1988

Comprehensive Model of a Reciprocating Compressor Applicable to Component Design Issues

Thomas Morel
Integral Technologies Incorporated

Rifat Keribar
Integral Technologies Incorporated

COMPREHENSIVE MODEL OF A RECIPROCATING COMPRESSOR APPLICABLE TO COMPONENT DESIGN ISSUES

Thomas Morel and Rifat Keribar

Integral Technologies Incorporated
Westmont, Illinois

ABSTRACT

A comprehensive reciprocating compressor simulation has been developed, capable of analysis of a variety of performance issues as well as issues related to component design. The thrust of its development is towards an integrated engineering software package, which can simulate many of the important compressor processes and describe their essential interactions. Then if changes are made in one part of the hardware or in operating conditions, the effects of these changes on performance and on other components of the system can be assessed. Among the component issues that the code can address are temperature distributions in the compressor structure, piston/cylinder friction, blowby, valve flow and dynamics, and pressure dynamics in compressor piping.

INTRODUCTION

The design and development of reciprocating compressors is an increasingly sophisticated process, driven by competitive pressures and technical factors such as new materials and controls. In addition, the complexity of compressor systems dictates that this process encompasses a wide variety of design and development subtasks such as those indicated in Table I. As a result, the development becomes a sophisticated process bringing together a multitude of specialists. It is important that all of these specialists work together as a team and communicate effectively. In a steadily growing fashion, computer based methods are being applied to assist in this activity. These include focused simulation codes developed specifically for the analysis of compressors. An example of such a simulation is e.g. the work of Singh (1).

Table I. Some Compressor Design and Development Issues

Performance	Structural
Efficiency	Component thermal loads
	Component temperatures
Breathing	Thermal stresses
Volumetric efficiency	Mechanical stresses
Valve dynamics	Cooling system design
Effective flow areas	
Suction/discharge manifolds	Friction and Lubrication
	Ring and piston design
Dynamics	Minimum friction
Compressor balance	Lubricant distribution
Crankshaft vibration	
Bearing loads	

In this paper we describe a comprehensive approach to compressor simulation which is based on a long-term development undertaken in the related internal combustion engine area, extended now to reciprocating compressors. The original engine-related work has been described in several publications (2-5). A salient point of the methodology is the introduction of dimensionality, spatial resolution, previously not available. This development was specifically undertaken to address a deficiency of previous simulation methodologies which, by their nature, separate consideration of performance from component design issues. In a new approach to simulation codes, these domains have been integrated into a single overall computer-based methodology so that the design and development issues enumerated in Table I can be approached from a comprehensive system perspective. We believe that this approach provides a much more effective engineering tool in arriving at an optimized new or upgraded compressor.

In this paper we describe the overall capabilities of the simulation code, named IRIS-C. The details of the methodology used for calculation of heat transfer are described in a companion paper (6).

OVERVIEW OF THE SIMULATION

As already indicated, the simulation that has been developed is meant to be broad based, incorporating models of many different compressor processes and solving them in an integrated fashion, taking into account all of their interactions. Most of the issues enumerated in Table I have already been addressed in the code, and at the present time the code has the following capabilities:

- performance analysis
- thermal analysis of structure
- valve dynamics
- piping dynamics
- mechanical friction: piston/rings/cylinder, bearings
- blowby
- single or multi-cylinder configuration
- steady-state or transient operation.

In the code development, every effort was made to make the use of the computer code convenient in order to remove barriers to its frequent use. Thus, the code is highly efficient computationally. The input is modular, with on-line instructions for interactive input file generation. The output includes menu-driven plotting for several hundred variables, plotted as a function of crank angle for steady-state runs, and also against time for transient runs. This allows rapid analysis of results and permits extraction of the details of many processes.

The code is computationally efficient, executing in one to five minutes per one shaft revolution, depending on the detail and extent of the piping and on the detail of the FEM structural representation.

THERMODYNAMIC FORMULATION

Standard first law of thermodynamics is used to calculate the evolution of in-cylinder pressure and temperature:

$$(me)^{\cdot} = -p\dot{V} + \Sigma \dot{m}_i h_i + \text{heat transfer}$$

where e is internal energy, m is mass, p is pressure, V is cylinder volume, $\dot{m}$ is mass flux through valves and blowby, and h is enthalpy associated with the mass flux. The equation of state used currently is for perfect gas with temperature and pressure dependent properties. An extension to non-perfect ("real") gases such as refrigerants is planned.

