1988

A Cautionary Note on Reciprocating Compressor Valve Design

A. B. Tramschek
University of Strathclyde

R. J. Boyle
National Engineering Laboratory

A CAUTIONARY NOTE ON RECIPROCATING COMPRESSOR VALVE DESIGN

A.B.Tramschek,
University of Strathclyde,
Glasgow
Scotland, U.K

R.J. Boyle,
National Engineering Laboratory,
East Kilbride,
Scotland, U.K.

ABSTRACT

This paper highlights the authors' experiences during a project involving the re-design of the valve system of a multi-cylinder reciprocating refrigeration compressor. Starting with a statement of the initial design brief, the paper traces the project development from the conceptual design stage through preliminary assessment procedures to the selection of a prototype design. Details of the prototype design studies and initial operating experiences with the prototype under laboratory conditions are cited. The paper gives details of the steps taken to overcome problems which arose with the prototype and concludes with a comparison of the performance of a compressor fitted with the replacement valve system against that of the performance predicted in the design calculations.

INTRODUCTION

The operating characteristics of the self-acting valves in any reciprocating compressor can have a profound effect upon the overall performance of the compressor and a poorly designed valving system can result in an inefficient compressor An analytical study of the performance of a particular multi-cylinder semi-hermetic refrigeration compressor fitted with cantilever reed type suction and discharge valves (flapper valves) had predicted significant power losses associated with the valves under certain operating conditions and had also suggested that the retention of the flapper valve concept in this particular machine offered little scope for performance improvement. Accordingly a design brief was formulated whereby the performance of the compressor might be uprated. The brief required that:

a) a valve design be evolved for use with an existing range of compressor cylinder blocks

b) the suggested design should not increase and should ideally reduce the cylinder clearance volume

c) the performance (throughput for a given power input) of compressors fitted with the new valve system be superior to that of unmodified compressors

d) the reliability of the replacement valve system be at least as good as and preferably better than that of the existing valve system

e) the proposed valve system should be economic in manufacturing terms.

INITIAL DESIGN STUDY

An examination of the layout of the valves on the valve plate in the existing machines (see Figure 1) indicated that the valve arrangement made poor use of the cross sectional area of the valve plate directly above the cylinder bore. In fact, the total valve port area represented only 25% of the piston cross sectional area. A preliminary study was made in which the effects of changing a number of valve design parameters (in particular valve port area) upon compressor performance were examined using a simple mathematical model (1). Results of this study (2) indicated that there was a premium on minimising "waste area" and that at given operating conditions there was an optimum combination of suction and discharge valve port areas which would result in minimum total valve power loss. Figure 2 illustrates the nature of these findings.

As a result of these preliminary studies a number of possible valve design arrangements were conceived, all of which met the broad design requirements. At this stage relatively little attention was given to manufacturing detail as it was argued that if one of the conceptual design arrangements showed clear performance advantages, then suitable manufacturing techniques could be devised to ensure economic production.

Five conceptual designs which comprised combinations of circular and annular ports were subjected to further analysis in which the following assumptions were made:

(a) the maximum valve flow area was made equal to the valve port area

(b) valves sat on knife-edged seats

(c) valve spring stiffness could be selected albeit somewhat arbitrarily but nevertheless in a manner which would yield satisfactory valve action

(d) plenum chamber pressures were constant

(e) oil stiction effects could be neglected

The application of assumption (a) occasionally resulted in a valve lift which was considered to be excessive and when this occurred the performance of the valve was examined for a range of values of maximum permitted valve lift. Calculations were performed for each of the proposed valve designs at two extreme operating conditions using R502 as the working fluid. Details of the test conditions are given in Table I. In each case the refrigerant was assumed to be superheated by 20°C at entry to the compressor.

