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SOME RECENT RESEARCH IN GAS DYNAMIC MODELLING OF
MULTIPLE SINGLE STAGE RECIPROCATING COMPRESSOR SYSTEMS.

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INTRODUCTION

Models for simulating single cylinder reciprocating compressors have been suggested by Costagliola (1), Wambsganss et al (2) and Bannister (3). These have either neglected the dynamics of the intake and delivery systems (1,2) or neglected the inertia of the compressor valves (3). Several authors (4-9) have suggested methods for predicting the pressure pulsations in compressor delivery systems. These methods are essentially based on small wave theories. These lead to an insight into the problems of pulsating flows in compressor systems, and give reasonable predictions of pulse frequency, but pulse amplitude predictions, however, become steadily worse as the amplitude of the pressure pulse increases. Extensive research (10-13) on non-steady flows in internal combustion gas exchange systems has led to the development of powerful computer programs for the study of non-steady flows. Excepting for the mode of operation of compressor automatic valves these techniques can be readily applied to compressors and their intake and delivery systems. This paper reviews some recent researches on the application of these methods to single cylinder compressors in multi compressor installations. Space precludes a detail review of this work, this has been given elsewhere (14-16).

As a first step an experimental and theoretical investigation was carried out on non-steady flow through non return disk valves (17). This was used as the basis for modelling the valve movement. Since the generalized non-steady flow equations including friction, heat transfer, gradual area changes and entropy gradients required extensive computing time with the generation of computers available at the commencement of the project, a modified theory was developed in which the flow was assumed to be isentropic in the pipe, but friction was included. Simplifying assumptions were made and adjustments included in the calculations to allow for these assumptions (14-16). To test the model, an extensive test programme was carried out with a large number of compressor systems with one, two and three single cylinder compressors and receivers of different capacity.

GAS DYNAMIC MODEL

The development of the algebraic expressions used in the gas dynamic model have been given elsewhere (14-

16). It is proposed therefore to briefly summarise the treatment. For calculation purposes compressor-pipe systems may be subdivided into two parts, the primary part comprising the intake and delivery pipes and the secondary part the compressor cylinder, the suction and delivery valves, the pipe junctions and the receivers.

The calculations of the conditions in the primary parts of the system involve the numerical solution of the one-dimensional non-steady flow equations by the method of characteristics (14). For each pipe the distance-time field is subdivided into a mesh system. At each point the non-dimensional speed of sound A and particle velocity U are calculated through the Riemann variables λ , β ($\lambda = A + \frac{k-1}{2} U$, $\beta = A - \frac{k-1}{2} U$). At each pipe end either λ or β is known but not both. The secondary or boundary calculations evaluate the unknown Riemann variable and the gas dynamic conditions at the pipe ends. If the volumes at the boundaries can be neglected the flow conditions are determined using quasi steady equations of continuity and energy. If, however, the volume at the boundary is significant then the generalized first law of thermodynamics (equation (2.20), ref.(18)) is used relating the conditions within the volume to the flow rates as well as the quasi steady equations of continuity and energy.

At the time of the commencement of the research project, the allocated computer run times were limited, it was therefore decided to simplify the characteristics equations by neglecting heat transfer and entropy changes and include friction only. Thus, a modified homentropic theory was developed (14). It was subsequently found that the difference in entropy levels in the various parts of the system as well as the heat transfer in the pipe had an important influence on the calculations within the pipe and cylinder. By obtaining the mean entropy level in the system and adjusting the pipe length for the temperature variations due to heat losses the modified homentropic theory gave good results. It was arranged that the mean entropy level could be automatically calculated in the calculation (15), whilst the pipe length adjustments could be predicted in advance (16).

With the present generation of computers time allocations become reasonable and there is no difficulty using the full non-homentropic theory which allows for entropy gradients and heat transfer and the approximations referred to are no longer necessary. Work is proceeding on these lines.