VALVE FLOWS AND VALVE DYNAMICS

Flow through valves is calculated using standard isentropic compressible orifice flow equations. Those equations are used on quasi-steady basis, i.e., the instantaneous mass flow rate past the valve is calculated as the steady flow through an orifice of a given effective area at the instantaneous values of upstream pressure and temperature and of downstream pressure. The effective valve areas can be determined experimentally in steady-state bench tests, in which a known upstream pressure is applied to the port, and the resulting mass flow is recorded as a function of valve lift.

If the suction port/valve configuration produces a measurable angular momentum flux into the cylinder, the resultant in-cylinder swirl has to be taken into consideration in the flow model. This is done by integrating the incoming angular momentum flux, using experimentally determined swirl coefficients. These can be measured by a honeycomb straightening device equipped with a torque meter, installed in the steady-state test rig. The instantaneous valve effective areas are calculated from the instantaneous valve lift through simulation of the valve dynamics. This involves the integration of the equation of motion for each valve, as done e.g. Singh (1):

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

where D is the damping force, K is the spring constant, F is the pressure force acting on the valve and x is the instantaneous lift. All three quantities D, K and F may be non-linear and are functions of lift. They can be calibrated by steady-state tests.

HEAT TRANSFER AND THERMAL ANALYSIS

One of the most unique features of the IRIS-C is its capability to calculate accurately heat transfer and component temperatures. The model it uses for calculation in-cylinder heat transfer is based on a well established boundary layer representation which utilizes a flow model describing all of the major in-cylinder fluid motions. The model allows for bowl-in- piston and recessed head gemetries, dividing the cylinder volume into four regions and solving for the flow motions in each. The predicted gas-to- wall heat transfer is coupled with full structural (FEM) calculations of heat conduction in the compressor structure to provide a full solution for temperature distributions within it. All performance parameters are computed at the same time, and thus each simulation produces a full description of the compressor operation as needed for design purposes. As already mentioned, this aspect of the simulation is described in detail in a separate paper (6).

WAVE DYNAMICS IN SUCTION/DISCHARGE SYSTEMS

If pressure wave dynamics in the suction/discharge system and piping design are of interest, one has to solve more detailed gas dynamic equations describing the flow and pressure pulses. This requires calculations tracing pressure waves, mass flows and energy losses in ducts, plenums and manifolds of suction and discharge systems. The code solves the governing equations of compressible flow (conservation of mass, momentum and energy) in networks of pipes, volumes and junctions. The model provides for gradual as well as abrupt area changes, as well as for Y-junctions where more than two runners or ducts meet. It can analyze a comprehensive set of suction/discharge geometrical configurations (schematically shown in Figure 1), including:

constant area runners,
runners with gradually varying area,
junctions of several runners,
runners with controllable geometry,
elbows,
orifices,
valves, and
plenums.

The details of the flow in the suction and discharge systems are obtained as a solution of quasi-one dimensional compressible flow equations governing the conservation of mass, momentum and energy. There have been numerous publications describing methods for calculation on pressure dynamics, either by the method of characteristics (e.g. Benson and Woods, (7)), or by finite difference techniques (e.g. Chapman (8)). In the present approach, the entire system is discretized into a multitude of small volumes and the three governing equations are written in a finite difference form for each of these elementary volumes. A staggered mesh system is used, with equations of mass and continuity solved for each volume, and momentum equation solved for each interface between two volumes. As such the method is similar to that of Chapman et al (8)), but it uses different energy equation formulations and different handling of flux terms. The equations are written as follows:

mass: $$\frac{dm}{dt} = \Sigma\dot{m}$$

energy: $$\frac{dme}{dt} = \Sigma\dot{m}h + \text{sources}$$

momentum: $$\frac{dmu}{dt} = -A\frac{dp}{dx} + \Sigma\dot{m}u - \text{losses}$$

A variety of sources and loss elements are considered in order to accurately represent practical systems: heat transfer, wall friction, elbow losses, distributed pressure losses, and pipe entrance losses. Losses at abrupt area changes are accounted for directly by the finite difference equations.