TABLE I

Condition	Evaporating Temperature °C	Condensing Temperature °C	Suction Pressure bar	Discharge Pressure bar	Pressure Ratio
1	−40	−20	1.3	2.9	2.2
2	−40	55	1.3	23.4	18.0

It was thus possible to determine values for parameters such as valve impact velocities, valve power loss, cycle work, volumetric efficiency, maximum pressure drop across a valve and thereby effect a comparison of the performance of each of the conceptual designs with the performance of the existing valve system. An evaluation scheme was employed in which points were awarded or deducted when the above parameters showed an improvement or a deterioration with respect to the values pertaining for the existing design.

A design comprising a centrally placed disc shaped discharge valve surrounded by an essentially annular ring shaped suction valve emerged as that demonstrating the greatest overall performance improvement. Other features of the design, i.e., the slightly crowned piston, the incorporation of suction valve springs in pockets in the cylinder block are shown in Figure 3.

Having identified the most promising candidate design, this design was subjected to a detailed optimisation procedure covering a wide range of operating conditions.

VALVE DESIGN OPTIMISATION

The task of determining the combination of valve design parameters such as valve mass, valve spring stiffness, maximum permitted valve lift which would yield optimum compressor performance was performed using a constrained direct search technique based on the Complex Method of Box (3), as modified by Hoare (4) and the simple mathematical model (1).

It is extremely difficult to design self-acting compressor valve systems which will operate completely satisfactorily over a wide range of operating conditions. A valve system optimised for operation at one condition will almost inevitably have an inferior performance when made to operate at other conditions. This aspect of the problem is clearly illustrated in Figure 4, which shows the P-V diagrams for designs which had been respectively optimised (made to produce minimum power losses for maximum throughput) at the test conditions 1 and 2, Table I, when the compressor runs at test condition 1, i.e. at a low pressure ratio condition. The design optimised for the high pressure ratio condition, condition 2, displays significant cylinder over pressure and excessive discharge valve power losses when actually run at condition 1 compared with the design specifically optimised at condition 1.

Actual compressors seldom operate exclusively at one given condition and an element of compromise is necessary in translating the findings of an optimisation study into a practical design. Having determined the optimum values of the design parameters of the selected design for a wide range of operating conditions a final choice of values was made for the production of a prototype valve assembly.

PROTOTYPE VALVE DESIGN

Figure 5 shows a cross sectional view of the valve arrangement whilst Figures 6 and 7 give details of the valves and the valve plate respectively. The valve port area amounting to approximately 50% of the piston cross sectional area was evenly divided between the suction and discharge valve ports. All valves were manufactured from 0.9 mm thick spring steel sheet using a spark wire erosion technique followed by a tumbling process.

Valve springing was provided by 4 small helical coil springs per valve in the case of the suction valves and by single semi-elliptical leaf springs in the case of the discharge valves. These latter springs were produced in the Departmental workshop and inadequate heat treatment probably contributed to subsequent operational difficulties.

It is worth noting that the prototype valve design incorporates a horizontal separation of the valve plenum chamber regions whereas the original valve design used a vertical separation between the plenum chambers.

INITIAL TESTS OF PROTOTYPE VALVE DESIGN

Breakage of the discharge valves and springs occurred early in the initial testing programme and resulted in damage to the valve seats in cylinder No.2. Figure 8 shows photographs of the broken components. Prompted by these valve and spring failures, the discharge valve assembly was re-designed so that it incorporated a modified valve stop and a helical coil compression spring. Details of the modified assembly are shown in Figure 9.

The compressor was re-assembled and testing was resumed. During the testing programme a loss of pumping capability was suspected and when the cylinder head assembly was removed for inspection purposes it was discovered that two of the discharge valve coil springs were broken but all the valves were intact. It was inferred that during operation the discharge valve springs had been rotating and had worn grooves in the radial spring support ribs located at the upper end of the spring support/valve stop assembly. The wear problem was overcome by the introduction of a hardened steel wear ring, See Figure 10, on which the upper end of the discharge valve spring could bear. As a precaution all the seals in the prototype cylinder head were renewed and additional head bolts were provided in the vicinity of the central cylinder. An examination of the solenoid unloading valve fitted to the machine revealed two small pieces of coil spring near the unloader valve seat. It was judged that the unloader valve had been prevented from seating correctly and that refrigerant had thus passed directly from the discharge plenum chamber back to the suction valves. When the compressor was next run a significant increase in refrigerant mass flow rate was recorded, thereby justifying the various steps taken to minimise any internal leakage within the compressor.