The secondary or boundary conditions have been based on theories (and experiments) used in internal combustion engine gas dynamics (12,13). For the compressor cylinder it is assumed that the pressure and temperature are uniform throughout the cylinder. The generalized first law of thermodynamics (equation (2.20) ref.(18)) is applied to compute the pressure and temperature changes from a knowledge of the mass flows into and out of the cylinder as well as the piston movement. For the compressor automatic valves two problems arise. These are the movement of the valve (which depends on the pressure difference across the valve as well as the spring stiffness, the valve and spring mass, pressure drag and viscous damping) and the flow mechanism through the valve. From the work of Kaddah and Woollatt (17) it is possible to set up the equation of motion of the valves. A second order differential equation can be formulated and solved by Kutta-Merson techniques. Experimental data are required to evaluate the drag. The flow mechanism through the valve is extremely complex, but based on internal combustion engine experience it is possible to simplify the flow system. The model is similar to the model used for poppet valves (19). For outflow from the cylinder to a pipe, it is assumed that the flow through the valve is isentropic up to the minimum effective area (or vena contracta). At this point the pressure is assumed to be either equal to the pipe pressure (for subsonic flow) or to the critical pressure (for choked flow). The flow from the minimum area to the full pipe area is assumed to be adiabatic but irreversible at either constant pressure (for subsonic flow) or with a pressure drop to pipe pressure (for choked flow). For flow from a pipe to the cylinder another model is used. In this case the flow is assumed to expand isentropically to the cylinder pressure (for subsonic flow this pressure occurs at the minimum area; for sonic flow the pressure at the minimum area is the critical pressure). Four sets of equations are set up for these four possible flow conditions in terms of the pressure ratio across the valve, the velocity in the pipe, the upstream stagnation speed of sound and the effective area of the valve (expressed as a ratio of the minimum area to the pipe area). From steady flow blowing tests across the valve the effective area can be computed as a function of the overall pressure ratio and the valve lift, (expressed as the non-dimensional parameter $\eta = \text{lift}/\text{pipe diameter}$). Typical experimental curves are shown in fig. 1.

Pressure receivers can be simulated as either pipes of large bore with lengthwise wave action or by vessels of constant volume with uniform pressures and temperatures. Since in practice the pipe connections may not always be in the end planes of a receiver the latter model is preferred. In this model the generalized first law of thermodynamics (equation (2.20) ref.(18)) is used for the receiver. At the pipe ends adjacent to the receiver the pressures are assumed to be equal to the receiver pressure for inflow. For outflow from the receiver to

the pipe the ellipse of energy (equation (4.8) ref. (18)) is used to relate the receiver pressure to the gas velocities at the pipe end. At pipe junctions, for example, a three way branch, the pressure is assumed to be the same at each pipe end and the volume neglected (12). The valve pockets are included in the pipe system by equivalent pipe lengths (16).

The complete model is set up as a digital computer program in a series of subroutines. A block diagram is shown in fig. 2 which is self explanatory. To reduce computation time the starting boundary conditions in the delivery system correspond to the set delivery pressure for the system, the controlling parameter being a nozzle of known area at the outlet from the delivery system. The size of the nozzle is calculated from the compressor capacity and speed. The calculation proceeds until there are no substantial changes in the pressure diagrams in two successive cycles. In practice this is found to be four cycles, the third and fourth cycle being almost similar.

EXPERIMENTAL PROGRAM AND TEST RESULTS

To test the model an extensive research programme has been carried out with three small air-cooled compressors. Photographs of the system are shown in figs. 3 and 4. Four different compressor systems have been tested. These comprise a single compressor with a delivery pipe, a single compressor with delivery pipe and receiver, two compressors with delivery pipes and receiver and three compressors with delivery pipes and receiver. Four different sized receivers have been examined ranging in volume from approximately twice to twenty times the compressor displacement volume. The receivers' diameter to length ratios followed normal commercial practice. All the compressors were driven from a single electric motor through a vee tooth belt drive with variable speed control. In the multi compressor tests various combinations of phasing between compressor cranks have been examined. Altogether some thirty eight different combinations of speed, compressors, pipes and delivery systems have been tested.

