Simulation of a Two Cylinder Compressor for Discharge Gas Pressure Oscillation Prediction

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The paper deals with the prediction of unsteady flows in the discharge system of a reciprocating compressor. A new concept of modeling fluid dynamic interactions between cylinders is developed. Discharge system acoustic impedances, in distributed parameters format, are coupled to the time variant cylinder kinematics and thermodynamics, valve fluid flow, and valve dynamic mathematical simulation models. Excellent correlations between measured and predicted gas pressure cyclic variations and frequency spectra are obtained. To supplement the model, a lumped parameters analysis is also performed to provide physical insights for design purposes.


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ABSTRACT

The paper deals with the prediction of unsteady flows in the discharge system of a reciprocating compressor. A new concept of modeling fluid dynamic interactions between cylinders is developed. Discharge system acoustic impedances, in distributed parameters format, are coupled to the time variant cylinder kinematics and thermodynamics, valve fluid flow and valve dynamic mathematical simulation models. Excellent correlations between measured and predicted gas pressure cyclic variations and frequency spectra are obtained. To supplement the model, a lumped parameters analysis is also performed to provide physical insights for design purposes.

NOMENCLATURE

A, B, C & D Fourpole elements
c Speed of sound
\( \mathbf{J} \) Imaginary number
\( \mathbf{L} \) Length
\( \mathbf{m} \) Mass flow rate
\( \mathbf{n} \) Number of harmonic
\( \mathbf{N} \) Total number of harmonics
\( \mathbf{P} \) Pressure
\( \mathbf{S} \) Cross sectional area
\( \mathbf{t} \) Time
\( \phi \) Mode Shape
\( \mathbf{X} \) Volume displacement
\( \mathbf{X} \) Volume velocity
\( \mathbf{z} \) Acoustical impedance
\( \gamma \) Propagation constant
\( \theta \) Crank angle
\( p \) Fluid mean density
\( \phi \) Phase between cylinders
\( \psi \) Phase of a complex quantity
\( \omega \) Compressor running speed, circular frequency

INTRODUCTION

Fluid fluctuations in the suction and discharge systems of a positive displacement compressor are inherent as the primary source mechanism in the reciprocating action of the piston. Furthermore, valve and pipeline dynamics may either cause or modulate flow pulsations \(^1\). The effects of these flows on the compressor operation are multifaceted. Gas pulsations can increase energy requirements \(^2,4\), cause valve vibrations and flutter \(^3,4\) and may even be a potential cause for the noise radiation and structural vibrations \(^1,2\). Generally the lower harmonics of the running speed influence compressor performance and capacity. Furthermore, the dynamic behavior of a positive displacement compressor and its associated system is highly interactive in nature, and gas pulsations play a major role in determining it. Therefore, the study, analysis and prediction of these deleterious flows have become an integral part of modern design procedures \(^1-17\).

Early efforts were mainly directed towards the analysis of the suction and/or discharge piping systems only \(^2,6,7,9,10,12\), but recent investigators have focused their attention on the overall system studies \(^3-5,8,11,13-17\). They have concluded that for the prediction of suction and/or discharge gas oscillations, one requires a full fledged compressor mathematical model. It should consist of the following models: cylinder process, valve dynamics, fluid flow through valves and flow pulsations. Suction and/or discharge line pulsation models usually consist of equations of motion describing fluid behavior, and are solved to satisfy piping geometric configurations and boundary conditions. Various modeling philosophies have been used with varied degrees of accuracy and completeness for pulsation prediction. Linear acoustic theories \(^2,3,6-12,15\) and the gas dynamic method of characteristics \(^9,12\) are the two most widely accepted modeling concepts. Practically all of the pulsation models are based on a one dimensional description of the unsteady flow motion. This assumption is valid as only the lower frequencies are of importance and furthermore, one often encounters small diameter components in reciprocating compressor installations. Unsteady flow mathematical models have been solved by both transient \(^3,6,8,13,15\) and steady state \(^7,9-12\) techniques, and simulations have been performed on analog \(^2,8\), digital \(^3-7,9,11-16\) and hybrid \(^13\) computers.

