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Element Interactions and Dynamic Behaviour of Multistage Intercooled Reciprocating Compressors - an Analytical and Experimental Study

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ELEMENT INTERACTIONS AND DYNAMIC BEHAVIOUR OF MULTISTAGE INTERCOOLED RECIPROCATING COMPRESSORS - AN ANALYTICAL AND EXPERIMENTAL STUDY

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ABSTRACT

This paper describes a model which can simulate the dynamic behaviour of a multistage, intercooled reciprocating compressor. A mathematical model was derived for each element of such a system, accounting for entropy changes due to friction or other irreversible processes. Numerical solution of the model was effected on a digital computer using discrete time steps and a quasi-steady approach. Results predicted using the model are compared with corresponding experimental results obtained from a simple two-stage intercooled air compressor.

INTRODUCTION

Some of the problems associated with the presence of pressure pulsations due to the essentially intermittent nature of the flow in a reciprocating compressor system are (a) valve malfunction (flutter or slamming), which may lead to early valve failure and reduced performance, (4): (b) loss or gain in volumetric efficiency due to ramming effects (3), (8): (c) loosening of pipes and fixtures as a result of unbalanced pressure forces, with associated vibration and high noise levels (16). The complex processes that take place during the operation of a compressor made necessary in the past an empirical approach to compressor design. Ready access to computers and advances in numerical and simulation techniques have changed this situation in recent years. Most early investigations followed the pattern of setting up a mathematical model similar to that developed by Costagliola (5) and programming the equations for solution by a digital computer using an iterative method. These models accounted for the interactions between the cylinder and the valves but neglected any pressure variations in the valve plenum chambers. Such investigations were conducted by MacLaren and Kerr (6), Wambsganss and Cohen (17), Touber and Blomsa (15) and others: a review was presented by MacLaren (7) to the 1972 Purdue Compressor Conference. Each investigator claimed that theoretical predictions showed agreement with experimental results over a range of conditions and that the model could be used with some confidence to predict compressor and valve behaviour. However their validity was limited to situations in which pressure pulsations in the suction and discharge plenums were negligible and often this is not so (4), (6).

Consequently more advanced models were developed which accounted for pressure pulsations in the valve plenum chambers and pulsating flow in the pipes. Models of this type were developed by Brablik (3), Blankespoor and Touber (2), Benson and Ucer (1), Schwerzler and Hamilton (14), MacLaren et al (8) and others. The agreement obtained between experimental and analytical results was improved and information became available on the way in which pressure pulsations affect the behaviour of the compressor, the valves and the system generally. A review was presented by Singh and Soedel (12) to the 1974 Purdue Compressor Conference. These models were limited to the simulation of single stage compressors due to certain simplifying assumptions made.

SIMULATION MODEL FOR A MULTISTAGE INTERCOOLED COMPRESSOR

A model was developed to predict the dynamic behaviour of a multistage reciprocating compressor within its associated pipework. To this end aspects which had been neglected in previous models had to be incorporated. In particular: (a) entropy variations to which the gas may be subjected were accounted for and (b) unsteady gas flow in the pipes and plenum chambers was described using one-dimensional finite amplitude theory accounting for friction and heat transfer (non homentropic flow). These advances were necessary because of the limitations imposed by the use of gas equations in two variables of state only (homentropic flow) in which pressure and temperature are not independent variables. In the case of a multistage compressor it is required to predict with accuracy the state of the gas at outlet from a previous stage and to take into account the temperature variation along the intercooler so that the state of the gas at inlet to a stage is accurately known.

DESCRIPTION OF THE SIMULATION MODEL

The model consisted of a set of equations which describe each of the processes that occur during a compressor cycle. (A summary of the equations is given in Appendix A) Wave action in cylinders and receivers was neglected. To account for heat transfer during the compression and re-expansion process in the cylinder, a polytropic law for the change of state of the ideal gas was assumed, the indices being estimated from experimental data. (This approach proved easier to apply than the alternative of using a heat transfer correlation in the cylinder which required estimation of cylinder wall temperature and certain empirical coefficients.) Compressor valves were treated as single degree of freedom damped spring-mass systems, with damping assumed to be proportional to valve speed and average gas density. The force on the valve plate was assumed to be a function of the pressure difference across the valve and valve lift. An "effective force area" was determined experimentally from steady flow tests. (The analytical procedure suggested by Schwerzler and Hamilton (13) was also used to estimate the values of these coefficients, however results obtained were not quite satisfactory.) The differential equation which described valve plate motion was integrated using the Kutta Merson numerical integration method.

