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FLUID DYNAMIC EFFECTS IN THE MULTICYLINDER COMPRESSOR SUCTION AND DISCHARGE CAVETIES

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INTRODUCTION

Recent investigations [1-8] have demonstrated the interactive nature of the basic processes of a positive displacement compressor. Fluid pressure pulsations play an important role in influencing it. For example, consider a single cylinder with suction and discharge pipes. Unsteady flows in suction/discharge pipes are generated by the reciprocating action of the piston, aided by the rapid opening and closing of pressure actuated automatic valves. These pressure fluctuations, in turn, affect valve displacements, cylinder pressure and instantaneous fluid flow rates. If these processes are modeled mathematically, just by glancing at the equations, one can see the strong interactions between the various components [1-6]. Because of this reason, a simultaneous solution of all mathematical models is a recommended procedure [1-8]. In multicylinder compressors, these dynamic interactions are further complicated by the addition of cylinder interactions. Suction/discharge piping of a cylinder influences flows in the suction/discharge pipings of other cylinders. These fluid dynamic effects are discussed in this paper. Cylinder interference mechanisms will be defined and explained, and then the results of a two cylinder compressor investigation shall be presented.

Gas fluctuations in both suction and discharge lines consist of a periodic train of mass pulses. Suction and discharge flows could be considered as composed of mean and fluctuating parts. The fluctuating part exhibits a wave phenomena. The pressure pulses in both suction and discharge systems travel through and ahead of the mean fluid flow with the speed of sound, and are reflected back and forth by discontinuities and end conditions. With frequent opening of valves, pressures are altered rapidly and are functions of both spatial coordinates and time. But in general, one dimensional model description is sufficient for fairly accurate analysis.

Fluctuating fluid pressures are generally much smaller than the mean pressures. Thus, acoustical theories could be utilized to describe the fluctuating fluid motion [1, 2, 4-9]. Mean flows can be predicted by using standard steady flow formulations. Therefore, both steady and unsteady motions could be analysed separately and then added together to provide the total fluid pressures. Note that only the unsteady flows are capable of dynamic interactions. Also note that only the lower harmonics are detrimental to compressor performance and behaviour. Thus the present study concentrates on the acoustic plane waves formulations. It should be pointed out that the philosophy and modeling technique is the same for both suction and discharge systems. Thus the formulations for one are applicable to the other.

As is evident from the literature survey, the work in the area of mathematical modeling of the multicylinder compressor lines' pulsations is very limited. The only investigation which dealt with the dynamic coupling of multicylinder cavities is by Soedel, Padilla, and Kotalik [4]. It applies an acoustic lumped parameters type of approach to model a two-cylinder discharge system. The model is solved in the time domain with the rest of the compressor simulation equations. A lumped parameters technique relies heavily on an easy identification of the inertia and elastic elements which may not be possible in all the cases, and it also suffers from the accuracy viewpoint. Also, the Soedel, et al [4] model could not account for the piping exiting from a common plenum. In a recent effort, Soedel [9] has added an equation for the exit pipe. But, it is not general as it describes only an anechoic termination.

Schwerzler [10] modeled suction and discharge system of a multicylinder compressor by using only a quasi-steady mean pressure variations technique which ignores gas inertia

coupling effects. The literature search has not revealed any other efforts in either compressors or its close parallel, internal combustion engines.

MULTICYLINDER FLUID DYNAMIC INTERACTIONS

The gas pressure oscillations are more complicated in a multicylinder discharge (or suction) system because the cylinders interact with each other. In a single cylinder case, mass flow rate through the discharge valve is the source function for pulsations. The waves generated by it propagate through the system and get reflected because of the impedance mismatches. All these considerations are applicable here also except that there are some more complications. The interactions can be thought of as consisting of two type of couplings, namely the kinematic and the cavity couplings.

(i) Kinematic Coupling

The kinematic arrangement of a multicylinder reciprocating compressor is such that the instantaneous crank angles of all the cylinders are not the same. Since the compressor operation is a cyclic phenomena, all the processes in a particular cylinder will be ahead or behind the other cylinders by some phase differences. During a cycle, every cylinder discharges gas to the discharge system through the automatic valves. As the respective processes in all of the cylinders are not taking place at the same time, the discharge mass flow rates of all the cylinders also will not be simultaneous. Their time lag, in every cycle, will correspond to the crank phasing between the cylinders. In the frequency domain, this time lag becomes the phase lag (or lead) and has to be accounted with each and every harmonic.

(ii) Cavity Coupling

The mass discharges from all cylinders are accumulated and then sent to the condenser. The cylinders are generally connected to each other through passages and plenums. The mean fluid flow follows the pattern of these geometries and is pumped downstream. The acoustic waves not only propagate down the discharge line of one cylinder but also influence each other through the connecting elements. The impedance inequalities all over the lines would reflect the acoustic waves back. Thus, in a discharge plenum there are incident and reflected waves corresponding to not only its respective cylinder but also to all the other cylinders. Hence, the resulting pressure in front of any discharge valve has components corresponding to all of the mass flow rates. Of course, the main component is due to its own mass flow rate input. The other cylinders may enhance or subdue it. Similarly, at any other point in the system, the waves from all the cylinders interfere with each other.

