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Component Losses in Refrigeration Compressors

A Thesis presented for

The Degree of Doctor of Philosophy

by

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Abstract.

The purpose of this investigation was to examine the relative importance of factors affecting the volumetric efficiency of small refrigeration compressors with particular reference to blowback effects at the suction valve. The overall volumetric efficiency being recognised as an unreliable indication of the relative importance of component losses, the separate losses associated with the action of unclamped, reed type, compressor suction valves were analysed and measured.

The main factors affecting volumetric efficiency were assumed to be heat transfer between fluid and cylinder, re-expansion of clearance volume vapour, throttling at the valves, blowback past the valves while closing, leakage past the piston and leakage due to imperfect seating of the valves. A review of previous work indicated that the loss due to blowback was worthy of attention though no experimental results were available owing to the difficulties attending the measurement of this quantity.

A theoretical study of the problem of blowback past the suction valve was made and methods of analysis devised by which blowback quantities could be predicted. The approach of previous workers to the dynamics of valve action was not followed as it proved invalid when applied to those parts of the cycle where blowback occurred.

New experimental techniques were developed in order to make direct measurements of blowback losses in a small open type compressor. Initial tests were performed in which the suction valve of an otherwise conventional compressor was operated by electro-magnetic means. This method did not prove sufficiently accurate and an A.C. operated constant temperature

hot-wire anemometer was developed to indicate gas valocities in the suction port.

The anemometer, in conjunction with valve motion measurements, was used to investigate the effects of speed, valve lift, valve thickness, valve end freedom and fluid density on blowback. From the records obtained it became evident that the anemometer diagrams could also be used to give rapid and accurate measurements of re-expansion and throttling losses. The results showed that a combination of light valve reed and low lift produced the greatest efficiencies in practice. The loss due to blowback was also much reduced when pumping dense fluids.

Some tests were carried out on a small modern "hermetic" compressor and these indicated that the anemometer could be applied to investigations on such sealed units without undue difficulty.

Details are also given of a new type of compressor which was designed on the basis of the data obtained in the investigation. The main features of the design are the reduced throttling and blowback losses.

CHAPTER INDEX

INDEX OF FIGURES AND GRAPHS

Nomenclature used in connection with valve movement.

D_v	diameter of valve disc
D_t	diameter of valve port
D_p	diameter of piston
r	radius
V_v	gas velocity parallel to valve plate at edge of valve disc
V_t	gas velocity parallel to valve plate at port diameter
V_p	instantaneous piston velocity
V_m	maximum piston velocity
P_c	pressure in cylinder
P_v	pressure between valve and plate at diameter D_v
P_t	pressure between valve and plate at diameter D_t
ω	gas density
W_v	weight of valve
l	distance between valve and valve plate
t	time
Θ	angle between crank position and bottom dead centre
K	constant

See inside back cover for fold out list.

Nomenclature used in connection with anemometer.

V	gas velocity
I	hot wire current
R_a	wire resistance at gas temperature
R_e	wire resistance at equilibrium
R	instantaneous wire resistance
Θ_a	gas temperature
Θ_e	equilibrium temperature of wire
Θ	instantaneous temperature of wire
α	temperature coefficient of wire resistance
c	specific heat of gas
σ	density of gas
k	thermal conductivity of gas
d	diameter of wire
J	Joules' equivalent
f	frequency
M	time constant of wire
h	heat transfer coefficient from surface of wire
μ	viscosity of gas
A, C	constants

See inside back cover for fold out list.

COMPONENT LOSSES IN REFRIGERATION COMPRESSORS

CHAPTER 1

INTRODUCTION AND GENERAL REVIEW

1. 1 Introduction.

The development of the science and practice of refrigeration benefited greatly from the work previously carried out in the field of thermodynamics and applied to improvement of the theory and practice of power engineering. Thus, refrigeration techniques kept pace with the machinery and working fluids available in their day in a manner which is a distinct contrast to the early gropings of steam power engineering. In recent years, however, the increasing market for compact refrigeration equipment and the new materials and manufacturing techniques available have led to rapid changes in design which involve increased speeds and reduced dimensions. The effects of these design changes are not yet fully understood and modern compressor efficiencies are often surprisingly low. (16b).

It is generally conceded that the losses affecting refrigeration compressors are interdependent and of a complex nature so that overall efficiency figures do not usually give a clear indication of the relative importance of the various losses. Thus, in order to provide data for the rational design and development of compressors, it is necessary to separate and measure the component losses associated with these machines.

