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A MODEL OF A SINGLE STAGE RECIPROCATING GAS COMPRESSOR ACCOUNTING FOR FLOW PULSATIONS

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ABSTRACT

An analytical model is described which simulates a reciprocating gas compressor, its valves, the working fluid and operating conditions together with the pressure pulsations inherent in the intermittent flow. The model couples the conservation equations for compressor cylinder and valves with the hyperbolic equation which describes one-dimensional non-homentropic unsteady flow in the pipework. Numerical solution is effected on a digital computer by the method of characteristics using a mesh technique. Results predicted by the model are compared with corresponding experimental records from a single stage air compressor system. The model is then used to predict the effect of induction ramming on the performance of this compressor and its valves.

INTRODUCTION

In early compressor models one-dimensional quasi-steady flow theory was used to describe the flow of gas through the valves into and out of the compressor cylinder. These models, developed in several countries, were reviewed at the Compressor Conference (1) held at Purdue University in 1972.

Mathematical models of complex physical situations invariably require simplifying assumptions to be made and in the early models one major assumption often made was that infinite receivers existed at inlet and discharge so that inlet and discharge plenum chamber pressures remained constant. However, models are now available which account for the gas pulsations inherently present due to the suction and discharge processes. The state of this development at the time of the Compressor Conference at Purdue University in 1972, was conveyed in papers by Elson and Soedel (2), Brablik (3), Benson and Ucer (4) and MacLaren and Tramschek (5). The analysis by MacLaren and Tramschek (5), which assumes that the flow in the pipework is one-dimensional, non-homentropic and with finite pressure amplitude has been rescrutinized and the computer programs have been reorganised and re-written in Fortran IV. Results predicted by this model are now compared with those observed experimentally over a range of conditions for a single stage air compressor in a simple pipework system. The system has a series configuration of atmosphere/inlet pipe/inlet valve/compressor cylinder/discharge valve/discharge pipe/receiver/nozzle/atmosphere. In this initial investigation pipe junctions and dampers (mufflers, snubbers) have been specifically excluded.

PULSATING FLOW EFFECTS

Pulsating flow in reciprocating compressor systems causes mechanical vibration which results in noise, failure of pipework and fixtures, loosening of fastenings, overloading of the compressor and problems of measuring non-steady gas flow rates. These effects have been discussed extensively in published literature but there is little information available on the resulting interaction with compressor valves, possibly the most delicate components in the system. It is known that there is such an effect, sometimes disastrous. Carpenter (6) reported that the reduction of a peak to peak pulse from 33 lbf/in^2 (228 kN/m^2) to 10 lbf/in^2 (69 kN/m^2) in a particular 100 lbf/in^2 (690 kN/m^2) line "stopped valve breakage completely". He reported another case history in which broken discharge valves occurred once every two weeks in a synthetic ammonia compressor: by reducing pulsations between the cylinder and discharge manifold by 70% the compressor ran for more than three years without breaking a valve. In the discussion at a Conference (7), Bauer, Harris and Parry each illustrated the effect of pipelines and dampers on valve displacement diagrams. Obviously the earlier mathematical models, in which the compressor plenum chamber pressures were assumed constant, are over simplified if pulsation effects are significant. Hence a more general model which will simulate a compressor, its valves, the working fluid together with the pulsations inherent in the intermittent flow, appears to be justified despite the increased complexity of the model and the additional computer time required.

DESCRIPTION OF ANALYTICAL MODEL

Subroutines were written to analyse the behaviour of individual components in a compressor system and the subroutines may then be assembled in a program to describe a particular system: the various boundary conditions encountered were also programmed in subroutines.

The equations which described one dimensional finite amplitude unsteady flow in the pipework also accounted for small changes of cross-sectional area, heat transfer and pipe friction. Inclusion of entropy gradients (i.e. considering the flow to be non homentropic) increased the complexity of the model and the computer time required. This refinement was undertaken following the investigation by Benson and Ucer (4) who concluded that discrepancies between their analytical and experimental results, in particular in the discharge pipe, were due to their assumption of homentropic flow. The technique used for numerical solution was the method of characteristics in which the hyperbolic partial differential equation describing the fluid flow is replaced by a pair of total differential equations. This pair represents the compatibility conditions along (a) the wave characteristics and (b) the path line characteristics (Shapiro (8)). The procedure used for solution by computer was the rectangular mesh method developed by Benson, Garg and Woollatt (9).