In addition to the conventional pressure, temperature and mass flow records the transient pressure was measured at various points in the pipe system as well as the cylinder. The delivery valve movement was measured with a contactless inductive transducer. The results were recorded on two four channel Tectronics oscilloscopes and analysed using a chart reader with a five times full size optical enlarger and outputted in digital form on paper tape. In general, good agreement was obtained between predictions and measurements. Some typical results are shown in figs. 5 to 11. The pressure diagrams for the receiver and the tail pipe are shown as pressure differences from the mean pressure level. All the other pressure measurements are in the conventional form. It has not been possible to obtain a dynamic calibration of the valve movement transducer due to zero drift. The results shown are based on the static calibrations with the maximum dynamic lift set equal to the maximum static lift. Comparisons should therefore be made mainly of the predicted frequency of valve flutter and the

measured flutter. Since the valve movement is critical to the pressures in the cylinder and the delivery and intake systems, the good agreement between the pressure predictions and measurements would indicate that the calibration technique used for the valve lift is not unreasonable and a quantitative assessment of the valve movement is not out of order.

A full interpretation of the results has been given elsewhere (15,16). Figs. 5 to 10 give a representative range of results which have not been published and supplement the results given elsewhere. Fig. 5 shows a typical result for a single compressor pipe system without receiver. The model over-predicts the valve flutter, nevertheless, the pressure diagrams are quite good. The influence of a receiver added to the delivery system is shown in fig. 6. The valve flutter is now reduced as predicted as well as the pressure fluctuations in the tail pipe (station 6). The influence of compressor speed is quite clearly seen in fig. 7. The results for the two compressor combinations are shown in fig. 8 for the two compressor cranks in phase and in fig. 9 for the two compressor cranks 180° out of phase and there is good agreement between the predicted phenomena and the measured results. Finally, fig. 10 gives the results for three compressors, compressors B and C in phase and compressor A 180° out of phase. Once again, there is good agreement between the predicted results and the experiments.

COMMENTS AND CONCLUSIONS

In order not to duplicate previous published work (14-16) and to keep within the space available, this paper has been a descriptive exercise of the development of a gas dynamic model to represent single stage compressors in multi compressor installations. The model is fairly simple, being based on extensive gas dynamic work on internal combustion engines. For overall predictions of pressure pulsations in compressor systems and valve movements, the method is reasonably accurate and should prove a useful design tool. A more sophisticated model for the pipe system would include entropy gradients and heat transfer. Such a model is already available for internal combustion engines; its adaptation to compressors is underway. The weakest link in the model is the representation of the valve. At present, the model is grossly oversimplified and depends on data obtained from blowing tests. What is required is some method to predict the data in advance from the valve geometry. This is complex because it will inevitably involve two or possibly three dimensional models. The calculation clearly shows, as with the internal combustion engine work, that steady flow models are adequate for the boundary conditions in non-steady flow calculations. Thus, non-steady effects need not be included in model predictions of flow through valves based on valve geometry.

The work described in this paper was completed over two years ago, pressure of other work has prevented a continuation of the research. Should funds become available it is hoped to examine two-stage compressors with and without intercooling using similar techniques to that described in the paper.

There is no limitation to the fluid being compressed provided it is gaseous and obeys the gas laws. A possible extension to gases obeying complex gas laws is possible but would require changes in the basic equations. It is hoped that this review paper will lead to stimulation of interest in applying the methods outlined in this paper to the difficulties and complexities referred to above.

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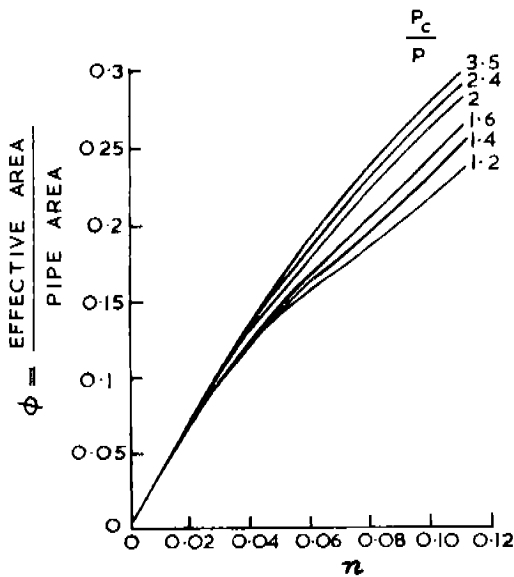