The unsteady non-homentropic flow (assumed one-dimensional) in the pipes was described by a set of hyperbolic differential equations which account for heat transfer, friction and gradual change of cross sectional area. The equations in characteristic form were solved by the method of upstream differences, similar to that used by Benson (1). (Other methods of solution were investigated with good results and are reported in another paper (9) to this Conference.) The mean pipe wall temperature and heat transfer rate at each station along the discharge and intercooler pipes was estimated using a steady flow heat transfer model. This model was based on the average flow conditions and individual heat transfer modes present at a given pipe section.

The boundary equations relate the conditions found at the pipe ends to those existing in adjacent items (cylinders, receivers). The solution of these equations defines the values of the variables at the end of the pipe and the amount of mass entering or leaving the cylinder or receiver. These boundaries were at pipe ends which were fully open, partially open or closed (closed compressor valve). The solution of the boundary equations was based on the assumption that quasi-steady flow existed at the boundary.

Simulation of the operation of the compressor consisted of the solution of a number of initial value/boundary condition problems, one for each element of the compressor system. Thus, knowing the values of all variables at time t and the interactions which take place across the boundaries of each element during a time interval Δt, the values of all variables at time $t + \Delta t$ can be uniquely determined. Successive application of this process allowed the study of the sequence of events which occur during a compressor cycle. When starting the analysis it was necessary to select arbitrary initial conditions at a point in the cycle and continue the computation through a number of cycles. Sufficient convergence to an approximately repeatable cycle was obtained at the end of the third or fourth cycle of computation.

The computer program for the simulation model was intended to be as general as possible. Each system element (cylinder, receiver, pipe, valve, etc.) was programmed as a separate module. A control program input the data which described the configuration of the particular system being studied and then called the appropriate sequence of subroutines. Logic subrountines continually monitored the conditions at pipe ends in order to solve the corresponding boundary condition equations. A flow chart for the compressor simulation program is given in Figure 1.

SIMULATION OF A TWO STAGE INTERCOOLED COMPRESSOR

The model was first tested on each stage singly of a two-stage water cooled air compressor (1st stage, 6 in bore x 4.5 in stroke; 2nd stage, 3.25 in bore x 3.5 in stroke; speed range 350 - 700 rev/min). Tests were conducted over a wide range of speed, compressor pressure ratio and inlet and discharge pipe lengths. Results obtained for the first stage working as a single stage compressor were reported (8) to the 1974 Purdue Compressor Conference. The two stage compressor (Figure 2) is now examined. The system had a simple geometry, avoiding flow complexities due to acute pipe bends, pipe junctions or sudden large changes in cross sectional area. The experimental intercooler was a 21 ft length of 1.375 in internal diameter pipe, having large radius bends and fitted externally with a water jacket. No dampers were included, the only damping which had to be accounted for was that due to friction in the pipes.

Results for the simulation of the two stage intercooled compressor are shown in Figure 3 which illustrates one test in the series conducted. The cranks in the two stages were displaced by 180°, i.e. L.P. top dead centre (0° on the L.P. diagrams) and H.P. bottom dead centre (180° on the H.P. diagrams) occur at the same instant of time. In the two diagrams shown at the top of Figure 3 the predicted pressures in the suction plenum, discharge plenum and cylinder are shown to the same scale. Valve plate displacement diagrams are included. The lower diagrams in Figure 3 show the same predicted pressure/time histories to a larger scale with the experimental records superimposed. Correlation between experimental and analytical results was considered to be satisfactory. However, the experimental methods used (11) have since been developed further (10).

EFFECT OF UNSTEADY FLOW ON COMPRESSOR AND VALVE PERFORMANCE

The study (8) of the first stage acting as a single stage compressor had shown the effect that pressure pulsations in the inlet pipe may have on the volumetric efficiency of the compressor, causing induction ramming and anti-ramming. A particular aspect of the present study was the effect that pressure pulsations in the intercooler have on the behaviour of the compressor valves. From the traces for the discharge valve plenum pressure of the L.P. stage and the suction valve plenum pressure of the H.P. stage in Figure 3, the following may be observed. The compression pulse generated by the L.P. stage during the discharge process at about 300 crankangle degrees after T.D.C. (120^{o} for the H.P. stage) arrived at the suction plenum chamber of the H.P. stage when the piston was near B.D.C. and the valve was starting to close, forcing the suction valve to open again and ramming more air into the cylinder. On the other hand, the rarefaction pulse generated at the start of the suction process in the H.P. stage is reflected back (again, as a rarefaction wave), arriving at the suction plenum chamber approximately 20^{o} before B.D.C. and momentarily closing the suction valve. A strong pattern of pulsations is being reinforced in this case because the rarefaction and compression pulses are occurring in rapid succession.