MODELING OBJECTIVES

The present work focuses on the prediction of fluid pressure oscillations in the discharge system of a two cylinder refrigerating compressor. The objective is to develop a mathematical model which would clearly account for the cylinder dynamic interactions. Since the investigation is more ambitious towards the study of discharge unsteady flows, the basic compressor models are to be simplified. But, the simplifications should not sacrifice the incorporation of discharge system interactive dynamics with the compressor cylinder and valves. The locations at which pulsation predictions are considered

\(^1\) Numbers in brackets designate references at the end of paper.
important, are discharge valve exit and discharge piping end [2]. As only the first few harmonics of the running speed are detrimental to the compressor performance, the investigation concentrates on the lower frequencies, up to approximately 700 Hz. The investigation is geared to establish the computed simulation program as a design tool in the compressor industry. And as such emphasis has been laid on the computational ease and modeling simplifications without sacrificing any accuracy. Furthermore, it should be possible to obtain data predictions and physical insights necessary for design purposes.

The only other investigation which has dealt with the dynamic coupling of multicylinder cavities is by Soedel et al. [15]. It applied an acoustic lumped parameter approach to describe a two cylinder discharge system. The model is solved in the time domain along with the rest of the compressor simulation equations. A lumped parameter approach relies heavily on an easy identification of components as either inertial or elastic cases. It also suffers from accuracy and is limited to a shorter frequency range. Furthermore, Soedel et al. [15] could not account for the pipe exiting from the collector.

Schwizer [16] has modeled suction and discharge plenums of a multicylinder compressor by using a quasi-steady mean pressure variation technique which ignores gas dynamic coupling effects.

The present study attempts to define and model cylinder interference mechanisms. The discharge system is modeled here with steady state acoustical impedances in distributed parameter format. The impedance concept was first tried by Elson and Soedel [11] on a single cylinder compressor. They determined that it is more efficient and general than some classical approaches. The procedure is interactive in nature as it simulates the effect of discharge back pressure on the automatic valves. Elson and Soedel [11] tried it for a very simple system (valve chamber connected to long lines) and obtained good results. However, they could not generalize the procedure for a multicylinder case, and also did not show its applicability to complex practical configurations. The present study undertakes such efforts.

BASIC COMPRESSOR MODEL

The basic mathematical model for the compressor processes consists of the following [18]:

1. cylinder kinematics model
2. cylinder thermodynamics model
3. valve fluid flow model
4. valve dynamics model

A summarized form of the simulation equations for the multicylinder case is presented in [24, 25]. Note that for the present investigation, the emphasis is more on the elaborate simulation of the discharge line unsteady flows. Therefore, a number of simplifications, based on the previous investigations [17,18], are made in the basic compressor model. Theoretical engineering approximations, instead of the usual experimental inputs [18], are utilized to construct a simple mathematical model for the prediction of cylinder process operating variables. For the suction line, a steady and constant flow condition is assumed. Because of the strong dynamic interaction between the discharge line and valve, special attention has been given to the modeling of the discharge valves. It has simplified to the point where it can no longer predict details of the wrapping of the valve reed around the stop [11]. But, it is still accurate enough to predict

The mass flow rate through the valves. First mode approximation (a symmetrical one) for the valve ring beam, shown in Figure 1(a), is made as follows:

\[ u_{\text{ring}}(n) = A - B \cos 2\eta \]  

\[ u_{\text{beam}}(x) = \frac{2(A-B)}{L^2} x^2 - \frac{2(A-B)}{L^3} x^3 \]  

The mode shape of the valve ring is illustrated in Figure 1(b). The Rayleigh-Ritz procedure, using (1) and (2), is applied to calculate the first natural frequency of the valve. It also provides an estimation of a higher natural frequency corresponding to the symmetrical mode shape. The single mode approximation results in only small errors as only the first mode is dominant. Also, it is compatible with the frequency limitations dictated by pulsations. For the ring valving system, shown in Figure 1(c), an analytical procedure is utilized to determine the effective valve force and flow areas [19,20]. It should be noted that the accuracy does not suffer appreciably if the areas for normal and back flow conditions are considered to be the same.