FIG. 1a. DELIVERY VALVE FLOW CHARACTERISTICS - OUTFLOW FROM CYLINDER

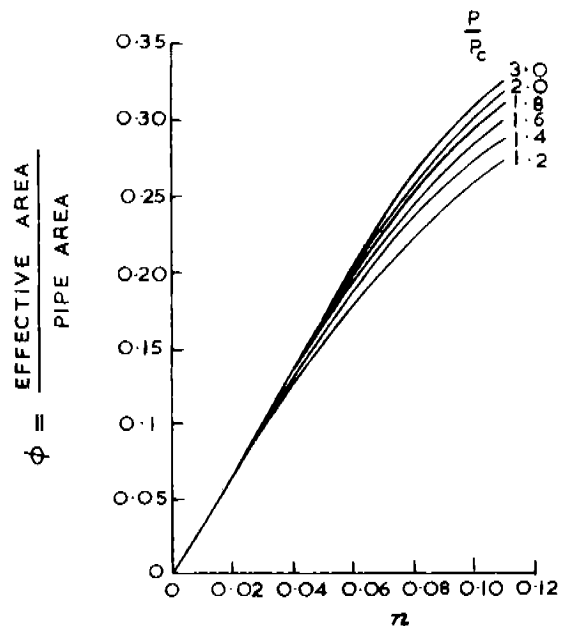


FIG. 1b DELIVERY VALVE FLOW CHARACTERISTICS - INFLOW TO CYLINDER

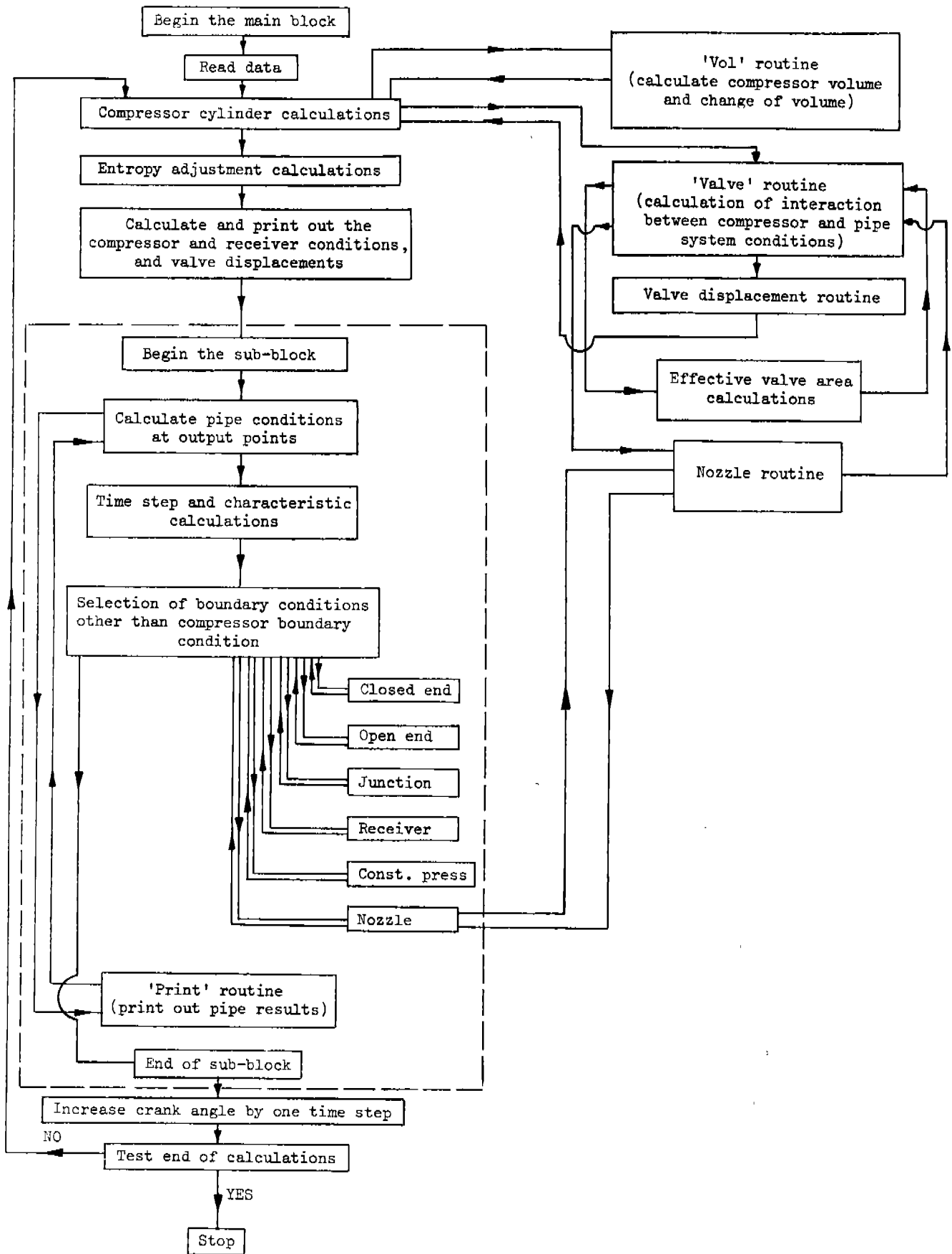


FIG. 2 Block Diagram.