The marked effect that interstage pulsations have on the behaviour of the H.P. suction valve is illustrated in Figure 4. The dotted lines correspond to the results predicted when the H.P. stage compressor was simulated under exactly the same operating conditions as those in Figure 3 but without an inlet pipe (constant plenum chamber pressure). Both the early closing of the valve due to the arrival of the rarefaction pulse and the second opening due to the compression pulse are now absent.

The gas column in the interstage pipe is subjected to two forcing pulsations: a compression pulse generated during the discharge of the L.P. stage and a rarefaction pulse generated during the suction process in the H.P. stage. The effect of the pulsations on the behaviour of the H.P. stage suction valve (the L.P. stage discharge valve is less sensitive to these) will depend on their amplitude and their phasing with respect to the suction process in the H.P. stage. It was found that this interaction can be expressed in terms of the phasing between crankshafts and a non-dimensional pipe length ϕ. (This non-dimensional pipe length was defined as the ratio of the pipe length to the gas column length which would resonate in its fundamental mode at the frequency of the compressor cycle. That is $\phi = NL/30\underline{a}$ for a closed/closed pipe; $\underline{a}$ being the mean speed of sound in the pipe.) For the case in which the cranks are displaced by 180^{o} a particularly strong pattern of pulsations will exist in the intercooler pipe for: (a) the third order pulsation of the findamental mode ($\phi = \frac{1}{3}$), corresponding to a situation like that shown in Figure 3 and (b) the first order pulsation of the fundamental ($\phi = 1$), corresponding to a situation similar to that for anti-ramming in a single stage compressor. A peak in volumetric efficiency of 92% was measured when the conditions for induction ramming occurred in both the L.P. and H.P. stages.

CONCLUSIONS

Multistage reciprocating gas compressors and their associated pipework can be adequately simulated using a digital computer model. Results predicted by the simulation model described here compared satisfactorily with those obtained experimentally from a simple two-stage intercooled air compressor. The model was capable of simulating the processes that occur during the operation of the compressor and the interactions between the various elements of the system. The model provides a powerful design aid which can be used inter alia to reduce the experimental programme required during the development of a new or modified design.

REFERENCES

1. BENSON, R.S. and UCER, A.S. "Some Recent Research in Gas Dynamic Modelling of Multiple Single Stage Reciprocating Compressor Systems". Proc. 1972 Compressor Technology Conference, Purdue University, p. 491

2. BLANKESPOOR, H.J. and TOUBER, S., "Computer Simulation of a One Cylinder Reciprocating Compressor using a Hybrid Computer", Proc. 1972 Compressor Technology Conference, Purdue University, p. 506.

3. BRABLIK, J., "Gas Pulsations as Factors Affecting Operation of Automatic Valves in Reciprocating Compressors", Proc. 1972 Compressor Technology Conference, Purdue University, p. 188

4. CARPENTER, A.B., "The Use of Analog Simulator Studies to Reduce or Eliminate Objectionable Pulsations in Reciprocating Compressors and Associated Piping", A.S.M.E. Paper No. 67-Pet-30.

5. COSTAGLIOLA, M., "The Theory of Spring Loaded Valves for Reciprocating Compressors", J.Appl.Mech., 17 (4), 1950, p. 415

6. 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, 1970

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

8. MACLAREN, J.F.T., KERR, S.V., TRAMSCHEK, A.B. and SANJINES, A., "A Model of a Single Stage Reciprocating Gas Compressor Accounting for Flow Pulsations", Proc. 1974 Compressor Technology Conference, Purdue University, p. 144.

9. MACLAREN, J.F.T., TRAMSCHEK, A.B., PASTRANA, O.F., SANJINES, A., "Advances in Numerical Methods to Solve the Equations Governing Unsteady Gas Flow in Reciprocating Compressor Systems", Proc. 1976, Purdue Compressor Technology Conference.

10. MACLAREN, J.F.T., KERR, S.V., CRAWFORD, R.A. HOARE, R.G., "A Computer Controlled System for the Acquisition and Processing of Experimental Data from Reciprocating Compressors", Proc. 1976 Compressor Technology Conference, Purdue University.

