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J.F. T. MacLaren

A. B. Tramschek

R. G. Hoare

J.V. Soulis

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Professor J F T MacLaren : Dr A B Tramschek : Mr R G Hoare : Mr J V Soulis
Mechanical Engineering Group, University of Strathclyde, Glasgow

ABSTRACT

The presence of pressure pulsations in the suction and discharge plenum chamber regions of a reciprocating compressor can have a significant effect upon the operation of the compressor valves and hence upon the compressor performance. Although comprehensive models are available which will predict both the action of compressor valves and the associated plenum chamber pressure variations these models require a detailed description of the system in which a compressor operates and use large amounts of computer time. A technique is described whereby experimentally determined plenum chamber pressure variations are superimposed on a relatively simple mathematical model of a compressor, thereby allowing more realistic valve actions to be described than those predicted by the simple model alone and in a fraction of the computer time taken by a more comprehensive model. The use of experimental data limits the technique to the study of compressor systems already in existence.

INTRODUCTION

In early mathematical models of reciprocating gas compressors the effects of the pressure pulsations in the suction and discharge plenum chambers were neglected. However, it is recognized that these pulsations can have a significant influence on the performance of the compressor and the life and behaviour of its valves. During the last decade pulsation effects have been accounted for in more comprehensive mathematical models which however are much more expensive of computer time. This paper reports the application of a hybrid technique whereby experimentally measured plenum chamber pressure-time histories are superimposed on a relatively simple model, with very little addition to the small amount of computer time required to run the simple model.

THE MATHEMATICAL MODEL

The model used was based on that developed by Costagliola (1). The events in a compressor cycle may be designated re-expansion, suction, compression and discharge (Figure 1). In a compressor fitted with automatic valves, the valve lift during suction and discharge is an unknown function of the unknown pressure difference across the valve. Hence both the lift and the pressure difference are dependent on compressor speed, valve inertia, spring stiffness, etc. Two simultaneous non-linear differential equations which relate the relevant variables provide a mathematical model to describe the suction and discharge events in the cycle. One is a "flow" equation which expresses the flow through the valve as a function of the pressure difference across the valve during its movement, the other is a "dynamic" equation which expresses the displacement of the valve in terms of the relevant forces acting upon the moving element in it. Both equations can be expressed as functions of time and hence of crankangle. Solution of the non-linear equations, by iterative methods using a digital computer, yields the displacement of each valve and the pressure difference across it. The pressure in the cylinder is obtained by subtracting (during suction) or adding (during discharge) the pressure difference across the valve to the appropriate plenum chamber pressure. Integration of the mass flow and work during the cycle permits the evaluation of volumetric efficiency and power consumption. A major simplification which makes this simple model economical to use accrues from the assumption that the plenum chamber pressures, P_i and P_d (Figure 1), remain constant during the cycle.

However, the gas flow through a plenum chamber is both variable and intermittent because of the sinusoidal motion of the piston and the cut-off action of the valves. Hence during a cycle the pressure in the plenum chambers and compressor cylinder varies in a complex way. A valve may start to open at point c (Figure 2) which may be at a plenum chamber pressure above or below the mean pressure, P_i or P_d and earlier or later in the piston stroke than point 1. An initial pressure difference (ab) to overcome spring preloading and oil stiction at the valve seat may have to be accounted for. In either case a higher frequency pressure pulsation is imposed on the fundamental wave generated by the piston motion. The pressure variations in the plenum chamber and cylinder mutually interact with each other and with the valve movement. Hence to account for the variation of pressures during suction and discharge requires information on the geometry of the plenum chambers and the associated pipework of the system: the model thus becomes larger and requires more computer time.