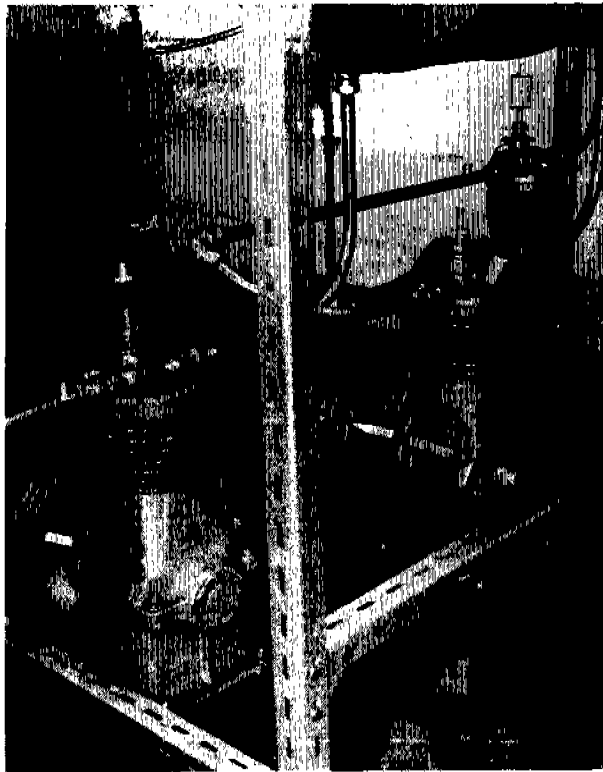


FIG. 3 ARRANGEMENT OF COMPRESSORS

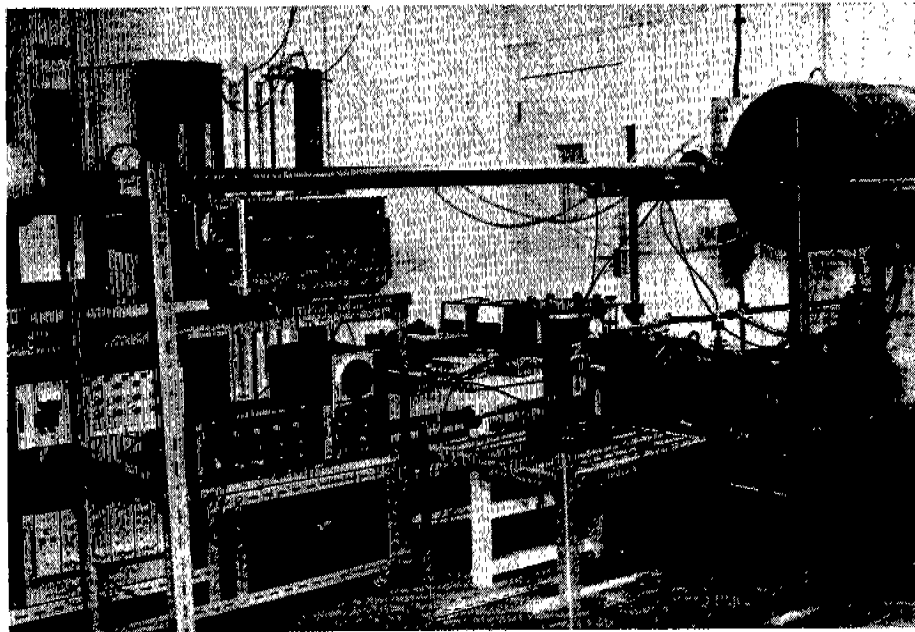


FIG. 4 GENERAL VIEW OF PIPE SYSTEM

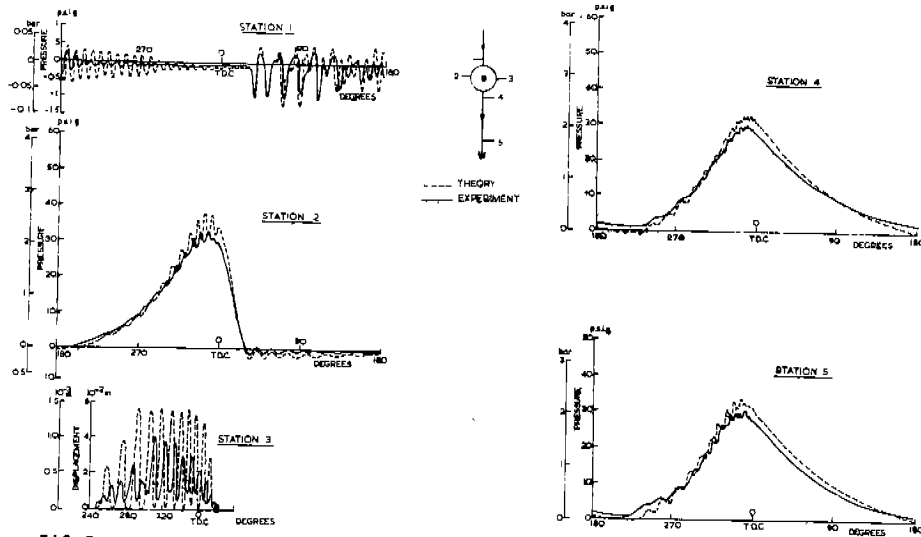


FIG. 5 COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS (TEST A6)
SINGLE COMPRESSOR 300 r.p.m.