The boundaries at the pipe ends could be closed pipe ends (closed inlet or discharge valve), open pipe ends or restricted areas (open valve or nozzle). Quasi-steady conditions were assumed at these boundaries.

For flow out of a pipe, quasi-steady isentropic flow was assumed to the throat, if this existed, and using equations of continuity and energy the values of entropy, temperature and velocity were determined at the exit section. If the flow was subsonic it was assumed that the pressure at the throat was equal to the exit pressure. For sonic flow it was assumed that the Mach number at the throat was unity.

For flow into a pipe, quasi-steady isentropic flow was assumed to the throat followed by an irreversible change to pipe conditions. Since the flow in the pipes was non-homentropic the entropy change of a particle as it crossed the boundary had to be compatible with known conditions outside the pipe and the conditions which exist within the pipe. This necessitated an iterative process since the reflected wave characteristic depends on the entropy value calculated at the boundary section. Again the cases of subsonic and sonic flow at the throat were accounted for.

Volumes at the pipe ends, i.e. compressor cylinder (time variable volume), receiver (fixed volume) and atmosphere (infinite volume) were treated in a unified manner for computational purposes. The properties of the fluid were assumed to be uniform throughout the volume.

The volumes of the inlet and discharge plenum chambers were accounted for by equivalent lengths of inlet or discharge pipe. Although these plenum chamber volumes could have been treated as finite volumes, the simplification appeared justified in the particular system studied judging from the results obtained.

The time increment used in the computations was that which would satisfy stability criteria anywhere in the system. An integer number of sections for each pipe was obtained by modification of the non-dimensional time increment.

The valves were considered to be one degree of freedom damped spring-mass systems with damping assumed to be a function of valve plate velocity. The force on the valve plate was taken to be a function of the pressure difference across the valve, the valve lift and an "effective force area" which was determined experimentally by steady flow tests. The differential equation describing the valve motion was integrated using a modified Kutta-Merson method. Any valve plate bouncing on seat or stop within the integration step was allowed for. Values of coefficients of restitution at the seat or stop were assumed from published information - experience indicates that a precise value may not be necessary.

Mass flows into and out of the cylinder through the valves were calculated when solving the boundary conditions at the appropriate pipe ends. The effective valve throat to pipe area ratio used in the boundary calculations is primarily a function of valve opening. This ratio, termed the "effective flow area" was determined by steady flow tests. Alternatively, the "effective force area" and "effective flow area" could have been estimated by calculation using the method of Schwerzler and Hamilton (10).

COMPARISON OF ANALYTICAL AND EXPERIMENTAL RESULTS

The experimental program was conducted using a variable speed single-stage single-acting water cooled air compressor with cylinder diameter 6 inches (152 mm) and piston stroke $4\frac{1}{2}$ inches (114 mm) delivering about 20 ft^3/min (0.567 m^3/min) free air at 400 rev/min. The moving element in both inlet and discharge valves was a 2 $3/4$ inches (70 mm) o.d. single annular ring plate loaded by three helical coil springs. The lengths of the 2 in (51 mm) diameter pipe used were 18 ft (5.5 m) or 9 ft (2.75 m) on the inlet side and 13 ft (3.97 m) on the discharge side. Piezo-electric transducers sensed pressure; inductive transducers recorded valve displacement; signals were fed to a multichannel U.V. recorder. More detail of the measuring techniques is contained in reference (11).

The analytical computations were commenced at piston top-dead-centre assuming that the discharge valve was closed and that (a) both the cylinder and discharge pipe were at the nominal discharge pressure and (b) the inlet pipe was at atmospheric pressure. It was found that sufficient convergence to an approximately repeatable cycle was obtained by the third compressor cycle. Convergence to exact repeatability cannot be expected since the forcing frequency of the compressor and the oscillating frequency in the pipes are not exact multiples. Small variations from cycle to cycle were observed also in the experimental records.