The resulting wave action in the system can be accounted for by the methods discussed in the Short Course "Introduction to the Linear Mathematical Simulation of Gas Pulsations" held prior to this Conference. If the amplitudes of the pressure pulsations are large and if heat transfer effects are significant, as in an intercooler for example, the more elaborate methods described (2), (3) at the Conference held at Purdue in 1976 may be employed. Although such models simulate the complex physical situation more closely they are still only approximate descriptions, since simplifying assumptions have to be included if mathematical solutions are to be obtained. If the models are sufficiently detailed to permit certain complex problems to be examined, for example, the interaction between cylinders, plenum chambers and pipework in a multi-cylinder compressor, then the model is expensive both to program and to run. The majority of the computer time is spent in solving the partial differential equations which describe the unsteady flow in the system and much of this time could be saved if accurate experimental records of the pressure time histories of P_i and P_d were obtained and used to provide values of P_i and P_d for each time (crankangle) interval of the iterative solution of the two simultaneous differential equations which are the basis of the simple model which relates to Figure 1. In addition, by running the computer programs for the model without superimposing the plenum chamber pressure-time histories, information would be provided at small extra cost on the effect of the plenum chambers and pipework system on the performance of the compressor and its valves.

MODIFICATION OF THE MATHEMATICAL MODEL

The computer programs for the simple basic mathematical model were altered to allow for variation of the pressures P_i and P_d. Experimentally determined pressure variations could then be used as data in the model either

(a) from a digital store of experimental values of P_i and P_d, and the corresponding values of time (crankangle). A simple interpolation procedure was used to obtain the values of P_i and P_d at each time (crankangle) interval used in the iterative solution of the two simultaneous equations or

(b) from a Fourier series which described the particular pressure-time history of P_i or P_d during a complete compressor cycle.

The coefficients for the terms in the Fourier series were obtained by a program which analysed the digitised version of the experimental analog record. (This analysis could have been obtained also by processing the experimental analog signals on a Fourier Analyser.) The program might have been incorporated as a subroutine in that for the model but was kept separate for reasons of flexibility and economy. The modified mathematical model allowed either P_i or P_d to be held constant if desired. For example, if the behaviour of a suction valve was the centre of interest then P_d could be assumed constant for reasons of economy, it being accepted that the variation of P_d has very little effect on the suction process.

APPLICATION OF THE HYBRID TECHNIQUE

The technique was applied to the study of the suction valves in a six cylinder single-stage compressor having a cylinder bore 82 mm and piston stroke 65 mm, the cylinders being disposed in a 90° Vee arrangement with three cylinders on each side of the cylinder block (Figure 3a). A single cylinder head provided a common plenum at suction and at discharge to each bank, the cylinder block containing galleries through which the refrigerant (R502) passed as it entered or left the compressor. The discharge valves were of the cantilever reed type with a backing plate as a stop, each cylinder having three reeds. For each cylinder there was a single suction reed valve (Figure 3b) which covered three circular ports in the valve plate, the lift being limited at the tip by a recess in the cylinder wall. The compressor operated in the speed range 1450 - 1750 rev/min.

From the complexity of the cylinder disposition and the irregular shape of the plenum chambers, cavities, and galleries, it will be realised that while the system could be modelled by the methods of MacLaren et al (2) or Soedel et al (3), a model having sufficient detail to simulate adequately the difference in behaviour between the individual cylinders would be large and expensive to develop and to use. Hence, piezo-electric transducers were fitted and pressure-time histories measured in the three plenum chamber regions associated with bank A (Figure 3a) of the six cylinders. A phase marker was provided by an electro magnetic pick up and a notch on the compressor drive pulley.

The compressor tested was many miles from the computer facilities and so the experimental pressure-time records were recorded in analog form on magnetic tape and transported to an analog - digital conversion and storage system controlled by a Hewlett-Packard 2100A mini computer (4). Following digitisation the records were stored in a disc memory system with each store file containing 500 discrete digitised pressure readings. These readings were at one degree intervals of crankangle and hence spanned more than one compressor cycle. For cylinder A2 the record started 90° before TDC and the corresponding angles for cylinders A1 and A3 were 330° and 210° before TDC respectively. The records could be recalled from these store files and, following interpolation, used to update the values of P_i and P_d at the discrete time intervals employed in the iterative process to solve the equations constituting the model. In the alternative procedure a Fourier analysis program was applied to provide a file of the coefficients for a preselected number of terms in a Fourier series which described the particular pressure-time history during exactly one compressor cycle. When running the model this series was evaluated at each time (crankangle) interval employed in the iterative process. Table II lists some of the output from the model for Test 1 in Table I with the variation of P_i (a) neglected and (b) accounted for (by the Fourier series procedure). Table III permits comparison between some results for the three cylinders A1, A2 and A3 for Test 1: P_d is assumed to be constant and again the experimentally recorded variation of P_i is accounted for by the Fourier series procedure. Graphical output from this test, Figure 4, illustrates that the suction plenum chamber pressure-time histories and the interrelated suction valve movement were discernably different for the three cylinders. Hence other parameters available from the model (volumetric efficiency, power consumption, etc.) were different to some extent - a feature which, of course, cannot be studied by the simple model alone, where the variation of plenum chamber pressures is neglected. When interpreting Figure 4 note that the cranks of the three cylinders are displaced relative to each other so the diagrams plotted to a base of crankangle after TDC for each cylinder, do not pertain to the same instant of time.

