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A Digital Reciprocating Compressor Simulation Program Including Suction and Discharge Piping

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1. ABSTRACT

A digital reciprocating compressor simulation program has been developed which fully accounts for cylinder thermodynamics, inlet and discharge valve dynamics, and pressure pulsations in suction and discharge piping in an integrated manner. The piping network can extend to any length and may include typical elements such as orifices or choke tubes. The program is valid for any mixture of gases and most major types of valves. The program's structure permits global simulation of the complete compressor system or individual components such as compressor or piping separately. Many other programming features make it a very versatile and user-friendly program. This paper presents functional details of the program and summarizes the underlying theory. Some test results and their comparison with the program's predictions are also presented.

2. INTRODUCTION

Reciprocating compressors have been used in industry for gas compression longer than any other type of machinery. The performance prediction for such compressors has always been based on a mixture of rational analysis and a good measure of empiricism. However, high energy costs and tight market competition demand not only better performance prediction capability but also better understanding of valve dynamics and compressor-piping interaction. Several investigators (1 - 3) have suggested methods to calculate cylinder performance, valve dynamics, or piping pulsations individually. Others (4 -7) have combined many of these elements and applied them to a particular compressor. However, a general scheme for complete simulation of compressor-piping systems in the industrial environment has always been lacking.

This paper presents a user-oriented, general-purpose, reciprocating compressor simulation program which fully accounts for cylinder thermodynamics, valve dynamics and interaction of inlet and discharge piping with the cylinder. Despite the program's complex logic and mathematical sophistication, the program is structured to be flexible, versatile and extremely user-friendly as explained later. These features were deemed absolutely necessary if a large simulation program was to become a working tool for day to day use.

Digital simulation of steady-state pulsations in piping is one important feature of this program. Analog simulation is currently a standard industry practice for large compressor applications. While the analog simulation(14) has its own advantages, the digital simulation proposed here has many of the same merits plus many others. The digital method, as presented here, is rapid and flexible, but requires little set-up time or training to use. The method is based on computing four-pole matrices for each element and combining all the matrices in a proper order to yield system impedance matrices for different points along the piping network. The program can also compute system natural frequencies in order to help identify the source of high pulsations.

This program provides details of both inlet and discharge valve dynamics including valve HP loss, seat and stop plate impact velocities, and valve opening and closing angle. Valve motion studies are gaining increasing importance in the industry and analytical methods have become indispensable in valve design and selection, particularly for problem machines. Special emphasis was given to the valve dynamics part of the program and a steady flow tester was used to examine flow and lift characteristics of different types of valves. The program's results have been checked against test data with good agreement.

3. METHOD OVERVIEW

Figure 1 shows a schematic of a general compressor-piping system that can be analyzed by the program's use. The program is divided into two distinct parts: compressor simulation and piping simulation. The intermittent flow through compressor valves acts as a driving force or the forcing function to generate pulsations in the piping. These pulsations then influence valve motion and gas pressure in the cylinder and therefore modify the flow through valves in turn. This iterative process proceeds in the following fashion:

o Calculate cylinder pressure, valve motion, and flow through valves assuming constant nominal inlet and discharge pressures.

o Calculate natural frequencies of the inlet or discharge piping system and their driving impedances at valve cover and at other defined points along the system.

o Calculate pressure-time response at the valve covers.

o Recalculate cylinder pressure, valve motion and flow through valve based on the new valve cover pressures.

o Use previously calculated impedance matrices to determine pressure response at the valve covers and follow the above steps iteratively until valve cover pressures converge. Typical convergence is achieved in three of four iterations.

The above procedure is illustrated through a flow chart in Figure 2. The program also offers various other options to perform only selected tasks. For example, the driving force for the piping system can be based on a valveless, ideal cylinder analysis similar to the one used in analog simulation.

The gas properties are calculated by either the BWR method or the Nelson-Obert method depending on the type of gases in the gas mixture. The first law of thermodynamics is used in computing cylinder pressure and single degree of freedom systems are used in valve motion analysis. The program is valid for many kind of valves such as channel or plate and all practical gas mixtures.

4. THEORY

Only a brief description of theory is presented here because of space limitations. In addition, many of the basic equations have been reported in the literature earlier (10 - 12).

4.1 Gas Properties

A data base of about 80 individual gases common to process industry is used to calculate thermodynamic gas properties such as compressibility and specific heat at a given pressure and temperature. The BWR method is used to find properties of individual or a mixture of gases including most hydrocarbons, CO_2, N_2, and O_2 while the Nelson-Obert method is used for the rest. Thus,

$$Z = Z(P, T; P_{cr}, T_{cr}) \tag{1}$$

The gases are assumed to follow the perfect gas law with compressibility correction as follows:

$$P = \rho R Z T \tag{2}$$

Real gas effects are taken into account by redefining the adiabatic compression exponent γ as follows:

$$\gamma = \ln r_c / \ln(r_c^k z_s / z_d) \tag{3}$$

The speed of sound in the gas is determined from the following relationship:

$$c = \sqrt{\gamma_L g R Z T} \tag{4}$$

The symbols are explained in the Appendix.

