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ABSTRACT

Experimental records of the displacement during opening of a ring plate suction valve in an air compressor showed that the plate did not lift symmetrically. Values of velocity at the centre of gravity of the plate were compared to those calculated using three mathematical models (a), (b) and (c). The simplest, model (a), assumed that a uniformly increasing pressure difference (ramp function) existed across the valve. Model (b) accounted for the change of the pressure difference due to variation of the interrelated valve displacement and piston motion but assumed that the pressure in the plenum chamber remained constant. Model (c) described the complex physical situation in greater detail, accounting for the variation of the pressure difference across the valve due to the interacting effects of valve displacement, piston velocity and pressure changes in the finite volume plenum chamber caused by the wave action in the pipework of the system in addition to the pressure changes in the cylinder. Models (a) and (b) predicted valve velocities during suction valve opening greatly in excess of those measured. While model (c) predicted values of maximum valve velocity close to those observed, much more computer time was required by this model. While the experimental technique used provided accurate measurement of the maximum valve plate velocity it was concluded that it is difficult to predict this using a simple model. In addition, the relationship between this maximum velocity and the effective impact velocity is required, since it is the latter which is the more important criterion in relation to valve failure.

INTRODUCTION

The mathematical models of reciprocating compressors developed during the previous 25 years include sufficient detail of the complex physical situation which they describe that they may be used with some confidence as an aid to design. However, within these general models of the compressor (which may also account for the system in which the compressor operates) the description of some aspects (eg. valve dynamics and stressing, valve and piston leakage, real gas properties, heat transfer effects, etc.) are often neglected or accounted for only in a most rudimentary manner. These aspects are usually of lesser significance if the model is required primarily for the study of thermodynamic performance (compressor power and capacity), so that inclusion of them would unnecessarily increase computer time and cost. However, if the important criterion of durability is of prime concern then a more detailed analysis is required, particularly of the valves.

Most models predict valve displacement as a function of crankangle or time and hence predicted values of valve plate velocity are readily available by differentiation. This parameter is of relevance to valve durability yet the methods used to predict displacement and hence velocity are usually crude. For example, assumptions are often made that the drag force of the gas on the valve, F, can be described by incorporating an empirical drag coefficient C_D in a simple expression of the form $F = C_D A \Delta p$, that C_D can be evaluated by steady flow tests and thereafter assumed a constant independent of valve lift or of the effective area of plate normal to flow, A, or of Δp, the pressure difference across the valve. Similarly to simplify description of the dynamics of the valve it is common to assume that the valve is a single degree of freedom spring-mass system. These simplifying assumptions may be justified when only the overall thermodynamic performance of the compressor is of concern.

However if valve failures occur, durability tends to take precedence over thermodynamic performance and models with adequate detail are not yet available to supply reliable data on valve plate velocity, effective impact velocity at seat or stop, the dynamics of multi-degree of freedom valves, the bending, torsional, impact and fatigue stresses and strain energies involved during movement or impact. Only after such reliable data becomes available can the relationship be sought between these parameters and valve failures experienced in the field.

PREVIOUS INVESTIGATIONS

Some contributions towards providing a fuller description of the valves in compressor models have been made. In 1967 Wambsganss and Cohen (1) accounted for the drag which provides the accelerating force on a valve plate by including an empirical coefficient, C_D, as a function of valve lift and employed the appropriate value of this coefficient for direct or reverse flow through the valve port. Brown, Davidson and Hallam (2) at the 1976 Conference at Purdue University, showed that the drag force determined in a steady flow rig was not significantly different, except at low valve lift, from that measured in a much more elaborate dynamic test rig. Hallam (3) makes a further contribution at the present Conference by describing a finite element method to calculate the forces on a disc valve.

Detailed modelling of the dynamics of cantilever valve reeds was undertaken by Gatecliff and Henry (4) and programs to evaluate stresses in such flexing reeds have been developed, using finite element methods, by Friley and Hamilton (5).

The velocity of the valve plate is an important parameter relevant to valve dynamics and to stressing. Sufficient knowledge of even this basic parameter is lacking, partly because accurate experimental evidence has not been available from which to assess the validity of the various analytical models capable of predicting it. This paper compares measured values of valve velocities with those predicted using three mathematical models. These models give an increasingly detailed description of the physical situation and hence are progressively more demanding of computer time.

VALVE PLATE VELOCITY

In a paper (6) presented at the 1976 Conference at Purdue University a computer controlled high-speed data acquisition and processing system was described which provided experimental values of valve plate velocity both accurately and conveniently. Figure 1 is reproduced from that paper and illustrates the displacement and velocity at one circumferential location of a single annular ring plate valve backed by three helical coil springs, fitted to both the suction and discharge sides of an air compressor. The same valves and compressor (6 in bore x 4.5 in stroke), delivering about 20 ft^3/min of free air at 400 rev/min, was used in the present study. In a group of five adjacent points on the plot of valve displacement each point in Figure 1 is separated in time by only 20 μs. From these displacements the valve plate velocity was evaluated.