11. SANJINES, A., "An Analytical and Experimental Study of a Two-stage Compressor Installation ", Ph.D. Thesis, University of Strathclyde, 1975

12. SINGH, R. and SOEDEL, W., "A Review of Compressor Lines Pulsation Analysis and Muffler Design Research", Proc. 1974 Compressor Technology Conference, Purdue University, p. 102

13. SCHWERZLER, D.D. and HAMILTON, J.F., "An Analytical Method for Determining Effective Flow and Force Areas for Refrigeration Compressor Valving Systems", Proc. 1972 Compressor Technology Conference, Purdue University, p. 30

14. SCHWERZLER, D.D. and HAMILTON, J.F., "Simulation and Design Studies of a Multiple Cylinder Reciprocating Compressor", Proc. 1974 Compressor Technology Conference, Purdue University, p.5

15. TOUBER, S. and BLOMSA, E.C. "Theoretical and Experimental Investigation of Valve Movement and Instationary Gas Flow in a Reciprocating Compressor" Proc. XIII Int.Cong.Refrig., Washington, Paper No. 3,14, 1971

16. VON NIMITZ, W.W. "Reliability and Performance Assurance in the Design of Reciprocating Compressor Installations", Proc. 1974 Compressor Technology Conference, Purdue University, p. 329

17. WAMBSGANSS, M.W. and COHEN, R., "Dynamics of a Reciprocating Compressor with Reed Valves", Paper No.3.06, Proc.XIII Int.Cong.Refrig., Madrid, 1967.

NOTATION

- A Non-dimensional speed of sound (a/a_{ref})
- A_a Non-dimensional speed of sound after isentropic change to reference pressure p_{ref}
- $\bar{B}$ Vector of non-homogeneous terms
- $C_{1,2}$ Pseudo Riemann variables $A \pm \frac{k-1}{2} U$
- C_{ff} Flow drag coefficient
- C_p Specific heat at constrant pressure
- D Pipe diameter
- f Friction factor
- F Friction term = $2f x_{ref}/D$
- $G(\bar{V})$ Vector function of $\bar{V}$
- k Ratio of specific heats
- K_s Spring stiffness
- m Mass
- M_v Effective valve plate mass
- P Non-dimensional pressure (p/p_{ref})
- $\dot{q}$ Heat transfer rate per unit mass
- q Thermal energy
- Q Heat transfer term = ($\dot{q}\, x_{ref}/a^3_{ref}$)
- R Non-dimensional density (ρ / ρ_{ref})
- S Area
- t Time
- U Non-dimensional particle velocity (u/a_{ref})
- V Cylinder volume
- $\bar{V}$ Vector of dependent variables
- X Non-dimensional length (x/x_{ref})
- Y Non-dimensional valve displacement (y/y_{max})
- Z Non-dimensional time ($t . x_{ref}/a_{ref}$)
- α_D Damping coefficient
- ϕ Effective throat/pipe flow area

Subscripts

- g Guess
- in Into the boundary
- j Subindex denotes space
- n Subindex denotes time
- o Stagnation
- out Out of the boundary
- p Pipe
- ref Reference
- t Throat
- v Volume (cylinder, receiver or atmosphere)
- w Intersection of path line with constant time line

APPENDIX A - Equations used in the Model

Only the final form of the equations employed in the simulation is given here. The derivation of the equations and the methods of solution used are given elsewhere (9, 11).

1. Unsteady, non-homentropic flow in constant cross-sectional area pipes

1.1 Equations in conservation-law form $\frac{\partial V}{\partial Z} + \frac{\partial G(V)}{\partial X} = B$

$$\frac{\partial}{\partial Z}\begin{bmatrix} R \\ RU \\ \frac{RU^2}{2} + \frac{P}{k(k-1)} \end{bmatrix} + \frac{\partial}{\partial X}\begin{bmatrix} RU \\ RU^2 + P/k \\ \frac{RU^3}{2} + \frac{UP}{k-1} \end{bmatrix} = \begin{bmatrix} 0 \\ -RFU\,|U| \\ RQ \end{bmatrix} \qquad \ldots (1)$$

This form was used to solve the internal meshes in the pipes in conjunction with the Two-Step Lax-Wendroff method.

