5.8}$$

Thus the average pressure drop from θ_1 to θ_3 is given by:

$$\overline{\Delta P_J} = \frac{\int_{\theta_1}^{\theta_2} A(\theta-\theta_1)\,d\theta + \int_{\theta_2}^{\theta_3}\frac{B}{(\theta-\theta_1)^{5.8}}\,d\theta}{\theta_3-\theta_1}$$

$$= \frac{0.708A^{.706}\,B^{.294}}{(B/\overline{\Delta P_\alpha})^{1/5.8}} - \frac{B}{4.8\,(B/\overline{\Delta P_\alpha})} \qquad \ldots\ldots (11)$$

The pressure drop from θ_3 to the end of the stroke is calculated by the methods given in earlier sections.

CALCULATION FLOW DIAGRAM

A flow diagram for the calculation is given as Fig. 5 and shows the order in which the various corrections should be applied. A different order is used for suction and discharge valves. The only justification for the use of the order given is that it gives the most accurate results. Any change in the method used to apply the corrections should be approached with care and not made without checking the effect on accuracy throughout the complete range of parameters.

RESULTS

Typical results of the analysis described above are compared with results from the numerical integration method (Ref. 3) in Fig. 6. The results of the conventional analysis (Equn 2) are also shown on these curves. The new method gives a greatly improved estimate of the true valve loss compared to the conventional method with only a small fraction of the complexity and calculation time of even the simplest integration method.

CONCLUSION

The method predicting valve loss described here gives a good estimate of the true values and is far superior to the methods normally used for compressor sizing calculations. If, as is usually the case, a computer is used for the sizing calculation, the implementation of this method is simple and causes only a small increase in computer costs. If a computer is not available, the work involved in estimating the valve loss by this method can be reduced to an acceptable level by plotting curves of the basic correction equations.

The method correctly predicts the effects of spring load and valve element natural frequency and has lead to an improved understanding of the importance of these factors when designing valves for a given application.

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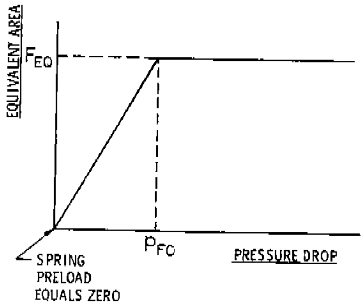
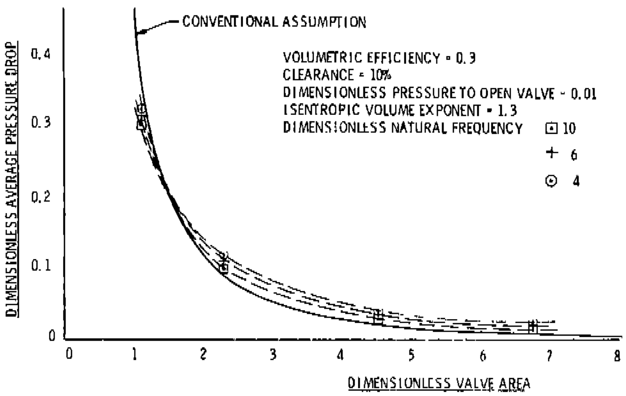