However, after just a few hours further running another discharge valve breakage occurred. The breakage pattern suggested that the discharge valves were approaching the discharge valve stop obliquely, experiencing a point contact with consequent high impact stress levels. This hypothesis was confirmed by a simple finite element stress analysis in which the valve was assumed to have been supported over a very small region. A final design modification was introduced in which an additional valve stop was provided, see Figure 11. The testing programme was resumed and no further valve breakages were experienced.

RESULTS OF MAIN TEST PROGRAMME

The modified compressor was tested over a wide range of operating conditions and the test results were compared to those obtained from an identical compressor fitted with the standard valve system and operating under similar conditions. Figure 12 shows typical pressure volume diagrams obtained from the two machines. Similar diagrams were recorded at various test conditions and the following general conclusions could be drawn:

1) The compressor fitted with the modified valve system has a smaller clearance volume than the unmodified compressor and as a consequence the suction valves open earlier in the compressor cycle.

2) As a consequence of the reduced clearance volume and an extended suction process the indicated cycle work is greater for the modified compressor than that of the original compressor.

3) The modified compressor experiences reduced valve power losses compared to the original compressor

Judged from the point of view of volumetric efficiency and valve power loss the modified compressor is superior to the original compressor. However, when compared on a basis of specific capacity the performance of the modified compressor was not significantly different from that of its predecessor.

A possible explanation for the observed effects is believed to lie in the paths taken by the refrigerant prior to its entry through the suction valves in the two machines. In the original machine the suction gas entered a relatively large plenum chamber through two holes in the valve plate, dispersed through the chamber and was drawn into the cylinders through pairs of suction valve ports. Any heat transfer to the suction gas was largely by conduction through the vertical wall which separated the suction and discharge plenum chambers.

In the modified machine suction gas entered the plenum chamber through holes in the valve plate as before but in dispersing through the plenum chamber the suction gases came into contact with the large horizontal plate which separated the suction and discharge plenum chambers. The suction gases were also drawn over the vertical cylindrical surfaces which separated the suction and discharge valve assemblies and could receive heat from the walls of an additional transfer passage which was required to convey hot discharge gases from the discharge plenum chamber to a pre-existing discharge passage in the compressor cylinder block.

It was calculated that there was three times as much heat transfer surface area between the suction and discharge regions of the cylinder head in the modified design compared with that in the original cylinder head. The heat transfer from the hot discharge gas to the cooler suction gas was believed to have raised the effective suction temperature and thus reduced the specific volume of the refrigerant actually entering the compressor cylinders. This argument was reinforced by measurements of gas temperature close to the suction valve ports. Measurements of the mass flow rates downstream of the compressor compared well with those calculated from the knowledge of the pressure and temperature at the suction valve ports and the indicated volumetric efficiency deduced from the P-V diagram.

Finally, Figures 13 and 14 compare measured and predicted P-V and valve displacement diagrams for the modified compressor. There is general agreement between the measured and calculated pressure variations and valve actions. A full account of this research is to be found in reference (5).

CONCLUSIONS

Computer modelling and optimisation studies can be of great help to compressor designers in pointing the way to the achievement of performance improvement. Whilst modelling exercises should be kept as simple as possible, the authors' recent experiences highlight the necessity for an awareness of possible deficiencies in any modelling program which may undermine the results produced by that program. In the present case a detailed heat transfer analysis was not part of the modelling process and in consequence gains in compressor performance arising from reduced clearance volume and a reduction in valve power losses were negated by unlooked for heat transfer effects.