4.2 Cylinder Thermodynamics

The first law of thermodynamics as explained in Reference 8 is used to calculate instantaneous cylinder pressure and indicated horsepower.

$$\frac{dP_c}{dt} = \frac{\gamma - 1}{V_c}\dot{q} - \frac{\gamma P}{V_c}\frac{dV_c}{dt} - \frac{\gamma P}{V_c}\dot{Q}_o + \frac{\gamma}{V_c}\sum P_i \dot{Q}_i \tag{5}$$

The volume change rate as a result of piston motion is given by:

$$\frac{dV_c}{dt} = \frac{\pi D_c^2 r \omega}{4}\left[-\sin\theta + \frac{r}{2l_c}\frac{\sin 2\theta}{\sqrt{1 - \frac{r^2}{l_c^2}\sin^2\theta}}\right] + A_{fd}\dot{x}_d - A_{fs}\dot{x}_s \tag{6}$$

The last two terms in (6) represent the volume change as a result of valve motion. This effect normally ignored by most investigators affects valve motion during opening and closing phases and the impact velocities. Flow through valve ports or due to ring leakage is determined from the usual compressible flow through an orifice equation as follows:

$$\dot{Q} = CA\sqrt{\frac{2\gamma_L g R T_{us}}{(\gamma_L - 1)}}\sqrt{(r_o)^{\frac{2}{\gamma_L}} - (r_o)^{\frac{\gamma_L + 1}{\gamma_L}}}\ ;\ r_o = \frac{P_{ds}}{P_{us}} \tag{7}$$

C is an orifice or flow coefficient and A is the nominal area. The product CA is called effective flow area and its value for inlet or discharge valve is determined experimentally from steady state flow tests. The effective flow area is a function of valve lift and the direction of flow. In the program, the flow area varies as a polynomial function of lift, x, as follows:

$$C(x) = \sum_{0}^{4} C_i x^i \tag{8}$$

The coefficient C_i may be different for backflow (reverse to the normal direction). The steady flow assumption made here is not strictly valid during valve opening and closing but it primarily affects impact velocities unless the valve flutters a great deal.

Heat transfer between the gas in the cylinder and cylinder walls is calculated from the following relationship:

$$\dot{q} = h A_\omega (T_c - T_\omega) \tag{9}$$

The heat transfer coefficient, h, can be based on several proposed models but Adair et al correlation (15) has specifically been proposed for compressors. The various models are currently being examined and a new model based on our own testing is under consideration. Heat transfer has generally been neglected for test cases up to now.

4.3 Valve Dynamics

A single degree of freedom, damped mass-spring model is used to describe valve motion. Different types of valves - single plate or buffered plate, channel, multiple ring - can be modeled. For a multiple element valve, the mass of spring rate of all elements are appropriately added. In reality, all elements do not act in phase and even in the case of a single element such as a ported plate, the lift or spring rate may vary along the circumference. However, the single degree of freedom model predicts average behavior with good accuracy for practical use.

The following equation is used to calculate valve motion:

$$m\ddot{x} + D(\dot{x}) + K(x) = F(x) \tag{10}$$

The spring force, K, and the damping force, D, are generally nonlinear in nature as follows:

$$K(x) = \sum_{0}^{4} K_i x^i \tag{11}$$

$$D(x) = D_0 \frac{\dot{x}}{|\dot{x}|} + D_1 \dot{x} + D_2 \dot{x}|\dot{x}| \tag{12}$$

For channel valves which use arch springs, a special relationship for spring constant is used (9) as given below:

$$K(x) = K_0 + K_1 x + K_2 x^2 + K_3 / \sqrt{(1 - x/x_{max})} \tag{13}$$

The values K_i are determined from actual spring measurements under steady loading. Damping parameters D_i are based on the nature of damping in the valve and matching of computer predictions with test data. For example, for channel valve, damping constants as reported in (9) are:

$$D(\dot{x}) = 3.5 \rho A \dot{x} + 92 \rho A \dot{x}|\dot{x}| \tag{14}$$

The force trying to open or close the valve is determined by multiplying effective force area with the net pressure difference ΔP across the valve. The effective force area is determined by multiplying nominal plan area with a force coefficient determined experimentally from steady force tests as follows:

$$F(x) = \left(\sum_{0}^{4} F_i x^i\right) A_f \Delta P \tag{15}$$

The boundary constraints on valve element motion are defined by,

$$\dot{x}(t)\big|_{st} = -\xi_{st}\, \dot{x}(t-\Delta t)\big|_{st} \tag{16}$$

$$\dot{x}(t)\big|_{se} = -\xi_{se}\, \dot{x}(t-\Delta t)\big|_{se} \tag{17}$$

In the case of buffered valves, additional constraints apply.

$$\dot{x}(t)\big|_{bu} = \dot{x}(t-\Delta t)\big|_{bu} \frac{m_v}{m_v + m_{bu}} \tag{18}$$

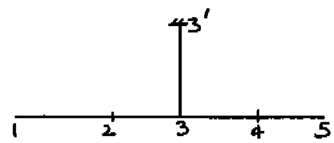
The equations (5) to (18) are then integrated by a two-step modified Euler method.