The maximum valve plate velocity and the velocity of the plate as it approached its stop are obviously different. In the published literature the maximum velocity is not usually distinguished from the velocity of approach to the stop and both are often considered to be synonymous with effective impact velocity. It can be observed from Figure 1 that at the low compressor speed of 360 rev/min the suction valve did not reach the stop during the first opening and clearly the maximum velocity reached during opening exceeded the approach velocity which died to zero. At higher speeds the valve struck the stop and there was an effective impact velocity but the magnitude of it cannot be quantified from such diagrams. Whilst there may be a "squish" effect as a thin layer of gas and oil decelerates the valve plate just before impact with the stop (or seat) the magnitude of the damping effect would depend significantly on local geometry and the manner in which the plate approaches the stop (or seat). The evidence in Figure 1, was based on the output of a single displacement transducer which could not show whether the ring plate was inclined. Damage associated with impact velocity effects is influenced by such geometrical factors as the area of contact at the stop or the width of the valve seat. A crude rule based on experience, states that the valve plate velocity at the stop for a discharge valve is twice that for the suction valve, yet in the case of cantilever valve reeds, the suction reed may fail more frequently since it does not have the backing plate which can be incorporated with the discharge valve to provide the large area of contact which will reduce impact stresses: the suction reed energy at impact has to be absorbed over a small area at the reed tip.

The effective impact velocity of large valves (eg. Hoerbiger type) is often reduced mechanically by damper plates, and this mode of deceleration can be readily accounted for in mathematical models (7). In other designs (eg. Worthington and Nuovo Pignone) deceleration is enhanced by incorporation of pneumatic damping which could be accounted for in a mathematical model if its magnitude was known.

Sandvik in Sweden (8, 9) has developed an impact fatigue testing rig to examine steels used in the manufacture of reed valves. Information from this source is available on the effective impact velocity which can be tolerated. However, it is difficult to simulate operating conditions accurately in such a rig, and it is difficult to introduce the experimental technique used into the confines of a compressor.

It is not easy, therefore, to measure maximum valve plate velocity, velocity of approach to stop (or seat) or effective impact velocity with an accuracy sufficient to allow the validity of predictions by mathematical models to be assessed. Yet the matter should be pursued since each of these velocity parameters, related one to the other, are in turn related to valve life in a way which cannot yet be quantified. As has been often stated in general terms, these velocity parameters are of relevance both to the dynamics and to the stressing of a valve. They also relate to compressor performance: as a paper (10) to this Conference shows, a knowledge of the upper limit of valve impact velocity which can be tolerated is a constraint of major importance when attempting to design a valve for optimum thermodynamic performance of the compressor.

MEASUREMENT OF VALVE PLATE VELOCITY

The single annular ring plate valves in the compressor used were not concentric with the compressor cylinder. It was suspected therefore that the valve might tilt in operation so three displacement transducers were incorporated circumferentially round the valve plate. By superimposing the displacement diagrams at three locations (Figure 2 and 3 bottom left) non-uniform lifting was demonstrated. From these displacement diagrams the inclination of the valve plate was computed. It was established that tilting took place along a preferred axis, as one might expect from the geometrical arrangement of the valve in the cylinder head. Figures 2 and 3 (top right) illustrate that significant tilting occurred once or more often during the suction process. On occasions this tilting could be the maximum possible, in this case about 1.8 degrees, i.e. one point of the ring plate was touching the stop while the diametrically opposite point was on the seat.

From the differing valve displacements at the three locations the movement of the centre of gravity of the ring plate was computed (Figures 2 and 3 bottom right) and used as a single experimental record for comparison with valve displacement calculated by the three mathematical models, all of which assumed that the valve had only one degree of freedom. With model (c) the pressure-time history in the plenum chamber could be predicted so experimental records of it and of the cylinder pressure were made also (Figures 2 and 3, top left).

MATHEMATICAL MODELS

Models of compressors can be expensive to use in terms of computer time so some investigators have attempted to provide a simpler procedure to estimate the velocity of the valve plate.

Nagaoka and Hotani (11) in 1955 provided experimental data from tests on a methylchloride compressor (3 in bore x $3\frac{1}{2}$ stroke, 265 rev/min). The mean velocity of the reed type valves during opening and closing was measured. It was found that the velocity of the reed at impact on the stop was about twice that at closure on the seat. (We have found this to be more widely applicable than might be supposed and have suggested (Ref. 7, Table X) further crude rules which state that the velocities at stop and seat of a discharge valve are about twice those of the corresponding values for the suction valve and that a suction valve reaches a velocity during opening which is not much in excess of the mean piston speed.)

Creswick (12) in 1967 provided two relatively simple analytical methods to estimate valve velocity near the stop. These accounted for the effect of valve mass, valve plate area normal to flow and permitted lift. Spring forces were neglected in the first method but included in the second. It was recognised that the rate of cylinder pressure rise was an important operating parameter, but a major simplifying assumption was made namely that cylinder pressure increased (or decreased in the case of the suction valve) at a constant rate during valve opening (an assumption made again in model (a) of the present study). No experimental evidence was presented but in the discussion on this paper Payne and Cohen showed that the model by Creswick overestimated the velocity of a discharge valve by 1.2 to 3 times the measured value. The largest discrepancy was at the highest permitted valve lift, as would be expected because the rate of change of cylinder pressure at higher valve lifts will have decreased, perhaps to zero or even reversed in sign (Figures 2 and 3).

At the 1974 Conference at Purdue University, Woollatt (13) provided an economical method to calculate the displacement of a valve and pressure drop across it. A step by step process permitted progressive evaluation of cylinder pressure and the rate of change of cylinder volume by solving six algebraic equations. The predicted and measured displacement diagrams for a suction valve during a cycle compared well but sufficient detail was not presented to permit comparison between predicted and measured valve plate velocity. It would be interesting to compare the predictions of velocity using Woollatt's model with those from the three models used in the present investigation.

MODEL (a) (HUSSEIN)

Hussein (14) developed a simple model to predict impact velocities in a valve treated as a single degree of freedom damped spring mass system. In many respects the model is similar to that suggested by Creswick (12) but allows damping effects to be analysed. The model assumes that the pressure difference causing the initial valve movement varies linearly with time (crankangle) and consequently separates the flow and valve dynamic effects which are interrelated in practice. Providing that the initial rate of change of the pressure difference can be found (either analytically or experimentally) the model yields a second order total differential equation which can be solved by standard methods.