The solution of the governing equations is obtained by the application of a finite difference technique utilizing the finite-volume approach to the discretization of the partial differential equations. The time-differencing is based on the explicit technique, with the time step governed by the Courant condition. This approach is superior to the method of characteristics, because it does not require the inaccurate and cumbersome ad hoc treatment of many of the terms of the governing equations and of the geometry, such as:

a) source terms such as heat transfer, friction, and distributed losses;

b) boundary conditions at locations of abrupt area change;
c) junctions of multiple ducts; and
d) bends.

The present approach, on the other hand, because of its basic formulation and solution technique is able to handle these effects accurately and with no special difficulty.

The basic methodology has been extensively tested against a set of reference test cases, including shock wave propagation in a duct, pressure wave reflection from closed and open ends of a duct, steady state flow through a duct with an abrupt change of cross-sectional area, flow through an orifice, pipe flow with friction, pipe flow with heat transfer and flow through junctions of three ducts.

BLOWBY PAST RINGS

The land crevice between piston and cylinder (above top compression ring), and the gas flow into and out of the land crevice, has an effect on the compressor delivery rate and efficiency. The mass flow of gas entering (or leaving) the crevice from or to the cylinder, and the crevice gas temperature are tracked by simultaneous solution of mass and energy equations for crevice gases, as in a number of prior studies e.g. Ruddy et al (9) and Namazian and Heywood (10). The solution accounts for the enthalpy flux from the cylinder and from the inter-ring spaces below, as well as for heat transfer to crevice walls.

Gas flow through the piston ring pack (blowby) is simulated as a flow through a series of orifices (ring gaps) and volumes (inter-ring spaces). Unsteady isentropic flow is assumed through gaps. At each crank angle during the computation the gas in each inter-ring space is assumed to be at the temperature of the walls it contacts (due to large heat transfer coefficient and large surface to volume ratio). The exception is the volume above the top ring for which energy equation is solved to calculate the gas temperature and the heat transfer to the adjacent walls. The instantaneous wall temperature is calculated for each land using the position of the piston and the cylinder temperature profile.

An example of the blowby and inter-ring calculations is shown in Figure 2. It shows the cumulative mass flow past the top of the piston and past three rings of the piston which will be discussed later (Figure 2a). It is seen that there is a strong outflow from the cylinder during the compression, followed by a return flow as pressure in the cylinder decreases and falls below the pressure trapped in the crevice volumes (Figure 2b). This trend is the most visible in the top crevice volume above the first ring. The residual value of the cumulative mass flow equals the time-mean blowby from the cylinder past the bottom ring. Both the blowby as well as the volume of the crevice volume above the top ring have a significant effect on volumetric efficiency and on the isothermal efficiency and must be minimized by proper ring piston design.

COMPRESSOR FRICTION

Friction is modeled in order to compute the input shaft work and also to predict the magnitude, distribution and effect of frictional heat generation. The model is quite comprehensive, involving a detailed crank-angle-resolved calculation of ring and piston friction, with the contributions of bearings (crankshaft, connecting rod, wrist pin) lumped in a cycle-averaged manner.

For piston rings, the Reynolds equation is solved for the oil layer under the ring and it is coupled with the calculation of the radial force balance on the ring, consisting of gas pressure, ring pre-tension and oil pressure forces. This approach is similar to that employed by Ting and Mayer (11) and Rhode (12). From these, one obtains the oil film thickness, oil hydrodynamic pressure/shear stress distributions, and thus hydrodynamic friction. This computation uses as input the ring dimensions and ring face profile. When film thickness is of the order of the mean height of surface asperities, the model includes the calculation of the asperity contact (elastic deformation) forces in the ring radial force balance and computes boundary friction due to asperity contact. An example of the calculated oil film thickness is shown in Figure 3. The oil thickness is seen to vary under the influence of instantaneous piston velocity, which tends to increase it, and gas pressure near TDC to a low value where it encounters a mixed lubrication regime.