A parameter which may be indicative of valve life is the impact velocity of a valve at its stop (5). A plot of this parameter for the suction reed valve, to a base of compressor pressure ratio, is shown in Figure 5. Although this particular model overestimates (perhaps by a factor of 2) the velocity of a valve as it approaches the stop (6), the impact velocity was very high, a feature which the authors had observed previously when examining cantilever type reed valves employed in such relatively large cylinders. Figure 5 shows that the variation of this high impact velocity between cylinders A1, A2 and A3 was small and hence the study did not provide a prima-facie reason for the higher incidence of failure which had been occurring in one of the cylinders. Figure 6 shows the marked effect, at higher compressor pressure ratios, of accounting for the variation of the suction plenum chamber pressure, P_i, on the point of final valve closure and hence on the valve blow-by loss and volumetric efficiency.

It is considered that a sufficiently detailed wholly mathematical model of the wave action in this complex suction side geometry may not have produced any more positive results so that the necessary investment of time and money to develop and run it could not be justified. A model with such detail might be justified when studying such a design prior to any manufacture.

COMPARISON OF FOURIER SERIES AND DISCRETE VALUE PROCEDURES TO ACCOUNT FOR VARIATION OF PLENUM CHAMBER PRESSURES

The computer program could account for a variable plenum chamber pressure either directly from experimental data or by means of a Fourier series derived from this experimental data. The experimental data was recorded either directly from a compressor plenum chamber or, if the compressor was remote from the computer, recorded and transported on magnetic tape. In either case the analog records were converted by an ADC while being read into the computer controlled data handling system (4). The digital values of pressure were stored at pre-specified discrete intervals (1 degree of crankangle).

The advantage of subsequently expressing this data as a Fourier series, compared to discrete values of pressure at each interval of crankangle, is that the pressure-time history is then available in a continuous form and that the pressure can be readily calculated at any value of crankangle. Its disadvantage is that longer computer time is required to generate the value of the pressure at each value of crankangle compared to the time taken to calculate a pressure by interpolation of the pressures recorded at adjacent discrete points. The matter was examined by applying both procedures to account for the variation of the pressure P_d in the discharge plenum chamber of an air compressor, 6 in bore x 4.5 in stroke (152 mm x 114 mm) fitted with spring loaded single annular ring plate valves and operating in the speed range 350 - 700 rev/min. A record of 1000 readings for P_d was taken at intervals of $\frac{1}{2}$ degree crankangle. Subsequently 720 readings, corresponding to one compressor cycle, were used as discrete data in the mathematical model. The coefficients for a Fourier series were evaluated based on 1024 (2^{10}) pressure values taken over one cycle, the values of pressure at the additional points being obtained by interpolation of the experimental data. A comparison was made by running the model (with constant suction plenum chamber pressure) but with the experimentally observed variation of P_d during one cycle represented by (a) 720 discrete points and (b) a Fourier series having 512 terms (1024/2). The differences between the results were almost nil. However there was a difference in the computer time required and hence the cost of running the model. A computation time of 26 units was required for (a) and 58 units for (b). (The time taken if P_i and P_d were both assumed constant (eg by using 1 Fourier term) was 15 units).