Kinematic coupling in the absence of cavity coupling would not cause any dynamic interactions between the cylinder. The only result would be that the discharge pressures of all the cylinders would have the same phase relationships amongst them as that of the rest of the compressor processes. On the other hand, the cavity coupling in the absence of kinematic coupling would cause fluid dynamic interactions. The waves, from all of the valve exits, would start simultaneously and propagate in the system. And, the interference would take place. When the kinematic coupling is superimposed on the cavity couplings, then the waves from the valve exits do not start simultaneously. Now, the interference phenomena is going to be rather complicated. One has to keep track of both the amplitude and phase of each and every harmonic of a cylinder. The similar harmonics cause a constructive interference and the dissimilar harmonics cause a destructive interference.

One can picture multicylinder discharge (or suction) case as a dynamic system with multiple periodic nonsimultaneous input signals. It is shown in Fig. 1. It depicts a multicylinder compressor discharge/suction system as a dynamic system which consists of m subsystems, where m is the number of cylinders. Mass flow rates through valves could be considered as inputs, and time delays correspond to their relative kinematic crank phasings. The output is the unsteady fluid pressure, and it is influenced by subsystem interactions, which correspond to cavity (or geometric) couplings.

MODELING PROCEDURES

The philosophy of the present modeling is the linear acoustical theory. Plane wave formulations are utilized to construct the mathematical models because it is sufficient to cover the frequency range of interest. The assumption is valid from the fact that most of the compressor suction and discharge components are much smaller than the lowest significant wavelength in question.

The following procedures are used to model and analyse discharge (or suction) systems.

(i) Distributed Parameters Modeling

Acoustical system characteristics are described in the distributed parameters format. These are in hyperbolic form, and their arguments include the following: (a) wave number (b) attenuation factor (c) geometrical dimensions and (d) additional energy related to the velocity field due to the disturbing influence of the discontinuties. Composite systems are analyzed by building fourpole matrices. It is a "building block" type of approach, and involves simpler mathematical operations. Fourpole matrices are

system characteristics and could be solved to provide other system characteristics, namely, acoustical impedances. The present solution technique utilizes these steady state impedance formulations for fluid pressure predictions. It was first tried by Elson and Soedel [7] on a single cylinder compressor. They determined that it is more efficient and general than some classical approaches. The procedure is iterative in nature as it simulates the interaction of the lines and valves, i.e., the effect of back pressures on the automatic valves. Elson and Soedel [7] tried it for a very simple system (valve chamber connected to long lines) and obtained good results. But, they could not generalize the procedure for a multicylinder case. Also, they did not show its applicability to complex and practical geometries.

The present study generalizes the impedance modeling concept by extending it to (a) complex composite systems and (b) multicylinder cases [12]. To account of the multicylinder interference mechanisms, the following procedure is used:

a. The source function for the excitation of suction/discharge acoustic system is the volume velocity. Refer to Fig. 2. For each cylinder, volume velocity is defined differently and the respective crank phasing is associated with it. Thus the kinematic interactions are accounted for in the volume velocity formulations.

b. To take into account the acoustical characteristics of the suction/discharge system, input and transfer impedances are defined and computed. Whereas the input impedances are defined only at the valve exits, the transfer impedances are defined not only at valve exits but also at intended pressure prediction locations. Input and transfer impedances are operated on by their respective volume velocities to obtain pressures. Thus the cavity interactions are incorporated in the definition and calculation of impedances.

(ii) Lumped Parameters Modeling

Suction/discharge system components can be identified and modeled as gas inertia, elastic and resistance elements. Such an approach is potentially attractive and powerful in the case of irregular cavities and passages, often encountered in the suction/discharge systems. Distributed parameters analysis of such cases may be difficult and tedious. Also, a lumped analysis provides a good and sound understanding of the fluid dynamic effects. Lumped parameters models have been analysed using both transient and steady state solution techniques.

a. Transient Analysis: Equations for gas pulsations are directly coupled with the rest of the compressor mathematical models, and are simulated in time domain. Fluctuating pressure force was considered as the excitation forcing function in the equation of acoustic motion [4, 6, 9].

b. Steady State Analysis: Unlike transient analysis, it was not used for the simulation purpose, rather the intention was to carry out only the acoustic analysis of the cavities and passages. The objective was to attain an insight into the phenomena of dynamic interactions [12]. The following are the various steps: (1) Fourier analysis of the mass flow rates and then incorporation of relative kinematic phasings into mass flow rate harmonics. (2) Identification of gas mass, spring and damping elements and development of the equations of motion in terms of acoustic volume displacement. Harmonic components of the mass flow rates are treated as input excitations. (3) Solution of the eigenvalue problem. It involves a solution of the equations of free motion. Natural frequencies and mode shape of gas oscillations are computed. Natural frequencies are predicted in terms of geometric dimensions of the cavities and sonic speed of the gas medium. (4) Computation of the forced acoustic response. It is done by using the modal expansion technique. Using the above mentioned procedure, both kinematic and geometric interactions could be evaluated separately. Time variant pressures could be computed by the Fourier synthesis of the pressure, or volume displacement, harmonics.