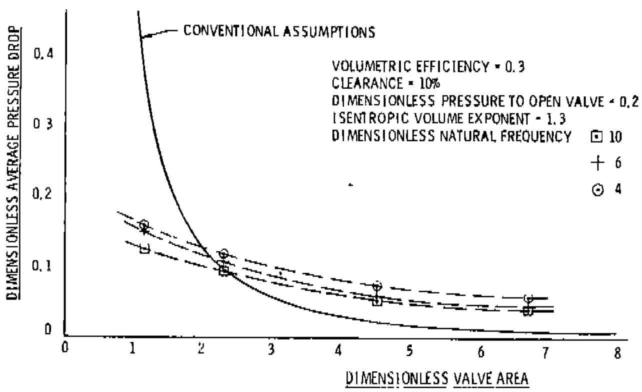
STEADY STATE VALVE PERFORMANCE

FIG. 1



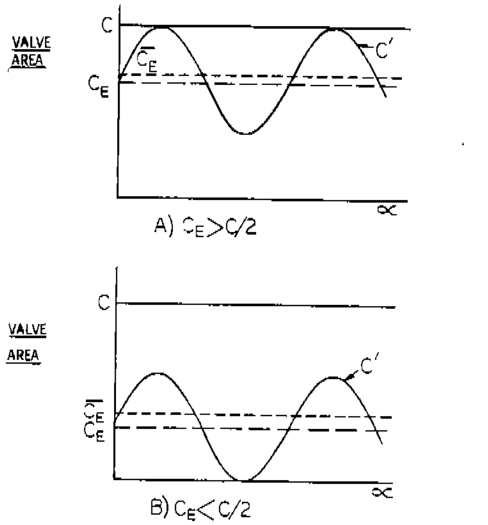
TYPICAL CURVE OF DISCHARGE VALVE LOSS - VALVE AREA (LOW SPRING FORCE)

FIG 2A



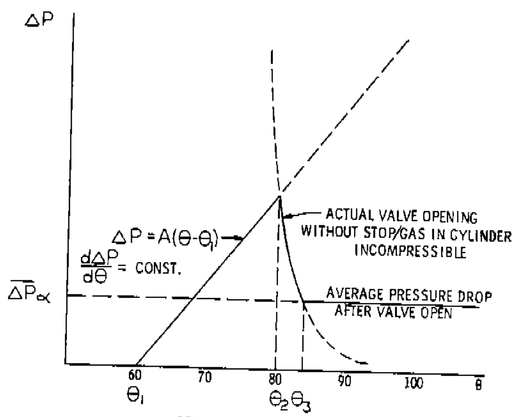
TYPICAL CURVE OF SUCTION VALVE LOSS - VALVE AREA (HIGH SPRING FORCE)