4.4 Piping Dynamics

The piping system impedance matrix is calculated by properly combining four-pole matrices of each element. The four-pole matrix for a pipe element is given by:

$$M_{12} = \begin{bmatrix} A_{12} & B_{12} \\ C_{12} & D_{12} \end{bmatrix} = \begin{bmatrix} \cosh \beta L & \frac{j\omega A_p}{\rho_p c_p^2 \beta} \sinh \beta L \\ \frac{\rho_p c_p^2 \beta}{j\omega A_p \sinh \beta L} & \cosh \beta L \end{bmatrix} \tag{19}$$

For a series of elements as shown in the figure below, the matrices can be combined to link the ends 1 and 5 as follows:



(20)

$$P_{3'3} = \begin{bmatrix} 1 & B_{3'3}/D_{3'3} \\ 0 & 1 \end{bmatrix} ; \beta = \alpha + j\frac{\omega}{c_p}$$

$P_{3'3}$ is the matrix for the junction element 3 given by:

$$P_{3'3} = \begin{bmatrix} 1 & B_{3'3}/D_{3'3} \\ 0 & 1 \end{bmatrix} \tag{21}$$

The pressures and flows in the frequency domain can be linked by,

$$\begin{bmatrix} Q_1(n\omega) \\ P_1(n\omega) \end{bmatrix} = M_{15} \begin{bmatrix} Q_5(n\omega) \\ P_5(n\omega) \end{bmatrix} \tag{22}$$

Further details of this technique are available in References 10 and 12. There is also a direct relationship between an impedance matrix and a four-pole matrix. The program calculates impedance matrices for different points along the piping system only once and multiplies the flow with the impedance to yield pressure harmonics in the frequency domain as follows:

$$\begin{bmatrix} P_1(n\omega) \\ P_5(n\omega) \end{bmatrix} = \begin{bmatrix} Z_{11}(n\omega) & Z_{15}(n\omega) \\ Z_{51}(n\omega) & Z_{55}(n\omega) \end{bmatrix} \begin{bmatrix} Q_1(n\omega) \\ Q_5(n\omega) \end{bmatrix} \tag{23}$$

The pressure-time history is obtained by the inverse Fourier transform of (23).

5. DIGITAL PIPING SIMULATION

While the basic theory for piping simulation is presented in the previous section, further details on the use of piping simulation part of the program are presented here. The piping can be made of one of the following elements:

- o Pipe elements
- o Junction elements
- o Orifice or choke tubes

The junction element is just the point where the piping branches off. The boundary conditions for the piping ends could either be closed, open (large volume), or a non-reflecting end (long pipe of a fixed diameter). The user only has to enter the dimensions of various elements in a prescribed manner. The program is structured in such a way that other special elements such as perforated pipes can be easily incorporated if their four-pole matrix relationships are known. The convective effect of mean velocity has been neglected since the mean velocity for typical compressor applications is much smaller than the sonic velocity.

The impedance technique used here was selected after a thorough review of all the available techniques. This method was found to be rapid, flexible and reasonably accurate. The typical simulation run using ideal valves for one fixed set of operating conditions takes only 10 seconds of CPU time on the PRIME 750 computer.

The user can automatically scan a certain RPM range at specified steps. It is trivial to change the dimensions of one or more elements and it is quite easy to insert additional elements or remove some others.

The program was first verified against several test cases reported in the literature. Figure 3 shows the experimentally determined natural frequencies for a piping configuration as given in Reference 13 and the corresponding calculated frequencies using the above method. The results indicate excellent agreement between the two. Figure 4 gives a comparison of the pulsation pressure spectrum given in Reference 12 for the piping configuration shown therein and that calculated by the author. Although the latter results are based on ideal valve analysis, the agreement is quite good. Figure 5 summarizes the results of analog simulation and digital simulation at different points along the piping configuration shown. The analog simulation was carried out at the gas compressor division of the author's company using a Southern Gas Association (SGA) analog simulator. The agreement again is quite good.

6. PROGRAM FEATURES

As mentioned before, special emphasis was given to make the program easy yet versatile for the user. This section outlines some of the uncommon features of the program.

6.1 Program Input

The first time, input data is entered by the user interactively in response to a series of clear, concise questions. When the answers require some judgement on the user's part, the program provides helpful guidelines along with the question. If the data is not of the correct type or out of a valid range, the program asks for correct data again. Gas data is available on files while valve data can be entered manually or from data files. If valve data files are available, it will require typically less than 5 minutes to enter and run the program for compressor simulation without piping analysis.

