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D. D. Schwerzler

General Motors Research Laboratories

J. F. Hamilton

Purdue University

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SIMULATION AND DESIGN STUDIES OF A MULTIPLE CYLINDER
REFRIGERATION COMPRESSOR

D. D. Schwerzler¹ and J. F. Hamilton²
Mechanical Engineering Department, Purdue University
West Lafayette, Indiana

INTRODUCTION

The benefits of an analytical model of a refrigeration compressor are obvious in the amount of information that can be obtained quickly and inexpensively. With an analytical model, parameter studies can easily be performed in order to optimize the compressor design and its use in a given refrigeration system. The dynamics of certain parts of the compressor can be studied to determine possible modes of failure, and an important increase in the understanding of the compressor is obtained when the various phenomena occurring inside the compressor are put into a mathematical form.

Because of these benefits many researchers have worked on trying to develop a completely general, theoretical model during the past several years. This work is an extension of this research. Techniques will be proposed which can be used to expand the analytical model so that it will include the gas processes occurring in a multiple number of cylinders and in the suction and discharge plenums that are common to all the cylinders. This extension allows the pressure and temperature coupling between the cylinders and the suction and discharge plenums to be studied in greater depth. These intereffects can be of significant importance in determining the efficiency of a reciprocating compressor.

For example, a cylinder exhausting into a discharge plenum having a "small" volume will operate inefficiently and will create large unwanted pressure fluctuations in the discharge lines. If one cylinder is nearing the end of its exhaust stroke while a second cylinder has just begun, the large pressure fluctuation in the discharge plenum due to the second cylinder could force the closing of the discharge valve of the first cylinder before it is through exhausting. This action would reduce the efficiency of the compressor and could produce rapid deterioration of the discharge valve. Compressors having cooling jackets around the suction and the discharge plenums can change the conditions of the gas so as to significantly affect the compressor performance. A compressor designed for efficient operation at one speed or set of operating conditions of pressure and temperature may be quite inefficient at a different speed or set of operating conditions. Each of these effects can be studied when the analytical model is developed so

that it includes all the cylinders and the suction and the discharge plenums.

Since the necessary equations and the logic needed for the complete analytical model are well documented in previous literature, only the new concepts that are proposed will be presented. The detailed equations and logic for the complete model can be found in the literature listed in the references.

In this paper the accuracy of the analytical model is demonstrated by comparing the results of the model to experimental data. The analytical model is also used to demonstrate the effect of compressor speed on the performance of the compressor.

DESCRIPTION OF COMPRESSOR

The compressor that was used for comparison with the analytical model was a fifty horsepower, three cylinder, ammonia, reciprocating compressor. The three in-line cylinders, as shown in Figure 1, have a 4.375 inch bore with a 4.250 inch stroke and were connected together by common suction and discharge plenums. The compressor was belt driven by a fifty horsepower motor at speeds around 940 RPM. The discharge plenum, suction plenum and the cylinders were cooled by water flowing through passages in the walls surrounding these volumes.

In Figure 2, a side view of the valving system of one of the cylinders is shown. The ammonia gas enters at one end of the suction plenum, feeds through the suction ports around the suction valve and into the cylinder. The gas in the cylinder is compressed by the piston and forced out through the two discharge valves into the inner discharge plenum for that cylinder. The gas then passes through the restriction produced by the valve retainer ring and the valve clamp into the large discharge plenum which connects all three cylinders.

All three valves are thick ring valves (0.047 to 0.063 inches) made of hardened steel. They are spring loaded by coil springs symmetrically located around the circumference. The two discharge valves are independent of one another with the larger of the two discharge valves termed the outer discharge valve and the smaller one termed the inner discharge valve.

¹Presently Associate Senior Research Engineer, Engineering Mechanics Department, General Motors Technical Center, Detroit, Michigan

²Presently Professor of Mechanical Engineering, Purdue University, West Lafayette, Indiana

MAJOR CONTROL VOLUMES

In order to develop the model of the compressor, the compressor was sectioned into certain major control volumes. As shown in Figure 3, these control volumes consist of the suction plenum, the small volumes associated with the valving system of the suction and the discharge valves, the cylinders, the inner discharge plenums and the discharge plenum. It was assumed that the stagnation pressure and temperature were spatially independent inside each control volume. This is an acceptable assumption in view of the fact that the speed of sound of ammonia (1300 ft/sec) was significantly higher than the velocities of the compressor parts (maximum velocity of the piston was 35 ft/sec). It was also assumed that the control volumes associated with the discharge and the suction valving system could be neglected because of their size in comparison to the volumes of the other plenums.