The extent of the agreement between the analytical and experimental results may be assessed from Figs. 1, 2, 3 and 4. Figs. 1, 2 and 3 relate to a test series conducted at three compressor speeds, 371, 511 and 645 rev/min with inlet pipe length 18 ft (5.5 m), discharge pipe length 13 ft (3.97 m) and compressor pressure ratio 4.8. Fig. 4 relates to a test with the inlet pipe length halved to 9 ft (2.75 m) with operating conditions close to those listed in Fig. 1. The upper part of each figure shows, as functions of crank angle and to the same scale of pressure, the analytically predicted suction and discharge plenum chamber pressures and the cylinder pressure. The upper part of each figure includes the analytical and experimental displacement diagrams for both suction and discharge valves. The lower part of each figure shows the pressure/time histories redrawn to larger scales to permit comparison between the analytically predicted and experimentally observed records of discharge plenum chamber pressure (a), suction plenum chamber pressure (b), cylinder pressure during the suction process (c) and during the discharge process (d). The agreement between the analytical and the experimental results was considered to be good. Sources of discrepancy are (1) assumptions made in the analysis (2) experimental errors and (3) small variations from cycle to cycle in both the analytical and experimental results. Some refinements could have been made but were not included in the computations for this initial investigation. Thus the valve flow and force coefficients, which are variables, were approximated as linear functions of valve opening and were assumed to have the same values during any small reversed flow which can occur near valve closure. Also the pipe friction coefficient was assumed to be constant and heat transfer effects were allowed for in a simplified manner.

The inlet or discharge pipe length of a compressor operating at a fixed speed and compressor pressure ratio affects the crank angle at which a valve opens, the valve movement and the pressure/time history in both the cylinder and plenum chamber. These effects are due to the relative phasing of the pressure variations in the cylinder and plenum chamber. At the instant of valve opening the variable pressure in the plenum chamber may be a maximum or minimum or be increasing or decreasing. This is illustrated by comparing Fig. 1 (b) and (c) with Fig. 4 (b) and (c): the effect of halving the suction pipe length on the cylinder pressure during the suction process is readily observed. A similar effect may occur with a fixed pipe length by changing the compressor speed. For example, in Fig. 1 (a) (371 rev/min) the discharge plenum chamber pressure is near a minimum at discharge valve opening while in Fig. 3 (a) (645 rev/min) the discharge plenum chamber pressure is near a maximum at discharge valve opening.

INDUCTION RAMMING AND VALVE SLAMMING

The model was used to investigate (a) the effect of varying the inlet pipe length at a constant compressor speed (600 rev/min) and pressure ratio (7.8) for a fixed discharge pipe length: (b) the effect of varying the discharge pipe length for a fixed length of inlet pipe using the same speed and pressure ratio as in (a). Results for (a) are shown in Fig. 5 and clearly illustrate the effect on volumetric efficiency of induction ramming and anti-ramming. In Fig. 5 volumetric efficiency is plotted against "delay angle", which is defined as the angular rotation of the compressor crankshaft during the time taken for a pressure pulse to travel from the compressor to the atmosphere (or receiver) and return in a reflected form to the compressor. Assuming a mean wave propagation speed $@_o$ (ft/s) the delay angle (degrees) is given by $12\ NL/\ @_o$ where N is the compressor speed (rev/min) and L is the pipe length (ft). The "delay angle" concept was used by Bannister (12): a comparison of his results with the results plotted in Fig. 5 shows great similarity in the variation of volumetric efficiency due to induction ramming and anti-ramming. The non-dimensional pipe length $\emptyset$, used in Figures 1 to 6, is defined as the ratio of the natural period of pressure oscillation in the pipe to the period of the compressor cycle (hence $\emptyset = NL/15\ @_o$)

For comparison with the analytical results in Fig. 5, a series of experiments was conducted in which the inlet pipe length was varied from zero to 51 ft (15.5 m) in 3 ft (0.914 m) increments. The agreement between the analytical and experimental results was considered to be good: the main difference between the results suggests that the model slightly over-estimated the effect on volumetric efficiency of both induction ramming and anti-ramming.