The piping data is also entered interactively in a predefined logic. The logic is set in such a way that the program in the future may be able to read the data directly from computerized piping files like those stored in CAD/CAM systems. All input data once entered is available on the screen as well as in files for user review. Once the data has been entered, the program automatically stores data for future use.

6.2 Multiple Capability

The program is designed in a way as to allow different users to perform different tasks independent of each other. For example, the program can be used anywhere from a simple calculation of speed of sound for a gas mixture to complete simulation of a compressor-piping system. In each case, the program generally requires just enough information to do the particular task and performs only the needed computations. Figure 6 shows many capabilities of the program.

As another example, piping natural frequencies can be determined separately from the compressor simulation which can help in the selection of valve plates or springs as to keep the valve's natural frequencies apart from the piping's. Similarly, the piping sumulation can be performed before the final valve selection by assuming ideal valves in order to get an estimate of the expected pulsation levels.

6.3 Program Output

The program's output is available in several forms as follows:

o Output data files
o Graphic plots
o Screen display

A separate data file is generated for each different type of output such as compressor performance or piping pulsations. Furthermore, several hierarchial levels of output are available for each type; the lowest level gives detailed output at every integration step while the highest level contains a summary of a few important results. In this way, the user has access to all the information yet it is segmented to suit his particular needs.

All data that varies with time including PV diagram, valve lift and velocity, valve cover pressures, flow through valve, is available in the form of high-resolution plots on a Tektronix terminal along with hard copies. The user can design his own plots by selecting any two variables from a menu and which axis each goes on. In addition, frequency spectrum plot and pressure-time plots of piping pulsations are also drawn as a user option. A residue vs. frequency plot is available to verify the natural frequency numbers. At these frequencies, the residue is zero and changes sign after zero cross-over. Overall, about 100 meaningful plots can be drawn in a report-ready format. The program also displays other useful output data on the screen if the user wishes.

6.4 Modular Structure

The program is divided into about 80 modules or subroutines each of which performs a distinct but limited function. The subroutines can be added or taken out with great ease since they are linked to each other through common communication files. For instance, the digital piping simulation subroutines are connected to the rest of the simulation mainly through the driving source: flow through valves. If another driving source such as a reciprocating or a centrifugal pump was linked to the piping subroutines, the program can easily calculate pulsations due to the new source.

6.5 HELP! Function

If the user is still stymied, help is available at the touch of his fingers. The user need only type HELP and the program is likely to answer his specific questions. The menu-driven HELP function also guides a new user in explaining the program's features and how to use it most efficiently. HELP contains notes on program updates and helpful hints based on other users' experience. The program has no manuals to consult and no data sheets to fill.

The program helps integrate many functions such as performance calculation or piping simulation, typically handled by separate departments. Also, by performing these functions at various levels of complexity, it meets varying needs of different groups such as R & D, Engineering and Marketing.

7. COMPARISON WITH TEST DATA

The program is currently valid for a single-stage, single- or double-acting single cylinder compressor with unrestricted inlet and/or discharge piping. Its results have been checked against several compressors. This correlation work is still continuing and one of its purposes beyond checking the program's validity is to understand the causes for compressor performance losses.

In addition, the test data provides guidance for establishing or refining theoretical models for heat-transfer, valve dynamics, and acoustic damping in pipes.

The test data described here was obtained in the laboratory under carefully controlled conditions. The cylinder pressure was measured with a Kistler pressure transducer equipped with a switching adapter and referenced to the atmosphere just prior to the reading. Ported plate type valves with buffer plates were used both for inlet and discharge valves. The valve motion was recorded with four proximity probes equally spaced along the outer periphery. The pressure pulsations were measured by Kulite pressure transducers at several points along the inlet and discharge piping.

Figure 7 shows measured and calculated PV diagrams for the outer end of a double-acting 7.5 bore x 4.5 in stroke compressor running in a test loop at 847 rpm and compressing N_2 from 105 psia inlet pressure to 159 psia discharge pressure. Figure 8 and 9 show measured and calculated valve displacements for the same case. All the four proximity displacement probes showed nearly identical valve motion patterns in this particular test. Figure 10 shows a plot of the calculated suction valve velocity vs. valve lift. In this plot, the valve plate lifts until it hits a buffer plate and the velocity is damped. Both the valve and buffer plates then lift together until they hit the stop plate. The cycle is completed when the valve plate closes. The valve plate bounces back on hitting the stop plate and closes again as indicated by the second smaller cycle on the plot.

Figure 11 gives a comparison of the measured and calculated frequency spectrums at two points in the test loop. The agreement between the test data and calculations is reasonably good. The results at the valve cover show a slight shift in frequencies corresponding to pressure pulsation peaks which may arise from the differences in calculated and actual speeds of sound. If rpm or the speed of sound is varied over a small range as done in analog simulation, the agreement is likely to get even better. In addition, the complicated passageways in the compressor manifolds were grossly simplified to one pipe element in this example.