The assumption that the stagnation pressure and temperature are spatially independent in each control volume plus the fact that all of the cylinders are identically constructed and equally spaced with regard to crankangle leads to the assumption that at steady-state operation, each of the cylinders responds in the same manner. Thus, one cylinder can be used to represent the performance of all three cylinders. This concept was utilized by integrating only the necessary equations to describe the performance of the one cylinder which is called the characteristic cylinder.

The mass flow rates and the temperatures of the characteristic cylinder are delayed the appropriate time or crank angle to represent the effects of the other two cylinders on the suction and discharge plenums. Their effects on the plenums in turn affect the response of the characteristic cylinder during the next cycle. After the analytical model has gone through several cycles, the start up transient effects will die away and the response of the characteristic cylinder will approach a steady-state solution.

There are several distinct advantages that exist with this type of approach. First, it significantly reduces the number of differential equations that have to be integrated and thus decreases the amount of computer time necessary for a solution. Secondly, the modeling can be easily changed to include any number of cylinders. However it should be emphasized that this approach cannot be used when the cylinders are not equally spaced with regard to crankangle or where the effects of the suction and discharge plenum are different on each cylinder.

CONTROL VARIABLES

In order to extend the analytical model to include the discharge and suction plenums, a new concept of modeling had to be created. Previously the operating conditions which the compressor saw were modeled as constant values of stagnation pressure and temperature existing in the suction and discharge plenums. In reality, these operating conditions are time average values existing in these plenums. To

take into account the pressure and temperature variations that can exist in the plenums creates a problem as how to model the compressor so that the time average pressure and temperatures in the plenums are those at the operating conditions. Also the operation of a compressor is controlled by the pressure and temperature at the inlet, the pressure at the outlet and the speed of the compressor. It was therefore necessary to model the compressor in such a way that these conditions were the controlling variables to the analytical model.

It was found experimentally that there was essentially no pressure drop across the inlet or the outlet of the compressor. The average pressure measured at the inlet to the compressor was essentially the same as that measured in the suction plenum and the average pressure at the outlet to the compressor was the same as the average pressure in the discharge plenum. The controlling variables for the analytical model were therefore considered to be the average suction plenum pressure, the average discharge plenum pressure, the inlet suction temperature and the compressor speed.

To control the analytical model so that it would stabilize at given average pressures in the plenums, a technique was developed so as to control the mass flow rate into the suction plenum and out of the discharge plenum. Since an analytical model of the compressor only was the main objective, the rest of the refrigeration system, including the piping, the oil separator, the condenser and the throttling valves, was modeled as a series of pressure drops and a temperature change.

As a result controlling orifices were placed in the analytical model at the entrance to the suction plenum (A_{SI}) and the exit to the discharge plenum (A_{DO}) [see Figure 3]. Arbitrary values for the areas of the orifices were picked for the computer program. The initial values of the pressures, P_{PS} and P_{PD} , were estimated in the model by calculating a theoretical mass flow rate using an 80% volumetric efficiency and then using an orifice equation and the desired average plenum pressure values, the values for P_{PS} and P_{PD} were calculated. At the end of each cycle, the average values for the suction plenum pressure and the discharge plenum pressure were calculated and the values of P_{PS} and P_{PD} were adjusted so as to bring the average values of the suction and discharge plenum pressures closer to the desired values for the next cycle. When the average plenum pressures were sufficiently close to the desired average plenum pressures and when the average mass flow rate into the compressor equaled the average mass flow rate out, the analytical model was considered to have reached its steady state solution. Figure 4 is the flow diagram for this logic.

The effect of the size of the area of each controlling orifice on the analytical model was checked by varying the area while holding the operating conditions constant. The major effect was the amount of pressure variation that occurred in the plenums.

Figure 5 shows that if the actual areas of the inlet and outlet of the compressor were used, the pressure fluctuations would have been almost zero in both plenums. The reason for this is that in each plenum the orifice area was so large that for the given average mass flow rate the pressure drop across the orifice was very small. Any small change in the pressure drop caused a drastic change in the mass flow rate. For a very small area, the mass flow rate through the orifice was essentially constant. In this case the pressure fluctuations in the plenum caused only minute changes in the pressure drop across the orifice and hence small changes in the mass flow rate.