(iii) Modeling with Measured Acoustic Characteristics

It may not be always possible to create a mathematical model of some complex practical suction/discharge components. Also, the mathematical models could also become too cumbersome, difficult and time consuming for numerical computations. Under such cases, it is more attractive to go to a test bench and measure acoustical characteristics of either the whole suction/discharge system or a particular acoustic element. Also, a bench test would provide a quick check and hopefully the understanding of the acoustic resonances. The measurements could either supplement or replace mathematical models. Either of the following two types of experimental measurements could be conducted.

a. Determination of Natural Frequencies and Mode Shapes: Microphones or pressure transducers may be placed at different locations and the suction/discharge cavities can be excited by an acoustic speaker. The measured pressure responses would provide an indication of the natural frequencies and modes of gas oscillations like lumped parameters modeling. These modal characteristics could be expanded to provide system pressures.

b. Measurement of Acoustical Impedances:
For the present study, an efficient method
of measuring acoustical impedance charac-
teristics of a suction/discharge system was
employed [12]. The system was excited by
an oscillating piston, connected to an
electro-dynamic shaker. The input volume
velocity was monitored by a displacement
transducer. Pressure responses were picked
up by microphones, mounted at different
locations. Sine, random and pulse excita-
tions were fed to the shaker. A digital
FFT system was used for data acqustion
and processing. This method provided in-
put and transfer impedances efficiently
and directly. Since impedances are system
dynamic characteristics, one can easily
identify the effects of cylinder interac-
tions. These measured values could be used
in place of theoretically calculated imped-
ances, as shown in Fig. 4.

APPLICATION TO A TWO-CYLINDER CASE

The philosophy and procedures outlined in
the previous section were applied to a two-
cylinder refrigeration compressor [11, 12].
It was an opposed-piston type reciprocating
machinery. The objective was to simulate
discharge system gas oscillations. A num-
ber of simplifications were incorporated
accordingly into the construction of the
compressor mathematical models. However,
because of the strong interactions between
the discharge system and valves, special
emphasis was placed upon the modeling of
discharge valves. The aim was to predict
discharge mass flow rate accurately as it
was considered as the source function of
the discharge system excitation. Different
step sizes were explored and by comparing
the mass flow rates, an optimum simulation
step size was determined. The discharge
back pressure effect on the cylinder pres-
sure and valve motion was simulated by
employing an iterative procedure. Refer
to Figure. 2. The iteration was continued
until the pressure harmonics converged.

Figure 3 shows the schematic of a two-
cylinder compressor example case. The
discharge system consists of the following
components: discharge plenums, connecting
passages, collector, exit piping, two muf-
fler elements and an anechoic line. Mea-
surements were conducted on a load stand
which utilized a hot gas by-pass type re-
frigeration cycle. As outlined in Fig. 3,
pressure transducers were installed in the
cylinder, discharge plenum and anechoic
line. An electromagnetic pickup, mounted
on the crank, was used to identify the
crank angle. Data acquisition and proces-
sing were performed by a mini-computer
operated high speed digital system, with
the Fast Fourier Transform (FFT) capability.
The measurements were conducted for three
different operating conditions. The run-
ning speed of the compressor was 3580 rpm
and the fluid medium was R-12.

RESULTS AND DISCUSSION

Figure 4 shows the comparison between theo-
retical and experimental results of the
transfer impedance. It is defined as the
complex quotient of pressure at one cylinder
valve to the volume velocity at the other
cylinder exit. Thus it is clearly an indi-
cation of the interaction mechanism between
the cylinders. The results presented in Fig.
4 are for the air medium at room temperature
and pressure. Note that ordinate is the im-
pedance level which is defined as the dimen-
sionless acoustic impedance in decibels. The
abscissa i.e., frequency is not in the dimem-
sionless form. It is done because there is no
reference frequency. The frequencies here ap-
ply to the air medium. To make these plots ap-
plicable, the abscissa could be divided by the
air sonic speed to make it wave number.

A lumped parameters analysis of the inner
cavities (refer to Fig. 3) reveals that
there were two natural frequencies: slosh-
ing mode frequency is 390 Hz and compressive
mode frequency is 585 Hz. These resonances
are clearly shown in Fig. 4. Now the ques-
tion arises: what about the exit piping
and muffling system? If the discharge sys-
tem components belonging to one of the
cylinders are deleted, then the system is
reduced to a single cylinder discharge sys-
tem. An approximate analysis demonstrated
that there are two dominant low frequency
resonances, one at 140 Hz and the other at
200 Hz. Thus the 390 Hz and 585 Hz reso-
nances are due to the geometric elements
connecting the two cylinders, and 140 Hz
and 200 Hz resonances are due to the compo-
nents through which mean flows from one cylin-
der would pass. Therefore, if dynamic interfe-
rence mechanisms between cylinders were not
modeled, only 120 Hz and 140 Hz resonances
would have been predicted. Also, note that
390 Hz and 585 Hz resonances are more do-
minant than the 140 and 200 Hz resonances.

The natural frequencies of the inner cavi-
ties for R-12 medium are 190 and 285 Hz.
These were calculated by using lumped para-
meters analysis. Modal expansion demon-
strates that, for the present case of op-
posed piston compressor with a symmetrical
discharge system, out of the two natural
frequencies, only the first natural fre-
quency (190 Hz) is the resonance frequency.
It is associated with the sloshing mode of
gas oscillations i.e., inertia elements
in the connecting passages are oscillating
in phase with each other.

The computer simulation program included a
distributed parameters type of description
for the discharge system. Results of the
simulation model including comparisons with
the experiment are presented. The fluctua-
ting pressures at discharge valve exit and
at discharge system end i.e., anechoic line,
are of interest.