DISCHARGE SYSTEM MODEL

Impedance Model

Discharge system flows can be considered to consist of two parts: the steady (mean) part and the unsteady (alternating flow) part which is exhibiting wave phenomena. Unsteady flows are modeled using acoustical formulations. A critical discussion of this modeling philosophy will not be presented here as it is already available in the literature [1-3, 5-15,15,17].

The mass flow rate through the discharge valve is considered to be the source excitation of the discharge system. Since the steady state modeling technique describes discharge characteristics in the frequency domain, the mass flow rate data has to be converted to the frequency domain. As the computer simulation provides mass flow rate data at discrete points, a finite Fourier series formulation gives:

\[ m'(+, n) = \sum_{n=1}^{N} m(n) | \cos(nw_0 + \psi_n(nw)) | \]  

where

\[ \psi_n(nw) = - \arctan \left( \frac{b_n}{a_n} \right) \]  

and

\[ a_n = \frac{2}{SP} \sum_{\alpha=0}^{N} \frac{\hat{m}(\theta_\alpha) \cos n\theta_\alpha}{n} \]  

\[ b_n = \frac{2}{SP} \sum_{\alpha=0}^{N} \frac{\hat{m}(\theta_\alpha) \sin n\theta_\alpha}{n} \]  

where SP is the total number of sampling points and N is the total number of harmonics required to describe the mass flow rate accurately.

Volume velocity \( \hat{\mathbf{x}}(nw) \) is defined as

\[ \hat{\mathbf{x}}(nw) = \frac{m(nw)}{p} \]
\( p \) is the discharge system mean fluid density. Now if the system impedances \( Z(\omega) \) are known, then by definition the pressure, \( p(n\omega) \), for the \( n \)th harmonic is

\[
p(n\omega) = \hat{x}(n\omega) Z(n\omega)
\]  

(9)

Note that \( \hat{x}(n\omega) \) and \( Z(n\omega) \) and \( p(n\omega) \) are all complex quantities. The total pressure (in time domain) \( p(t) \) at the location where volume velocity and impedances are known, is given by the sum of the steady and the unsteady parts.

\[
p(t) = \bar{p} + p(t) + \sum_{n=1}^{\infty} p(n\omega) \cos(nut + \psi_p(n\omega))
\]  

(10)

where \( \bar{p} \) is the steady mean flow pressure and \( p(t) \) is the d.c. component of the pressure in the frequency domain. The nominal discharge pressure \( p_{\text{nom}} \) specified as the compressor operating condition is

\[
p_{\text{nom}} = \bar{p} + p(0)
\]  

(11)

The Cylinder Discharge System Model

Gas pressure oscillations are more complicated in the two cylinder discharge system because of the cylinder interactions. These interactions can be thought of as consisting of the two following types of coupling.

**Kinematic coupling.** The kinematic arrangement of a two cylinder compressor is generally such that the instantaneous processes in one cylinder are either ahead or behind the other cylinder, depending upon the crank phasing (9). Thus during a cycle, discharge mass flow rates of both cylinders will not be simultaneous. The time delay corresponds to the kinematic phasing (\( \phi \)) between the cylinders.

**Cavity coupling.** The mean fluid flow follows the pattern of cavities and is pumped downstream. But, waves not only propagate down the discharge line of a cylinder but also travel back and forth through the connecting components between the two cylinders. Thus the pressure at a particular valve exit is not only affected by its own mass discharge but is also affected by the mass discharge of the other cylinder.

Kinematic coupling alone would not cause any dynamic interactions. Along with the cavity couplings, it produces a complex oscillatory behavior and both constructive and destructive types of interference take place.