The lower part of Fig. 5 shows the analytically determined variation in volumetric efficiency when the discharge pipe length was varied. As might be expected, the variation of volumetric efficiency was small: the corresponding experimental series was not conducted.

The impact velocity of the valve plate against the valve stop (Fig. 6) was predicted by the model over a similar range of pipe lengths. Results presented in Fig. 6 suggest that in this particular compressor system, pulsations in the inlet or discharge piping did not result in excessive slamming at the stop of either discharge or suction valve. The discharge valve impact velocity at the stop was about twice that of the suction valve. The impact velocity of each valve plate at closure against the valve seat (not shown) was approximately half that recorded at the corresponding stop.

CONCLUSIONS

By comparing the results predicted by the analytical model with those obtained experimentally, of which Figs. 1, 2, 3, and 4 are a sample, it was concluded that the computer model provided a valid simulation of a single stage reciprocating gas compressor system without pipe junctions or dampers. Over a wide range of conditions the model adequately predicted the pressure time histories in the cylinder and plenum chamber, the displacement of the valve plates and the changes in volumetric efficiency due to induction ramming or anti-ramming. The model predicted that, for the particular system examined, variations in inlet or discharge pipe length would not result in excessive valve slamming.

REFERENCES

1. MacLaren, J.F.T., "A Review of Simple Mathematical Models of Valves in Reciprocating Compressors", Proc. 1972 Purdue Compressor Technology Conference, p. 180.

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

3. Brablik, J., "Gas Pulsations as Factor Affecting Operating of Automatic Valves in Reciprocating Compressors" Proc. 1972 Purdue Compressor Technology Conference, p. 188.

4. Benson, R.S. and Ucer, A.S., "Research in Gas Dynamic Modelling of Multiple Single Stage Reciprocating Compressor Systems", Proc. 1972, Purdue Compressor Technology Conference, p. 491

5. MacLaren, J.F.T. and Tramschek, A.B., "Prediction of Valve Behaviour with Pulsating Flow in Reciprocating Compressors" Proc 1972 Purdue Compressor Technology Conference, p. 203.

6. Carpenter, A.B., "The Use of Analog Simulator Studies to Reduce or Eliminate Objectionable Pulsations in Reciprocating Compressors and Associated Piping", ASME Paper No. 67-Pet-30.

7. Institution of Mechanical Engineers Conference "Industrial Reciprocating and Rotary Compressors" London 1972. (Discussion) Bauer F. (p. 227), Harris A.G., (p. 235) Parry J., (p. 242)

8. Shapiro, A.H., "The Dynamics and Thermodynamics of Compressible Fluid Flow", The Ronald Press, New York.

9. Benson, R.S., Garg, R.D. and Woollatt, D. "A Numerical Solution of Unsteady Flow Problems" Int.J.Mech.Sci., Vol. 6, pp. 117 - 144, 1964.

10. Schwerzler, D.D. and Hamilton, J.F., "An Analytical Method for Determining Effective Flow and Force Areas for Refrigerating Compressor Valving Systems", Proc. 1972, Purdue Compressor Technology Conference, p. 30.

11. MacLaren, J.F.T. and Kerr, S.V., "An Analytical and Experimental Study of Self-acting Valves in a Reciprocating Air Compressor", I.Mech.E. Conference, "Industrial Reciprocating and Rotary Compressors", London, 1972.