Figures 5-7 trace the pressure time history of the refrigerant flow in the cylinder, discharge plenum, and anechoic line over one cycle of compressor operation. Theoretically computed discharge valve displacements, plotted on the same figures, provide a correlation between the pressures and valve openings. All the plots show excellent agreements between measured and predicted results. The cylinder cyclic pressure variation, Fig. 5, agrees very well during the discharge valve opening time. But, there is a slight discrepancy during the suction process. It is compatible with the research objectives and goals. For better agreement during the suction, the same attention should be given to the modeling of the suction valves and plenums. Also, it should be noted that the valves are modeled entirely analytically with no experimental input data. A better agreement demands more experimental inputs. Although the results were compared for three different conditions, they all showed the same trends and tendencies. This repeatability of the experimental and theoretical data developed a confidence in the validity of the program. From the cyclic pressure variation at valve exit in the discharge plenum, Fig. 6, it is seen that three cycles of the gas pressure oscillation take place during one cycle of the compressor process. It indicates that the third harmonic is dominant at the valve exit. It confirms the results obtained by lumped parameters analysis. The anechoic line pressure, Fig. 7, is more or less steady and no substantial fluctuations are taking place.

The pressure frequency spectra is plotted in Figs. 8-9. They show an excellent correlation between the measured and predicted values. Fig. 8 illustrates discharge plenum pressure spectra at the valve exit. The results are compared for the first 20 harmonics, i.e., up to about 1200 Hz. The spectrum demonstrates the dominance of the lower harmonics in the discharge plenum. In fact, the third harmonic is the most dominant. Fig. 9 shows the pressure spectra at the anechoic line. Only the first and second harmonics are still strong here; the rest have been attenuated either by the discharge cavity configuration or by the mufflers. The results are compared for the first 15 harmonics, i.e., up to 900 Hz.

CONCLUDING REMARKS

No general statement can as yet be made regarding the suction/discharge pressure time history or frequency spectra of a multicylinder case. It all depends upon the crank arrangement and geometric configurations. In general, multicylinder compressors are encountered and the designer is faced with some of the most puzzling problems arising from pulsations. Now, the question is: how should he approach it? Well, for an initial guess, one can draw an analogous mechanical or electrical circuit of the suction/discharge system and try to study the dynamic behaviour. One has to think along and keep track of harmonic components of pressure, and not only their magnitudes but phases as well. Furthermore, it is hoped that the information presented in this paper would aid a designer in evaluating the significance of geometric, kinematic and fluid parameters. This should lead to a proper selection of suction/discharge components.

REFERENCES

1. J. Brablik, 1972, Purdue Compressor Technology Conference, Proceedings, p p. 188-195, "Gas Pulsations Affecting Operation of Automatic Valves in Reciprocating Compressors".

2. J.P. Elson and W. Soedel, 1972, Purdue Compressor Technology Conference, Proceedings, p p. 311-315, "A Review of Discharge and Suction Line Oscillation Research".

3. J.F.T. McLaren and A.B. Tramschek, 1972, Purdue Compressor Technology Conference, Proceedings, pp. 203-211, "Prediction of Valve Behaviour with Pulsating Flow in Reciprocating Compressors".

4. W. Soedel, E.P. Navas and B.B. Kotalik, 1973, Journal of Sound and Vibration, 30(8), pp. 263-277, "On Helmholtz Resonator Effects in the Discharge System of a Two Cylinder Compressor".

5. R. Singh and W. Soedel, 1974, Purdue Compressor Technology Conference, Proceedings, pp. 102-123, "A Review of Compressor Lines Pulsation Analysis and Muffler Design Research, Part I - Pulsation Effects and Muffler Criteria, Part II - Analysis of Pulsating Flows".

6. J.F. Hamilton, 1974, Short Course Text, Ray W. Herrick Laboratories Purdue University, "Extensions of Mathematical Modeling of Positive Displacement Type Compressors".

7. J.P. Elson and W. Soedel, 1974, Journal of Sound and Vibration, 34(2), pp. 211-220, "Simulation of the Interaction of the Compressor Valves with Acoustic Back Pressures in Long Discharge Lines".

8. R. Singh and W. Soedel, 1975, International Institute of Refrigeration, Congress of Refrigeration, Moscow, Paper No. B2.15, "Simulation of Compressor Lines Gas Pulsations".

9. W. Soedel, 1972, Purdue Compressor Technology Conference, Proceedings, pp. 136-139, "On the Simulation of Anechoic Pipes Helmholtz Resonator Models of Compressor Discharge Systems".

10. D. Schwerzler, 1971, Ph.D. Thesis, Purdue University, "Mathematical Modeling of a Multiple Cylinder Refrigeration Compressor".

11. E. Sandgren, 1974, M.S.M.E. Thesis, Purdue University, "Computer Simulation of a Two-Cylinder Reciprocating Compressor and Associated Discharge System Using Acoustical Impedances".

