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International Compressor Engineering Conference

School of Mechanical Engineering

1972

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PREDICTION OF VALVE BEHAVIOUR WITH PULSATING FLOW IN RECIPROCATING COMPRESSORS

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INTRODUCTION

Reciprocating gas compressors have evolved over the years from simple low-speed machines to highly developed high-speed machines. Efforts to increase output and to reduce size and cost have involved increasing crankshaft and piston speeds, while seeking to retain acceptable standards of performance, reliability, life, and, for some applications, noise level. A survey over a range of power input to compressors and power ouput of oil engines shows that, in general, the mean piston speed of compressors is much less than that of oil engines. It may be deduced that the automatic valves, used almost universally in compressors for reasons of simplicity and low cost, constitute a greater limitation on throughput of the working fluid than do the mechanically operated valves of an engine.

Compressor valves have been the subject of much study, especially as they are relatively delicate components with a reliability less than that of most other parts of the compressor. They operate automatically in a complex physical situation and intuitive processes are usually inadequate to assess the dominating parameters responsible for valve behaviour in a particular system.

MODELS WHICH NEGLECT FLOW PULSATIONS

In recent years an element of science has been added to the traditional empirical art of valve design by the introduction of relatively simple mathematical models to describe a compressor, its valves, operating conditions and the behaviour of the working fluid. The non-linearity of the differential equations which form the basis of most models, and the large number of variables involved, made it necessary to program such models for a digital computer. Almost all models developed during the last twenty years have assumed, together with other simplifying assumptions, that the pressure in the suction and discharge plenum chambers remained constant during the compressor cycle. In a survey of these models MacLaren (1) stated that much useful design information could be provided for valves in systems in which pressure pulsations were not significant.

ANALOG SIMULATION

It is well known, however, that in many systems there are pressure pulsations of considerable amplitude. (For example, Carpenter (2) quoted a case of resonant interstage conditions where a pulse amplitude of 330 lbf/in^2 was recorded in a line at a mean pressure of 660 lbf/in^2). The manifestation of the phenomena is vibration and noise, failure of pipework and fixtures, loosening of fastenings, over-loading of the compressor and failure of valves. Efforts to reduce pulsation phenomena at the design study stage have led to the use of analog system simulators to assess the effectiveness of proposed pulsation damping arrangements. In this connection the work of the Southern Gas Association, Pipeline and Compressor Research Council, U.S.A. is well known. In such analog studies the simulation of the compressor valves has usually been neglected or greatly simplified. A major simplification has been that the valve flow area was constant at that corresponding to maximum permitted lift. Brunner (3) simulated the valve by a non-linear resistor and an inductor in series in order to simulate the flow area as a function of valve pressure drop. With this, still simple assumption, it was demonstrated that the valve was an important part of the total system and its behaviour should not be neglected.

Usually, analog studies have been directed towards the reduction of pulsations at the line side of the damper. The primary function of dampers, snubbers or mufflers is to decouple the low kinetic energy sub-system (the pipework) from the high kinetic energy sub-system (the compressor). However, after doing so, the compressor valve "sees" a short length of connecting pipe between the valve plenum chamber and the damper. In this finite length there might be a pulsation spectrum with harmonics coinciding with the harmonics of the valve. The possibility exists, therefore, that a damper could reduce pipeline pulsations but aggravate valve malfunction.

EFFECT OF FLOW PULSATIONS ON VALVE BEHAVIOUR

According to Czaplinski (5), pulsations in the discharge line of a compressor are always harmful: pulsations in the suction line may cause supercharging. An early study of this "ramming" effect was made by Bannister (6) who demonstrated that supercharging could improve volumetric efficiency by about 15%. The same value was reported by Czaplinski (5) and was also obtained by the authors in a similar investigation. Even higher values are quoted elsewhere. However, it is more usual to "de-tune" systems. In large high pressure systems de-tuning is desirable to avoid the large pulsation amplitudes associated with resonant conditions which may lead to valve and pipeline failure. Smaller refrigerating plants are de-tuned to reduce noise levels. The studies reviewed by Czaplinski (5) , by Bannister (.6) and studies by Jaspers (7) and Wallace (8) concerned the effects on compressor capacity of pulsations and the effectiveness of damping arrangements and did not consider the effect on valve behaviour or the interaction between the pulsations and the valve movement.

Bauer (4), Fig. 1, showed the marked effect on the displacement of a discharge valve by variation of the length of the delivery pipe. This demonstrates that there would be significant inaccuracies if the valve movement was predicted by the simple analytical models reviewed by MacLaren (1), which neglect the effect of pipeline pulsations. Parry (4) showed that a marked change in the valve displacement diagram occurred when a discharge manifold was modified.

MacLaren and Kerr (9) extended a simple model, in which pipeline pulsations were neglected, to account for the effect on valve behaviour of pressure variation in a suction or discharge plenum chamber. The pressure-time history was not predicted analytically but was known from experimental observation. Fig. 2 (top) shows an experimental displacement diagram and an analytical diagram predicted by the simple model, for one of the four ($8\frac{1}{2}$ in diameter, Hoerbiger type) suction valves in the double-acting third stage of a five stage 3000 hp, 300 rev/min, air compressor. Both experiment and analysis indicated that the valve experienced a second opening towards the end of suction, thus subjecting the valve springs to an excessive number of cycles. The simple model was used to study the effect of some modifications which could have been implemented cheaply, e.g. change of valve lift, spring stiffness and spring preloading. The results showed that changes in these variables would not significantly modify the second unwanted opening of the valve. The model was thus helpful: it indicated that the expense of field trials with any of these simple modifications would not be justified. It also suggested that the second opening would be present whether or not significant pressure pulsations occurred in the plenum chamber. The experimental valve displacement diagram in Fig. 2 showed evidence of a valve "dwell" and a delay in final closure of 19° after piston reversal (20° after the point of closure predicted by the simple model). An experimental record, Fig. 2 (bottom) was obtained of the pressure at the suction plenum chamber. This was idealized as shown and a sinusoidal pulse, amplitude 11.5 % of the mean pressure, was superimposed on the constant line pressure assumed in the simple analytical model. The results obtained, Fig. 3, indicated that the third partial closing and reopening of the valve and the late closure could be accounted for.