Figure 5 also points out that as the area was decreased below a certain level the magnitude of the pressure fluctuations in the plenum remained a constant value. The point at which the magnitude started to remain constant was where the average pressure drop across the orifice was equal to one-half of the peak-to-peak value of the pressure fluctuations in the plenum. The area for this point will be termed the break area. For the discharge plenum, for example, the pressure downstream of the orifice, P_{pp} , was equal to the minimum pressure

occurring in the discharge plenum. For the suction plenum, the pressure upstream of the orifice, P_{ps} , was equal to the maximum pressure occurring in the suction plenum. In this case the mass flow rate varied from a zero value to a maximum value and back again over the cycle of the pressure variation. As the area was decreased the magnitude of the mass flow rate cycle decreased and approached an average value.

Although the magnitude of the pressure variation in the plenum remained the same, the shape of the curve changed slightly as the orifice area decreased. For the discharge plenum, as is demonstrated in Figure 6, the positive slope of the curve remained the same since this is controlled mainly by the rate at which the cylinder pushed the gas into the plenum. On the negative slope, the curve decreased more rapidly for the break area since the mass flow rate was higher than the average value encountered with the smaller area.

Good correlation between the experimental data and the analytical results was obtained using the break areas. This type of pulsation of the gas out of the discharge plenum seems logical when considering the mechanism involved in the real case. The mass flow rate is zero when the gas in the discharge plenum is equal to the pressure in the discharge line. As the pressure builds up in the plenum, the mass flow rate would increase proportionally. Hence, the mass flow rate out of the discharge plenum would be of the same cyclic form as that obtained using the theoretical model.

This type of model thus gives an excellent way of representing the effect of the rest of the refrigeration system on the compressor. So long as the areas are picked below the break areas, the magnitude of the pressure fluctuations occurring in the plenum due to the mass flow effect will probably be

very close to the actual case. Once the model of the compressor is programmed, the break areas can be determined and used for a more accurate model.

The only situation that the modeling possibly may not work for is the case in which the actual inlet and outlet areas are below the break areas. It is suggested that the model should then use the actual areas, but this situation was not checked. A compressor having a real pressure drop either across the inlet or the outlet of the compressor should be changed since the pressure drop reduces the efficiency of the compressor.

VERIFICATION OF THE ANALYTICAL MODEL

The time history curves for one set of operating conditions are shown in Figures 7 and 8. The curves with symbols represent the experimental data and the curves without symbols represent the theoretical values. The only major discrepancy between the analytical model and the experimental data is in the prediction of the closing times for the valves. For the given compressor at the times when the valves are starting to close the pressure drops across the valves are very small. Prediction of closing times where the pressure drop across the valve are very small means precise prediction of pressures. This is especially difficult on valves which are designed to work on very small pressure drops. Pressure variations in the suction and discharge plenums can effect the closing times of the valves. It is believed that if a more inefficient compressor was chosen, that is one in which larger pressure drops exist across the valves at the end of intake and exhaust, better correlation could have been obtained.

The plenum pressure comparisons of Figure 8 should be viewed with the idea that the only significant comparisons that can be made are the overall magnitude of the pressure fluctuations and the locations of the peaks of the pressure curves. The curves of Figure 8 show that fairly good agreement between the peaks of pressure fluctuations in the discharge and suction plenum is obtained with those produced by the characteristic cylinder. For the experimental curves the locations of the peaks of the pressure fluctuations are different for the other two cylinders. This effect may have been produced by the fact that the pressure in the plenums is not really a constant value and hence the extra distance the pressure wave has to travel from these other two cylinders to the transducer measuring the pressure which was right above the one cylinder could account for some of the time delay.

Figures 7 and 8 show the effects of overlapping intake periods. The rapid pressure drop in the suction plenum pressure at the 340° crank angle location was caused by one of the other cylinders beginning its intake of gas into the cylinder. This effect is responsible for the suction valve of the cylinder closing earlier than it normally would.

In Table 1 the time average results are given for three sets of operating conditions, both analytical and experimental. The analytical model predicted

values for the increase in the temperature of the gas after it had entered the suction plenum, 3° to 6° higher than the experimental data. The increase in the gas temperature ranged from 21° to 32°. The model did well on predicting the discharge plenum temperatures. The maximum difference is only 7°.