If there is a moral to this cautionary tale it is that it is always possible to be wise after an event.

REFERENCES

1. MACLAREN, J.F.T. and KERR, S.V. An Analytical and Experimental Study of Self-Acting Valves in a Reciprocating Air Compressor. IMechE Conference: Industrial Reciprocating and Rotary Compressors Design and Operational Problems. Proc. Inst. Mech. Engrs., 1969-70, Vol. 184, 3R.

2. BOYLE, R.J., TRAMSCHEK, A.B., BROWN, J. and MACLAREN, J.F.T. The Apportioning of Port Areas between Suction and Discharge Valves in Reciprocating Compressors. Proc. Purdue Compressor Technology Conference, 1982.

3. BOX, M.J. A New Method of Constrained Optimization and a comparison with other methods. The Computer Journal, Vol. 8, 1965.

4. HOARE, R.G. Optimization of Reciprocating Compressor Design, PhD Thesis University of Strathclyde, 1982.

5. BOYLE, R.J. Valve Design Optimisation for a 3-cylinder Semi-Hermetic Reciprocating Refrigeration Compressor. PhD Thesis, University of Strathclyde, 1985.

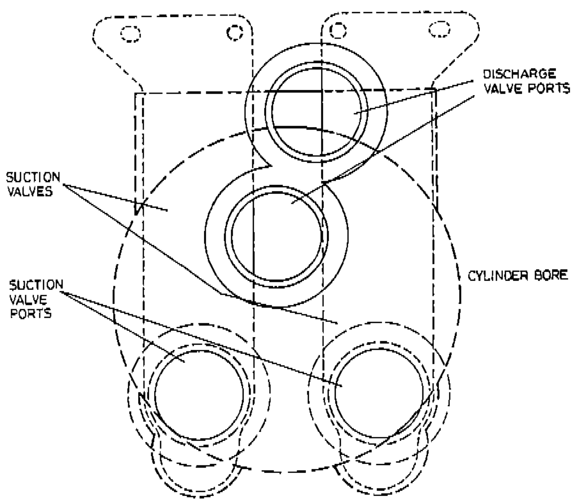

FIGURE 1. EXISTING PORT AND VALVE CONFIGURATION.