Hussein wrote the equation of motion of a valve as

$$M \frac{d^2y}{dt^2} + C \frac{dy}{dt} + ky = B_o t$$

where $B_o = C_D A\omega \left[\left(\frac{dp_i}{d\theta}\right)_o - \left(\frac{dp_c}{d\theta}\right)_o \right]$

the gradients $\left(\frac{dp_i}{d\theta}\right)_o$, $\left(\frac{dp_c}{d\theta}\right)_o$

were evaluated at the instant of valve opening.

The solution of the equation to give the valve displacement y for the case when

$$\left(\frac{C}{2M}\right)^2 < \frac{k}{M} \qquad \text{is}$$

$$y = \frac{B_o}{k}\left[t - \frac{c}{k} + \frac{1}{\omega_{nd}} e^{-\frac{c}{2M}t} \cos\left(\omega_{nd} t - \Psi\right)\right]$$

which yields the following expression for the valve velocity

$$V = \frac{dy}{dt} = \frac{B_o}{k}\left[1 - \frac{\omega_n}{\omega_{nd}} e^{-\frac{c}{2M}t} \cos\left(\omega_{nd} t - \frac{\Psi}{2} - \frac{\pi}{4}\right)\right]$$

where $\omega_n = \sqrt{\frac{k}{M}}$ (undamped natural frequency)

and $\omega_{nd} = \sqrt{\frac{k}{M} - \left(\frac{C}{2M}\right)^2}$ (damped natural frequency)

and $\Psi = \tan^{-1}\left[\frac{\omega_n^2 - 2\,\omega_{nd}^2}{2\omega_{nd}\sqrt{\omega_n^2 - \omega_{nd}^2}}\right]$

Thus the model calculates y and V for successive values of t until a time is reached at which $y = y_{max}$. The value of V when $y = y_{max}$ was taken to be the impact velocity. In the present study experimentally obtained pressure records were used to calculate the parameter B_o.

MODEL (b) (KERR)

The major simplifying assumption made in model (a) was that the pressure difference across a valve during opening or closing was a linear function of time. This is relaxed in model (b) by computing the variation of this pressure difference which depends on the motion of the valve, the motion of the valve depending in turn on the pressure difference across it. The inter relationship can be described by two non-linear differential equations which have to be solved simultaneously by iterative methods. A "flow" equation expresses the pressure difference across the valve and a "dynamic" equation describes the motion of the valve, both being written as functions of crankangle (time). By assuming that the plenum chamber pressure remains constant during the compressor cycle, the cylinder pressure may be evaluated by subtracting (suction) or adding (discharge) the pressure difference across the valve from or to the plenum chamber pressure. This model has been widely used. In references (1), (15), (16), (17) it is employed, inter alia, to predict valve velocities. Costagliola (17), suggested that discharge valve lift during opening was related to time by a cubic parabola, a conclusion reached again later by Creswick (12) based on his simpler model which was similar to model (a).

Problems of stability of solution of the equations do not arise so the time step used in the iterative solution can be selected to suit. For example, to economise on computer time, a relatively large step of say one crankangle degree may be used when proceeding round the compressor cycle but this may be reduced automatically to 1/10 crankangle degree during the few crankangle degrees when a valve is opening and closing, when it is desired to evaluate the valve velocity and changes of it. The model assumes a single degree of freedom valve system with valve plate motion normal to the valve seat.

MODEL (c) (PASTRANA)

In models (a) and (b) attention was focussed on the compressor valves and the cylinder but the effect of the plenum chambers and pipework associated with compressor are wholly neglected. In model (c) the focus is rather on the pulsating flow which occurs in any positive displacement compressor system: the valves are treated as orifices of variable area and form one of the several types of boundaries to be crossed by the unsteady flow. Hyperbolic type partial differential equations describe the unsteady compressible flow in which finite amplitude pressure pulsations and heat transfer effects were allowed for but the flow was assumed to be approximately one-dimensional. These equations were solved for the various boundaries encountered by numerical procedures described at the 1976 Conference at Purdue University (18, 19, 20). This model therefore provides a more complete description of the pressure time history in each plenum chamber due to the wave action in the associated pipework of the compressor, together with the pressure difference across the suction or discharge valve, the displacement of the valves and the pressure in the cylinder. Valves were once again treated as single degree of freedom spring mass systems.

COMPARISON OF EXPERIMENTAL RESULTS WITH PREDICTIONS BY THE MODELS

While valve displacement could be measured accurately, a problem remained of locating precisely the crankangle at which the valve started to lift. The problem, complicated by the tendency of the valve plate to tilt, had to be resolved before meaningful comparison could be made between the experimental results and predictions by the models. The problem is particularly acute with a suction valve, where the velocity of the valve as it leaves its seat is small compared to that of a discharge valve. It was arranged, therefore, for the valve plate to operate four make-and-break electrical contacts placed around the valve seat (Figure 6). The signal from each point of contact was observed via the computer in the data acquisition system. This technique permitted the commencement of lift to be determined with sufficient accuracy.

Figures 4 and 5 (bottom right) illustrate that the suction valve displacement on opening was less rapid than predicted and hence the observed velocity, plotted to a base of valve displacement (Figures 4 and 5 top right) or to crankangle (Figures 4 and 5 bottom right), was only half that predicted by model (a) and appreciably less also than the prediction by model (b). The maximum velocity reached by the valve plate was predicted well by model (c), but like the simpler models it contained no provision to account for the deceleration close to the stop. In the particular situation examined the valve springs were sufficiently weak, the permitted valve lift sufficiently low and the compressor speed sufficiently high that, according to the models, the valve plate was still accelerating just before impact, contrary to the experimental results which showed that there was significant damping occurring as the valve approached the stop.