The pressure difference across a suction valve is usually small compared to that for a discharge valve so during closure a suction valve in particular is in an unstable condition. This is demonstrated in Fig. 3, which shows that the movement of the suction valve during closure was very sensitive to the phasing of the pulse. It was found that changes in pulse amplitude (not shown) had much less effect.

MODELS WHICH INCLUDE ANALYSIS OF FLOW PULSATIONS

To date there have been two investigations, known to the authors ,in which models have beendeveloped to describe the compressor, its valves, the working fluid, the operating conditions and the pulsations inherent in the intermittent flow. These have been by Benson and Ucer (10) at the University of Manchester Institute of Science and Technology and by Brablik (11) at the C.K.D. Compressor Factory, Prague. Papers by Benson and Ucer (10a) and by Brablik (11d) are presented at this Conference.

Benson and several co-workers had previously developed extensive computer programs for the analysis of the flow in internal combustion engine systems. Benson and Ucer applied these analyses to study flow in compressor intake and discharge systems and included the valves among the many complex boundary conditions to be considered. Three air-cooled compressors were linked in a number of systems and studied analytically and experimentally at a number of compressor speeds.

Brablik (11a) presented a method of analysis for complex piping systems in large multi-cylinder compressor installations.The analytical predictions of pressure-time histories for a number of systems showed good agreement with experimental results, particularly at higher pulsation levels. These investigations were prompted by pipe line failures and the effect of pulsations on valve behaviour was not considered. It was stated, however, that "the mutual dependence (of valve behaviour and pulsating flow) has been fully confirmed by experiment".

In a thesis (11c) Brablik showed, Fig. 4, the computed closing phase of a 106 mm diameter spring loaded ring plate suction valve. The compressor (180 mm bore x 450 mm stroke) operated at 300 rev/min and pumped nitrogen at a suction pressure of 184 atmospheres. The difference in valve displacement due to change of inlet pipe length is evident. Other diagrams from this thesis (11c) were republished by Brablik in a more readily available paper (11b) and are reproduced in Fig. 5. These diagrams relate to a 134 mm diameter suction valve drawing air at a pressure of 10 kp/cm^2 (142 lbf/in^2) (i) with no inlet pipe (ii) with inlet pipe 1 metre long, and show the change with time (crankangle) of valve displacement, valve velocity, pressure drop and gas mass flow rate. With the inlet pipe fitted the amplitude of the pressure pulsations increased, particularly near valve closure. The pressure pulsations were sufficiently large to cause an inversion of pressure difference across the valve (marked on Fig. 5), near to piston reversal where the pressure difference across the valve was small. Hence a reversal of the direction of mass flow i.e. "blow-by" occurred which would cause a decrease in compressor throughput (reduced volumetric efficiency).

The final, late, point of closure of the suction valve, Fig. 5b, is not shown but the velocity of impact increased each time the valve struck the seat and was greater than the impact velocity at closure when no inlet pipe was in place. That is, pipe-line pulsations were causing valve "slamming". By reducing the permitted valve lift the pressure difference across a valve would be increased and the effect of pulsations reduced - a remedy commonly used to combat valve failure due to excessive slamming.

Cases have been reported where failure occurred in only some of the valves of identical cylinders operating in parallel, due to the effect on pulsations of the different lengths of pipes connecting cylinders and receiver.

Since there is usually a greater pressure difference across a discharge valve than across a suction valve it might be considered that reversal of flow through a discharge valve was less likely. However, pressure pulsations in the discharge line are likely to be more severe because the discharge valve opens more rapidly due to the greater rate of change of pressure difference across it. Hence discharge valve displacement may also be affected by pressure pulsations and the possibility exists of inversion of pressure difference and of mass flow. Any such discharge valve "blow-by" gas would re-expand with the clearance gas, delay the point of opening of the suction valve, and cause a decrease in compressor throughput.

Since pressure pulsations in the line affect valve behaviour, they affect compressor throughput, even without inversion of pressure difference occurring. The cylinder pressure, greater or less than the line (passage) pressure by the pressure difference across the valve, follows the line pressure in form. The possible arrangements are illustrated in Fig. 6: the bottom right-hand diagram illustrates the situation when induction "ramming" occurs.

Figs. 1, 3, 4 and 5 show experimental and analytical results which demonstrate that pressure pulsations have some effect on valve behaviour. However, there is a mutual interaction and a valve, in turn may stimulate pressure pulsations. Due to the intermittent nature of the flow there is a basic low frequency pressure oscillation in the line (passage) as indicated in Fig. 7. Ideally the valve should begin to open at point c but valve spring preloading, valve inertia, and oil stiction at the seat may delay the opening until the cylinder pressure has reached point a. The vertical distance between a and b represents the pressure difference across the valve at the actual commencement of valve opening. With the valve open the high frequency pulsation (full line) is superimposed on the basic lower frequency pressure oscillation (which if continued would have the form shown by the dotted line). The resulting pulsation spectrum in the passage may contain harmonics which coincide with frequencies of the valve movement and may lead also to reversed flow as illustrated in Fig. 5.