12. Bannister, F.K., "Induction Ramming of Small High Speed Air Compressor", Proc. I.Mech.E., 1959, Vol. 173, No. 13.

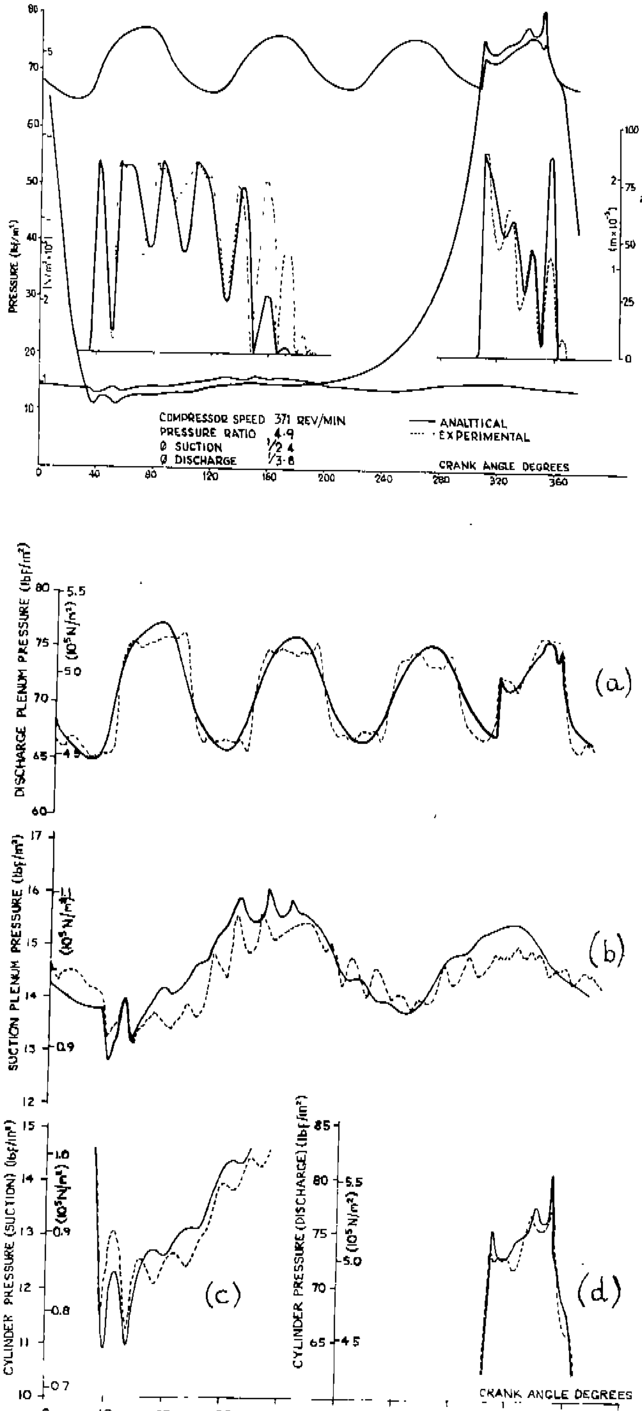

FIG. 1. COMPARISON OF EXPERIMENTAL AND ANALYTICAL RECORDS. (TEST B1)