An analysis of the experimental results for the suction valve displacement in the test compressor showed that the valve plate was generally inclined during its period of motion. It followed therefore that at the moment of impact, contact between the valve plate and the valve stop would occur at a single point on the periphery of the ring plate. Figure 8 shows the velocity and displacement of the point on the surface of the ring plate for which the initial opening impact occurs. At the moment of impact the impact velocity is almost equal to the velocity of the centre of gravity which as yet had not shown any significant deceleration attributable to a squish damping effect. The sudden change of plate inclination is indicative of a point contact and a reversal of the direction of rotation of the ring plate. After impact it is still possible for the centre of gravity to move towards the stop but with a reduced velocity. The valve plate velocities were computed from displacement transducer readings taken at $\frac{1}{2}$ degree intervals of crankangle. More precise data on impact velocity and of coefficients of restitution might have been obtained if the technique illustrated in Figure 1 had been extended to each of the three transducers.

Tests were conducted over a wide range of conditions and provided general confirmation of the particular observations made from the samples shown in Figures 4 and 5. Figure 7 (i) and (ii) illustrates the measured and predicted maximum valve plate velocities with (i) no air filter and the suction plenum open to atmosphere and (ii) an inlet pipe with perforations connected by another long pipe to the inlet. Other inlet conditions and compressor pressure ratios were used but the results did not add to the conclusions to be drawn from Figures 7 (i) and (ii). Only model (c) can differentiate between the compressor with and without inlet pipe work. The small differences between the predictions by models (a) and (b) in Figure 7 (i) and (ii) are solely due to the slightly different compressor pressure ratio used.

CONCLUSIONS

Knowledge of valve plate velocity is needed if progress is to be made in the understanding of valve behaviour and the specification of criteria for valve durability.

The present study has utilised a technique described previously (6) to measure accurately the velocity of ring plate valves in a relatively slow speed air compressor; the technique could be as readily applied at high compressor speeds. Experimental evidence showed that flow asymmetry had a tilting effect on the motion of the ring plate valve, justifying the use of more than one displacement transducer to monitor valve motion when accurate records are required.

Three models by (a) Hussein, (b) Kerr and (c) Pastrana, of progressively increasing detail and hence complexity, were used to predict valve velocities during opening of this ring plate valve. Model (a) overestimated the maximum velocity reached by the plate by 100%, a finding similar to that reached by Payne and Cohen regarding a similar model used by Creswick (12). Model (b) overestimated the maximum velocities by 50%. Model (c) predicted it well. Model (c) was the only one capable of accounting theoretically for the pressure variation due to the intermittent flow in the compressor plenum chambers but the effect of this on maximum valve plate velocity was small (Figure 7).

The maximum velocity reached by a valve plate is of interest in relation to valve durability only insofar as the effective impact velocity of the plate at its stop (or seat on closure) is related to maximum velocity. This relationship is not known: it cannot be deduced from experimental evidence in the form of Figure 1 or from any of the mathematical models since none of these accounted for damping as the valve approached its stop (or seat). The maximum impact velocity which can be tolerated depends not only on the properties and finishes of the valve components but on the valve geometry and the valve dynamics. For example, the tilting of ring plate valves or flexing of reed valves has an influence on the area of contact at impact. Thus much remains to be done to relate valve plate velocity, whether measured, or predicted by a valid model, with valve durability.

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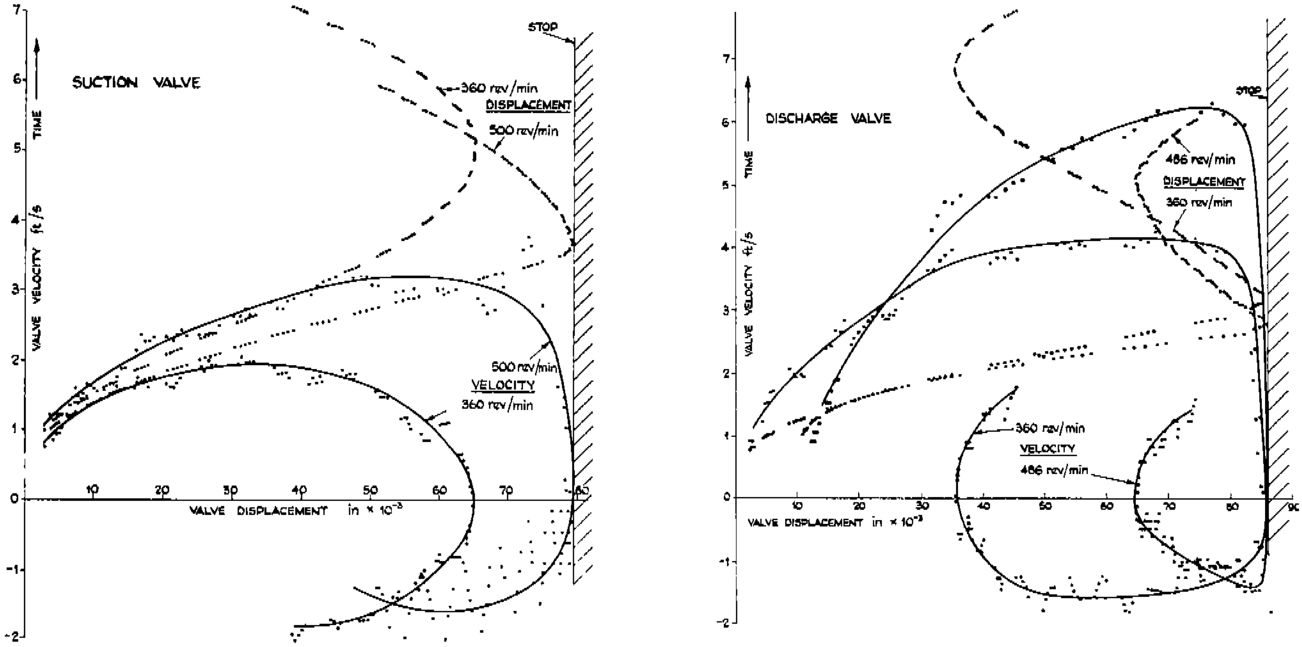