In his analytical investigations Brablik (11c) utilises a modified acoustic wave approach to include frictional effects and in many ways his method resembles the techniques developed by Benson and Ucer (10).

When developing models it is natural to study first relatively simple systems, e.g. a suction valve with an open ended pipe at inlet or a discharge valve with a pipe connection to a large receiver (or a still simpler boundary problem, a pipe connected through a nozzle to atmosphere). Fig. 8 shows the results of computations by the authors, using the method of characteristics, to illustrate the effect of different lengths of inlet piping on the behaviour of a single ring, spring loaded valve plate in a single stage single acting air compressor (6 in bore x 4.5 in stroke) operating at 400 rev/min. A large number of "flutters" was obtained and corresponded closely to the corresponding experimental diagrams (not shown). The marked effect on valve behaviour of pulsation patterns due to varying pipe length was demonstrated. In these computations initial starting conditions had to be assumed but it was found that the solution converged after only two or three compressor cycles. Fig. 9 shows the suction and discharge valve displacement for the compressor (with different spring characteristics and flow coefficients than in Fig. 8) with an inlet pipe length 18.25 ft and discharge pipe 13.5 ft. The mutual interaction between pressure pulsations and valve displacement can be observed, and again demonstrates the need to account for such pulsations in a general mathematical model which purports to describe valve behaviour.

CONCLUSIONS

In conclusion, more complete analytical models are required to describe fully a compressor, its valves, the working fluid, the operating conditions, and the complex pulsations inherent in the intermittent flow. The development of such models is a major undertaking and it may be regretted that the lack of cooperation in research results in much duplication of effort. However, the problem is very complex: many relevant variables have to be accounted for, their importance in each particular design assessed, and many possible boundary conditions have to be considered. There is ample scope for many investigators to contribute to the subject. Further, a large and varied experimental program must be undertaken to assess the validity of any model. (Inevitably there must be approximations in a model when a complex physical situation has to be described in mathematical terms). However, with the availability of large high-speed computers the development of such a general method is now possible. Ultimately the results from these researches should satisfy those whom Bauer (5) describes as "the customers who treat with scepticism most theoretical work to date".

REFERENCES

1. MacLaren, J.F.T., "A review of simple mathematical modelsof valves in reciprocating compressors". A.S.H.R.A.E./A.S.M.E. (Central Indiana Section)/Ray W. Herrick Labs. Compressor Technology Conference, Purdue University, July 1972.

2. Carpenter, A. Brooks., "The use of analog simulator studies to reduce or eliminate objectionable pulsations in reciprocating compressors and associated piping" A.S.M.E. paper 67-Pet-30.

3. Brunner, W. "Simulation of a reciprocating compressor on an electronic analog computer". A.S.M.E. paper. 58-A-146

4. Bauer, F. (page 228). Parry P.J. (page 242) I.Mech.E. Symposium. "Reciprocating and Rotary Compressor Design and Operating Problems".

5. Czaplinski, S. "Pulsations in the suction and discharge line of Reciprocating Compressors" I.I.R. Meeting, Cambridge 1961, Annexe 1961-3, pp 299-316.

6. Bannister, F.K. "Induction ramming of small high-speed compressor." Proc.I.Mech.E. 1959, Vol. 173, No.13

7. Jaspers, H.A. "Special suction lines influencing the volumetric efficiency of reciprocating compressors". XIII Int. Cong. Refrig. Madrid 1967 Paper 3.54

8. Wallace, F.J. "Pulsation damping systems for large reciprocating and free-piston gas generators". Proc. I.Mech.E. 1960, Vol. 174, No. 33, pp 885-915

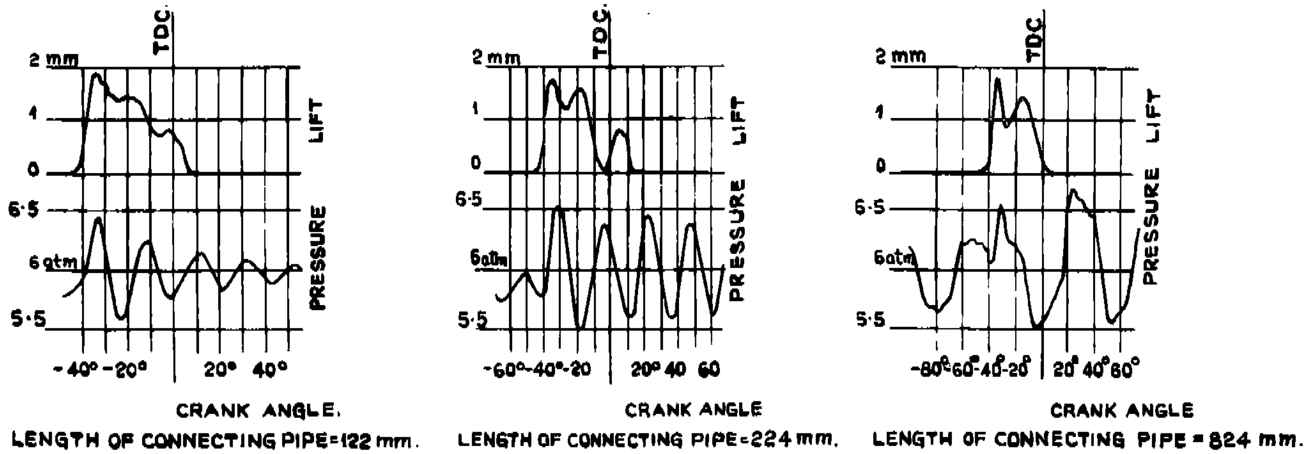
9. MacLaren, J.F T. & Kerr, S.V. "An analytical and experimental study of self-acting valves in a reciprocating air compressor". I.Mech. E. Symposium "Reciprocating and Rotary Compressor Design and Operating Problems" London, 1970. Paper No. 3.