The piston friction receives a contribution from hydrodynamic lubrication of the skirt. Instantaneous piston side force is tracked by solving piston-reciprocator dynamic equations at each time step. This force produces boundary and hydrodynamic friction between piston and cylinder, adding to the total piston friction force. The bearings are assumed to operate in the hydrodynamic regime and the friction they produce is calculated based on that assumption.

APPLICATION TO A SMALL AIR COMPRESSOR

An analysis was made of small three-cylinder single-acting air compressor with 75 mm bore and 56 mm stroke at nominal operating conditions defined by 1750 rpm, suction pressure of one bar, discharge pressure of 5 bars and clearance volume of 4 percent.

Operation at Baseline Conditions

At this operating point the compressor delivered 59.5 kg/hr of air, requiring input power of 6.1 HP. This corresponds to a volumetric efficiency of 68 percent and isothermal efficiency of 25.7 percent; compressor efficiency expressed at the discharge temperature of 514 K was 42.9 percent. These efficiencies are defined as

$$(\eta)i = \dot{m}\Delta p\ (RT_{suction}/p_{disch})/\text{input work}$$

$$(\eta)d = \dot{m}\Delta p\ (RT_{disch}/p_{disch})/\text{input work}$$

Some of the detailed compressor operating parameters are displayed in Figures 4 to 8. Of these, Figure 4 shows the pressure-volume diagram indicating the areas of useful compression work as well as the suction and discharge losses. The valve dynamics are seen in Figure 5. Worth noting is the dynamics of the discharge valve, which is not necessarily optimized for this operating point. It closes just before TDC to open again, and then it closes after TDC. During the period of the last closing there can be significant backflows into the cylinder, which affects the performance as discussed below. The magnitude of these backflows depends on the details of the discharge valve dynamics.

There is also a significant pressure dynamics that takes place in the suction and discharge piping. On the suction side, pressure oscillation is set up in the suction cavity with magnitude equal to 20 percent of the suction pressure (Figure 6). On the discharge side the pressure oscillation is about twice as large relative amplitude, or 4.5 bar absolute magnitude (Figure 7). The amplitude of the oscillation and its shape strongly depends on the details of the piping, i.e. the sizes of the volumes, pipe diameters and lengths. Figure 8 shows the result of a change in the length of the pipe connecting the discharge head cavity to a pressure tank from 0.9 m to 0.5 m. The differences seen between Figures 7 and 8 are due to wave reflections in the piping system which combine with the primary driving pulses from the compressor discharge valves.

Parametric Study of Some of the Design Parameters

The dependence of the compressor volumetric efficiency and isothermal efficiency on the pressure ratio (discharge pressure/section pressure) is shown in Figure 9 for several values of clearance volume (expressed in percent of displacement) at constant compressor speed of 1750 rpm. It can be seen that the volumetric efficiency decreases with increasing pressure ratio. This decrease is mainly due to the finite mass of air compressed in the clearance volume which cannot be discharged. This trend is accentuated by increasing the size of the clearance volume, and as can be seen at the pressure ratio of 12 and clearance volume of 10%, the effect on volumetric efficiency becomes very large. As pressure ratio is increased, the volumetric efficiency for any given clearance volume eventually reduces to zero as the trapped gas cannot be compressed to high enough pressures to be discharged; the compressor then operates as a gas spring compressing and expanding the same gas. The isothermal efficiency of the compressor increases at first, but then decreases monotonically with further increases in pressure ratio. This decrease is mainly the consequence of increasing discharge temperature, while $(\eta)d$ shows much less variation. The clearance volume has a negative influence on the $(\eta)i$ (Figure 10) and even stronger influence on $(\eta)d$.

The effect of compressor rpm is shown in Figures 11 and 12 for a fixed pressure ratio of five. The volumetric efficiency is seen to decrease monotonically with compressor speed as pressure losses at the valves increase. Increasing the clearance volume has a negative influence because it decreases the volumetric efficiency. The isothermal efficiency decreases by almost 20% over the speed range, and so does $(\eta)d$ (discharge temperature rises only by about 35 K over the speed range).

Further parametric studies, which are related to heat transfer, have been presented in Reference (6).

CONCLUSIONS

1. Design and development of compressors demands an increasing level of communication among those responsible for individual subsystems and components to improve the performance of the final product.