The computing time associated with procedure (b) could be lessened by reducing the number of terms used in the Fourier Series. Figure 7 shows the variation in computing time with the number of Fourier terms used when the model was employed to study the suction side of one cylinder of the six cylinder refrigeration compressor at test condition 2 of Table 1. For example, with P_d held constant and P_i variable the computing times were:

No of Fourier Coefficients	Computing Time Units
256	148
150	101
50	57

If however, a variable P_i was stored as 360 discrete values at 1 degree intervals of crank angle, and interpolated as necessary, the computer time required was only 35 units. If a variable P_d was also represented by 360 discrete values the computer running time with variable P_i and P_d was 37 units. These latter times were little different from that necessary (33 units) to run the basic model (Figure 1) where P_i and P_d are both constant.

The effect of varying the number of terms used in the Fourier series description of the pressure time history of the suction plenum chamber pressure P_i is shown in Table IV which gives a comparative list of several parameters calculated by the model. When P_d was held constant 256 terms yielded virtually the same results as 150 terms, but differences became apparent when a further reduction to 50 terms was made. While 150 terms satisfied accuracy considerations, the computer time was three times as much (Figure 7) as that required when using discrete pressure data and employing a simple interpolation technique.

A comparison was made of computing time required with this hybrid model, utilising experimental pressure time histories for either one or both plenum chambers, and a wholly analytical model which, being supplied with data describing the geometry of the plenum chambers and pipework system, was able to calculate the variations in P_i and P_d. The wholly analytical model required 15 times the amount of computing time used by the hybrid model.

When employing the Fourier series approach a further time of 90 units was required to generate the Fourier series coefficients. A subroutine to output results graphically, as in Figure 4, required an additional 40 units of computing time. The units of computer time were seconds but the times quoted should be regarded as relative rather than absolute since precise values depend on the computer available and the efficiency of the programs (the programs were developed by non specialist programmers).

Initial conditions at point 4, Figure 1 have to be assumed whichever model is used. If point 4' at the end of the first calculated compressor cycle is deemed sufficiently different from point 4 the computation may be repeated using point 4' as a revised starting point. Adequate convergence was achieved in three cycles and all the times quoted above are on this basis.

CONCLUSIONS

If the pressure pulsations inherent in a compressor system are considered to be having a significant effect on valve and compressor performance then a relatively simple mathematical model, wherein the plenum chamber pressures are assumed to remain constant, may be inadequate. To account analytically for these complex pressure pulsations by the methods of MacLaren et al (2) or Soedel et al (3) requires complex mathematical models which are expensive both to develop and to run. They may nevertheless be justified as an aid to design if employed prior to any manufacture.

If however, a compressor is already in production and problems arise, a meaningful study can be conducted economically using a hybrid analytical/experimental model. Also, such a study may be more reliable than one made using a wholly analytical model in cases when the geometry of the compressor system is very complex. The hybrid model requires that the pressure time histories in the compressor plenum chambers be measured experimentally so that variable pressures may be superimposed on a relatively simple mathematical model (1) in which, in the absence of variable plenum pressure data, plenum chamber pressures are assumed to be constant.

The experimentally determined variable pressure data was handled successfully either as a series of discrete data values or synthesised using a Fourier analysis of the experimental data. The use of discrete data values proved to be the most economical in terms of computer time.

When the technique was used to study a six cylinder, single stage, Vee refrigerant compressor it was found that the experimentally obtained pressure time histories for the individual cylinder plenum chamber regions led to significantly different predictions of each suction valve displacement. In each cylinder very high suction valve impact velocities were predicted but the differences between them were so small that they did not explain why suction valve failure occurred more frequently in one particular cylinder.