As it is desirable to keep the kinematic phasing a variable, it should be separate from the discharge system acoustical characteristics. It can be incorporated with the volume velocity. For the calculation of the pressure at the valve exits, refer to Fig. 2(a), an input impedance matrix is proposed. Fig. 2(b) illustrates the formulation of the problem in terms of the input impedance matrix which consists of input or driving point impedances \( Z_{ii} = p_{i}/\hat{x}_{i} \) and transfer or cross impedances \( Z_{ij} = p_{i}/\hat{x}_{j} \).

\[
\begin{bmatrix}
p_{1}(n\omega) \\
p_{2}(n\omega)
\end{bmatrix} =
\begin{bmatrix}
Z_{11}(n\omega) & Z_{12}(n\omega) \\
Z_{21}(n\omega) & Z_{22}(n\omega)
\end{bmatrix}
\begin{bmatrix}
\hat{x}_{1}(n\omega) \\
\hat{x}_{2}(n\omega)
\end{bmatrix}
\]  

(12)

where

\[
\begin{align*}
\bar{\xi}_{1} &= \omega t \\
\bar{\xi}_{2} &= \omega t + \phi
\end{align*}
\]  

(13)

Similarly for the pressure calculations at point (a), i.e. any other point inside the system, a transfer impedance matrix is proposed. Refer to Fig. 2(c).

\[
p_{a}(n\omega) = \begin{bmatrix} Z_{a1}(n\omega) \\
Z_{a2}(n\omega)
\end{bmatrix}
\begin{bmatrix}
\hat{x}_{1}(n\omega) \\
\hat{x}_{2}(n\omega)
\end{bmatrix}
\]

\[
\begin{bmatrix}
X_{11}^{(n\omega)} \\
X_{12}^{(n\omega)}
\end{bmatrix}
\begin{bmatrix}
\bar{\xi}_{1} \\
\bar{\xi}_{2}
\end{bmatrix}
\]  

(14)

Note that the transfer impedance matrix consists of only cross impedances.

**Impedance Computations**

For the computation of these impedance elements, acoustical transmission line formulations are used [21,22]. Since the interest lies only in the lower harmonics of running speed, a plane wave distributed parameters description is adequate. In this context, a fourpole solution technique [21,22] is the best method of analyzing any composite acoustic system as a whole because of the simpler mathematical operations. It is a 'building block' type of approach. The fourpole elements \( A, B, C \), and \( D \) represent relationships between input and output, as illustrated in Figure 2(d).

<table>
<thead>
<tr>
<th>Table 1. Fourpole Elements of Various Systems</th>
</tr>
</thead>
<tbody>
<tr>
<td>System/Configuration</td>
</tr>
<tr>
<td>---------------------------------------------</td>
</tr>
<tr>
<td>Uniform tube of length ( \lambda ) and cross sectional area ( S )</td>
</tr>
<tr>
<td>Impedance ( Z ) in parallel</td>
</tr>
<tr>
<td>Impedance ( Z ) in series</td>
</tr>
<tr>
<td>Two cylinder compressor discharge system impedance matrix elements</td>
</tr>
</tbody>
</table>

\[
\begin{bmatrix}
p \\
\hat{x}
\end{bmatrix}_{\text{Input}} =
\begin{bmatrix}
A & B \\
C & D
\end{bmatrix}
\begin{bmatrix}
p \\
\hat{x}
\end{bmatrix}_{\text{Output}}
\]

As both the fourpole element matrix and the impedance matrices represent system characteristics, relationships can be developed between them. Table 1 summarizes some of the fourpole elements, utilized in the present investigation. Reciprocity relationships for impedance matrices, and the properties of symmetrical fourpoles can be further used to simplify algebra.