For the average mass flow rate produced by the compressor, the analytical model predicted mass flow rates higher than the experimental values for the higher pressure ratios. If the analytical results are compared to the values measured by an uncalibrated orifice flow meter, the error can be as high as 15%, but if they are compared to the values measured by an energy balance method on the experimental data the maximum error is only 5%.

Table 2 shows the theoretical trends predicted by the analytical model that are useful in evaluating the performance of the compressor.

The volumetric efficiency shows a decrease in value as the pressure ratio increases. Part of the 6% drop is due to the extra 10° temperature rise of the gas in the suction plenum for the highest pressure ratio. The rest is due to the decrease in the mass flow rate.

The importance of valve performance is best indicated by the percentage of power caused by valve losses relative to the total power needed to compress the gas. For the suction valve of the 1st set of operating conditions, this percentage is about 5% and for the discharge valves 2%. This means that 7% of the power used is due to the valves. This is a significant amount of energy lost, some of which could possibly be saved by a redesign of the valving systems.

Also given in Table 2 are the impact velocities of the valves. There are two values given for each valve. One corresponds to the value when the valve first opens and hits its stop. The other value represents the closing of the valve as it hits its seat. The impact velocities are important in several respects. They can be used in calculating localized stresses in the valve at the time of impact. This information is therefore useful in predicting valve life. Also useful in predicting the life of the coil springs is the velocity of the end coil of the spring at valve impact, which can be assumed to be the same as the impact velocity of the valve. With this information, the amount of over-ride of the end coil over the other coils, which is the main cause of spring breakage in some compressors, can be determined. The noise generated by the impact of the valves is also a function of the impact velocity and a source of concern to manufacturers.

THEORETICAL TRENDS FOR DIFFERENT OPERATING SPEEDS

With an analytical model the performance of the compressor as a function of speed can be easily determined. To demonstrate how this can be done, the analytical model was used to generate the curves shown in Figures 9 and 10. For the given curves the operating conditions (inlet suction

temperature, average suction plenum pressure, and the average discharge plenum pressure) were held constant while the speed of the compressor was varied.

From Figure 9, it can be seen that as the speed increased the mass flow rate increased but not linearly. The nonlinearity of the mass flow rate curve is reflected in the volumetric efficiency curve which shows a drop of 26% from the speed of 500 RPM up to 2700 RPM. The main reason for the decrease in efficiency of the compressor at higher speeds was due to the sluggish action of the valves on closing which resulted in considerable backflow of gas. The temperatures in the plenums also show the effects of the increased mass flow rate. This is due to the fact that the gas did not stay in the plenums as long and therefore underwent less heat transfer. The leveling off of the suction plenum temperature above that of the inlet gas temperature is probably due to the extensive back flow that occurred through the suction valve for the higher speeds.

Figure 10 demonstrates how the impact velocities of the valves increased significantly with compressor speed. If the lives of the valves can be directly related to the impact velocities and the number of impacts, the lives of these valves which were considered to be satisfactory at 940 RPM could be shortened significantly by running this compressor at a much higher speed.

CONCLUSIONS

In summary, a large multi-cylinder ammonia compressor was successfully modeled. The modeling included the effects of the pressure fluctuations in the suction and discharge plenums, the heat transfer from the gas in the suction and discharge plenums and the cylinders, and the interaction of the cylinders through the common plenums. The rest of the refrigeration system was treated as a resistance by placing controlling orifices in the inlet and the outlet of the compressor. A feedback control was placed in the analytical model to force the average conditions in the suction and discharge plenums to the desired operating conditions. This was accomplished by controlling the pressure upstream of the inlet orifice and the pressure downstream of the outlet orifice so as to control the mass flow rates into and out of the compressor. The effects of the areas of the controlling orifices on the suction and discharge plenums were studied and conclusions were drawn as to what values of the areas gave the best correlation between the theoretical and experimental values. An explanation of why these particular values for the areas work well in simulating the resistance of the rest of the refrigeration system on the compressor was also suggested.

The results of the analytical compressor model were compared to experimental data taken on the compressor. These comparisons consisted of the cylinder pressure, the suction and discharge plenum pressures, the motion of the valves, the average mass flow rate, and the average temperatures of the

plenums. For most of the comparisons, the correlation was good. The only major point of poor correlation was on the closing times of the valves.