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Test No.	Suction Pressure (bar)	Discharge Pressure (bar)	Compressor Speed (rev/min)	File Reference for Pressure-Time History Cyl A1	Cyl A2	Cyl A3
1	1	16	1480	P1S04	P2S04	P3S04
2	3	16	1480	P1S02	P2S02	P3S02
3	5	16	1480	P1S01	P2S01	P3S01
4	1	13	1480	P1S10	P2S10	P3S10
5	3	13	1480	P1S08	P2S08	P3S08
6	5	13	1480	P1S06	P2S06	P3S06
7	1	16	1750	P1S54	P2S54	P3S54
8	3	16	1750	P1S52	P2S52	P3S52
9	5	16	1750	P1S50	P2S50	P3S50
10	1	13	1750	P1S60	P2S60	P3S60
11	3	13	1750	P1S58	P2S58	P3S58
12	5	13	1750	P1S56	P2S56	P3S56

TABLE I : RANGE OF TESTS

	(a)	(b)
Suction Pressure, P_1 (bar)	1 (constant)	1 (variable)
Pressure-Time History (data file number)	-	P2S04 (Table I)
Compressor Speed (rev/min)	1480	1480
Suction Temperature (^oR)	460 (constant)	460 (constant)
Nominal Pressure Ratio	16	16 (mean value)
Suction Valve Opens (degrees)	81.69	77.23
Suction Valve at Stop (degrees)	90.19	85.73
Impact Velocity at Stop (ft/s)	38.81	38.49
Suction Valve Leaves Stop (degrees)	130.19	142.73
Suction Valve Closed (degrees)	178.19	216.23
Impact Velocity at Seat (ft/s)	6.28	14.35
Actual Volumetric Efficiency (%)	51.84	40.35
Theoretical Volumetric Efficiency (%)	52.75	56.72
Loss of Volumetric Efficiency (%)	0.915	16.37
Throttling Loss (%)	0.915	0.75
Blowby (%)	0.00	15.61
Adiabatic Work (ft-lbf)	43.71	46.98
Excess Suction Work (ft-lbf)	1.98	1.70
Suction Power Loss (%)	4.54	3.62
Discharge Valve Opens (degrees)	341.73	342.23
Adiabatic Work (ft-lbf)	43.71	46.98
Excess Discharge Work (ft-lbf)	4.98	4.87
Discharge Power Loss (%)	11.39	10.36

TABLE II : COMPUTED RESULTS FOR CYLINDER A2 USING
(a) CONSTANT SUCTION AND DISCHARGE PLENUM PRESSURES
(b) VARIABLE SUCTION (FOURIER SERIES) AND CONSTANT DISCHARGE PLENUM PRESSURES.

	Cylinder A1	Cylinder A2	Cylinder A3
Suction Pressure, P_1 (bar)	1 (variable)	1 (variable)	1 (variable)
Pressure-Time History (data file number)	P1S04	P2S04	P3S04
Compressor Speed (rev/min)	1480	1480	1480
Suction Temperature (^oR)	460 (constant)	460 (constant)	460 (constant)
Nominal Pressure Ratio (mean value)	16	16	16
Suction Valve Opens (degrees)	80.00	77.23	80.94
Suction Valve at Stop (degrees)	88.50	85.73	89.44
Impact Velocity at Stop (ft/s)	39.40	38.49	40.42
Suction Valve Leaves Stop (degrees)	156.00	142.73	112.44
Suction Valve at Seat (degrees)	223.00	216.23	185.44
Impact Velocity at Seat (ft/s)	9.50	14.35	2.68
Actual Volumetric Efficiency (%)	46.54	40.35	52.64
Theoretical Volumetric Efficiency (%)	54.26	56.72	53.42
Loss of Volumetric Efficiency (%)	7.72	16.37	0.77
Throttling Loss (%)	0.88	0.75	0.77
Blowby	6.83	15.61	0.00
Adiabatic Work (ft-lbf)	44.96	46.98	44.27
Excess Suction Work (ft-lbf)	1.87	1.70	1.77
Suction Power Loss (%)	4.17	3.62	3.99
Discharge Valve Opens (degrees)	342.29	342.23	342.43
Adiabatic Work (ft-lbf)	44.96	46.98	44.27
Excess Discharge Work (ft-lbf)	4.94	4.87	4.60
Discharge Power Loss (%)	11.00	10.36	10.39

TABLE III : COMPUTED RESULTS FOR THE A1, A2 AND A3 CYLINDERS USING VARIABLE SUCTION (FOURIER SERIES) AND CONSTANT DISCHARGE PLENUM PRESSURES.