FIGURE 1 : EXPERIMENTAL DISPLACEMENT AND VELOCITY OF RING PLATE VALVE WHEN OPENING

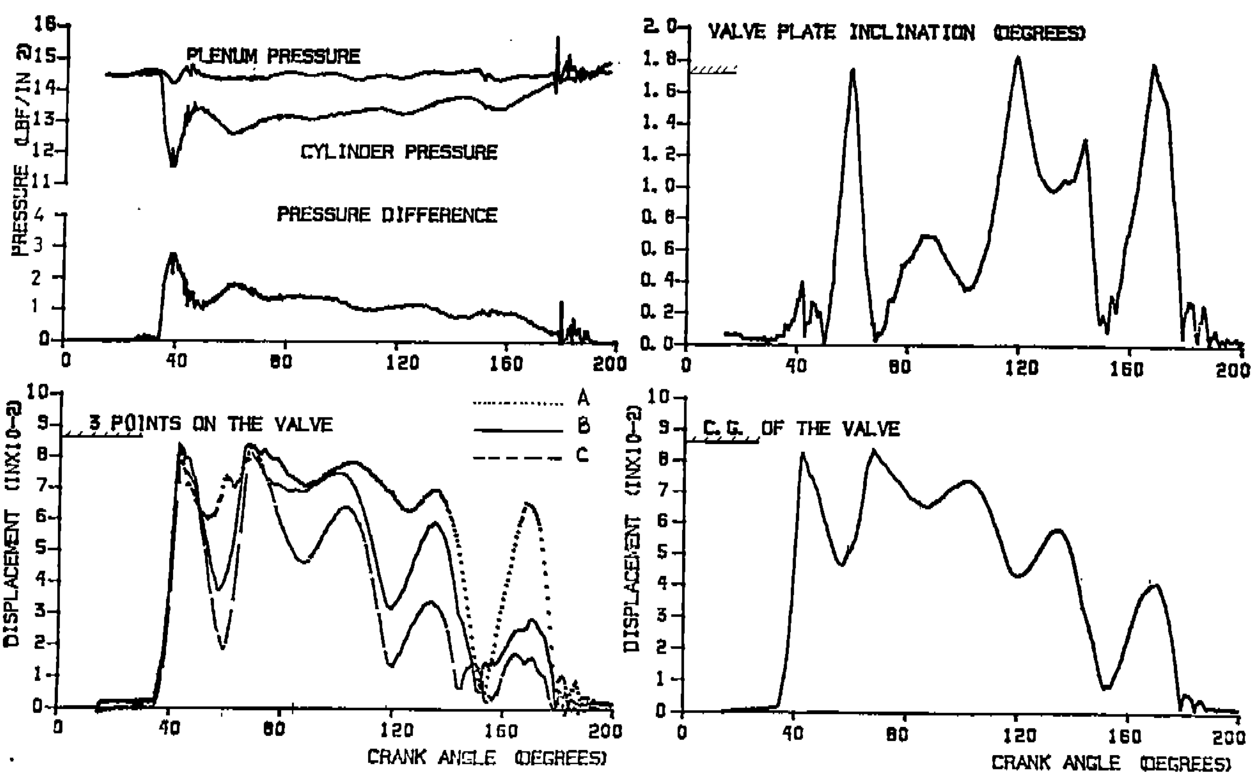

FIG. 2. EXPERIMENTAL RESULTS. (420 REV/MIN R = 6·1)