First Step: $$V_{j+\frac{1}{2}}^{n+\frac{1}{2}} = \tfrac{1}{2}(V_{j+1}^n + V_j^n) - \frac{\Delta t}{2\Delta x}(G(V)_{j+1}^n - G(V)_j^n) + \frac{\Delta t}{4}(B_{j+1}^n + B_j^n) \qquad \ldots (2)$$

Second Step: $$V_j^{n+1} = V_j^n - \frac{\Delta t}{\Delta x}(G(V)_{j+\frac{1}{2}}^{n+\frac{1}{2}} - G(V)_{j-\frac{1}{2}}^{n+\frac{1}{2}}) + \frac{\Delta t}{2}(B_{j+\frac{1}{2}}^{n+\frac{1}{2}} + B_{j-\frac{1}{2}}^{n+\frac{1}{2}}) \qquad \ldots$$

1.2 Equations in characteristic form; this form may be used to obtain solutions of the equations at the internal pipe meshes and was used at pipe boundaries in a composite scheme with the Lax-Wendroff scheme.

(i) Along the Mach lines $\frac{dX}{dZ} = U \pm A$

$$dC_{1,2} = A\frac{dA_a}{A_a} + \frac{(k-1)^2}{2A}Q\,dZ \mp \tfrac{1}{2}(k-1)FU|U|(1 \mp (k-1)\frac{U}{A})dZ \qquad \ldots (3)$$

(ii) Along the path lines $\frac{dX}{dZ} = U$

$$dA_a = \frac{k-1}{2}\frac{A_a}{A^2}\cdot(Q - FU|U|)\,dZ \qquad \ldots (4)$$

1.3 Stability Condition Both the Two-Step Lax-Wendroff scheme and the Mesh version of the Method of Characteristics employ a stability criterion, viz.

$$\frac{dZ}{dX} \le \frac{1}{|U| + A} \qquad \ldots (5)$$

2. Boundary Conditions

The determination of the conditions at the end of a pipe i.e. at the junction of a pipe and its adjacent volume (cylinder, receiver, or atmosphere) reduces to the problem of finding the value of the outgoing characteristic and entropy at the end of the pipe.

2.1 Flow out of the pipe (C_{out} unknown)

(i) Subsonic flow at the throat

$$C_{out} = 2A_A\,p\left(\frac{A_v}{A_{av}}\right) \cdot A_p^{*} - C_{in} \qquad \ldots (6)$$

where $A_p^* = A_p/A_t$ is found by iteration from

$$\frac{C_{in}}{A_{ap}}\cdot\frac{A_{av}}{A_v} = A_p^* + \Phi\sqrt{\frac{k-1}{2}\;\frac{A_p^{*2} - 1}{A_p^{*\,4/k-1} - \Phi^2}} \qquad \ldots (7)$$

(ii) Sonic flow at the throat

$$C_{out} = \left(\frac{1 - \Phi\frac{k-1}{2}A_p^{*\,k+1/k-1}}{1 + \Phi\frac{k-1}{2}A_p^{*\,k-1/k-1}}\right)\cdot C_{in} \qquad \ldots (8)$$

where $A_p^* = A_p/A_t$ is found by iteration from

$$\Phi = A_p^{*\,2/k-1}\sqrt{\frac{k+1}{k-1} - \frac{2}{k-1}\cdot A_p^{*2}} \qquad \ldots (9)$$

2.2 Flow into the pipe (C'_{in}, C_{out}, A_{ap} unknown)

$$C_{out} = C'_{in}\cdot\left(\frac{3-k}{k-1}\right) + \left(\frac{2A_v}{k+1}\right)\sqrt{(k-1)(k+1-2\left(\frac{C'_{in}}{A_v}\right)^2)} \qquad \ldots (10)$$

where $$C'_{in} = C_{in} + \frac{A_w}{A_{aw}}\cdot(A_{ap} - A_{ag}) \qquad \ldots (11)$$

and A_{ap} is given by:

$$A_{ap} = A_{av}\cdot\left(\frac{A_p}{A_t}\right)\left[\Phi\cdot\left(\frac{U_t}{U_p}\right)\cdot\left(\frac{A_p}{A_t}\right)^2\right]^{(1-k)/2k} \qquad \ldots (12)$$

(i) For subsonic flow at the throat, U_t is found from the roots of the equation

$$U_t^{2*}\left(k\Phi - \frac{k-1}{2}\right) - U_t^*\left[\frac{\Phi}{U_p^*}\left(1 - \frac{k+1}{2}U_p^{*2}\right)\right] + 1 = 0 \qquad \ldots (13)$$

where $U_t^* = U_t/A_v$, $U_p^* = U_p/A_v$; A_t is found from the energy equation, viz.

$$A_{vo}^2 = A_t^2 + \frac{k-1}{2}U_t^2 \qquad \ldots (14)$$

(ii) For sonic flow at the throat: $$\frac{U_t}{A_v} = \frac{A_t}{A_v} = \left(\frac{2}{k+1}\right)^{\frac{1}{2}} \qquad \ldots (15)$$