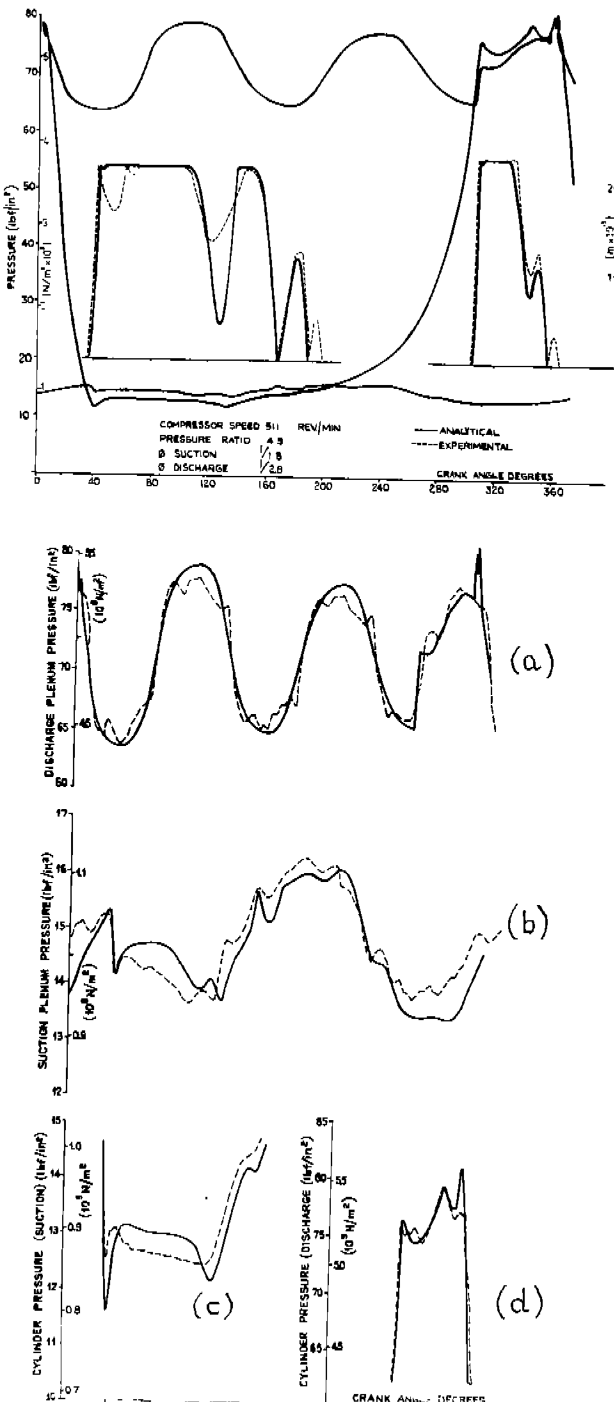

FIG. 2. COMPARISON OF EXPERIMENTAL AND ANALYTICAL RECORDS (TEST B3)

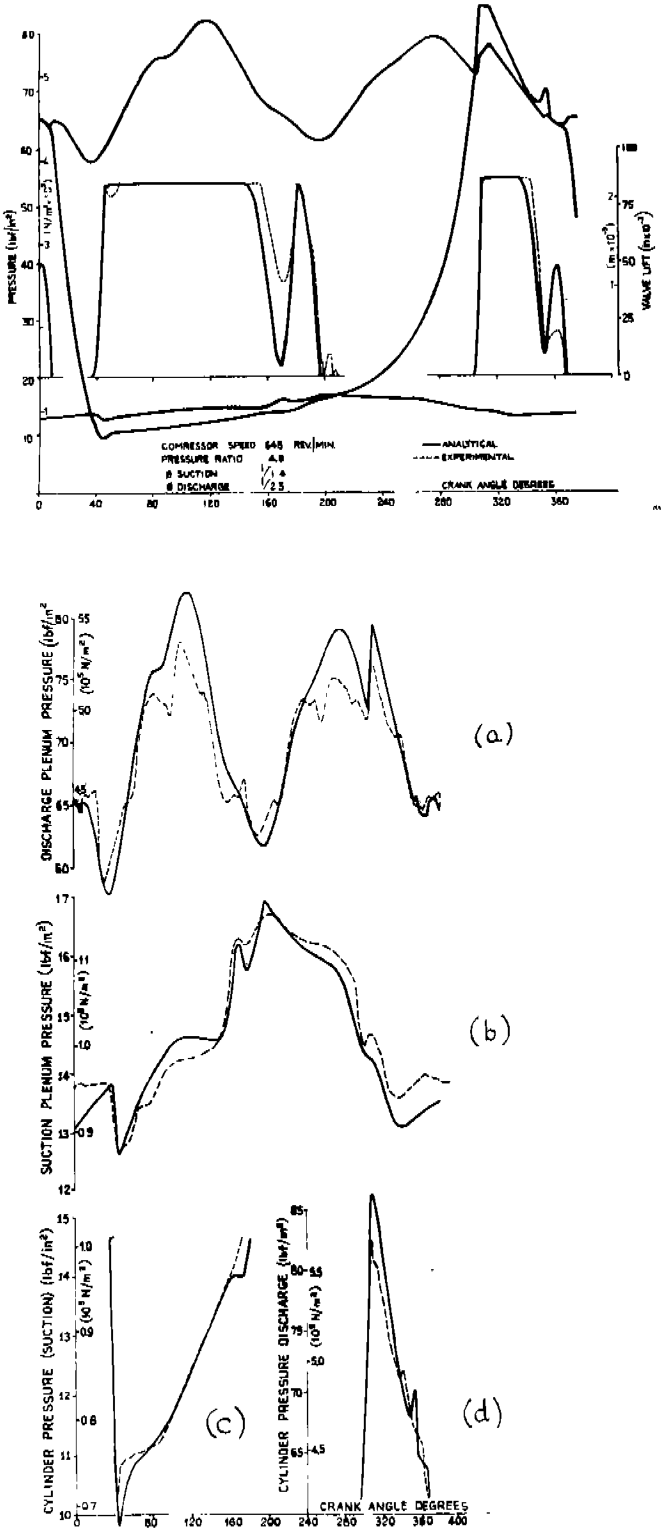

FIG. 3. COMPARISON OF EXPERIMENTAL AND ANALYTICAL RECORDS (TEST B5)