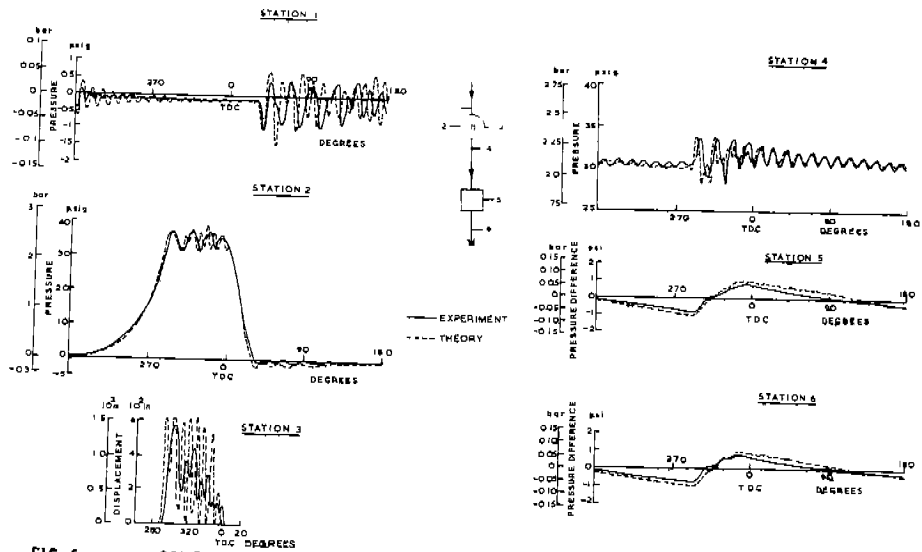


FIG. 6 COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS (TEST B3)
SINGLE COMPRESSOR AND RECEIVER VOLUME $2.359 \times 10^{-3} \text{ m}^3$ 300 r.p.m.

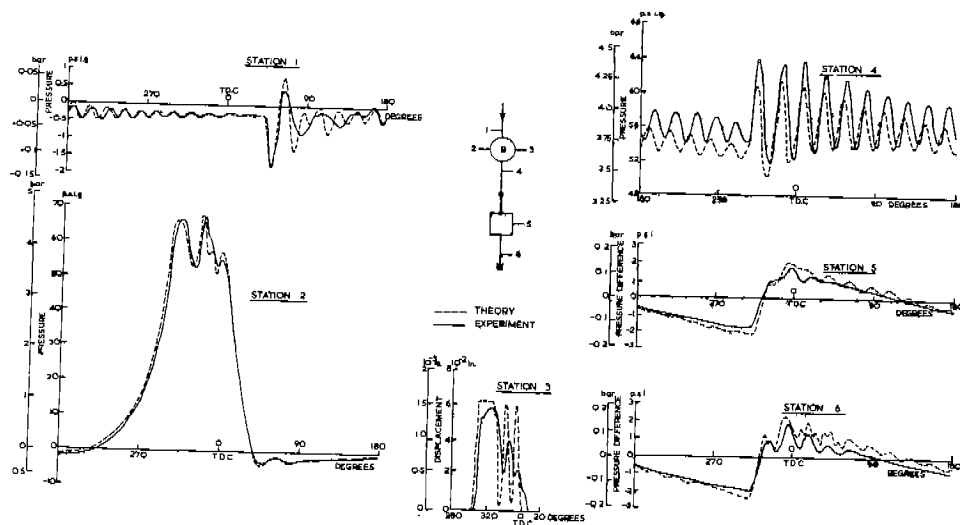


FIG. 7 COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS (TEST B5)