2.3 Mass flow at pipe boundary (in flow or out flow)

$$\dot{m} = \frac{k\,p_{ref}}{a_{ref}}(S_p)\cdot\left(\frac{U_p}{A_p}\right)\cdot\left(\frac{1}{A_p}\right)\cdot\left(\frac{A_p}{A_{ap}}\right)^{2k/k-1} \qquad \ldots (16)$$

3. Updating Conditions in Cylinders or Receivers

By considering the equations for the conservation of mass and energy applied to an arbitrary control volume together with the equation of state of an ideal gas the following equations may be established for the evaluation of changes of state variables within the control volume

$$\frac{dP_v}{P_v} = k\left[\frac{dq}{C_p(mT)_v} - \frac{dV_v}{V_v} + \Sigma\frac{dm_{in}}{m_v}\left(\frac{a_{oin}}{a_v}\right)^2 + \Sigma\frac{dm_{out}}{m_v}\right] \qquad \ldots (17)$$

$$\frac{dA_v}{A_v} = \frac{1}{2}\left[\frac{dP_v}{P_v} + \frac{dV_v}{V_v} - \frac{dm_v}{m_v}\right] \qquad \ldots (18)$$

$$\frac{dA_{av}}{A_{av}} = \frac{dA_v}{A_v} - \frac{k-1}{2k}\frac{dP_v}{P_v} \qquad \ldots (19)$$

For a receiver $dV_v = 0$ and the heat transfer term $\frac{dq}{C_p(mT)_v}$ was ignored. For a cylinder, it was assumed that only inflow or outflow could occur at one time. When applying these equations the mass flow rates into or out of the control volume were assumed to be constant over a given time increment and the calculations employed mass flow rates determined at the end of the previous time step.

4. Valves

Dynamic Equation: $$\ddot{y} = \frac{C_{ff}S_v\,\Delta P}{M_v\,y_{max}} + 2\alpha_D\sqrt{\frac{k_s}{M_v}}\,\dot{y} + \frac{k_s}{M_v}(y - y_o) \qquad \ldots (20)$$