The analytical model was used to demonstrate the effect of the compressor speed on the performance of the compressor. The compressor speed was varied from 500 RPM to 2700 RPM and the volumetric efficiency was found to drop from 87% to 60%. Changes in the temperatures of the suction and discharge plenums and the impact velocities of the valves were also shown.

NOMENCLATURE

- a_{SI} - Area of the inlet controlling orifice to the suction plenum (in^2).
- a_{DO} - Area of the outlet controlling orifice to the discharge plenum (in^2).
- F_c - Compressor speed (Hz).
- \dot{m}_e - Estimated mass flow rate (lbm/sec).
- \dot{m}_{SI} - Instantaneous mass flow rate through the inlet controlling orifice (lbm/sec).
- \dot{m}_{DO} - Instantaneous mass flow rate through the outlet controlling orifice (lbm/sec).
- \bar{m}_{SI} - Average mass flow rate through the inlet controlling orifice over one cycle (lbm/sec).
- \bar{m}_{DO} - Average mass flow rate through the outlet controlling orifice over one cycle (lbm/sec).
- P_c - Instantaneous stagnation pressure of the gas in the characteristic cylinder (psi).
- P_D - Instantaneous stagnation pressure of the gas in the discharge plenum (psi).
- P_S - Instantaneous stagnation pressure of the gas in the suction plenum (psi).
- P_{PD} - Average stagnation pressure of the gas downstream of the outlet controlling orifice (psi).
- P_{PS} - Average stagnation pressure of the gas upstream of the inlet controlling orifice (psi).
- P_{ID} - Instantaneous stagnation pressure of the gas in the inner discharge plenum (psi).
- P_{DD} - Desired average stagnation pressure of the gas in the discharge plenum (psi).
- P_{SD} - Desired average stagnation pressure of the gas in the suction plenum (psi).
- P_{DA} - Average stagnation pressure of the gas in the discharge plenum over one cycle (psi).
- P_{SA} - Average stagnation pressure of the gas in the suction plenum over one cycle (psi).

V_d - Displaced volume of the characteristic cylinder (in^3).

V_{CL} - Clearance volume of the characteristic cylinder (in^3).

VE - Volumetric efficiency (%).

θ - Crank angle (rad).

PS - Density of the gas in the suction plenum (lbm/in^3).

PD - Density of the gas in the discharge plenum (lbm/in^3).

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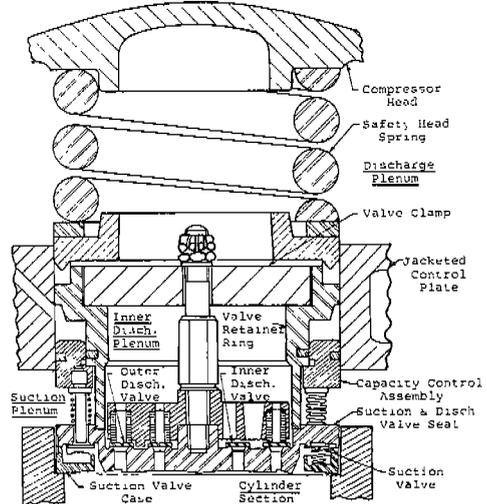


Figure 2 Side View of the Valving System of One Cylinder

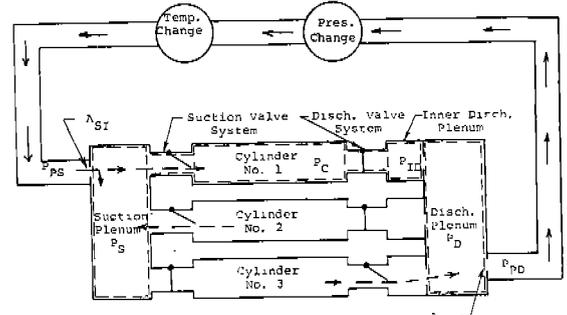


Figure 3 Basic Schematic of Compressor Model to Include Suction and Discharge Plenums