SINGLE COMPRESSOR AND RECEIVER VOLUME $2.359 \times 10^{-3} \text{ m}^3$ 500 r.p.m.

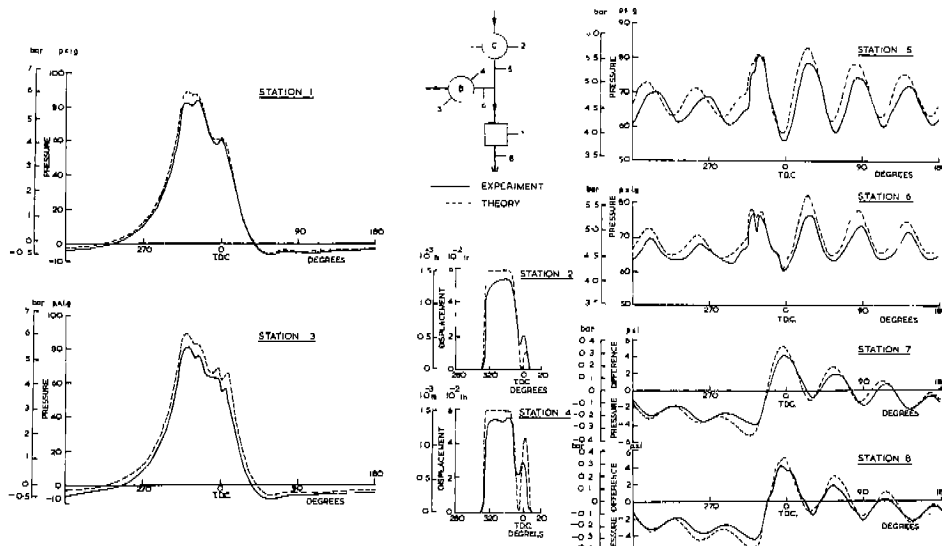


FIG. 8 COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS (TEST C8)
TWO COMPRESSORS IN PHASE 700 r.p.m. RECEIVER VOLUME $1.099 \times 10^{-3} \text{ m}^3$