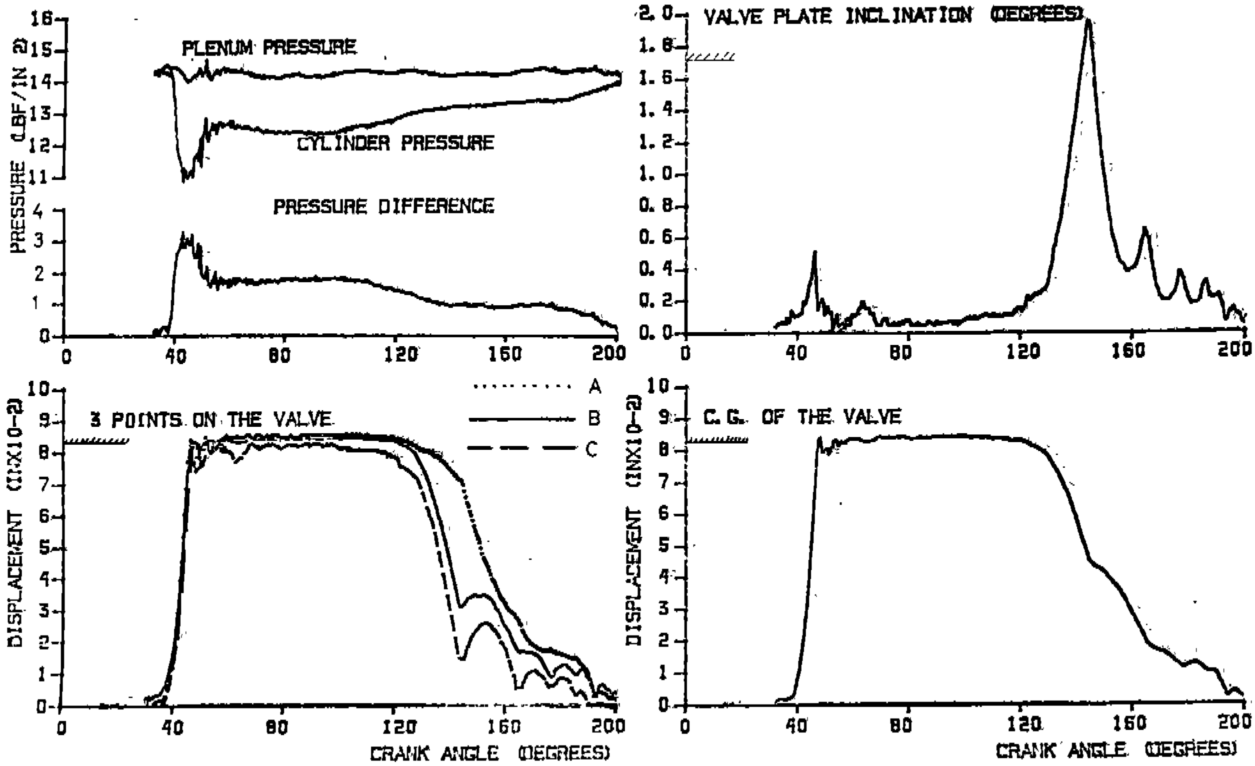

FIG. 3. EXPERIMENTAL RESULTS. (550 REV/MIN R = 6.1)

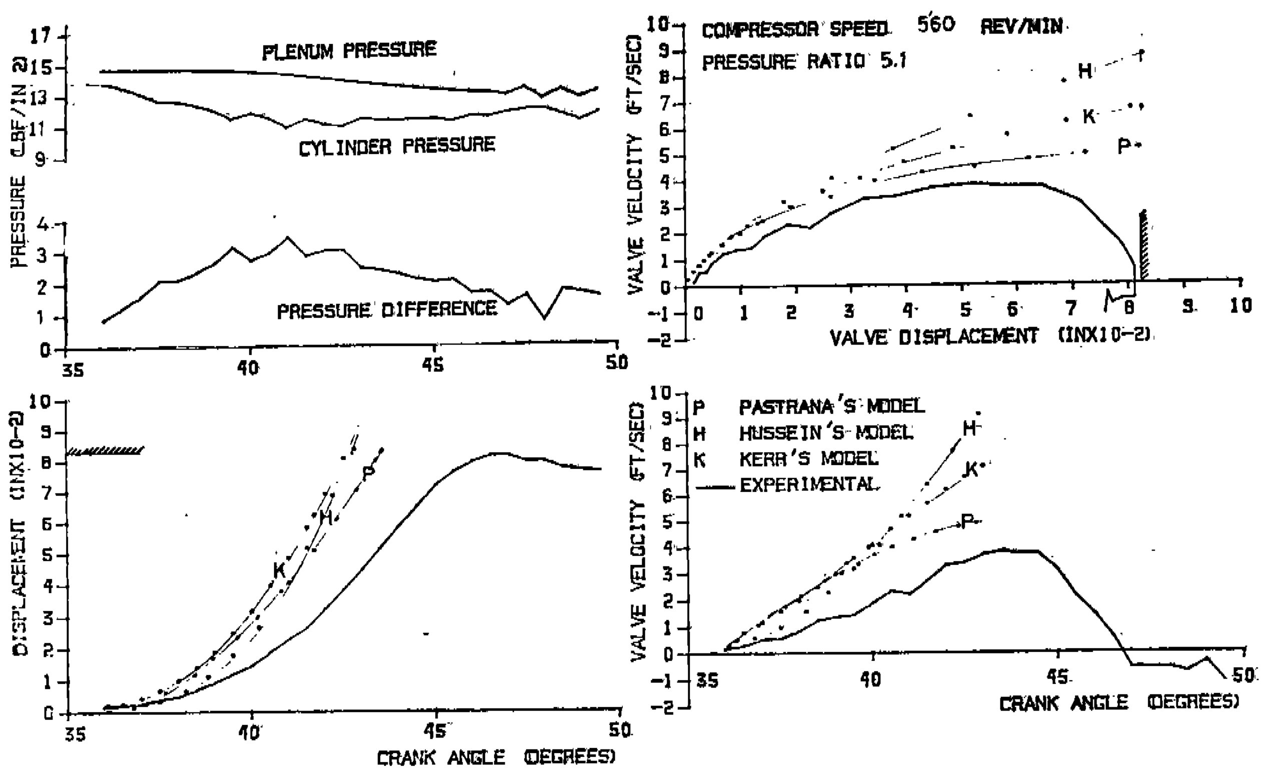

FIG. 4. COMPARISON OF ANALYTICAL AND EXPERIMENTAL RESULTS.