Number of Fourier Terms	50	150	250
Suction Valve Opens (degrees)	43.6	43.6	44.3
Suction Valve Impact Velocity (at stop, ft/s)	68.7	68.75	68.8
Suction Valve Closes (degrees)	191.2	192.1	192.2
Suction Valve Impact Velocity (at seat; ft/s)	12.1	13.4	13.5
Actual Volumetric Efficiency (%)	81.1	80.65	80.65
Suction Valve Throttling Loss (%)	0.75	0.75	0.75
Suction Valve Blow-by Loss (%)	1.7	2.2	2.2
Suction Valve Power Loss (%)	4.12	4.122	.124

TABLE IV VARIATION OF SOME COMPUTER RESULTS DUE TO CHANGING THE NUMBER OF TERMS IN THE FOURIER SERIES USED TO EXPRESS THE VARIATION OF P_1 DURING THE COMPRESSOR CYCLE

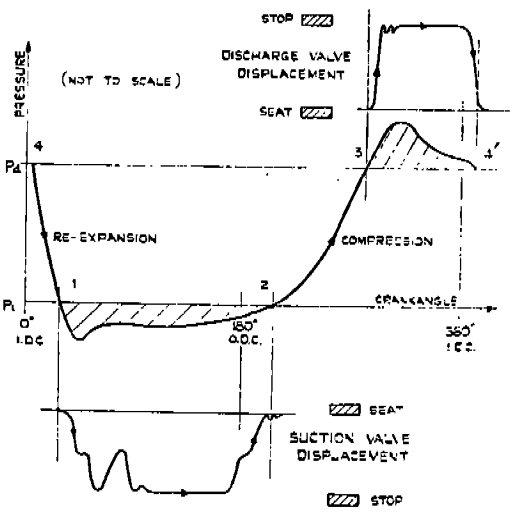

FIG. 1. PRESSURE - CRANKANGLE AND VALVE DISPLACEMENT DIAGRAM, FOR A COMPRESSOR WITH AUTOMATIC VALVES, ASSUMING THAT PLENUM CHAMBER PRESSURES (Pl AND Pd) REMAIN CONSTANT

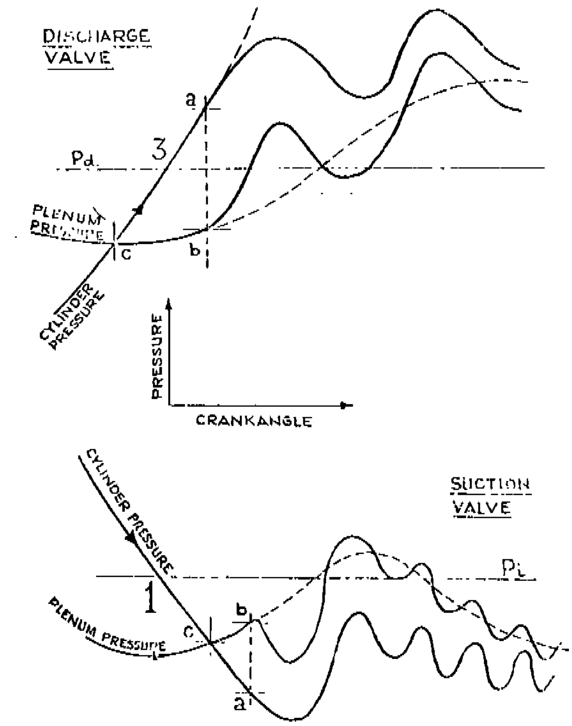

FIG. 2. EFFECT OF PISTON MOTION AND VALVE OPENING ON CYLINDER AND PLENUM CHAMBER PRESSURES.

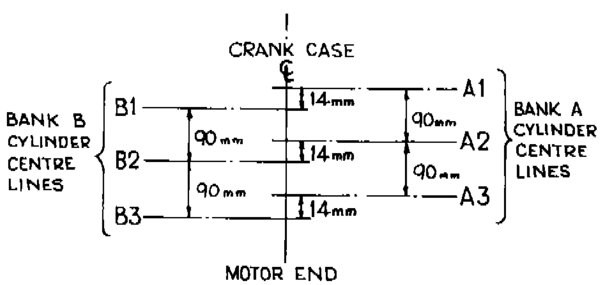

FIG. 3 a CYLINDER DISPOSITION OF SIX CYLINDER VEE REFRIGERANT COMPRESSOR.