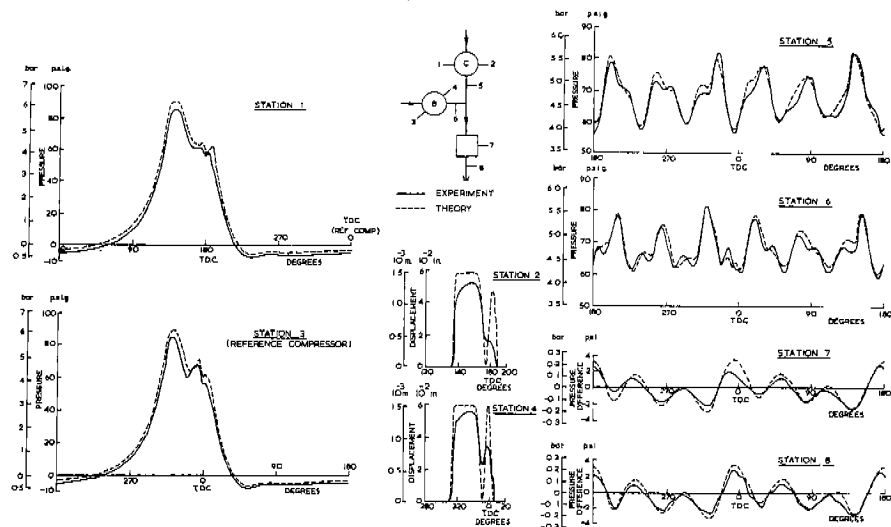


FIG. 9 COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS (TEST C9)
TWO COMPRESSORS 180° OUT OF PHASE 700 r.p.m. RECEIVER VOLUME $1.099 \times 10^{-3} \text{ m}^3$

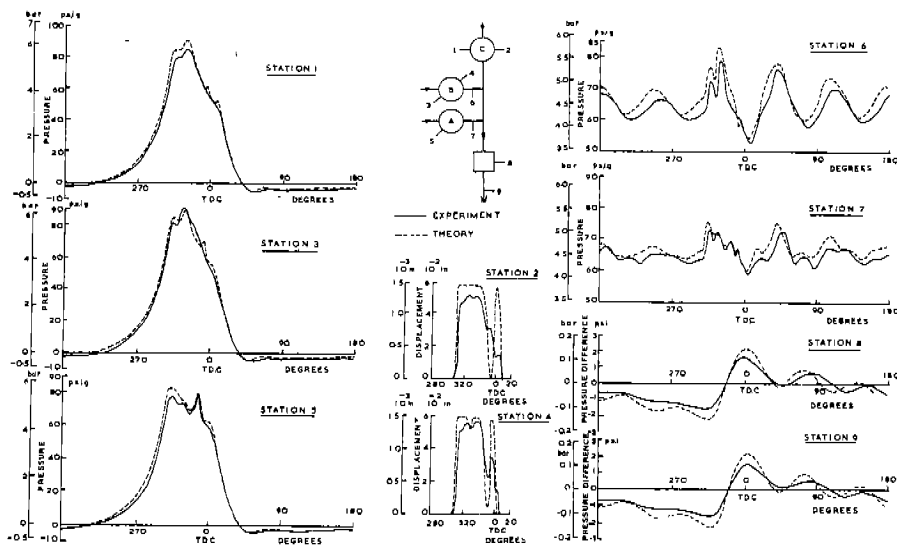


FIG. 10. COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS (TEST D7)
THREE COMPRESSORS IN PHASE 600 r.p.m. RECEIVER VOLUME $4.96 \times 10^{-3} \text{ m}^3$