FIG. 2B



CORRECTION FOR FLUTTER

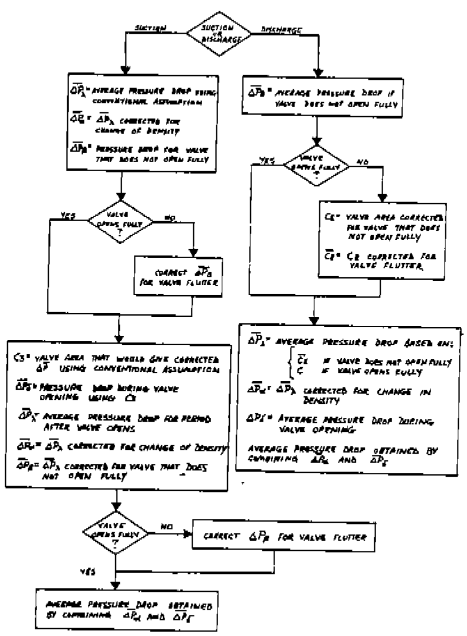
FIG. 3

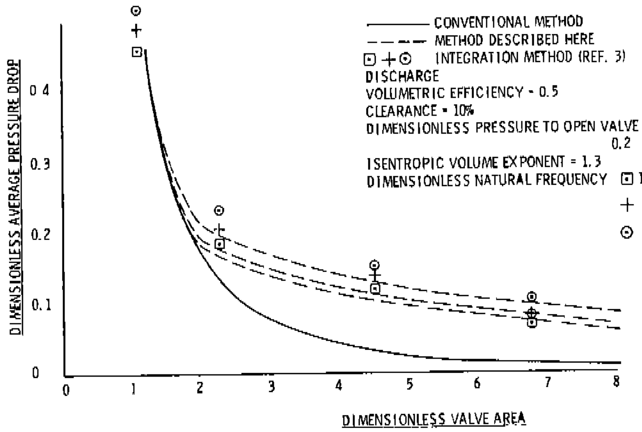


CORRECTION FOR EFFECT OF VALVE OPENING

FIG. 4

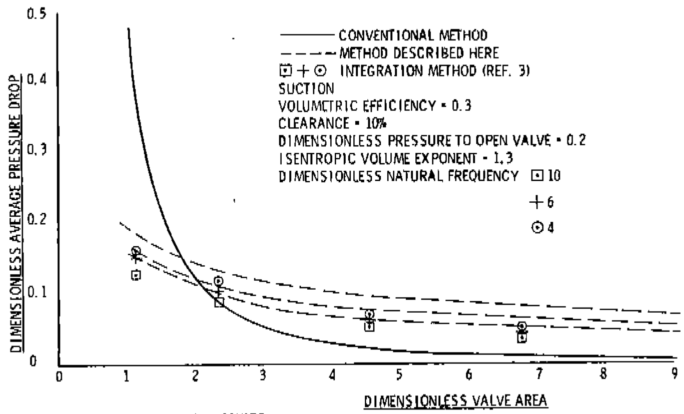
CALCULATION FLOW DIAGRAM





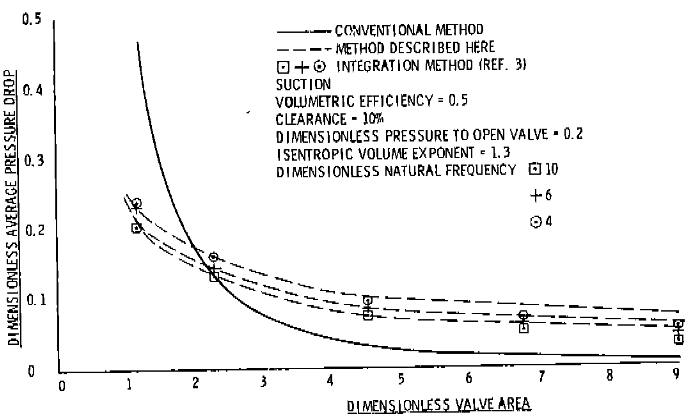
TYPICAL RESULTS

FIG. 6A



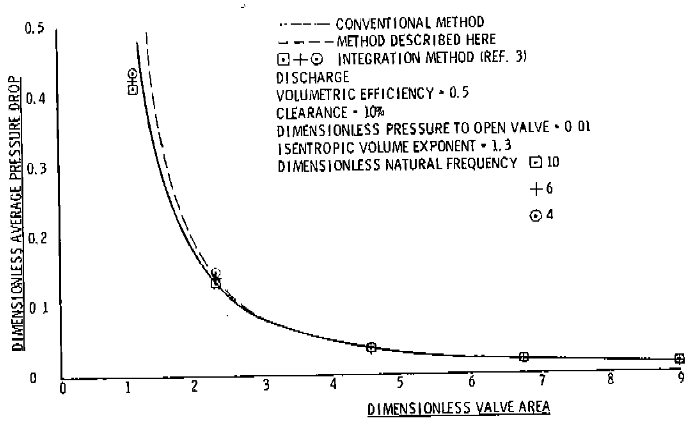
TYPICAL RESULTS

FIG. 6D



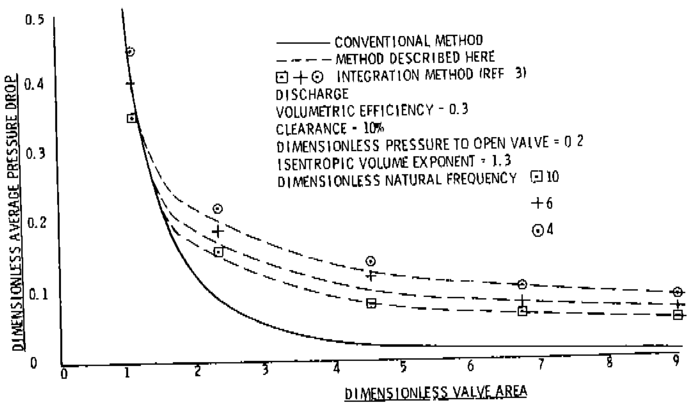
TYPICAL RESULTS

FIG. 6B



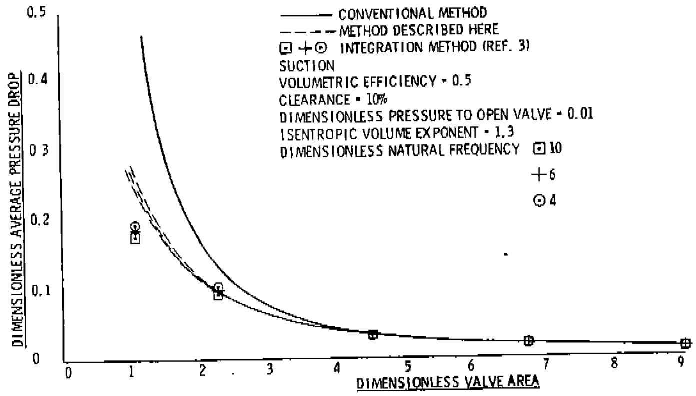
TYPICAL RESULTS

FIG. 6E



TYPICAL RESULTS

FIG. 6C



TYPICAL RESULTS

FIG. 6F