2. Such communication can be greatly improved by application of sophisticated systems software which integrates the analysis of all key subsystems and accounts for essential interactions among them. Changes in one subsystem can then be analyzed for their effects on other subsystems over a range of operating conditions. The software also can serve as a repository of up-to-date information about the entire compressor that all involved can share.

3. One approach to development of such systems software is the code described in this paper, structured to allow simultaneous analysis of compressor performance, temperature distributions in compressor components, piston/cylinder friction, blowby, valve flow and dynamics, and pressure dynamics in compressor piping.

REFERENCES

1. Singh, P. J. (1984), "A Digital Reciprocating Compressor Simulation Program Including Suction and Discharge Period", Purdue Compressor Technology Conference, 1984, pp. 129-138.

2. Morel, T., Keribar, R. and Blumberg, P. N. (1988), "A New Approach to Integrating Engine Performance and Component Design Analysis Through Simulation", SAE Paper 880131.

3. Morel, T. and Keribar, R. (1985), "A Model for Predicting Spatially and Time Resolved Convective Heat Transfer in Bowl-in-Piston Combustion Chambers", SAE Paper 850204.

4. Morel, T., Wahiduzzaman, S., Tree, D. L. and DeWitt, D. P. (1987B), "Effect of Speed Load and Location on Heat Transfer in a Diesel Engine--Measurements and Predictions", SAE Paper 870154.

5. Keribar, R. and Morel, T. (1987), "Thermal Shock Calculations in IC Engines", SAE Paper 870162.

6. Keribar, R. and Morel, T. (1988), "Heat Transfer and Component Temperature Predictions in Reciprocating Compressors", Purdue Compressor Technology Conference, 1988.

7. Benson, R. S. and Woods, W. A. (1960), "Wave Action in the Exhaust System of a Supercharged Two-Stroke Engine Model", Int. Journal of Mech. Sci., Vol. 1, pp. 253-281.

8. Chapman, M., Novak, J. M. and Stein, R. A. (1982), "Numerical Modeling of Inlet and Exhaust Flows in Multi-Cylinder Internal Combustion Engines", ASME Symposium on Flows in Internal Combustion Engines, November 14-19, 1982, T. Uzkan, Editor.

9. Ruddy, B. L., Dowson, D. and Economou, P. N. (1981), "The Prediction of Gas Pressures Within the Ring Packs of Large Bore Engines", J. of Mech. Eng. Sci. 23 #6, pp. 295-304.

10. Namazian, M. and Heywood, J. B. (1982), "Flow in the Piston-Cylinder- Ring Crevices of a Spark-Ignition Engine: Effect on Hydrocarbon Emissions, Efficiency and Power", SAE Paper 820088.

11. Ting, L. L. and Mayer, J. L. (1974), "Piston Ring Lubrication and Cylinder Bore Wear Analyses, Part I - Theory", Trans. ASME, Journal of Lubrication Technology, July 1974, pp. 305-314.

12. Rhode, S. M. (1980), "A Mixed Friction Model for Dynamically Loaded Contacts with Application to Piston Ring Lubrication", Proceedings of the 7th Leeds-Lyon Symposium on Tribology.

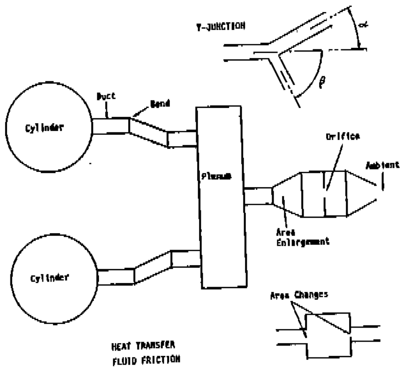

Figure 1. Schematic of suction/discharge configurations and elements that can be represented and analyzed.

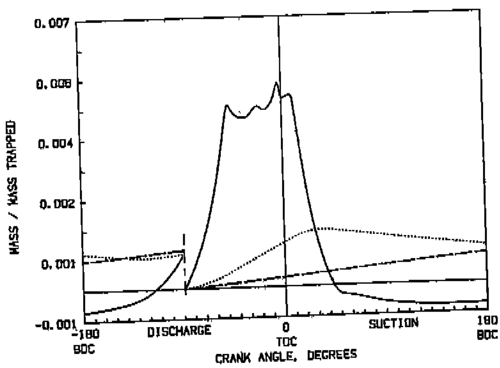

Figure 2. Cumulative mass flow (blowby) past top piston plane and past piston rings. ——— top crevice, ... top ring, - - - second and third rings.