10. Benson, R.S. & Ucer A.S.
(a) "Some recent research in gas dynamic modelling of multiple single stage reciprocating compressor systems." A.S.H.R.A.E./A.S.M.E. (Central Indiana Section)/Ray W. Herrick Laboratories, Compressor Technology Conference, Purdue University, July 1972,
Ucer, A.S. (b) "Unsteady flow in reciprocating compressor systems" Ph.D. Thesis, U.M.I.S.T. 1970

11. Brablik, J. (a) "Calculation of gas pulsation in the piping system of reciprocating compressors" Symp.International of Compressors, Prague, 1969. Edited by Dum Techniky C.S.V.T.S., Praha, Gorkeho nam. (b) Influence des pulsations de gaz sur le fonctionnement des soupapes automatiques de compresseurs. I.I.R. Commission 3, Meeting, Prague, 1969. (c) "Analysis of movement of valve plate of automatic valve under influence of pulsations of gas in the piping of reciprocating compressors. Thesis, State Research Institute for Manufacture of Machines Machines, Bechovice, Prague, Czechoslovakia.

(d) "Gas pulsations as factor affecting operation of operating of automatic valves in reciprocating compressors A.S.H.R.A.E./ A.S.M.E. (Central Indiana Section)/Ray W. Herrick Laboratories, Compressor Technology Conference, Purdue University, July 1972

12. Burgess-Manning Co. Pulsation Handbook



BAUER (REF. 4)

FIG. I. EFFECT OF PIPELINE PULSATIONS ON DISCHARGE VALVE DISPLACEMENT.

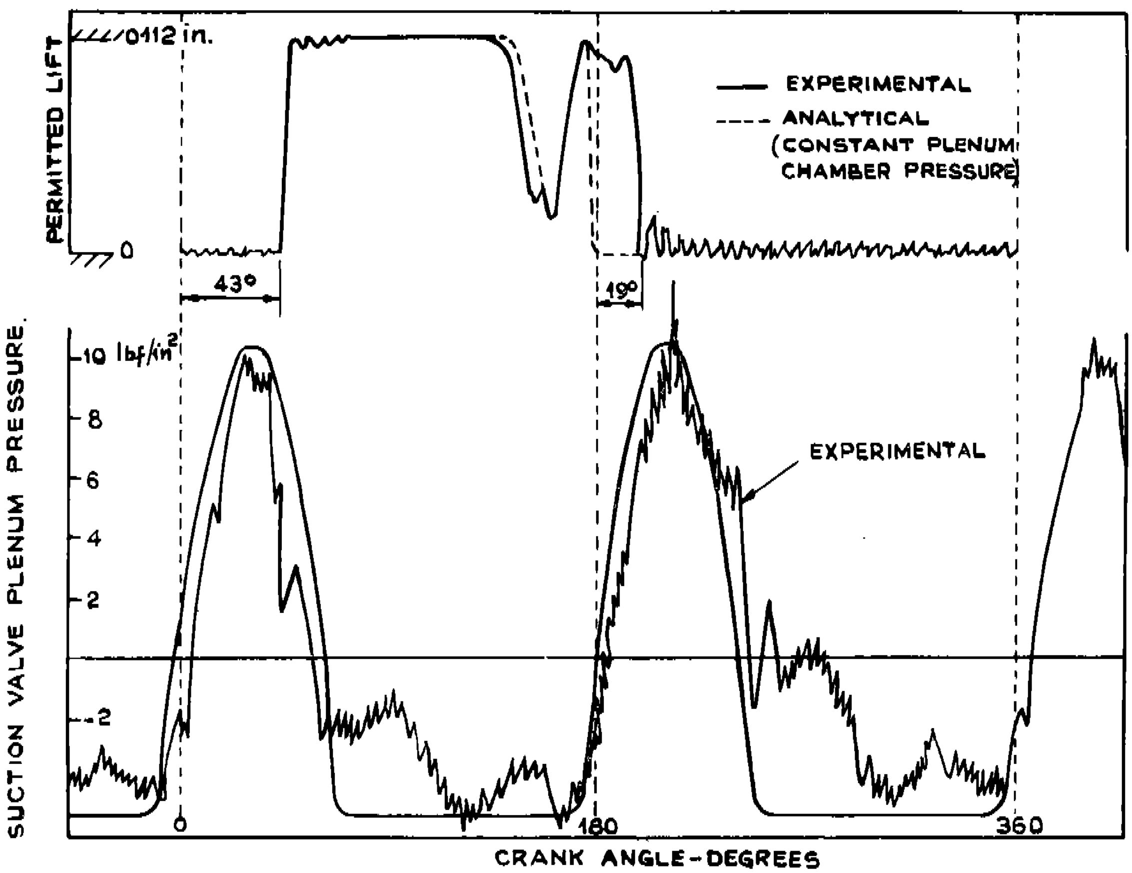


FIG. 2. MACLAREN AND KERR (REF. 9) 3RD STAGE 3000 hp, 300 REV./MIN. 5 STAGE AIR COMPRESSOR: SUCTION VALVE DISPLACEMENT AND LINE PRESSURE DIAGRAMS.