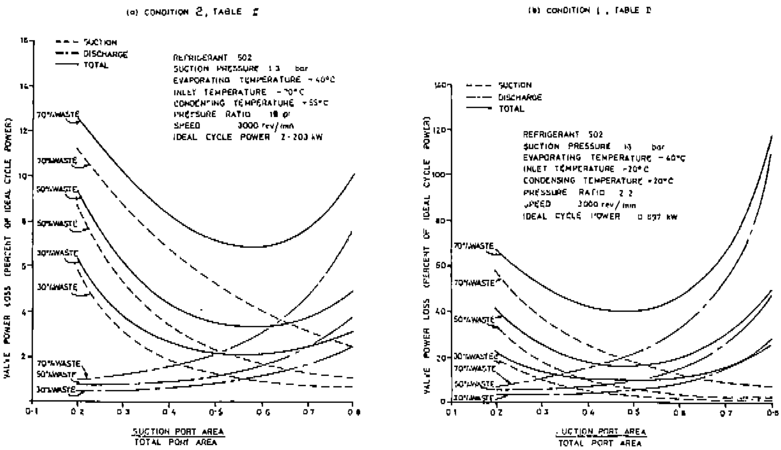

FIGURE 2 VARIATION OF VALVE POWER LOSS WITH PORT AREA DISTRIBUTION FOR VARIOUS WASTE AREAS

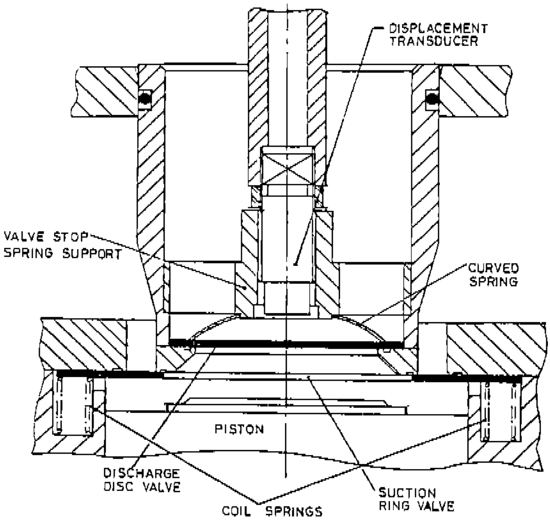

FIGURE 3. CROSS SECTION THROUGH SUCTION AND DISCHARGE VALVES

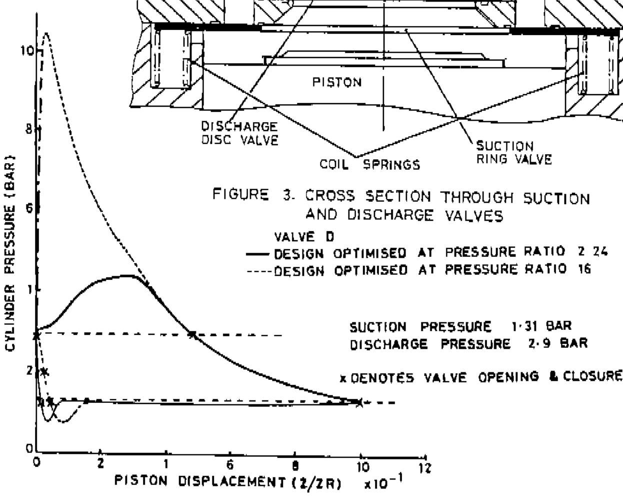

FIGURE 4. PERFORMANCE OF HIGH & LOW PRESSURE RATIO OPTIMISED DESIGNS AT A LOW PRESSURE RATIO OPERATING CONDITION

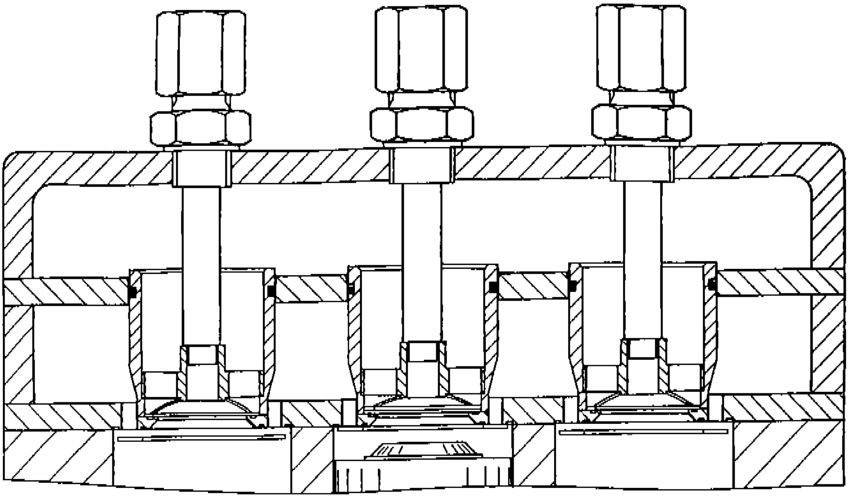