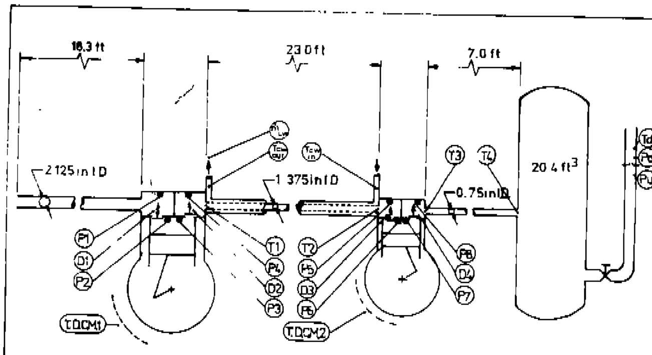

FIG. 2 SYSTEM CONFIGURATION FOR COMBINED COMPRESSORS (2 STAGE-INTERCOOLED) - TESTS G -

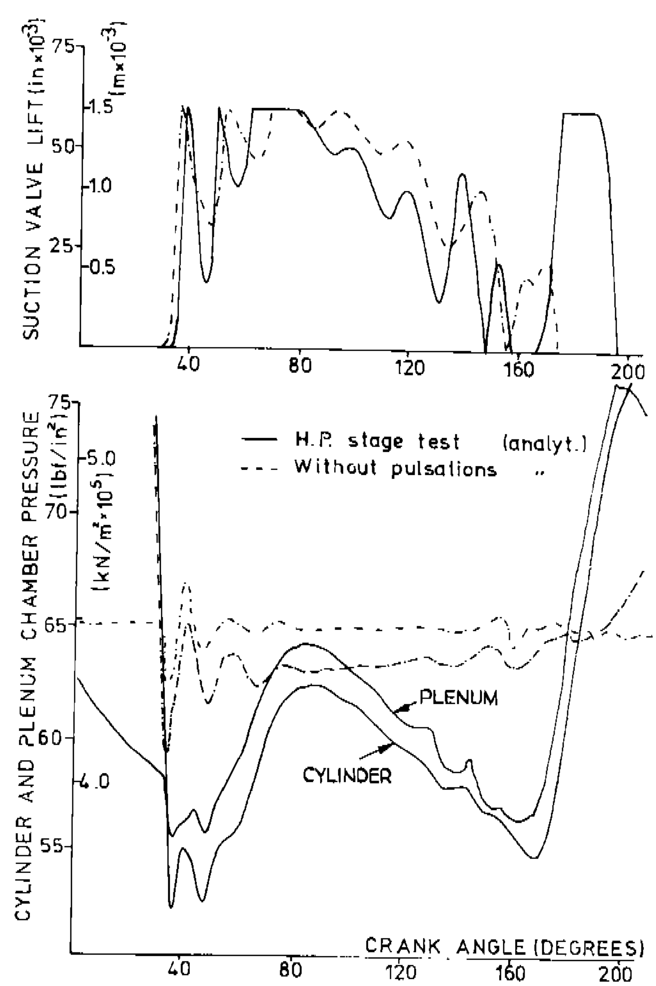

FIG. 4. EFFECT OF INTERSTAGE PULSATIONS ON THE BEHAVIOUR OF THE H.P. SUCTION VALVE