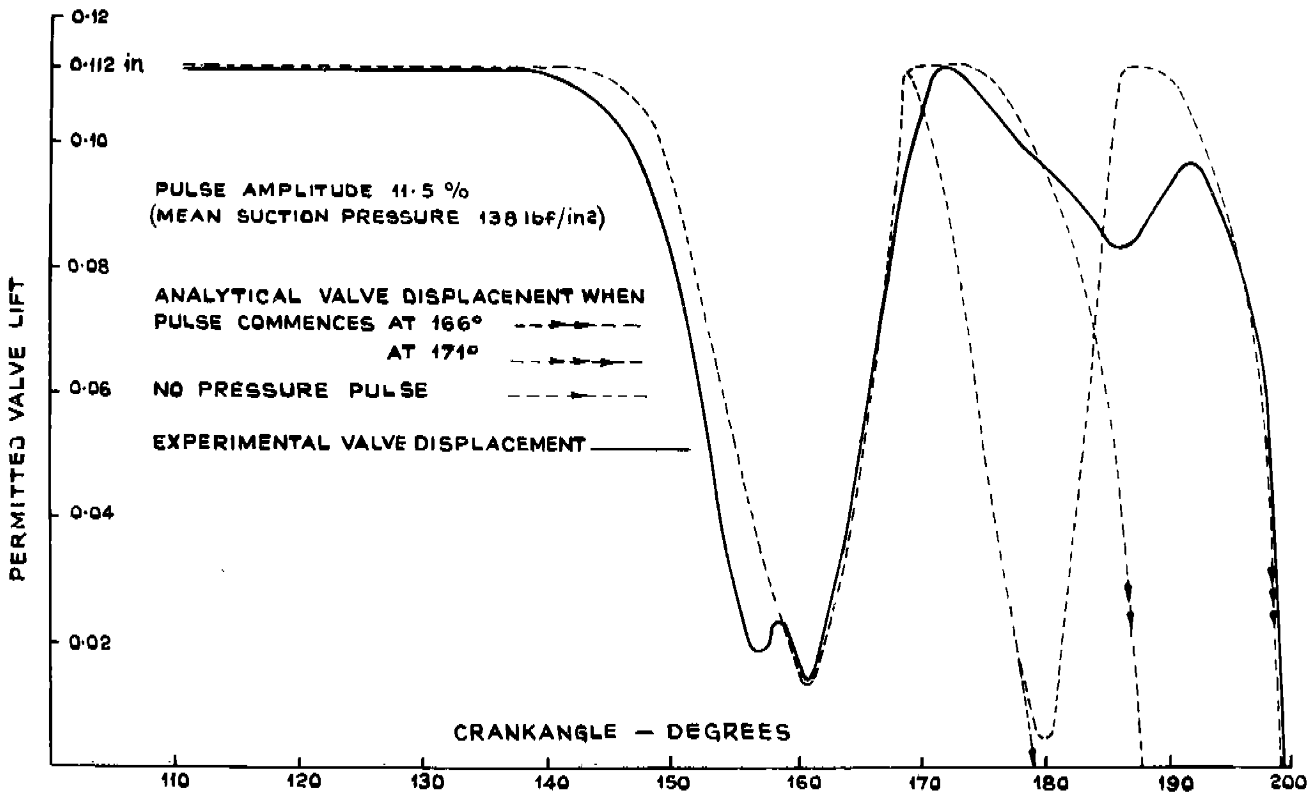


FIG. 3. MACLAREN AND KERR (REF 9) 3000 hp, 300 rev/min, 5 STAGE AIR COMPRESSOR: EFFECT OF PLENIUM CHAMBER PRESSURE PULSE ON SUCTION VALVE CLOSURE.