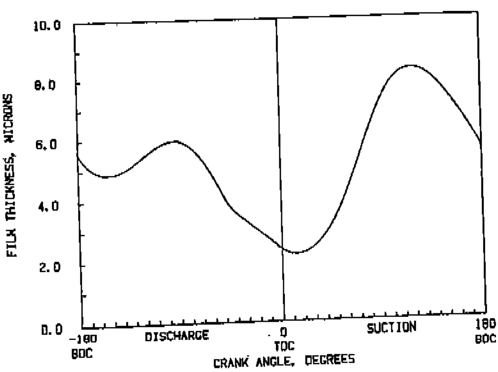

Figure 3. Oil film thickness variation under the top ring.

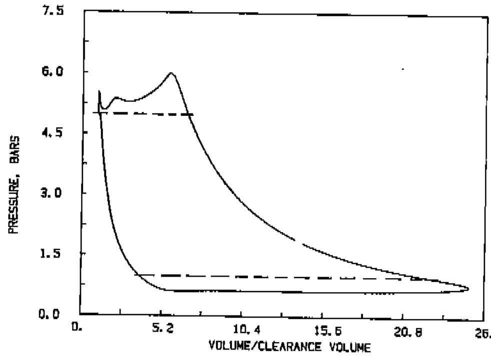

Figure 4. Pressure-volume diagram for the baseline compressor operating point.

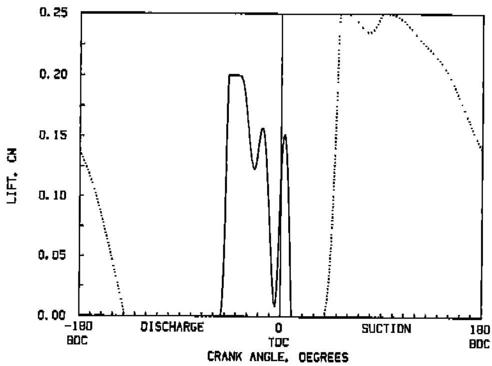

Figure 5. Suction and discharge valve lift profiles showing the valve dynamics. ——— discharge, ... suction.

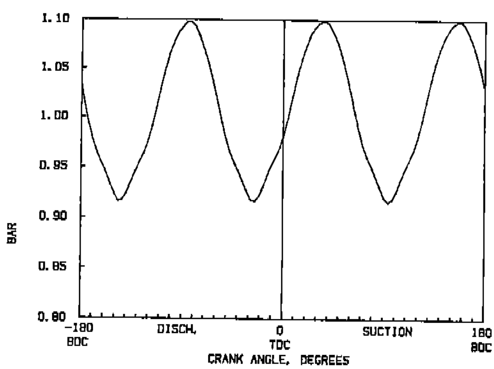

Figure 6. Pressure variation inside the head suction cavity.

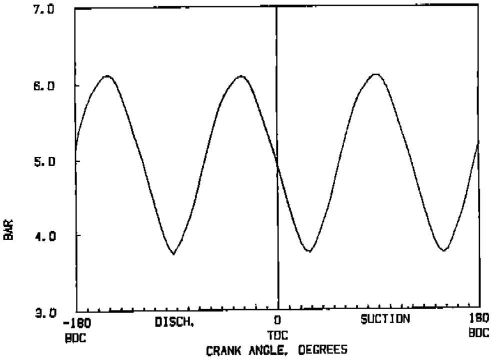

Figure 7. Pressure variation inside the head discharge cavity, connecting pipe length L = 0.9 m.