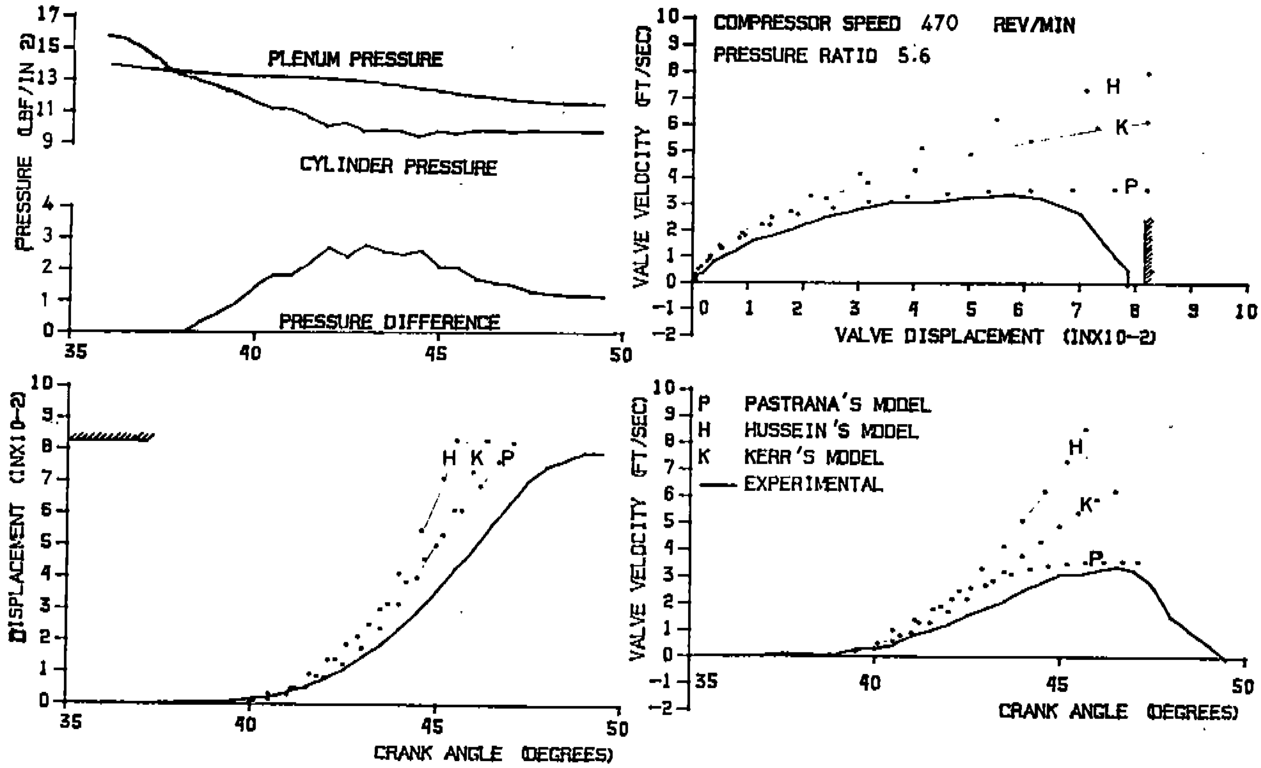

FIG. 5 COMPARISON OF ANALYTICAL AND EXPERIMENTAL RESULTS.