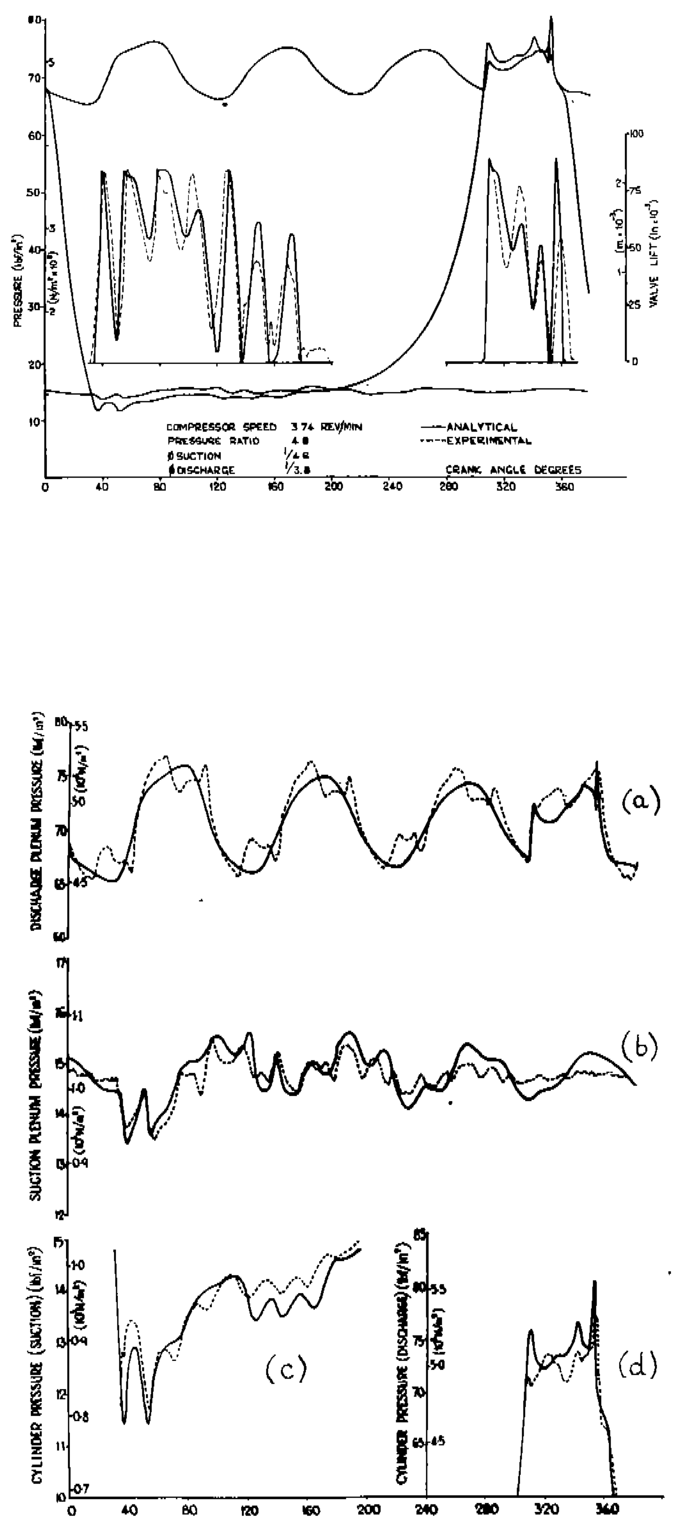

FIG. 4. COMPARISON OF EXPERIMENTAL AND ANALYTICAL RECORDS. (TEST C 2.)