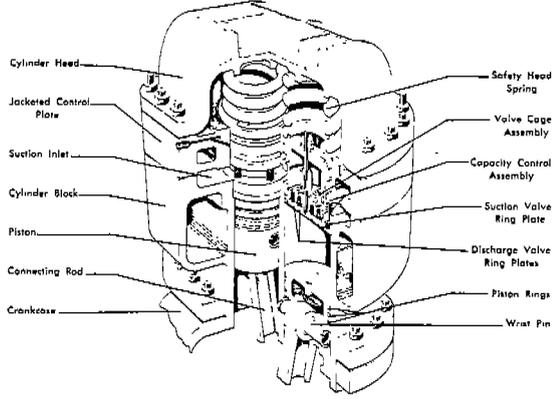


Figure 1 Cut-Away View of Compressor

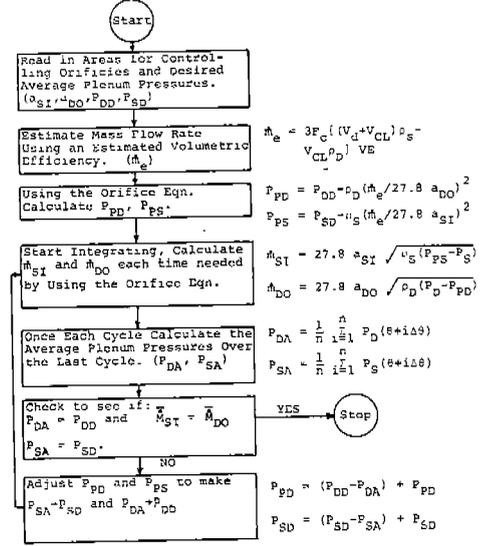


Figure 4 Flow Diagram of the Logic for Stabilizing the Average Plenum Conditions at the Desired Operating Conditions

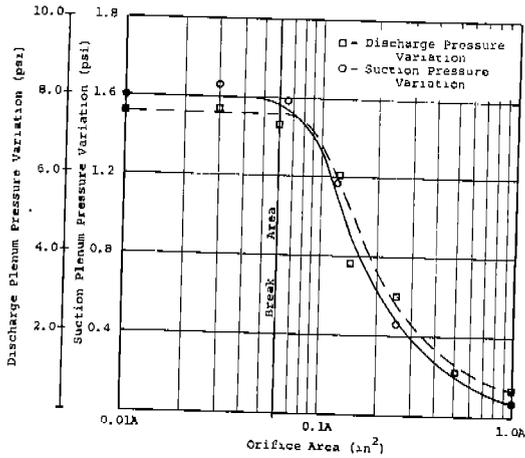


Figure 5 Effect of Controlling Orifice Areas on Pressure Variations in Plenums

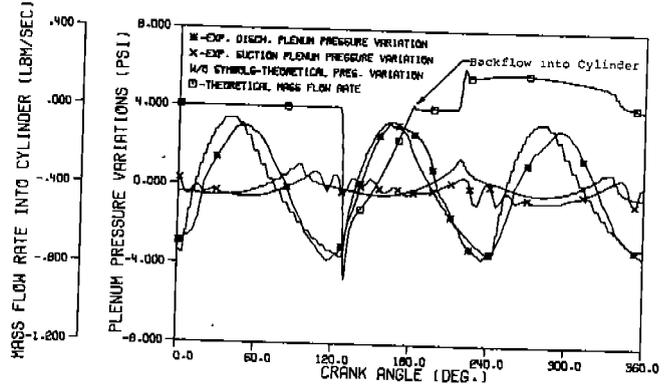


Figure 8 Comparison of Time History Values for One Set of Operating Conditions (Plenum Conditions)

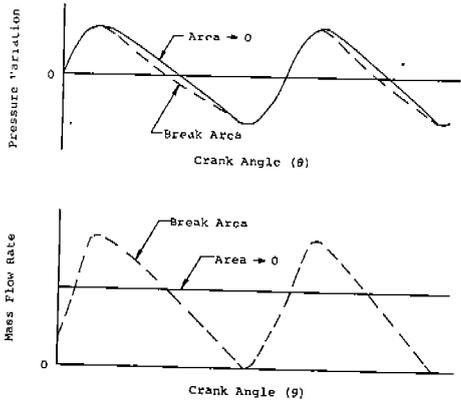


Figure 6 Limit of the Effects of the Area of the Controlling Orifice Below the Break Area on the Discharge Plenum

