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D. Woollatt

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ESTIMATING VALVE LOSSES WHEN DYNAMIC
EFFECTS ARE IMPORTANT

Derek Woollatt
Worthington Group
McGraw-Edison Company

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

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Constant $\left(\frac{\eta_c}{2\nu}, \sin \theta_c\right)$</td>
</tr>
<tr>
<td>B</td>
<td>Constant $\left(\frac{l}{0.00822 e^{2.89} \left(\frac{\nu_c P_{eq}}{n_c} \right)^2}\right)$</td>
</tr>
<tr>
<td>C</td>
<td>Dimensionless valve area</td>
</tr>
<tr>
<td>$\overline{C}_E$</td>
<td>Effective dimensionless valve area</td>
</tr>
<tr>
<td>$\overline{C}_E$</td>
<td>Effective dimensionless valve area corrected for flutter</td>
</tr>
<tr>
<td>$C_{op}$</td>
<td>Dimensionless valve area during valve opening</td>
</tr>
<tr>
<td>CL</td>
<td>Dimensionless clearance $(V_{cl}/V_{sw})$</td>
</tr>
<tr>
<td>E</td>
<td>Dimensionless valve natural frequency $(60f/N)$</td>
</tr>
<tr>
<td>f</td>
<td>Valve natural frequency</td>
</tr>
<tr>
<td>F</td>
<td>Valve equivalent area at lift L</td>
</tr>
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</table>
NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>Feq</td>
<td>Valve equivalent area at full lift</td>
</tr>
<tr>
<td>K</td>
<td>Constant (P^2 ΔP/ΔP)</td>
</tr>
<tr>
<td>l</td>
<td>Valve lift</td>
</tr>
<tr>
<td>M</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>MW</td>
<td>Molecular Weight</td>
</tr>
<tr>
<td>n</td>
<td>Constant (n^2/(n-1))</td>
</tr>
<tr>
<td>nV</td>
<td>Isentropic volume exponent</td>
</tr>
<tr>
<td>N</td>
<td>Compressor speed (rpm)</td>
</tr>
<tr>
<td>P</td>
<td>Pressure</td>
</tr>
<tr>
<td>Pc</td>
<td>Constant (Equn 5)</td>
</tr>
<tr>
<td>Pd</td>
<td>Constant (Equn 6, 7)</td>
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<tr>
<td>p</td>
<td>Gas compressibility</td>
</tr>
<tr>
<td>α</td>
<td>Angle</td>
</tr>
<tr>
<td>β1</td>
<td>Crank angle at valve opening</td>
</tr>
<tr>
<td>β2</td>
<td>Crank angle at end of constant rate of pressure rise (Fig. 4)</td>
</tr>
<tr>
<td>βf</td>
<td>Crank angle at end of valve opening period (Fig. 4)</td>
</tr>
<tr>
<td>ρ</td>
<td>Gas density</td>
</tr>
</tbody>
</table>

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 \((\Delta p = \Delta p/p)\) is a function of the following dimensionless parameters.

1) Dimensionless Equivalent Area *

\[
c = \frac{\text{Feq}}{\sqrt{2RT}} \frac{Vsw}{\pi N/60}
\]
or if \(\text{Feq}\) is in in.\(^2\); \(Vsw\) in in.\(^3\);
\(T\) in \(^\circ\)R;
\(R\) in ft.1b./lb. \(^\circ\)R and
\(N\) in RPM

then

\[
c = 72,280 \frac{\text{Feq}}{N\text{Vsw}} \sqrt{\frac{ZT}{MW}}
\]

2) Dimensionless pressure drop to fully open the valve

\[P_{FO} = P_{FO}/p\]

3) Dimensionless valve element natural frequency

\[E = \frac{60f}{N}\]

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.

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, 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.