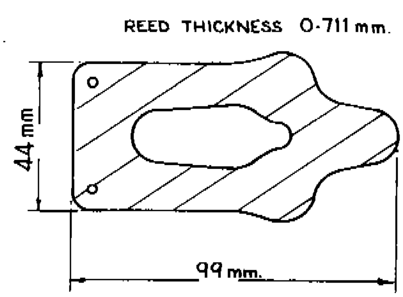

FIG. 3b. SUCTION VALVE REED.

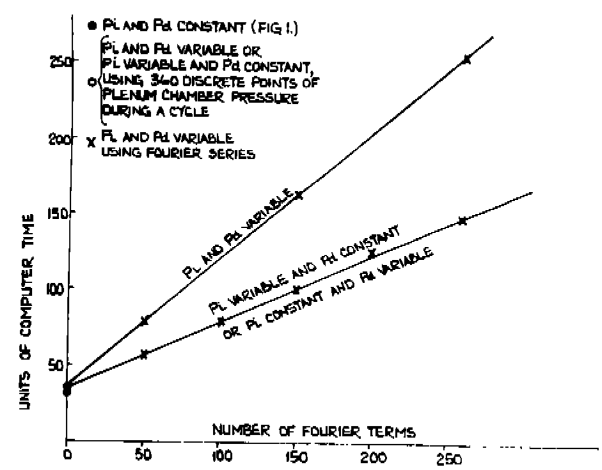

FIG. 7 COMPUTER TIMES REQUIRED.