12. R. Singh, 1975, Ph.D. Thesis, Purdue University, "Modeling of Multicylinder Compressor Discharge Systems".
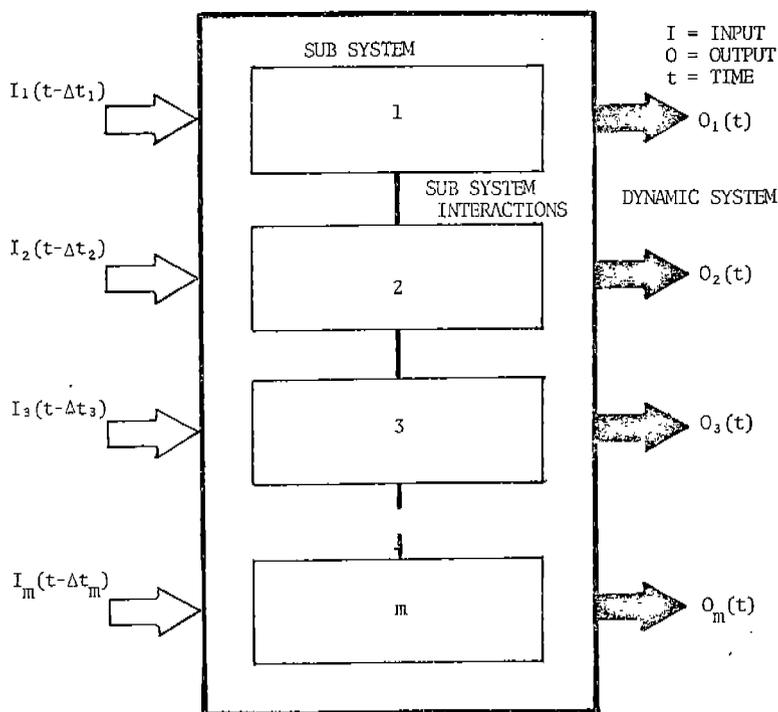
Figure 1   System Interactions