8. CONCLUSIONS

This paper has outlined the development of a comprehensive, interactive compressor-piping system simulation program. The program has many special features which make it versatile and multifaceted yet easy to use. It is currently being checked against the test data. Some test comparisons were presented which show that the program does simulate a compressor's test behavior quite reasonably.

ACKNOWLEDGEMENTS

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APPENDIX

Symbols

A	Nominal valve flow area
A_f	Nominal valve force area
A_p	Pipe element cross-sectional area
A_w	Cylinder wall area for heat transfer
c	Speed of sound in the gas
D	Damping force
D_c	Cylinder bore
D_i	Damping coefficients
F_i	Lift force coefficients
g	Gravitational constant
h	Overall heat transfer coefficient
j	Square-root of -1
k	Specific heat ratio
K	Spring force
K_i	Spring force coefficients
l_c	Connecting rod length
L	Pipe element length
m	Mass in (10); may vary with lift
m_v	Valve element mass
m_{bu}	Buffer element mass
n	Harmonic number
P	Gas pressure
P_c	Cylinder pressure
P_{cr}	Critical pressure for the gas
$P(n\omega)$	nth harmonic of pressure pulsations
$Q(n\omega)$	nth harmonic of flow variation
r	Crank radius
r_c	Nominal compression ratio
r_o	Pressure ratio across orifice in (7)
R	Gas constant
$\dot{q}$	Heat transfer into the cylinder
$\dot{Q}$	Volume flow rate through an orifice
$\dot{Q}_i$	Volume flow rate into the cylinder
$\dot{Q}_o$	Volume flow rate out of the cylinder
t	Time
T	Gas temperature
T_c	Cylinder temperature
T_{cr}	Critical temperature for the gas
T_{us}	Temperature upstream of orifice
T_w	Cylinder wall temperature
V_c	Cylinder volume at time t
x	Valve displacement
x_{max}	Maximum valve displacement
$\dot{x}$	Valve element velocity
$\ddot{x}$	Valve element acceleration
Z	Gas compressibility
$Z(n\omega)$	Piping impedance for the nth harmonic
α	Piping damping factor
γ	Adiabatic compression exponent
ω	Angular velocity
ρ	Gas density
Σ	Summation over subscript i
θ	Crank angle
ζ	Rebound coefficient at impact
Δt	Small time interval

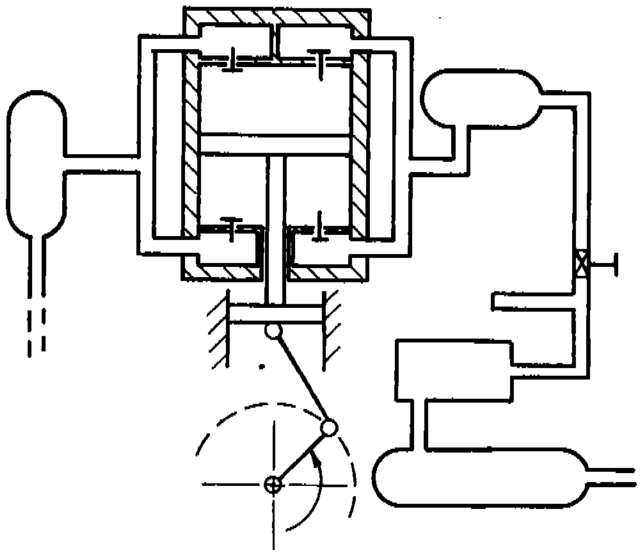
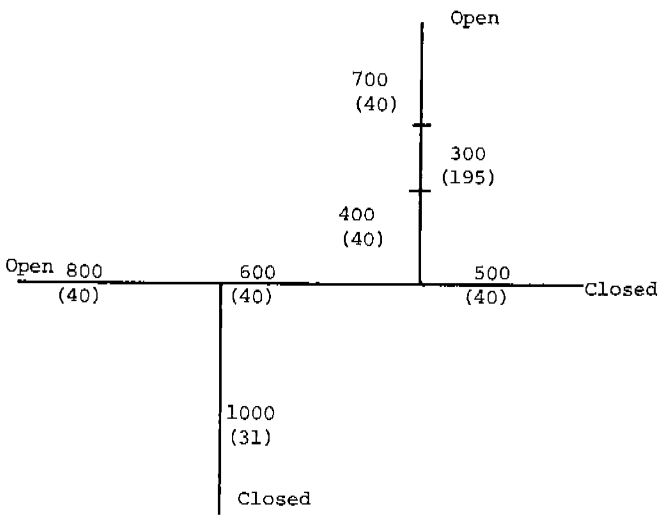


Figure 1 Compressor and Piping System Schematic



Order	Experimental (12)	Calculated
1	26	26.3
2	63	62.3
3	101	100.8
4	145	144.7

Figure 3 Comparison of experimental and calculated natural frequencies for the piping diagram shown. The dimensions shown in mm. are length and diameter (in parenthesis).