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

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

\[ V = CL \cdot V_{sw} + \frac{V_{sw}}{2} \left( 1 - \cos \theta \right) \]

and \( \frac{dv}{d\theta} = \frac{V_{sw}}{2} \sin \theta \)

Assuming that the gas in the cylinder is incompressible,

\[ \dot{m} = \rho \frac{dv}{d\theta} = \text{Feq} \int 2 \rho \, \Delta P \]

\[ \Delta P_X = \frac{\Delta P}{p} = \left( \frac{\sin \theta}{c} \right)^2 \]

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

i.e. \( \Delta P_X = \int \Delta P \cdot \frac{dv}{\int v_{in} v_{out}} \)

using (1)

\[ \Delta P_X = \frac{6 \text{VE} - 4 \text{VE}^2}{3c^2} \]

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.

\[ \frac{\rho_c}{\rho_s} = 1 - \Delta P \]

Then

\[ \dot{m} = \rho_c \frac{dv}{d\theta} \frac{d\theta}{dt} = \text{Feq} \int 2 \rho_c \Delta P \]

Or

\[ \Delta P_X = \frac{1}{2} \left( \frac{c}{\sin \theta} \right)^2 + 1 - \left( \frac{1}{4} \left( \frac{c}{\sin \theta} \right)^4 + \left( \frac{c}{\sin \theta} \right)^2 \right) \]

Applying this correction to \( \Delta P_X \) gives a better approximation to \( \Delta P \), namely:

\[ \Delta P_X = \frac{1}{2} \Delta P + 1 - \sqrt{\frac{1}{4} \left( \frac{c}{\sin \theta} \right)^2 + \frac{1}{\Delta P_X}} \]

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

\[ \Delta P_X = \frac{n_v}{n_v} - \frac{n_v}{n_v} + \sqrt{\left( \frac{n_v^2}{\Delta P_X} - \frac{2n_v^2}{n_v \Delta P_X} \right) + \frac{n_v^2}{n_v^2} + 2n_v} \]

where \( n = \frac{n_v^2}{n_v - 1} \)

or with the assumption that

\[ \frac{\Delta P_X}{2n_v} \ll 1 + \frac{\Delta P_X}{n_v} \]

\[ \Delta P_X = \frac{\Delta P}{1 + \frac{\Delta P}{n_v}} \]

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 \( A \) at pressure drop \( \Delta P \) is given by:

\[ F / \text{Feq} = \Delta P / \Delta P_F \]

During the time when the valve is not fully open

\[ \dot{m} = \rho_c \frac{dv}{d\theta} \frac{d\theta}{dt} = F \int 2 \rho_c \Delta P \]

The corresponding instantaneous valve pressure drop is

\[ \Delta P_p = \left( \frac{K}{c} \right)^{1/2} \left\{ (x+y)^{1/2} - (x-y)^{1/2} \right\} + \frac{H}{2} \]

where \( K = \left( \frac{P_{po} \, \sin \theta}{c} \right)^2 = P_{po}^2 \Delta P_X \)

\[ X = 3 - \frac{3}{2} \frac{K}{Y} \]

\[ Y = \frac{2}{9} K^2 - 2K + 3 \]
To obtain the average value of $\Delta P_p$, the average value of $\Delta P_x$ is used in the definition of $K$.

$$
\Delta P_x = \left( \frac{\overline{x}}{\epsilon} \right)^{1/2} \left\{ (\overline{x} + \overline{y})^{1/2} - (\overline{x} - \overline{y})^{1/2} \right\} + \frac{K}{6}
$$

where $\overline{x} = \frac{\Delta P_p}{P_{f0}}$

$$
\overline{x} = \sqrt{\frac{9 - \frac{4}{3} \overline{K}}{\epsilon}}
$$

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

For discharge valves, the correction gives:

$$
\Delta P_{p'} = \left( \frac{\overline{x}}{\epsilon} \right)^{1/2} \left\{ (\overline{x} + \overline{y})^{1/2} + (\overline{x} - \overline{y})^{1/2} \right\} - \frac{K}{6} \epsilon
$$

where $\overline{x} = \sqrt{\frac{9 - \frac{4}{3} \overline{K}}{\epsilon}} + \frac{3 - 2 \overline{K}}{2 \overline{y}^2}$

$$
\overline{y} = 3 - \frac{\overline{K}}{2 \overline{y}^2} - \frac{3 \overline{K}^2}{3 \overline{y}^2}
$$

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

$$
C_E = \frac{\Delta P_{p'}}{P_{f0}} C \quad \text{if} \quad \Delta P_{p'} \text{ is less than } P_{f0}
$$