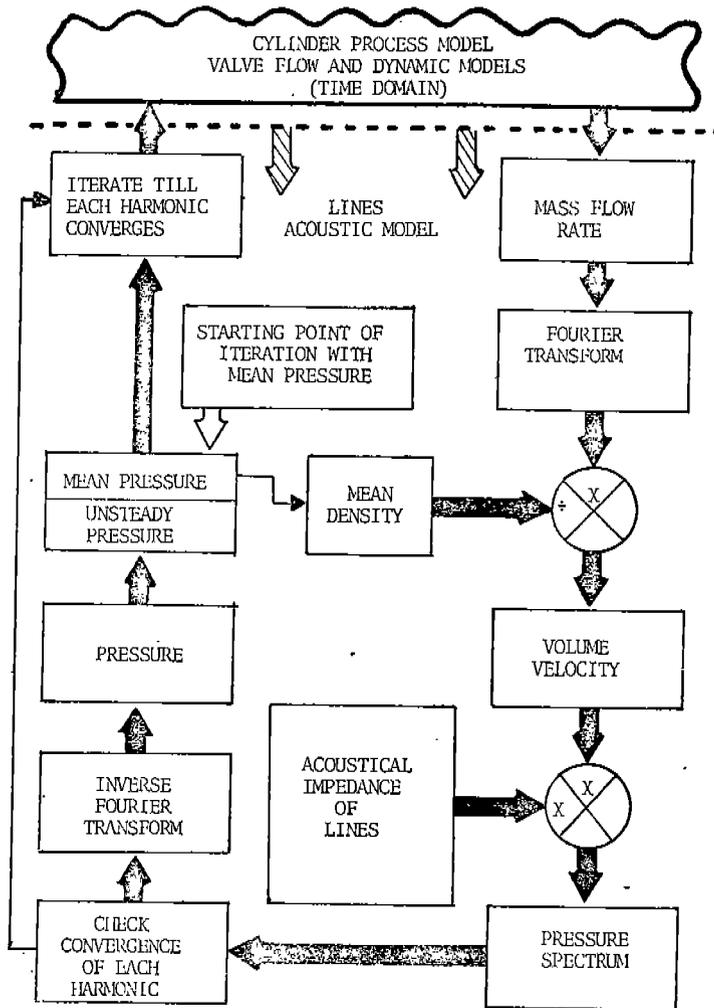
Figure 2 Overview of Computer Model
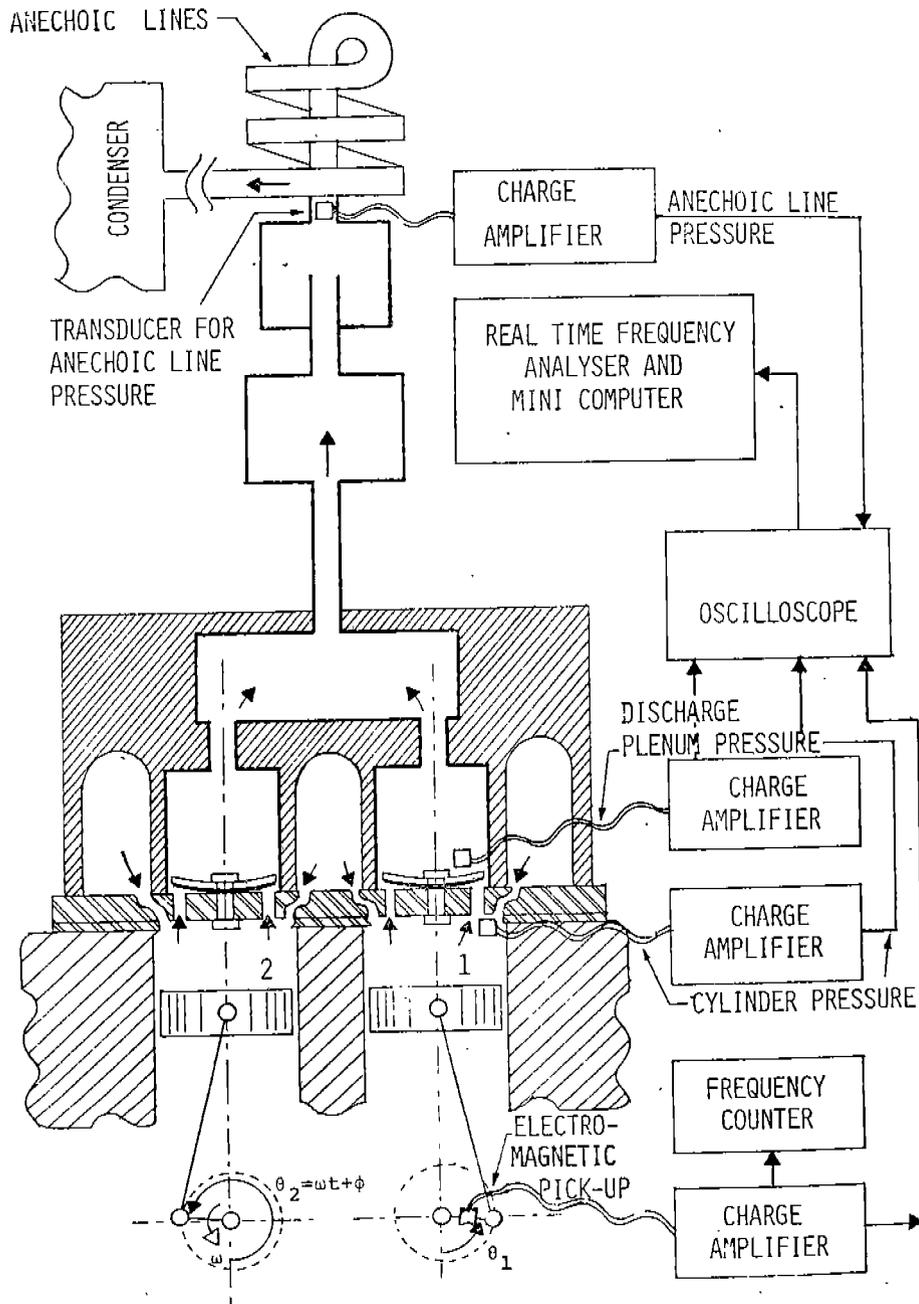
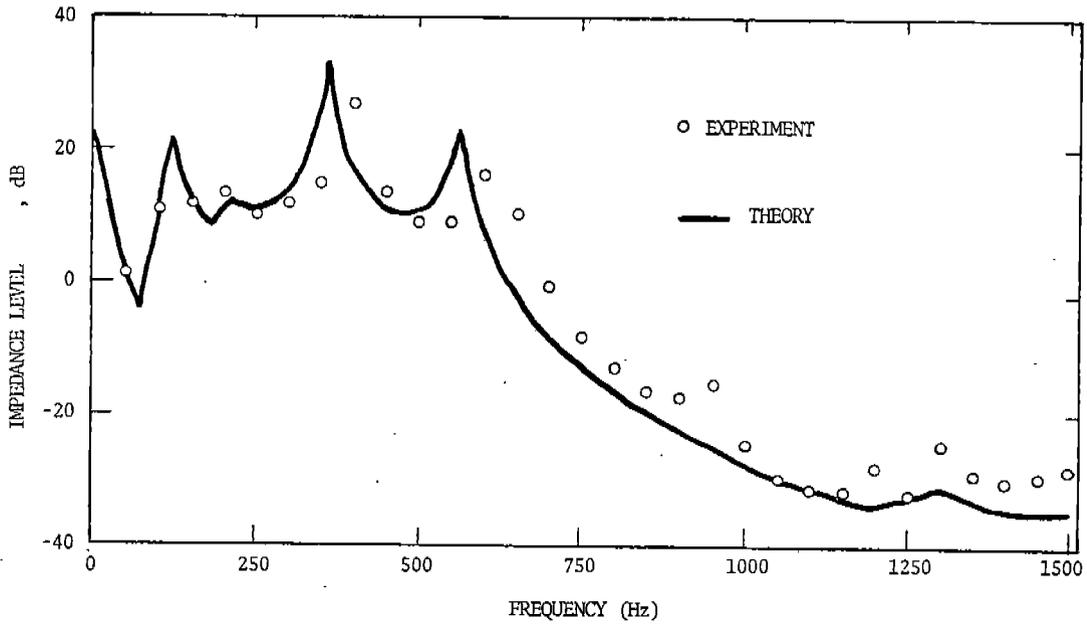Figure 3   System Used as Example Case