FIGURE 5 PROTOTYPE CYLINDER HEAD ARRANGEMENT

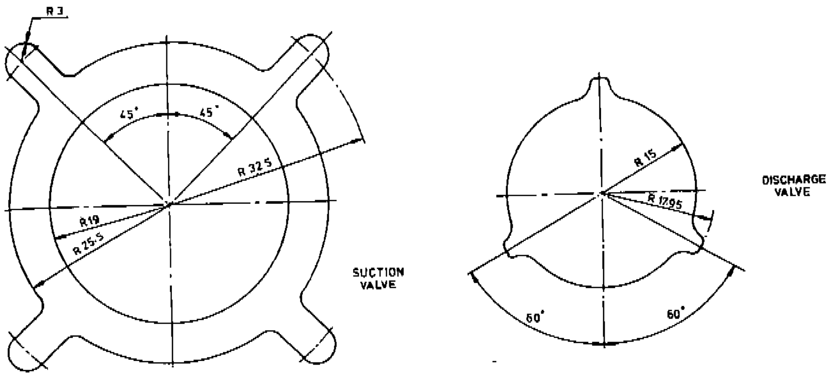

FIGURE 6 PROTOTOTYPE SUCTION AND DISCHARGE VALVES

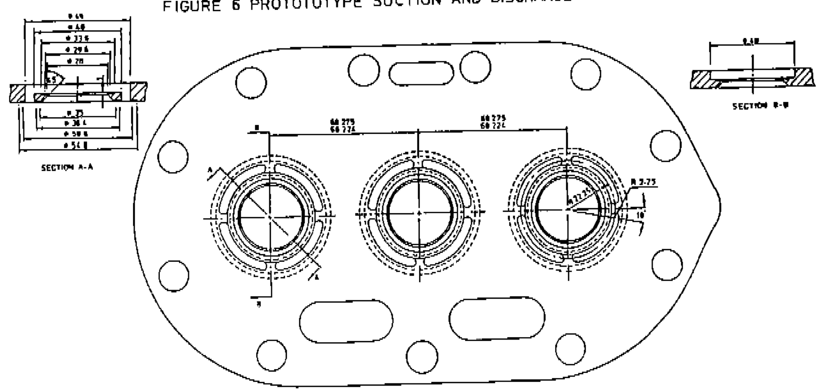

FIGURE 7. PROTOTYPE VALVE PLATE

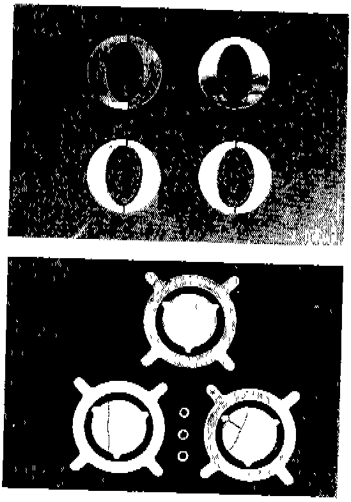

FIG. 8 BROKEN VALVE COMPONENTS.

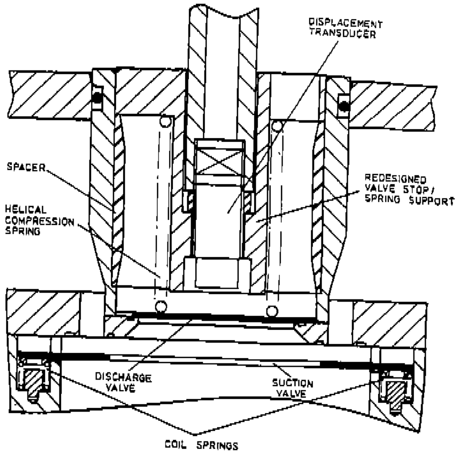

FIGURE 9. REDESIGNED DISCHARGE VALVE ASSEMBLY

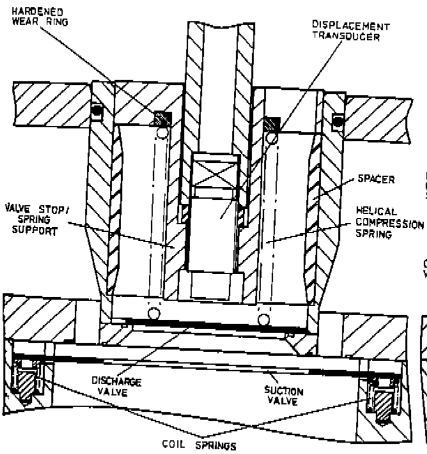

FIGURE 10. DISCHARGE VALVE ASSEMBLY INCORPORATING WEAR RING.