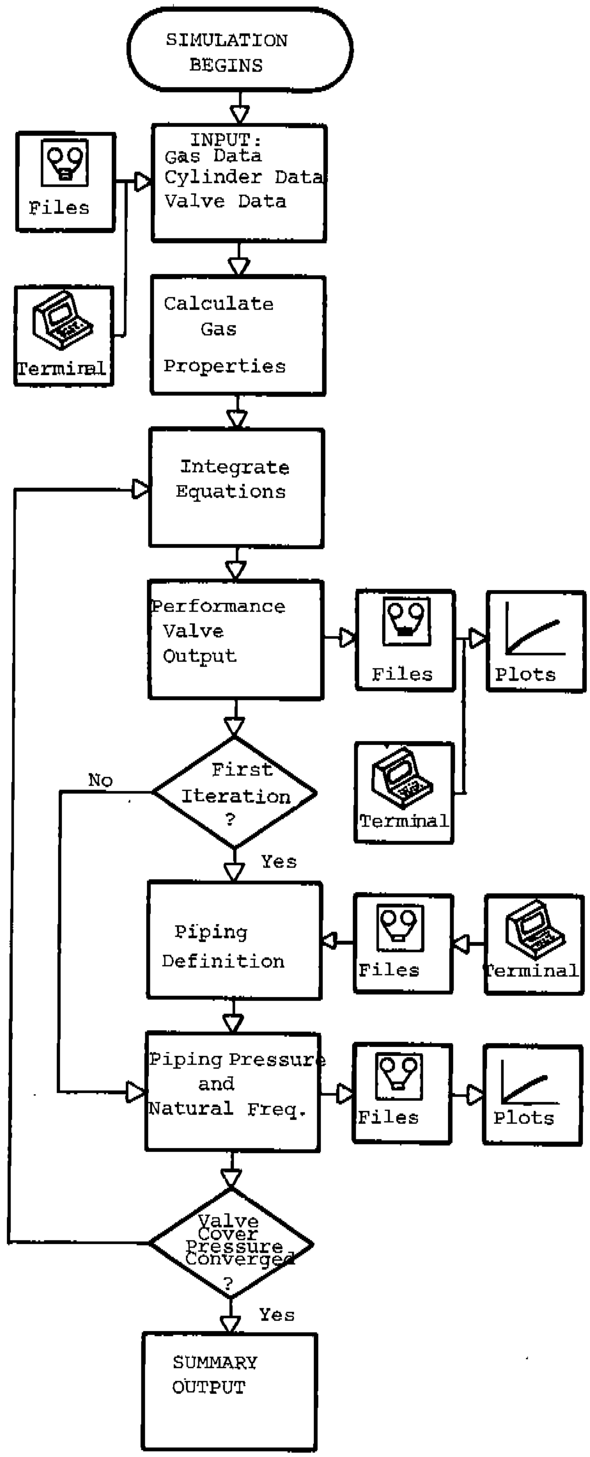


Figure 2 Program Flow-Chart

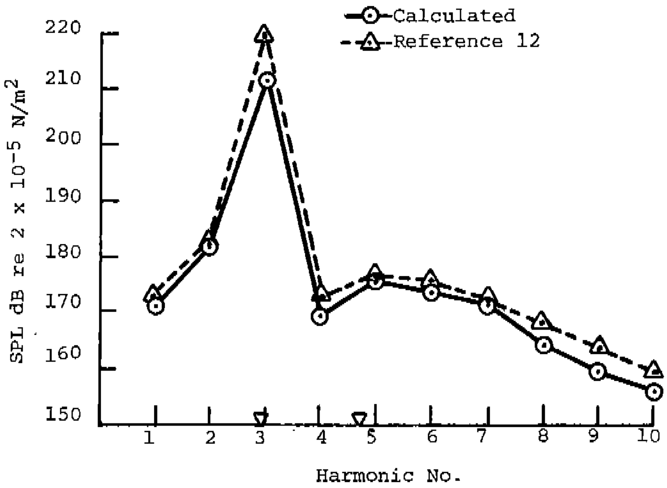


Figure 4: Comparison of calculated Sound Pressure level (SPL) with that from Reference 12. Only viscous damping is assumed in both cases. Arrows indicate natural frequencies calculated by the method presented here.