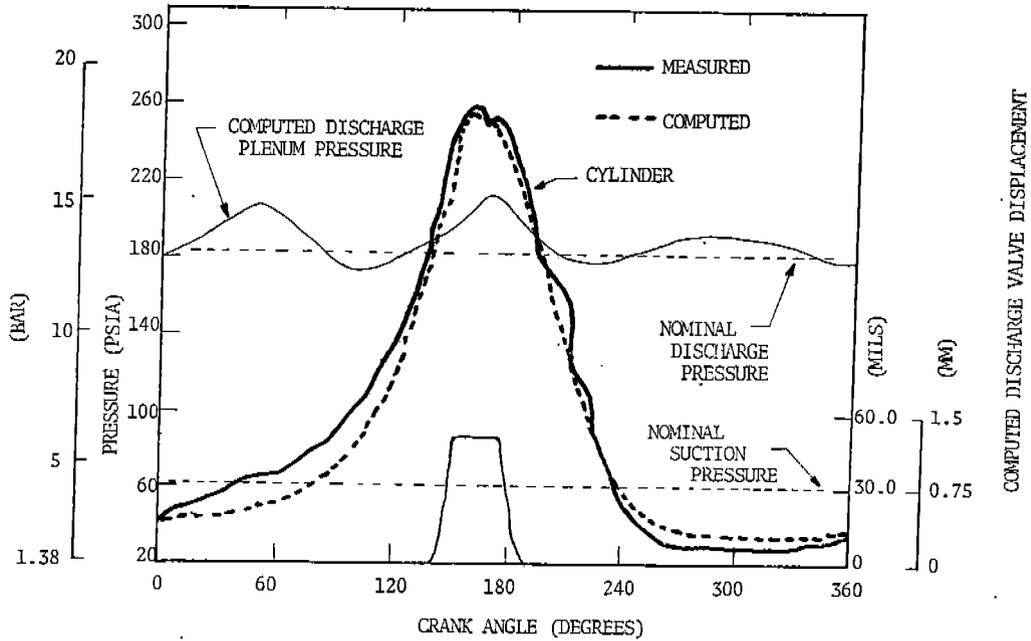Figure 4   Transfer Impedance Level

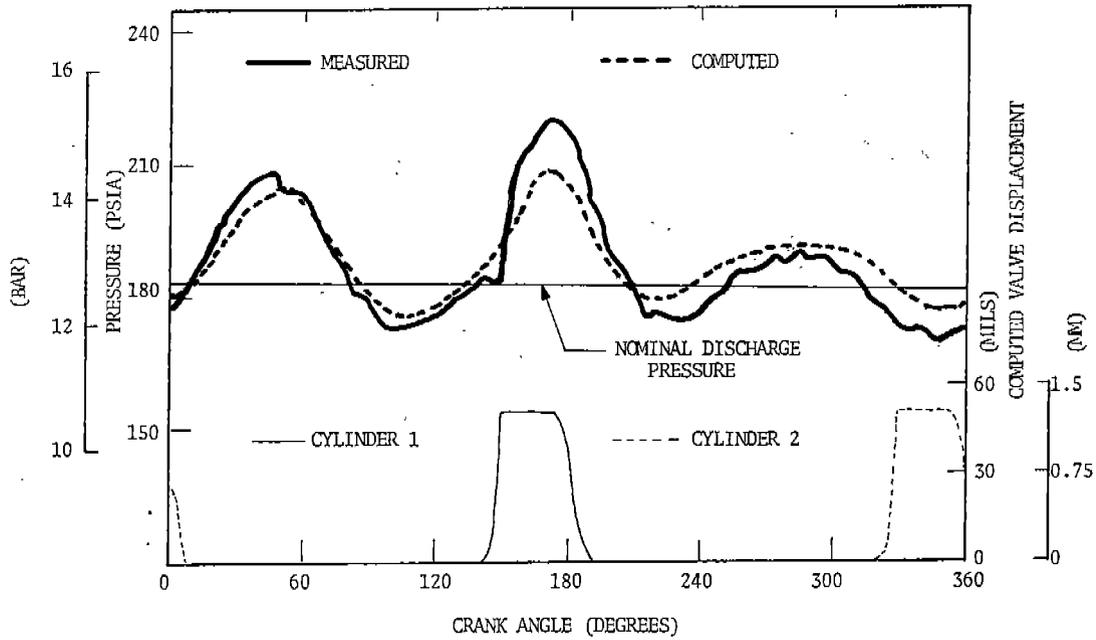Figure 5   Cylinder and Discharge Plenum Pressures
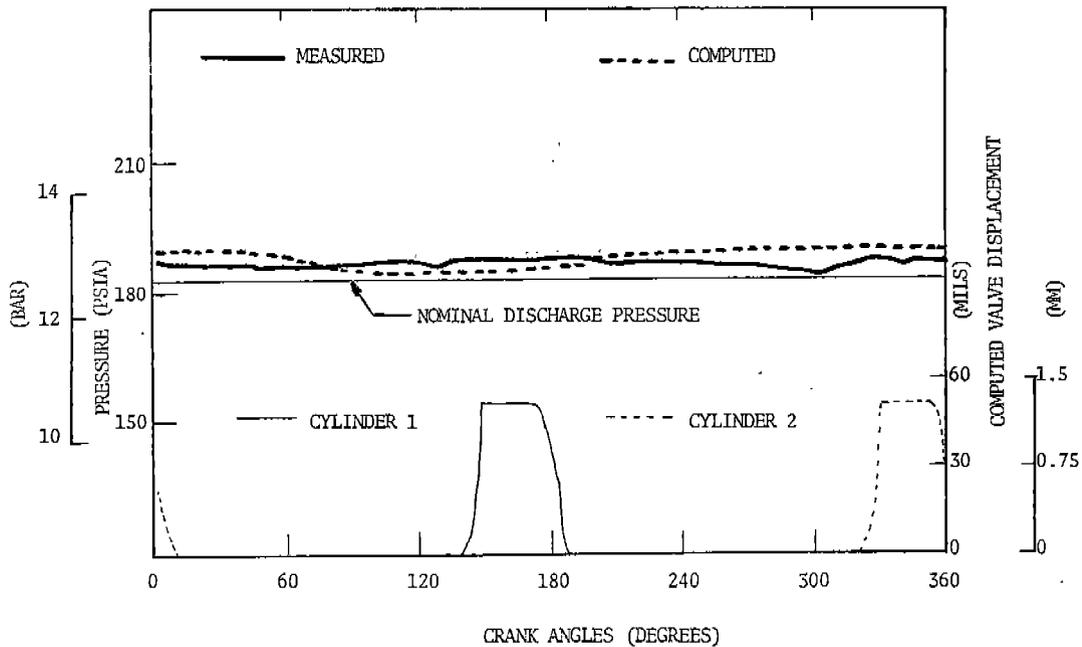
Figure 6 Discharge Pressure Behind Valves