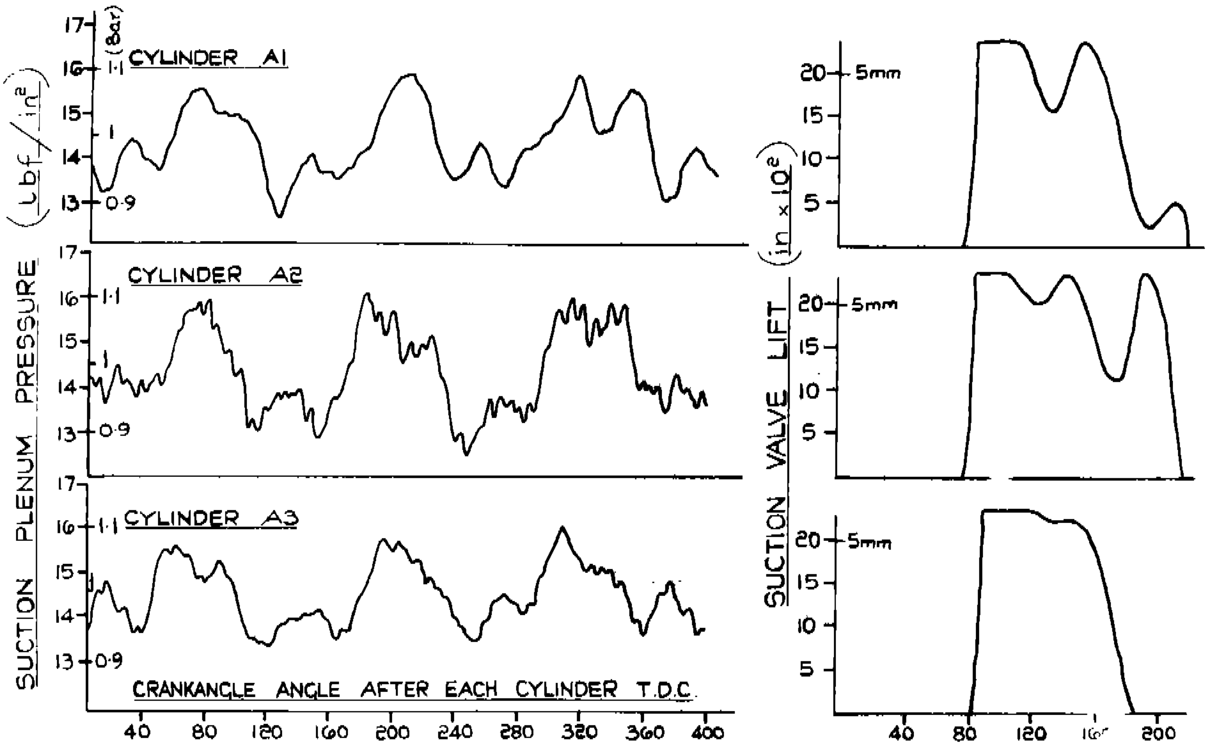

FIG 4 SUCTION PLENUM PRESSURE VARIATION AND VALVE DISPLACEMENT.
COMPRESSOR PRESSURE RATIO 16 : SPEED 1480 rev/min

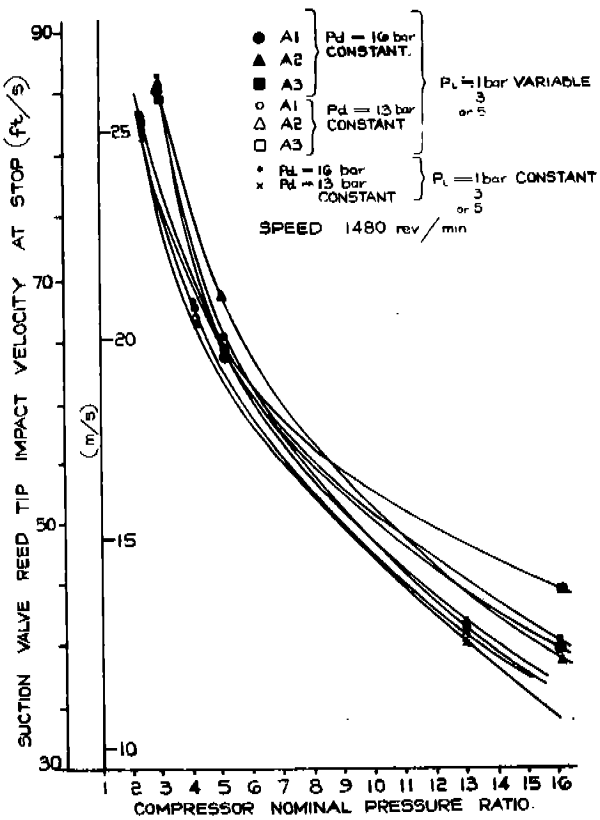

FIG 5 VARIATION OF SUCTION VALVE IMPACT VELOCITY AT STOP WITH COMPRESSOR PRESSURE RATIO FOR A1, A2 AND A3 CYLINDERS.

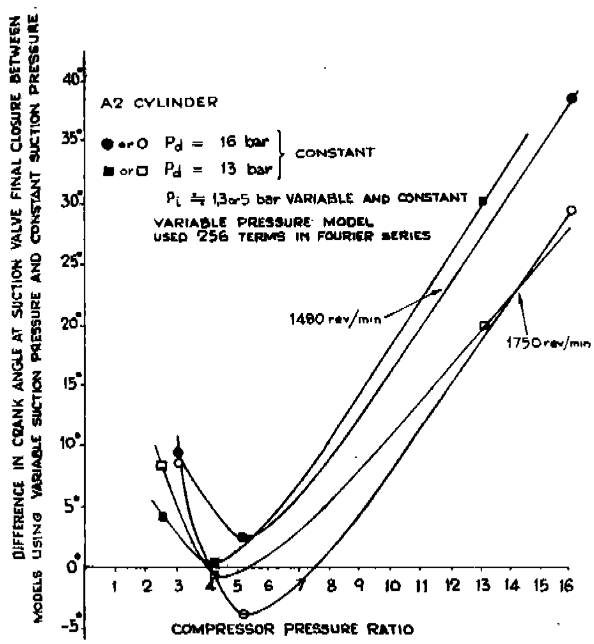

FIG. 6 DIFFERENCE IN SUCTION VALVE FINAL CLOSURE ANGLE WITH COMPRESSOR PRESSURE RATIO AS A CONSEQUENCE OF ACCOUNTING FOR THE VARIATION OF SUCTION PLENUM CHAMBER PRESSURE, P_L