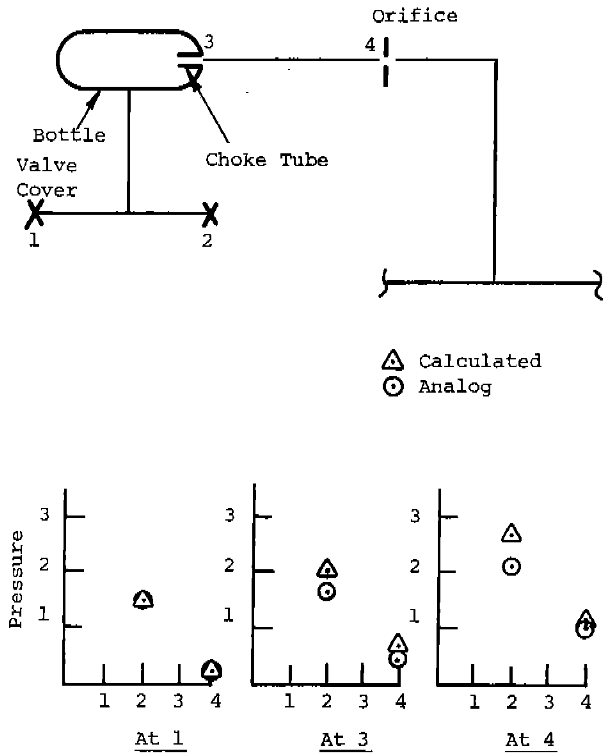


Figure 5: Comparison of analog and digital simulation for the piping diagram shown. The scale along x axis represents harmonic numbers. The pressure along y axis is peak to peak to in psi.