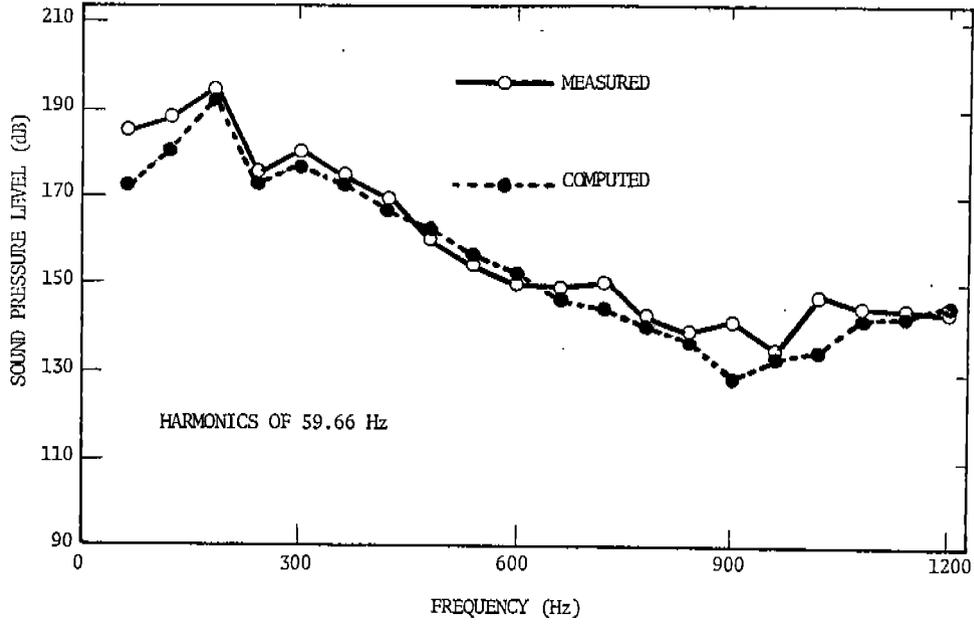Figure 7 Discharge Pressure at Anechoic Termination
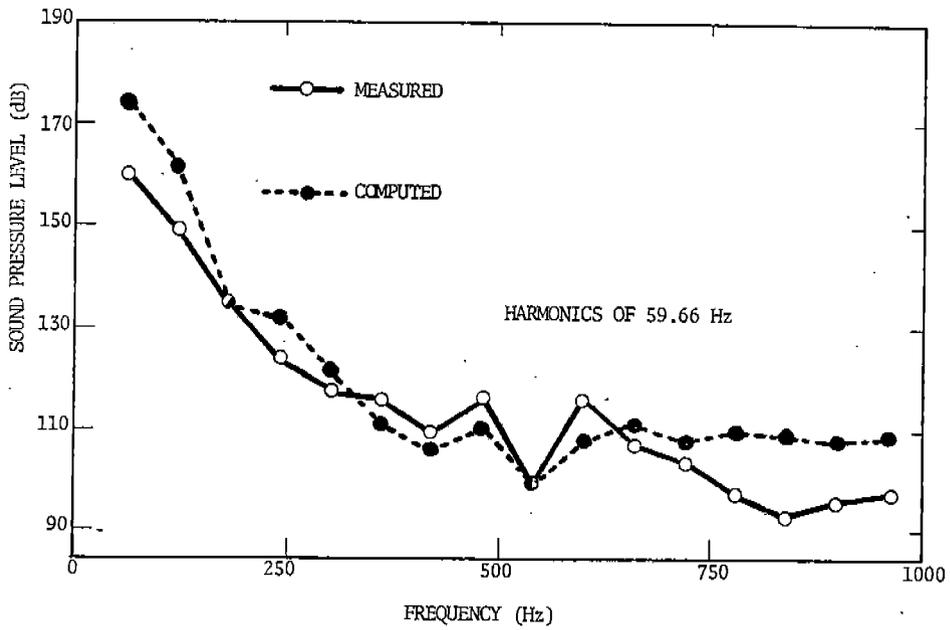
Figure 8   Pressure Spectrum Behind a Valve

Figure 9   Pressure Spectrum at Anechoic Termination