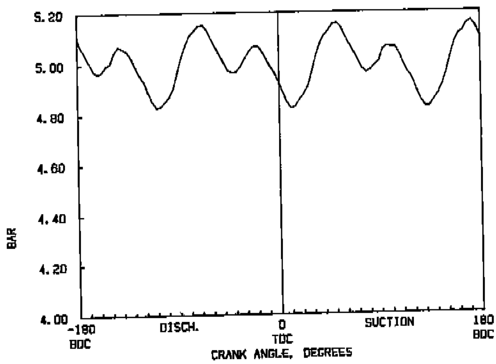

Figure 8. Pressure variation inside the head discharge cavity, connecting pipe length L = 0.5 m.

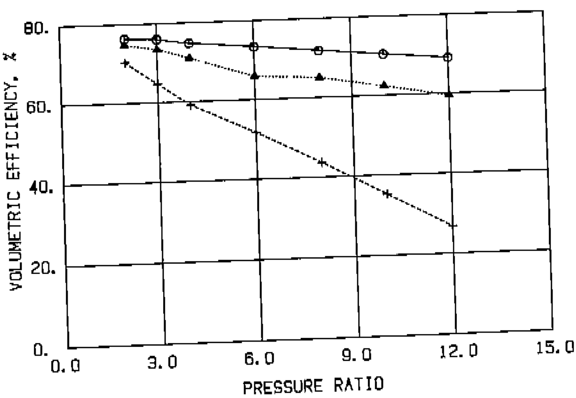

Figure 9. Effect of discharge pressure and clearance volume on volumetric efficiency. Clearance volume: ——— 2%, ... 4%, - - - 10%.

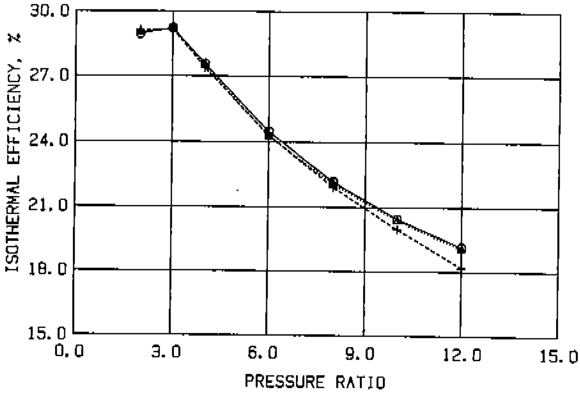

Figure 10. Effect of discharge pressure and clearance volume on isothermal efficiency. Clearance volume: ——— 2%, ... 4%, - - - ,10%.

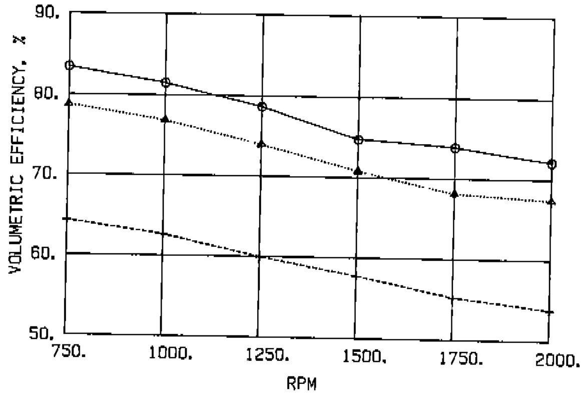

Figure 11. Effect of compressor rpm and clearance volume on volumetric efficiency. Clearance volume: ——— 2%, ... 4%, - - - 10%.

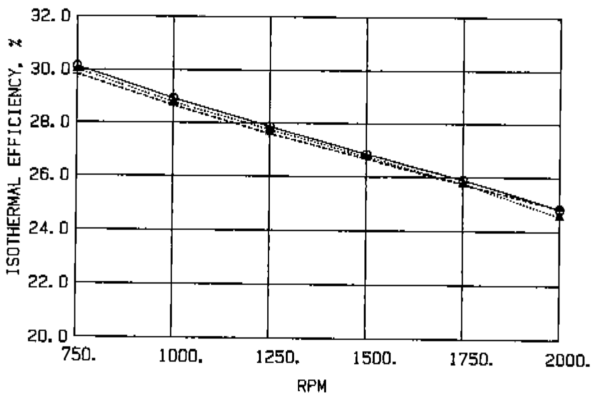

Figure 12. Effect of compressor rpm and clearance volume on isothermal efficiency. Clearance volume: ——— 2%, ... 4%, - - - 10%.