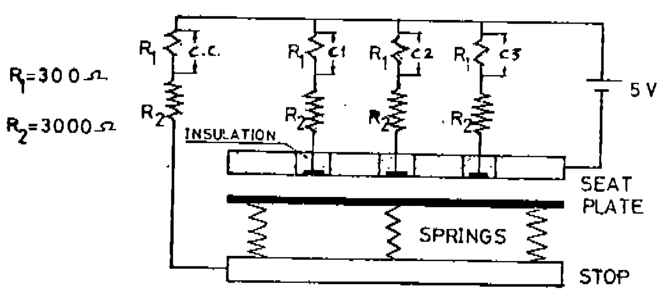

FIG.6 (a) SCHEMATIC DIAGRAM OF CONTACT ELEMENTS

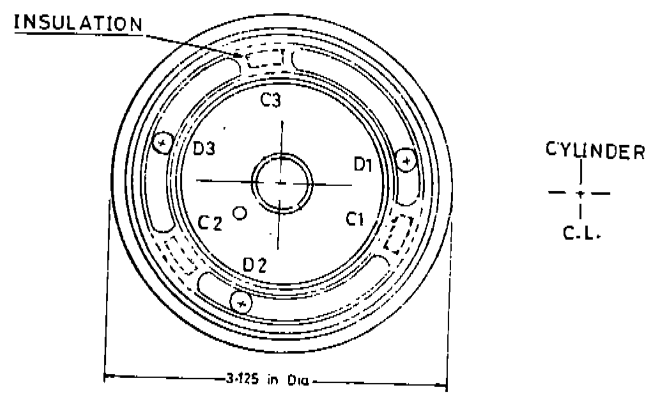

FIG.6 (b) LOCATION OF DISPLACEMENT TRANSDUCERS (D1 D2 D3) AND CONTACT ELEMENTS (C1 C2 C3) IN SUCTION VALVE SEAT

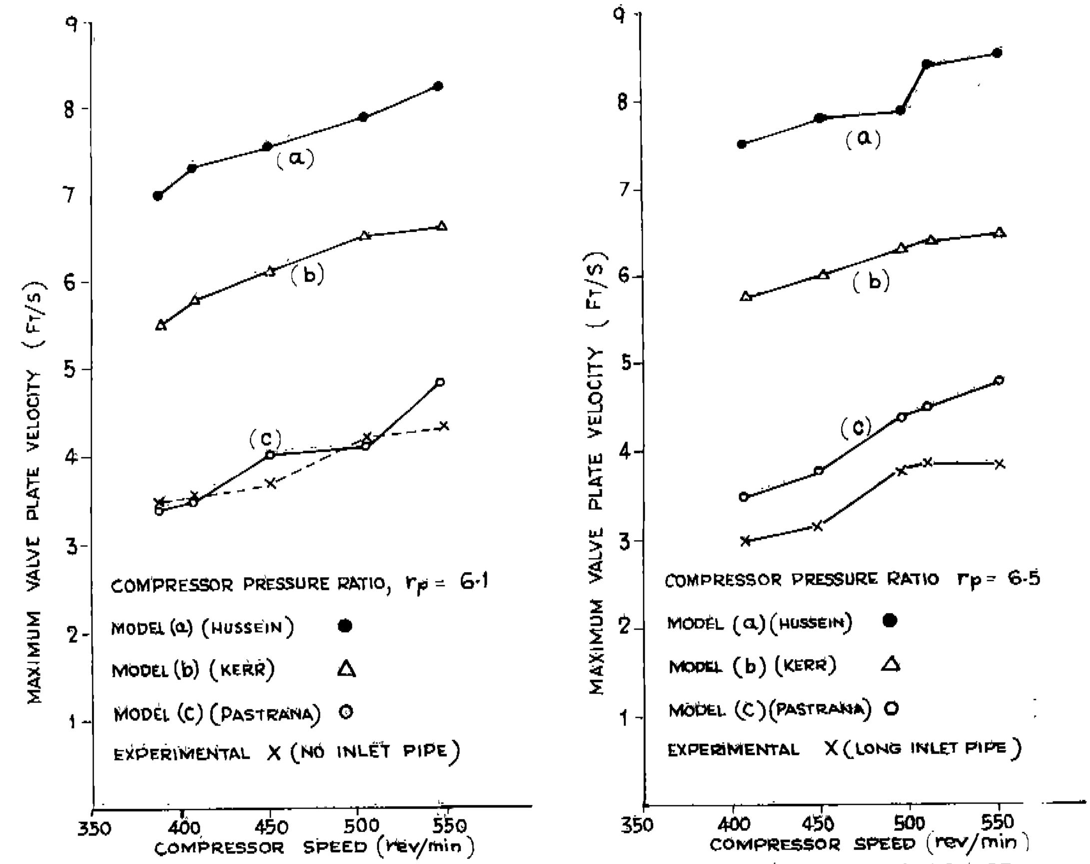

FIG. 7. COMPARISON OF MEASURED AND CALCULATED VALVE PLATE VELOCITIES

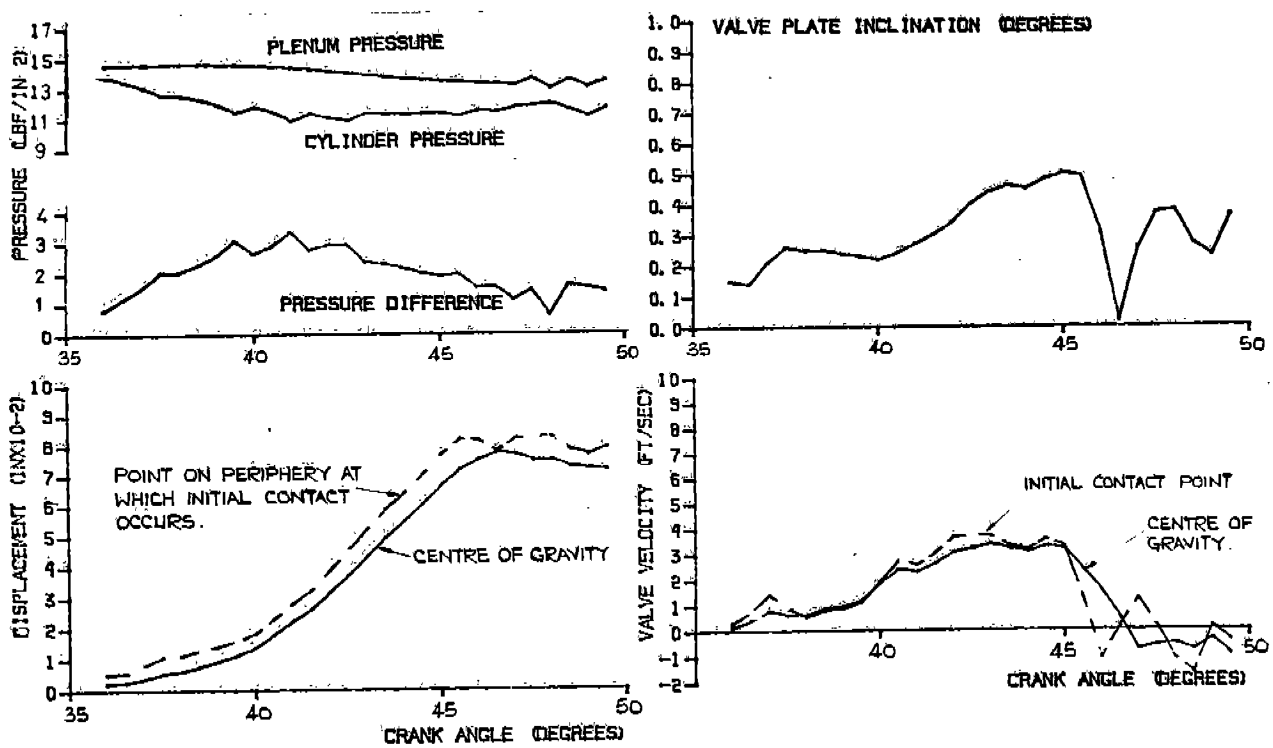

FIG. 8 EXPERIMENTAL RESULTS. (560 REV/MIN R = 5·1)