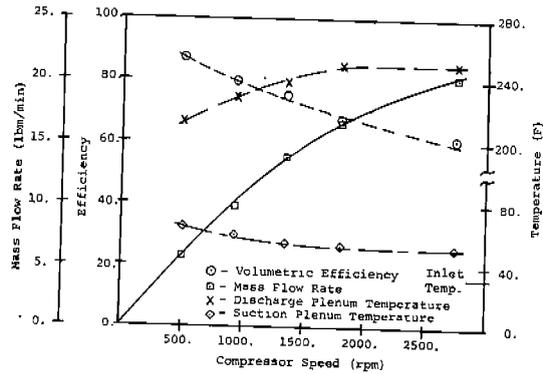


Figure 9 Variation of Theoretical Results with Compressor Speed

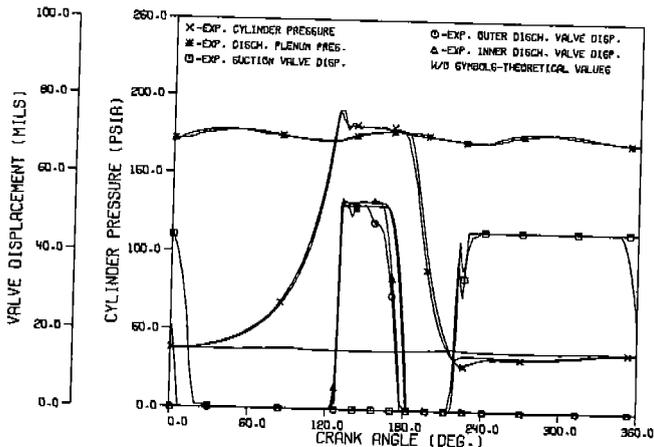


Figure 7 Comparison of Time History Values for One Set of Operating Conditions (Cylinder Conditions)

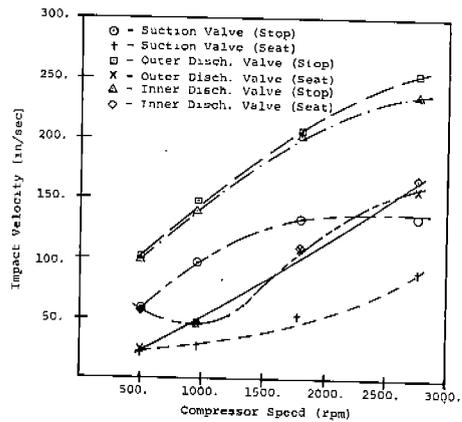


Figure 10 Variation of Valve Impact Velocity with Compressor Speed

Table 1 Comparison of Time Average Values for Three Sets of Operating Conditions

Measurement	Set No. 1		Set No. 2		Set No. 3	
	Exp.	Theo.	Exp.	Theo.	Exp.	Theo.
Discharge Plenum Pres. (psia)	174.7	174.7	195.1	195.1	214.7	214.7
Suction Plenum Pres. (psia)	38.0	38.0	35.3	35.3	29.5	29.5
Inlet Suction Temp. (°R) (Input Constant)	493.4	493.4	493.0	493.0	490.0	490.0
Suction Plenum Temp. (°R)	514.0	518.6	519.0	522.4	522.0	528.7
Discharge Plenum Temp. (°R)	694.0	688.2	721.0	714.6	746.0	741.6
Mass Flow Rate (Orifice) (lbm/min)	9.95	9.94	7.97	8.83	6.09	6.97
Mass Flow Rate (E.B.) (lbm/min)	10.35		8.46		6.93	

Table 2 Theoretical Results of the Analytical Model for Three Sets of Operating Conditions

Value	Set No. 1	Set No. 2	Set No. 3
Volumetric Efficiency	76.8	77.9	70.8
Power Required to Compress the Gas (H.P.)	27.3	28.0	26.3
Power Loss to Suction Valve (H.P.)	1.26	1.14	0.91
Power Loss to Discharge Valves (H.P.)	0.60	0.51	0.38
Suction Valve Impact Velocity with Stop (in/sec)	92.0	89.2	84.7
Suction Valve Impact Velocity with Seat (in/sec)	24.1	24.8	26.3
Outer Disch. Valve Impact Velocity with Stop (in/sec)	144.	154.	159.
Outer Disch. Valve Impact Velocity with Seat (in/sec)	37.9	40.3	43.8
Inner Disch. Valve Impact Velocity with Stop (in/sec)	139.	144.	150.
Inner Disch. Valve Impact Velocity with Seat (in/sec)	33.8	34.4	34.7