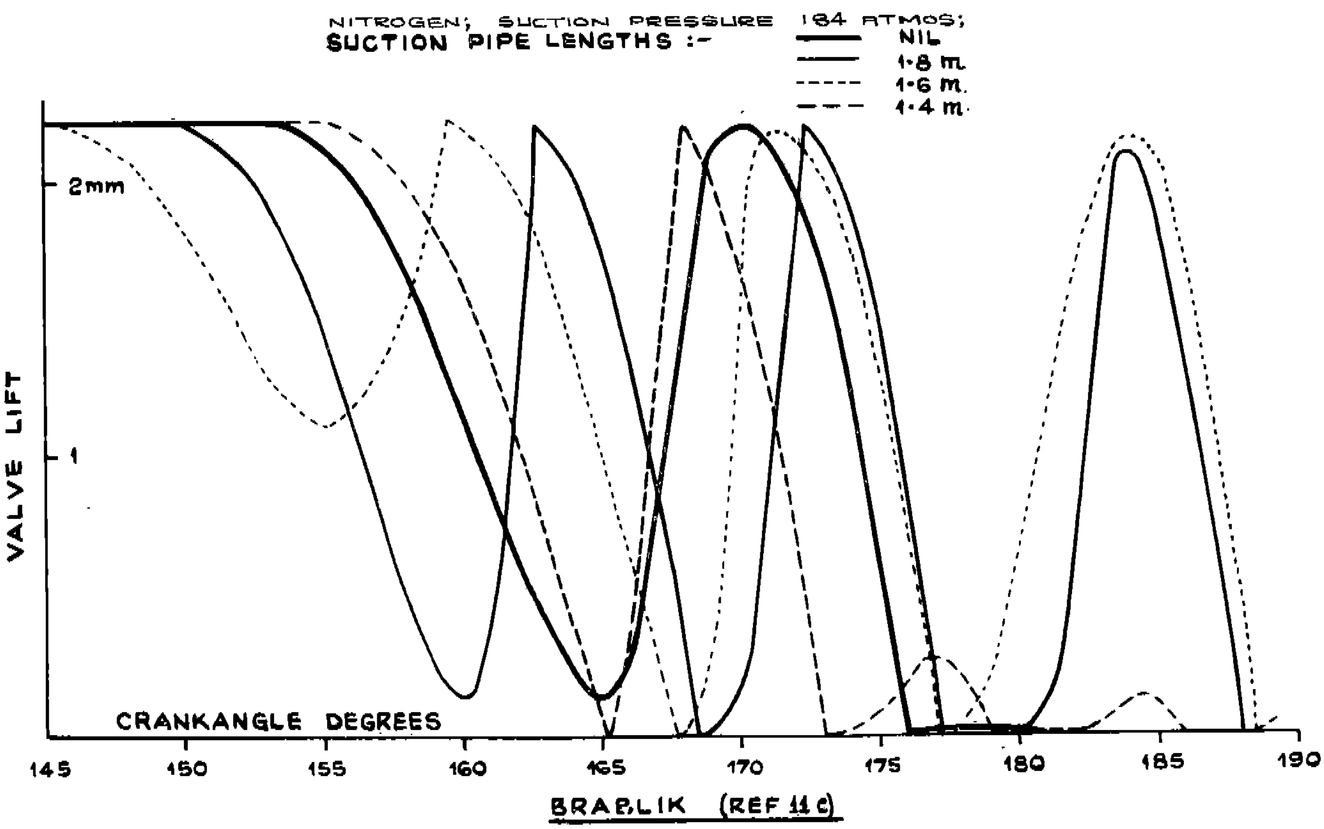
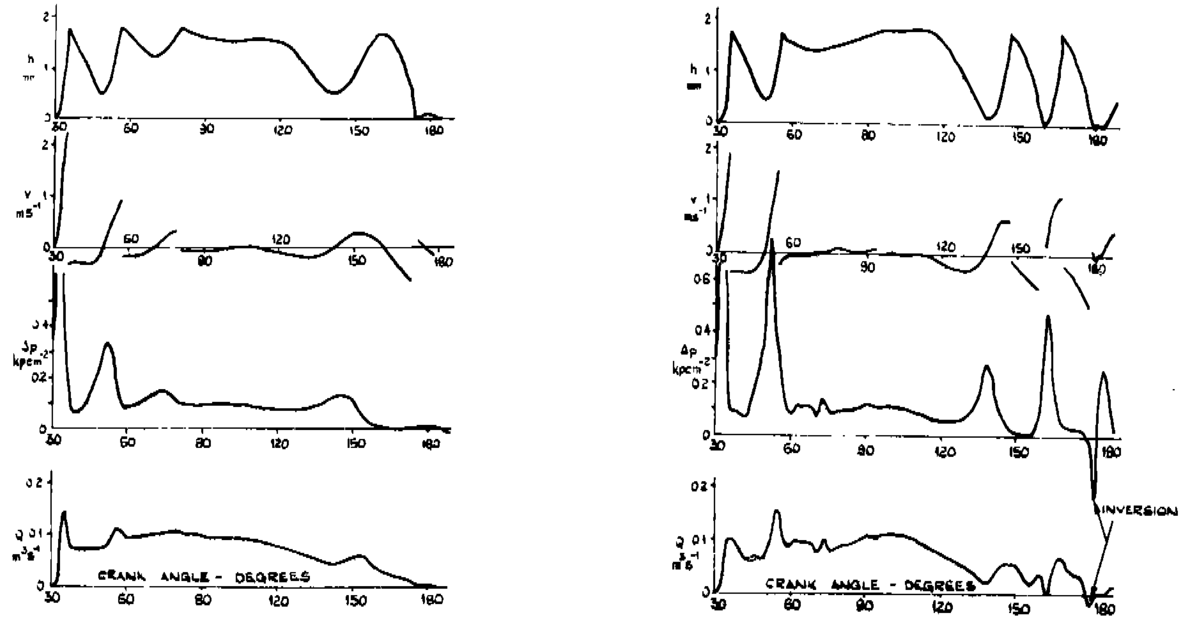


FIG. 4. - EFFECT OF SUCTION PIPE LENGTH: FINAL PHASE OF COMPUTED MOVEMENT OF SUCTION VALVE PLATE.



(a) WITHOUT INLET PIPE.

(b) INLET PIPE LENGTH, 1m

BRABLIK (REF 11b)

h = VALVE LIFT ; V = VALVE VELOCITY ; Δp = PRESSURE DIFFERENCE ACROSS VALVE ; Q = AIR MASS FLOW RATE.

FIG 5. COMPUTED EFFECT OF INLET PIPE LENGTH ON SUCTION VALVE BEHAVIOUR.