The propagation constant \( \gamma \) is a complex quantity and is given as

\[
\gamma = \beta + jk = \beta + j \frac{\omega}{c}
\]  

(16)

where \( \beta \) is the damping factor and \( k \) is the wave number. The expressions for linear viscous and thermal boundary layer dissipations are readily available [21, 22]. However, these may not be sufficient as nonlinear fluid-induced frictional effects could be witnessed [21].

Additional kinetic energy effects, related to the velocity field due to the disturbing influence of the discontinuity, have to be incorporated in the mathematical models [21,22]. This additional energy can be
described in terms of a certain fictitious attached mass [21,22]. Note that a detailed presentation of both nonlinear damping and attached mass effects is beyond the scope of this paper.

COMPUTATIONS

The mathematical models for the compressor are described in the time domain, but can be converted to the theta domain for the prediction of cyclic variations. The theta domain also accounts for the kinematic coupling between the cranks of both the cylinders.

The discharge system modeling philosophy incorporates an iterative procedure as illustrated in Fig. 3. The iteration is started by assuming the total discharge pressure to be equal to the nominal discharge pressure. The compressor model is simulated to generate the instantaneous mass for each step size. And then it is integrated over a cycle to determine the total mass flow rate which now acts as the source function for the discharge system. A finite Fourier series routine is used to convert the mass flow cyclic variations into harmonic data. The iteration is continued until the harmonics of the simulated discharge pressures do not vary significantly, from one iteration to another.

By comparing mass flow rates, different step sizes were explored for obtaining an optimum size. For the present case, the step size was finally determined to be 1/32, as sizes smaller than this brought negligible increase in the accuracy. However, it cannot be generalized as the step size is a strong function of the valve natural frequencies, and needs to be determined for each case. A Runge-Kutta subroutine could also be used to adjust the step size throughout the program.

EXAMPLE CASE AND MEASUREMENTS

Measurements were conducted as a part of the investigation to verify the validity of the simulation program. The example case is depicted in Fig. 4. The example case is an in-line type of a reciprocating refrigeration compressor. The discharge system is symmetrical for both cylinders. It consists of the following components: discharge plenum, connecting passages, collector, exit piping and muffling system. The anechoic line is connected to the end of the discharge system. The use of an anechoic termination enables the discharge system to be isolated acoustically from the downstream components e.g. condenser.

Instrumentation for the measurement of the following is shown in a block diagram format in Fig. 4.

1. cylinder pressure cyclic variation
2. discharge plenum (behind valves) pressure cyclic variation
3. discharge plenum pressure spectrum
4. anechoic termination (discharge system end)
   (pressure cyclic variation)
5. anechoic termination pressure spectrum

Table 2. Example Case Operating Conditions

<table>
<thead>
<tr>
<th>Fluid medium</th>
<th>R-12</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kinematic phase</td>
<td>180°</td>
</tr>
<tr>
<td>Cylinders</td>
<td>2880 rpm (59.6 Hz)</td>
</tr>
<tr>
<td>Nominal suction pressure</td>
<td>41 psi (2.82 bar)</td>
</tr>
<tr>
<td>Nominal suction temperature</td>
<td>61°F (16.1°C)</td>
</tr>
<tr>
<td>Nominal discharge pressure</td>
<td>183 psi (12.62 bar)</td>
</tr>
<tr>
<td>Nominal discharge temperature</td>
<td>194°F (37.4°C)</td>
</tr>
</tbody>
</table>

Three piezoelectric crystal type pressure transducers were installed in the system to measure the above mentioned. A sixty tooth gear with an electromagnetic pick-up was mounted on the crankshaft to establish the crank angle position. A minicomputer based digital system, with a Fast Fourier Transform (FFT) capability, was utilized for data acquisition and processing. The operating conditions for the example case are listed in Table 2.

Uncertainties in the measurements are mainly due to (a) pressure transducer sensitivities and (b) digital data processing procedures. The experimental data for the present case has the following uncertainty values:
1. Time variant pressures: ± 5%
2. Sound pressure levels: ± 1 dB
3. Crank position marking: ± 1°

RESULTS AND DISCUSSION

The validity of the simplifications involved in valve modeling is tested by comparing the present results with the elaborate valve modeling efforts of Elson & Soedel [11]. For both models, discharge plenum pressures are compared for a single cylinder case. They match each other quite well (Fig. 2).