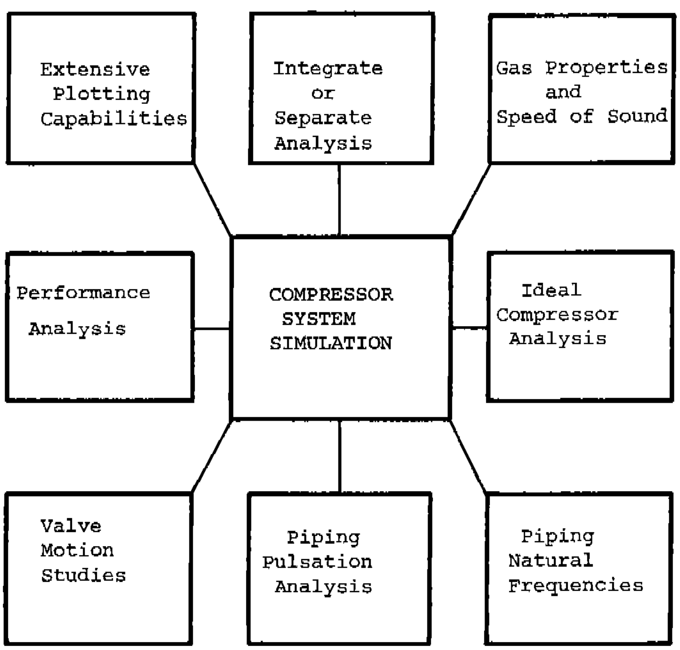


Figure 6: Multiple Capabilities of the Computer Program

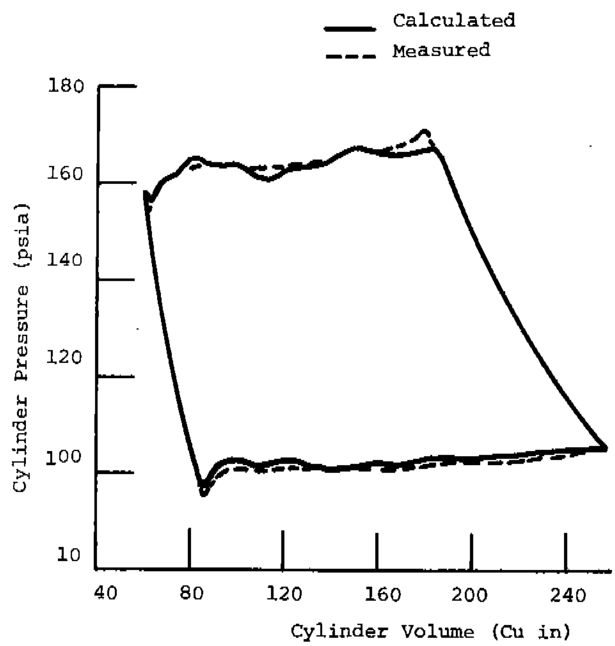


Figure 7: Measured and calculated PV diagrams

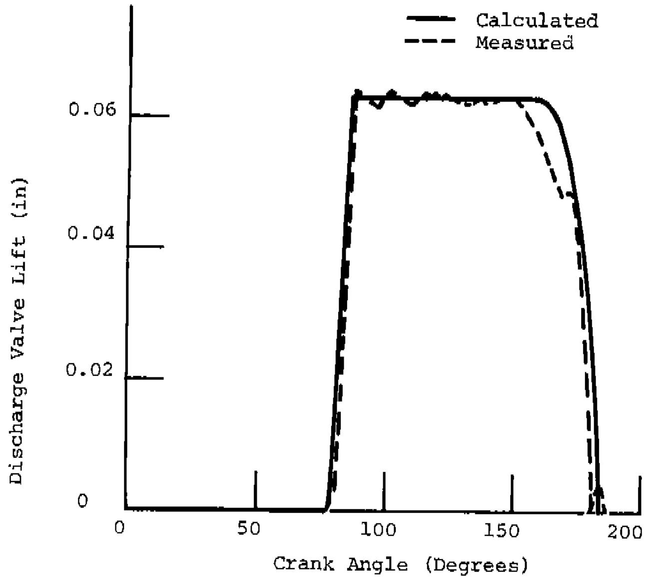


Figure 8: Measured and calculated discharge valve displacement

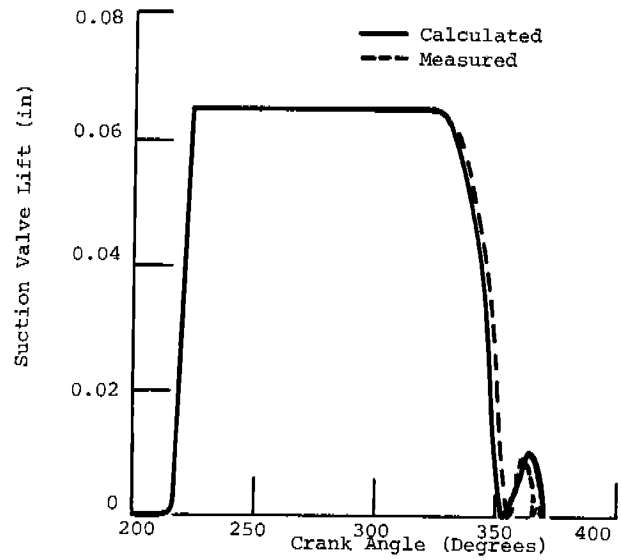


Figure 9: Measured and calculated suction valve displacement

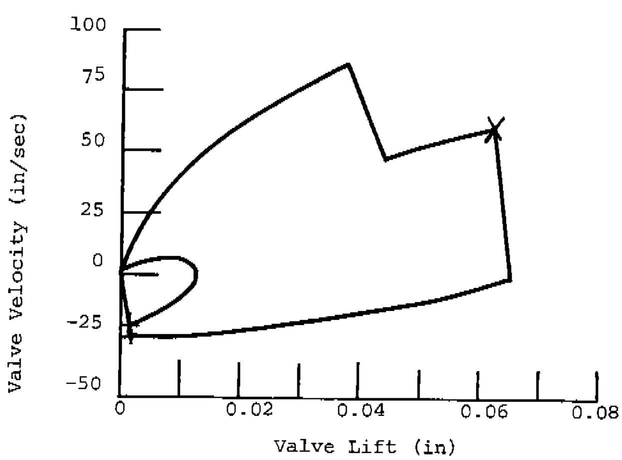


Figure 10: Calculated valve plate velocity vs. lift for the suction valve. X and + indicate impacts against stop and seat plates respectively.

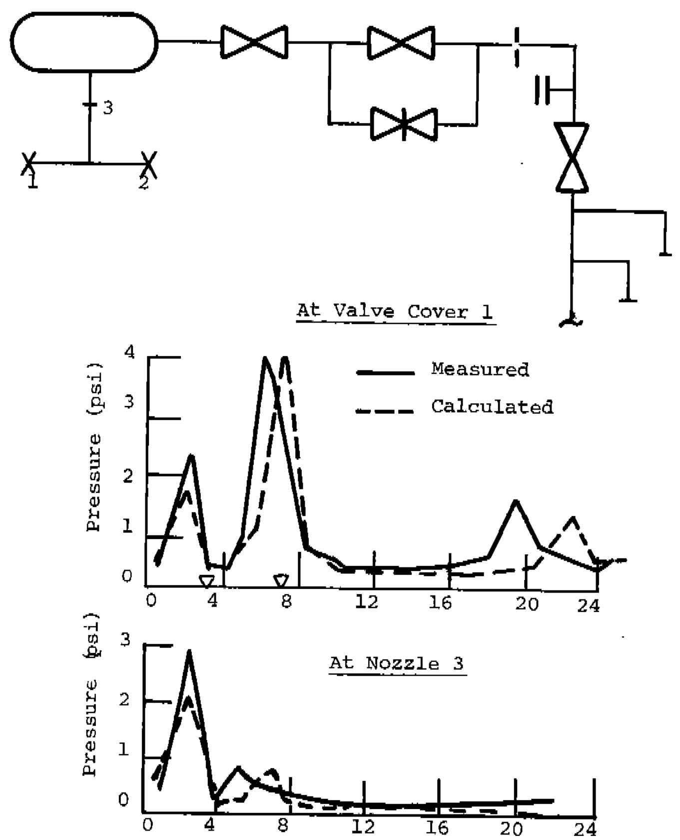


Figure 11: Measured and calculated pressure (peak to peak) at two points in the test loop. Arrows indicate calculated natural frequencies. Additional damping (50 times normal damping) was used for the seventh harmonic near resonance.

$A_{ij}..D_{ij}$	Four-pole matrix elements for the pipe element ij

Subscripts

bu	Valve buffer element
c	Cylinder
d,s	Discharge, suction condition
l	Local condition
p	Pipe element
se,st	Valve seat, stop plate
us,ds	Upstream, downstream condition

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