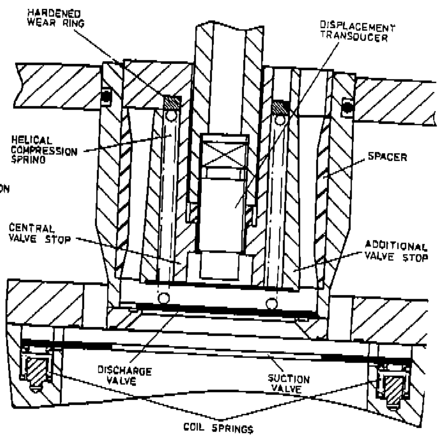

FIGURE 11 FINAL DISCHARGE VALVE ASSEMBLY

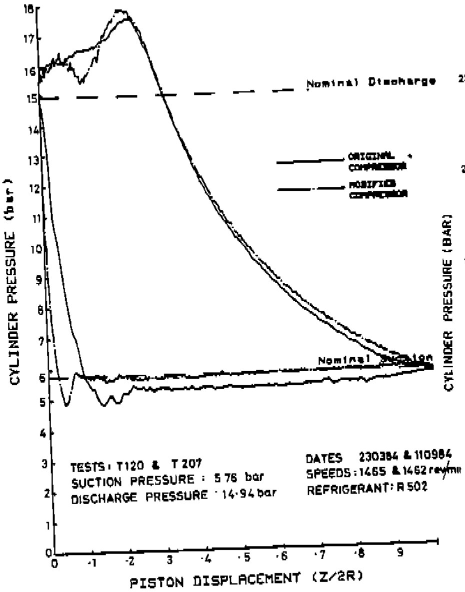

FIGURE 12. PRESSURE - VOLUME DIAGRAMS OBTAINED FROM ORIGINAL AND MODIFIED COMPRESSORS

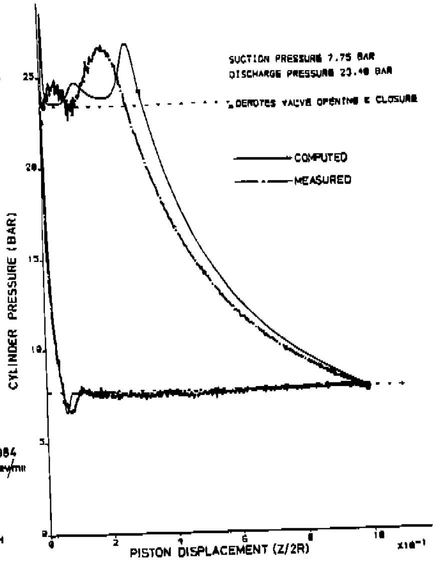

FIGURE 13. MEASURED AND COMPUTED PRESSURE-VOLUME DIAGRAMS FOR MODIFIED COMPRESSOR.

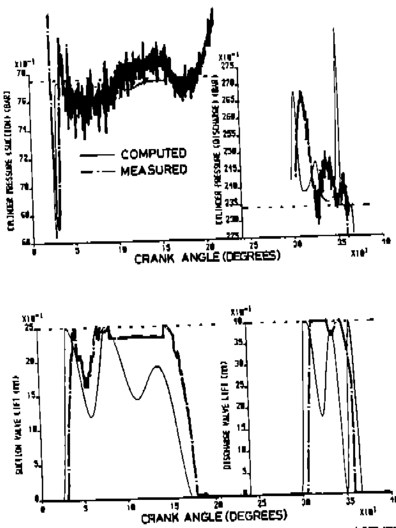

FIGURE 14 MEASURED AND COMPUTED VALVE DISPLACEMENTS AND CYLINDER PRESSURE FOR MODIFIED COMPRESSOR.