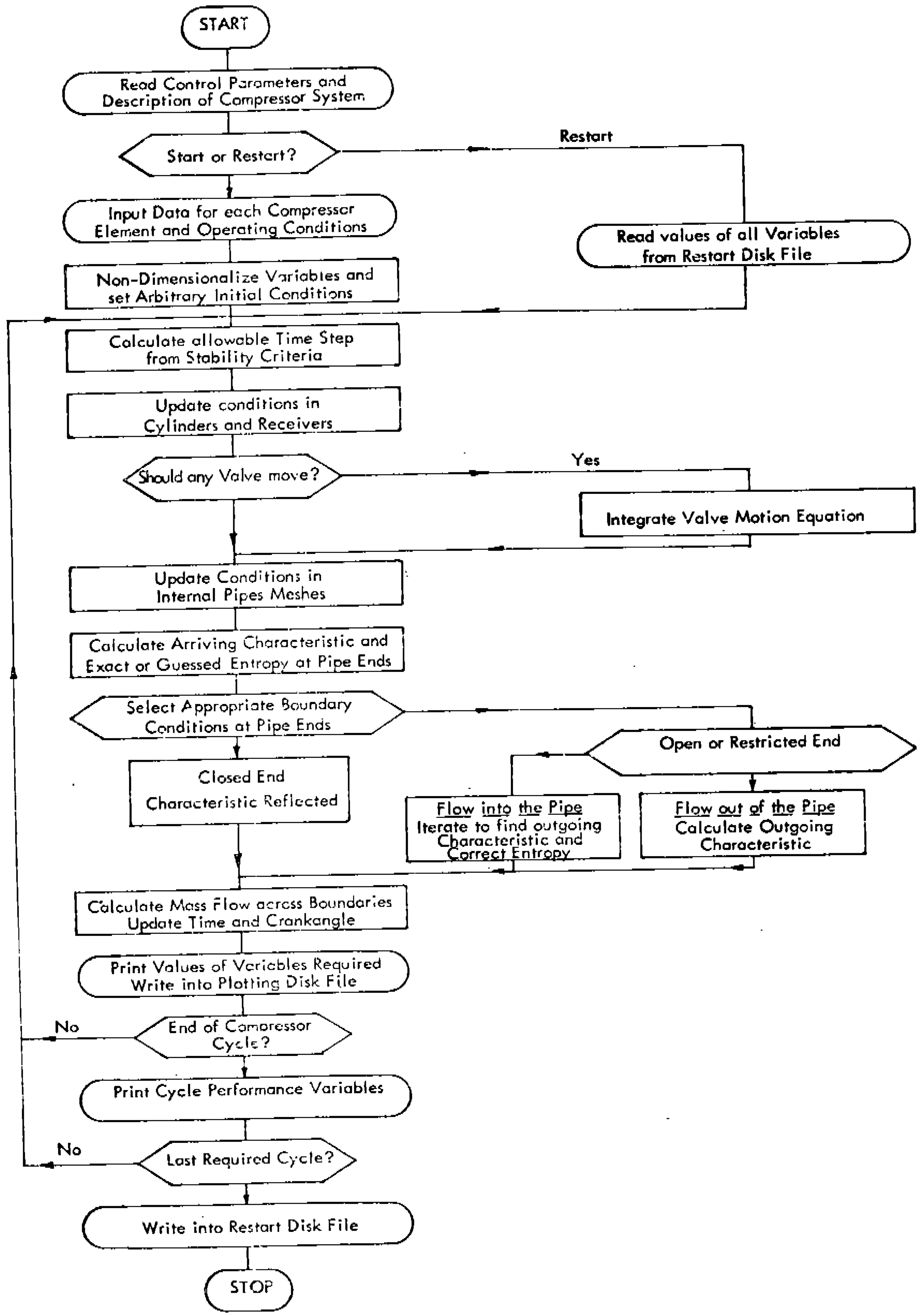

FIG 1 FLOW DIAGRAM OF COMPRESSOR SIMULATION PROGRAM

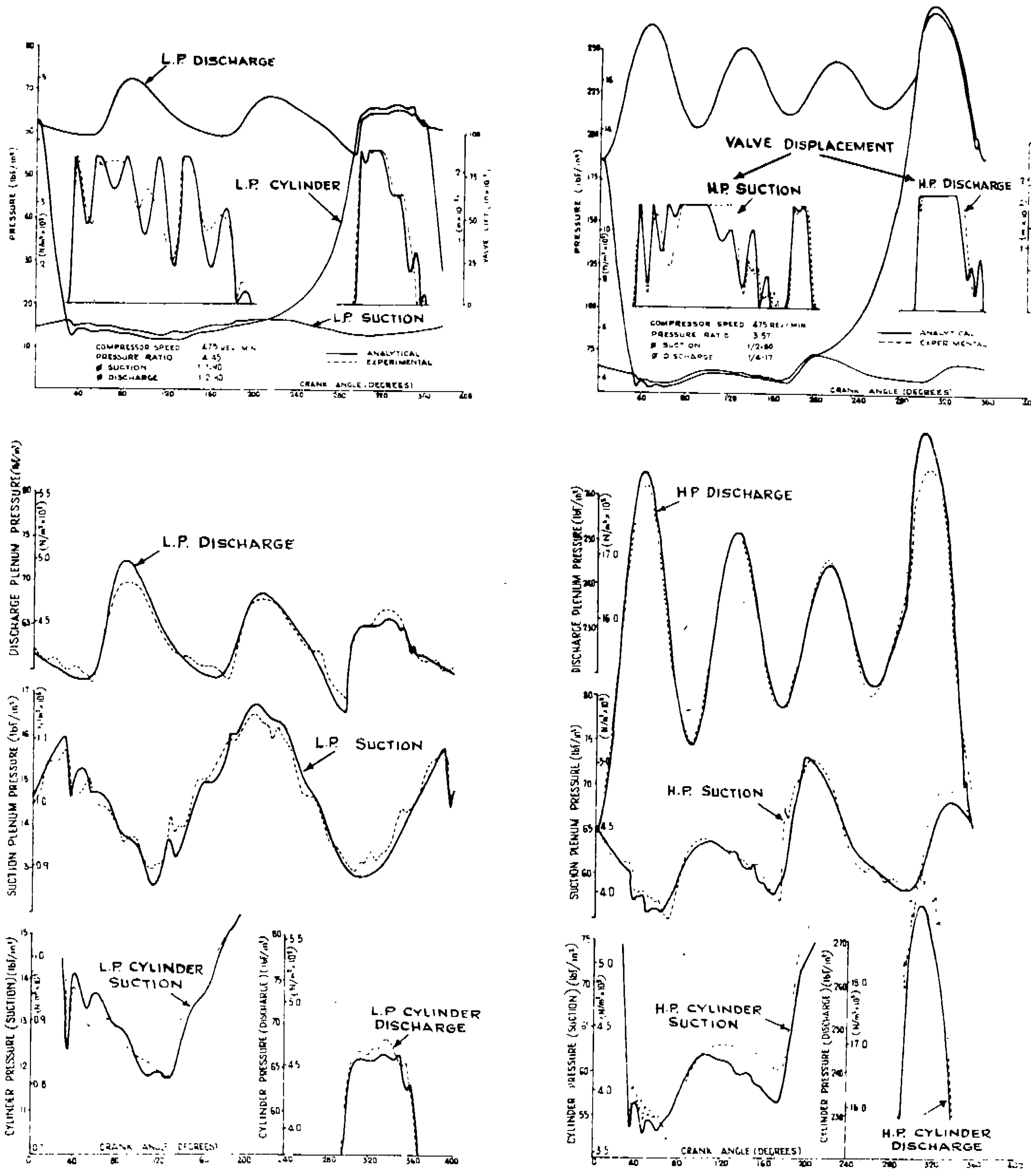

FIG. 3 COMPARISON OF EXPERIMENTAL AND ANALYTICAL RECORDS FOR A TWO STAGE INTERCOOLED AIR COMPRESSOR.