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:

$$
\overline{c_1} = \frac{\int_0^{2\pi} c_1^2 \, d\alpha}{2\pi}
$$

$$
= \frac{3C_E^2 + C_1^2 - 2CC_E}{2} \quad \text{...... (8A)}
$$

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

$$
c_1 = C_E (1 + \sin \alpha)
$$

or

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

The pressure corrected for this effect is:

$$
\overline{AP_{p'}} = \frac{C_E}{\epsilon} \Delta P_{p'}
$$

If $C_E > 0.5 C$ ($\overline{AP_{p'}} > 0.5 P_{f0}$)

$$
\overline{AP_{p'}} = \frac{2 \Delta P_{p'}}{3 + \left( \frac{C_E}{C} \right)^2 - 2 \frac{C_E}{C}}
$$

$$
= \frac{2 \Delta P_{p'}}{3 + \left( \frac{P_{f0}}{\Delta P_{p'}} \right)^2 - 2 \frac{P_{f0}}{\Delta P_{p'}}} \quad \text{...... (9A)}
$$

If $C_E < 0.5 C$

$$
\overline{AP_{p'}} = \frac{\Delta P_{p'}}{1.5} \quad \text{...... (9B)}
$$

CORRECTION FOR EFFECT OF VALVE OPENING

At the crank angle $\theta_i$, when the valve opens, the rate of change of cylinder volume:

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

and the rate of change of cylinder pressure is:

$$
\left( \frac{dp}{d\theta} \right) = \frac{n_v \rho}{v_i} \frac{V_{sw} \sin \theta_i}{2}
$$

$$
\left( \frac{d\Delta p}{d\theta} \right) = \frac{n_v \sin \theta_i}{2v_i} \sin \theta_i
$$

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:

$$
L = \left( \frac{1}{P_{f0}} \right) \left\{ \frac{\sin \theta_i (E(\theta - \bar{\theta}_i)) - (\theta - \bar{\theta}_i) n_v \sin \theta_i}{E} \right\}
$$

or using the approximation

$$
u - \sin u = 0.155 \, u^{2.9}
$$

$$
L = \frac{n_v \sin \theta_i}{2v_i} \left\{ 0.155 \left( E \theta - E \theta_i \right)^{2.9} \right\}
$$
The dimensionless valve area at angle \( \theta \), is given by:

\[
C_{op} = \frac{\eta \cdot S_i \cdot \theta}{2 \cdot V_i \cdot P_{fo} \cdot \varepsilon} \left\{ 0.155 \left( \varepsilon \theta - \varepsilon \theta_1 \right)^{2.9} \right\}
\]

Using the conventional assumptions (Eqn 1)

\[
\Delta P_{op} = \left( \frac{S_i \cdot \theta}{C_{op}} \right)^2 \frac{B}{(\theta - \theta_1)^{5.8}}
\]

where \( B = \frac{1}{0.006 \cdot \varepsilon^{2.9} \left( \frac{V_i \cdot P_{fo}}{\eta \cdot \varepsilon} \right)^2} \) ...... (10)

Eqn (7) because it is based on the conventional assumptions gives a pressure drop greater than that which would occur if the valve did not open. Indeed it will give infinite pressure drop at \( \theta \), when the valve area is zero. This is impossible and we assume that the pressure drop rises at a constant rate of \( \left( \frac{d\Delta P}{d\theta} \right) \), until it reaches the pressure drop given by equation 10 at crank angle \( \theta_2 \) (Fig. 4).

Thus from \( \theta_1 \) to \( \theta_2 \), \( \Delta P = A \left( \theta - \theta_1 \right) \)

where \( A = \frac{\eta \cdot S_i \cdot \theta}{2 \cdot V_i} \)

\( \theta_2 \) is given by \( A \left( \theta_2 - \theta_1 \right) = B / (\theta_2 - \theta_1)^{5.8} \)

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

We assume that the pressure drop is given by equation 10 from \( \theta_2 \) to the angle \( \theta_3 \) at which this pressure drop equals the average pressure drop calculated without this correction \( \Delta P_{av} \)

i.e. \( \theta_3 \) is given by:

\[
\frac{B}{(\theta_2 - \theta_1)^{5.8}} = \Delta P_{av}
\]

or \( \theta_2 - \theta_1 = \left( \frac{B}{\Delta P_{av}} \right)^{1/5.8} \)

Thus the average pressure drop from \( \theta_1 \) to \( \theta_3 \) is given by:

\[
\Delta P_{av} = \int_{\theta_1}^{\theta_3} A \left( \theta - \theta_1 \right) d\theta + \int_{\theta_3}^{\theta_2} \frac{B}{(\theta - \theta_1)^{5.8}} d\theta
\]

The pressure drop from \( \theta_2 \) 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 (Eqn 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.
REFERENCES