The convergence of the simulation program is quite fast as demonstrated in Fig. 6. Although only three iterations were sufficient for fairly good results, eight iterations have been used to get the final results.

Comparisons between theoretical and experimental results are presented in Fig. 7-11. Fig. 7, 8 and 9 trace the pressure time history of the refrigerant flow in the cylinder, discharge plenum (behind valves) and anechoic termination (piping end) over one cycle of compressor operation. Theoretically computed discharge valve displacements are also plotted on the same figures to provide a correlation between gas pressure and valve openings. Cylinder cyclic pressure variation as shown in Fig. 7 agrees very well during the discharge valve opening but there is a slight discrepancy during the suction process. But, it is compatible with the research goals and objectives. For better agreement during the suction, the same attention should be given to the modeling of the suction valve and plenum. Also, recall that discharge valves are modeled entirely analytically. A better agreement would certainly demand more experimental input [17,18]. Measured cylinder pressure is affected by the dynamics of the cavity in front of the transducer during expansion phase. From discharge plenum pressure cyclic variations, Fig. 8, it is seen that only three cycles of gas oscillations take place during one cycle of the compressor process. It indicates that the third harmonic is dominant at the valve exit. The anechoic termination pressure, Fig. 9, is more or less steady and does not show any substantial pulsations. Measured and predicted pressure frequency spectra are presented in Fig. 10 and 11. Fig. 10 demonstrates that the lower harmonics are dominant in the discharge plenum and in fact the third harmonic is the most dominant. Fig. 11 shows that only first and second harmonics are still strong at the anechoic termination; the rest have been attenuated by either inner cavity interferences or downstream mufflers.

Although a limit on the upper frequency prediction was placed at around 700 Hz, the results demonstrated here are shown for 1000 Hz (20 harmonics) for discharge plenum pressure and 900 Hz (15 harmonics) for the anechoic line pressure. All of these are still in the plane wave propagation region as the first cutoff mode for the largest diameter in the present example case was around 1350 Hz.
The comparisons between measured and predicted results show excellent agreements. The discrepancies between computed and measured spectra could very well be due to some measurement limitations. The computed spectra dynamic range could be beyond the instrumentation dynamic range (60-70 dB). Anechoic termination spectra levels were so low that the pressure signals were almost buried in the flow noise, and only an ensemble spectra averaging could unearth the signals. Furthermore, flow-induced oscillations may have been excited.

**Lumped Parameters Analysis**

The computer simulation program demonstrated that the geometric dimensions of the inner cavities are critical in determining the overall dynamic behavior. While the distributed parameters modeling is a good tool for precise calculations, lumped parameters analysis provides better physical insights into the system. To supplement the above-mentioned results with a clear picture of the dynamics of cavities, a lumped or discrete parameters modeling approach is attempted here. It differs from [15] as in [15] pressure differential is the forcing function but here the volume velocity (∇v) is the source excitation for the system. The reason for choosing such a function is that it is consistent with the distributed parameters modeling. Furthermore, the present available information [15,17] is advanced here by performing a modal expansion analysis for forced oscillations (This will be the subject of a future paper).

Fig. 12(a) shows the schematic of the discharge system, consisting only of inner cavities. Its mechanical analog is presented in Fig. 12(b). An eigenvalue problem formulation provided natural frequencies and modes of gas oscillations. These are documented in Table 3 for a symmetrical discharge system (present example case). Modes of gas oscillations are also shown in Fig. 12(c).