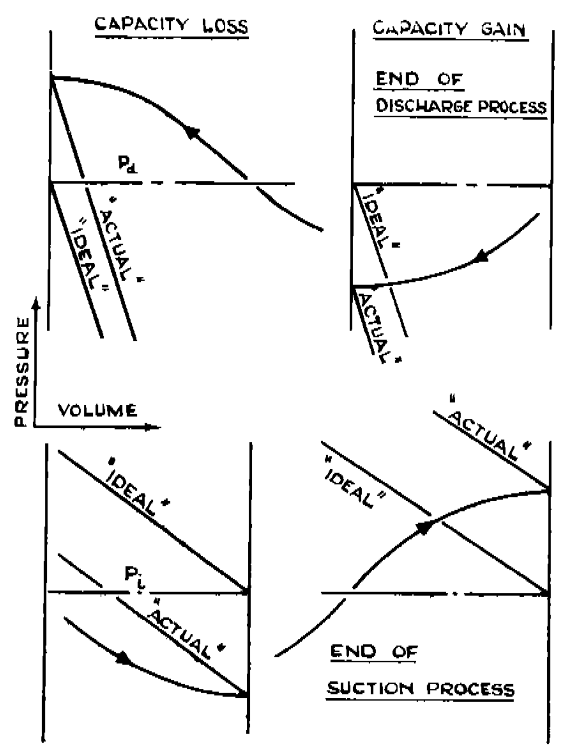


FIG. 6. PULSATION EFFECTS ON COMPRESSOR CAPACITY.

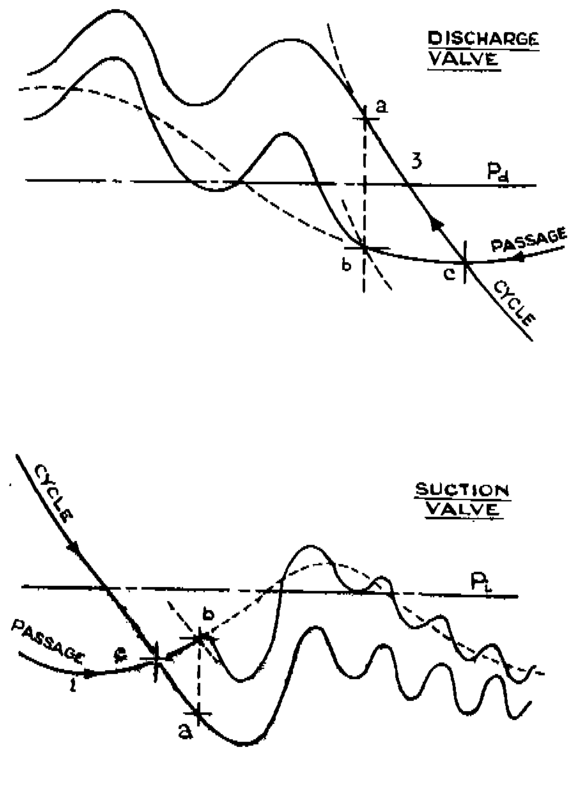


FIG 7 EFFECT OF DELAYED VALVE OPENING ON PASSAGE PULSATIONS.