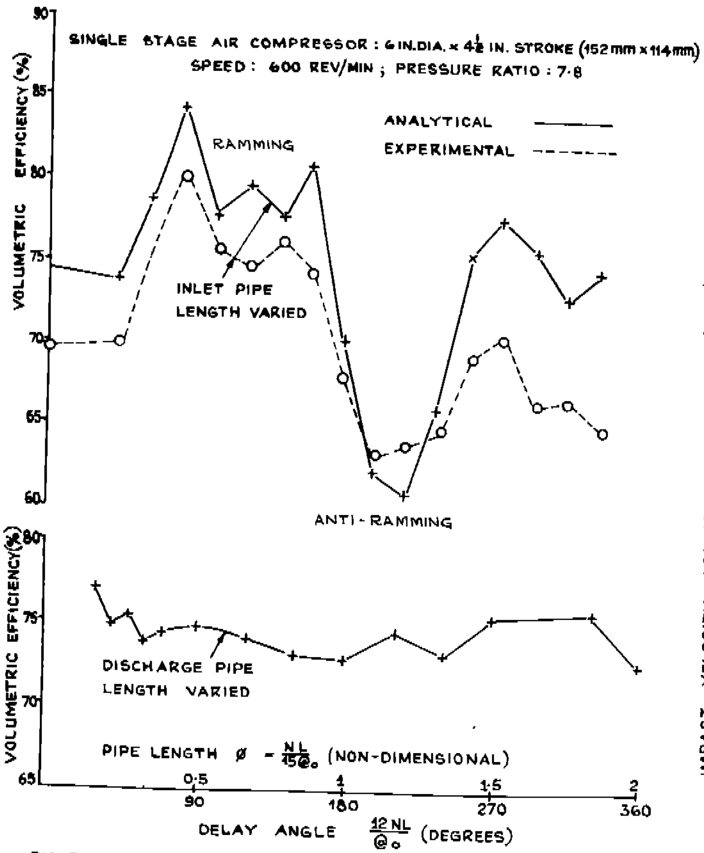

FIG 5. EFFECT OF PIPE LENGTH ON VOLUMETRIC EFFICIENCY

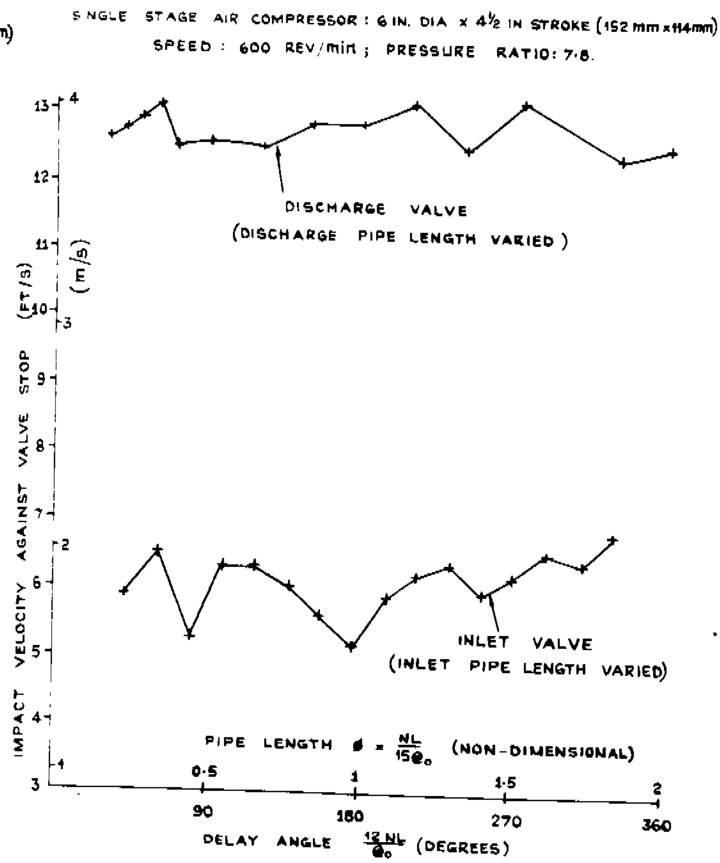

FIG 6 EFFECT OF PIPE LENGTH ON VALVE IMPACT VELOCITY ANALYTICAL