**Table 3. Natural Frequencies and Mode Shapes of a Symmetrical Discharge System (Refer Figure 12(a), (b) & (c))**

<table>
<thead>
<tr>
<th>Natural Frequency</th>
<th>Mode Shape (X₄/X₆)</th>
</tr>
</thead>
<tbody>
<tr>
<td>First</td>
<td>$c \sqrt{\frac{S_4}{k_4 \cdot V_3}}$</td>
</tr>
<tr>
<td>Second</td>
<td>$c \sqrt{\frac{S_4}{k_4 (\frac{1}{3} + \frac{2}{5})}}$</td>
</tr>
</tbody>
</table>

**Table 4. Effect of nth Harmonics Phasing on Symmetrical Discharge System Response**

<table>
<thead>
<tr>
<th>Phase</th>
<th>System Response</th>
</tr>
</thead>
<tbody>
<tr>
<td>0°</td>
<td>Only compressive mode excited</td>
</tr>
<tr>
<td>180°</td>
<td>Only sloshing mode excited</td>
</tr>
<tr>
<td>In between 0° and 180°</td>
<td>A combination of compressive and sloshing modes excited</td>
</tr>
</tbody>
</table>

The effect of kinematic phasing has been demonstrated with the use of the modal expansion technique. Table 4 presents the effect of phase difference between the nth harmonic individual pressure oscillations of both cylinders (as demonstrated by elements 4 and 6 in Fig. 12). Thus the interference mechanisms are explained.

Furthermore, the modal expansion reveals that out of the two natural frequencies in the present example case, only the first one is a resonance—because only the sloshing mode is excited in the system. The first natural (sloshing) frequency is very close to the third harmonic, and that is why it is dominant in the discharge plenum. It matches well with the results obtained by the simulation program using the distributed parameters approach.

**CONCLUDING REMARKS**

The major accomplishment of the investigation has been the development of a modeling philosophy for the two cylinder compressor discharge systems. Cylinder interference mechanisms are defined and classified into kinematic and cavity type of interactions. Also, the steady state impedance function approach has been established as a sound solution technique. It has been supplemented by a lumped parameters analysis of the discharge system inner cavities. Both elaborate modeling efforts and approximate analyses have pointed to the same direction, and critical design parameters have been identified. Perhaps more fundamental importance has been the question of using linear acoustic theories for such complicated unsteady flow processes. The answer to this certainly lies in the excellent agreements between theoretical and experimental results. At this point, it should be mentioned that linear acoustics is adequate for most cases. However, one has to identify and incorporate individual nonlinear effects into the easily analyzable linear formulations.

The investigation has mainly concentrated on the prediction of primary outputs such as gas pressures and valve displacements. The following secondary outputs can be related to and generated from the primary outputs: capacity, volumetric efficiency, performance, pulsations and vibration levels and stresses, etc. The building block approach of modeling discharge components makes the simulation program versatile. In the present format, any intermediate muffler element can be easily analyzed. Thus it is a good tool for design and performance studies.

Although the modeling procedure is presented only for a two cylinder discharge system, it can be extended to multicylinder (any arbitrary number of cylinders) compressor applications [25]. Also, the discharge system formulations can be readily applied to suction systems as well. Furthermore, it is interesting to note here the technique presented in the paper could be extended to any multicylinder/intercooled positive displacement fluid machinery.

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FIGURES

(a) MODE SHAPE FOR THE VALVE RING

(b) SCHEMATIC OF A TYPICAL RING TYPE VALVE

(c) SCHEMATIC OF A TYPICAL RING TYPE VALVING SYSTEM
FIGURE 7. CYCLIC PRESSURE VARIATION IN CYLINDER

FIGURE 8. CYCLIC VARIATION OF DISCHARGE PLENUM PRESSURE

FIGURE 9. CYCLIC VARIATION OF ANECHOIC LINE PRESSURE

FIGURE 10. DISCHARGE PLENUM PRESSURE FREQUENCY SPECTRUM

FIGURE 11. ANECHOIC LINE PRESSURE FREQUENCY SPECTRUM
Figure 12. Illustration of cavity interactions & modes of gas oscillations by lumped parameters approach.