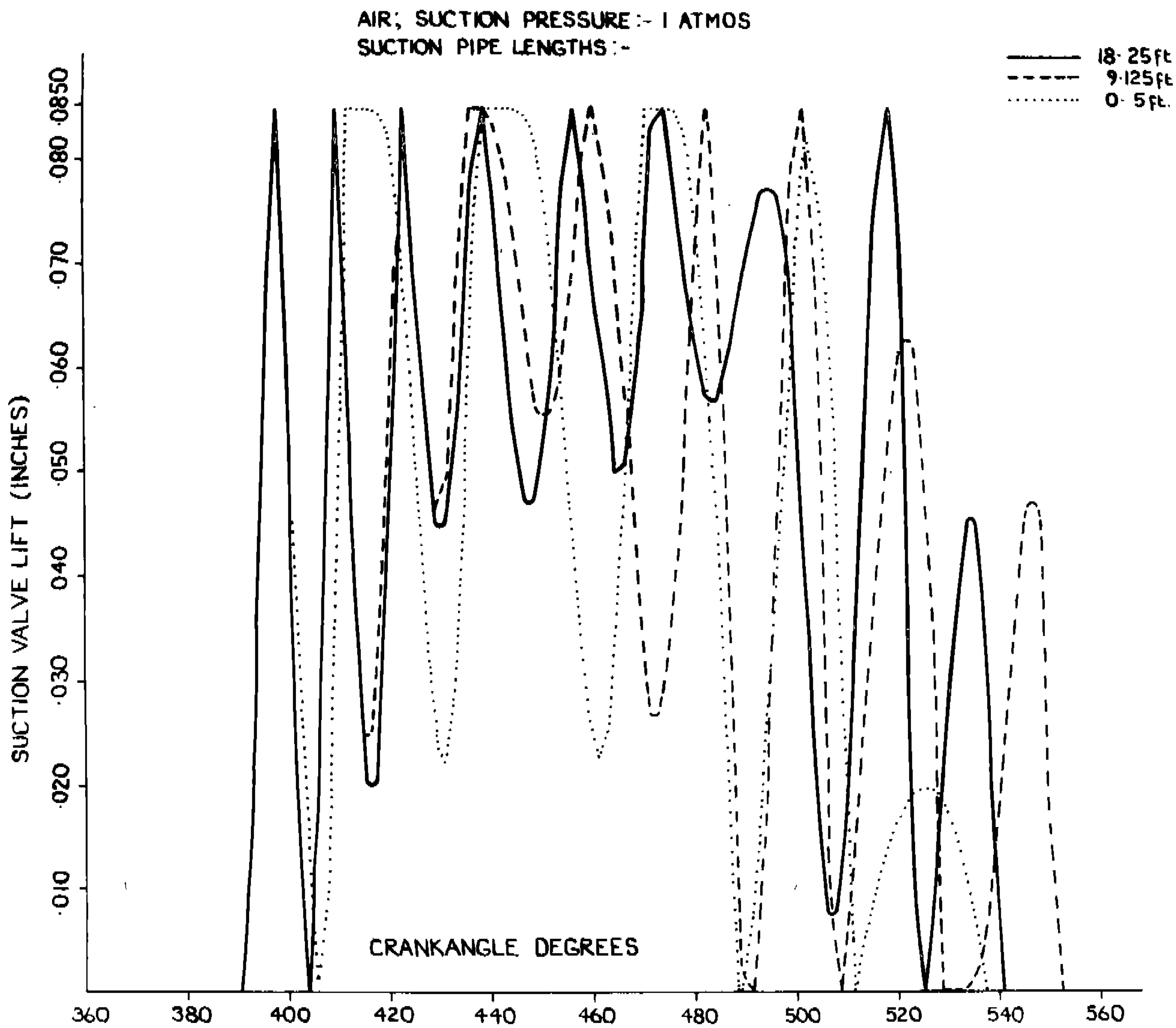


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FIG. 8. EFFECT OF SUCTION PIPE LENGTH ON COMPUTED MOVEMENT OF SUCTION VALVE PLATE. SINGLE STAGE AIR COMPRESSOR, 400 REV/MIN.

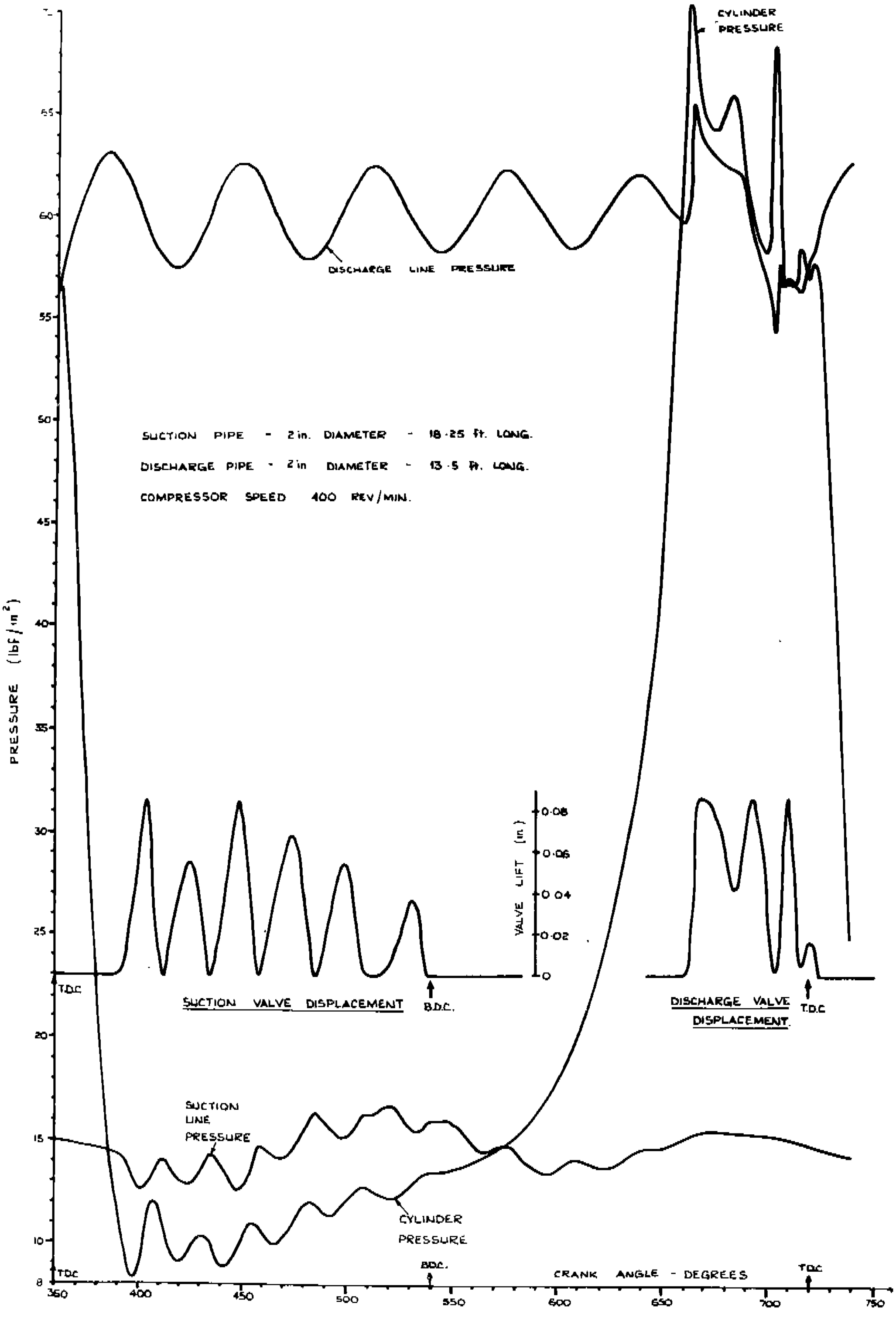

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FIG. 9. COMPUTED CYLINDER PRESSURE, PIPE LINE PULSATIONS AND DISPLACEMENT OF VALVES