The fact that refrigeration equipment must be designed to meet the conflicting requirements of cost, efficiency and reliability means that practical designs invariably embody a good deal of compromise. However, it is felt that investigations of the type described in this thesis are of most value when they approach the problem from the viewpoint of efficiency alone, providing the designer with sufficient information to enable him to work out the best compromise for himself. For this reason, though an experiment in compressor design has been carried out, it is not included in the main body of the thesis but can be found in Appendix VI.

1. 2 Losses in reciprocating compressors.

The main criteria of compressor performance are the coefficient of performance and the volumetric efficiency. The coefficient of performance, which is a function of friction losses as well as of the type of cycle employed, does not fall within the scope of the present work, but it should be noted that conditions which, according to simple theory, give high Thermal Efficiencies may be inadmissable owing to unfortunate secondary effects. "Wet Compression" is a case of this type. Volumetric efficiency which can be defined as

$$\frac{\text{Volume of vapour, at suction conditions, pumped per unit time}}{\text{Volume swept out by piston per unit time}}$$

is the property which is the main concern of this thesis.

The main losses affecting volumetric efficiency are caused by -

1. Heat transfer between fluid and cylinder
2. Re-expansion of clearance volume vapour
3. Valve throttling
4. Blowback past the valves while closing
5. Leakage past the piston
6. Leakage due to imperfect seating of the valves

The first investigation into the effects of cyclical heat transfer on the volumetric efficiency of Refrigeration Compressors was carried out by Wirth (26) 1932 using a large, slow running ammonia machine. This pioneer work was followed by three other investigations also using ammonia compressors - by Smith (22) 1934; Giffen and Newley (10) 1940 and by Lorentzen (16a) 1949. All these workers found that the volumetric efficiency improved to a marked degree as the enthalpy of the suction vapour was increased from initial values in the wet field. Following this work on relatively slow-speed ammonia machines there have been two publications on this effect in small high speed Freon 12 compressors; by Brown (4) 1951 and Gosney (11) 1953. A similar marked rise in volumetric efficiency as anthalpy increases is noted by each investigator and their results are remarkable in that the actual volumetric efficiency continues to rise as the enthalpy is increased within the superheat field. This feature is not so apparent in the results quoted for ammonia compressors and may indicate the presence of liquid droplets in the superheated Freon. It may, on the other hand, be the result of an action of the completely miscible Freon on the oil film coating the cylinder walls. An attempt has been made by Brown to determine the actual heat transfer coefficients obtaining between the charge and cylinder wall and thus to elucidate the mechanism of heat transfer.

Losses associated with the functioning of automatic compressor valves have been the subject of much study, though, in general, present day configurations are largely the result of an empirical approach. Early investigations of these losses attempt to relate static flow tests to actual operating conditions without conspecuous success. The tests of Lanzendorfer, (15) 1931, and Fuchs, Hoffman and Schuler (9) 1941, were of this type and their main value would appear to be in indicating desirable shapes and dimensions of porting arrangements. In passing, it should be noted that there is some doubt if tests using hydrodynamic models with suitably chosen fluids and velocities give results which can be directly applied to the flow of refrigerant vapours through valve assemblies, as it appears that such flow is a function of the dimensionless Mach and Froude Numbers as well as of the more familiar Reynolds' Number. Further results of tests with large wooden models in a low speed wind tunnel have been published by Hanson (12) 1945.

A fresh approach to the problem was made by Costagliola (6) 1949, who was able to express the dynamic valve losses mathematically by making a number of simplifying assumptions. Unfortunately, however, his method does not lead to linear differential equations and solutions have to be obtained by graphical methods. There appears to be good correlation between Costagliola's experimental and predicted results. Because the labour involved in valve analysis by this method is prohibitive, MacLaren (17) 1955, adopted a simplified form of Costagliola's equations though, in the author's own words, "The simplification is more apparent than real." MacLaren showed too, that valve flutter is likely to occur and this renders step by step solution of the equations invalid because of the cumulative effect that errors produce at points remote from the starting point of the solution. A surprising feature of MacLaren's results is the relatively small effect of valve throttling losses on volumetric efficiency and this has directed attention away from these losses to leakage and heat transfer losses.

The loss due to re-expansion of clearance volume vapour is well known to engineers and great care is taken to reduce the clearance volume of modern compressors as much as possible.

As far as can be ascertained there is no previous experimental work on the loss caused by blowback past the automatic valves while they are closing. In theory this loss can be deduced from the analyses of Costagliola and MacLaren but in practice any such computation is of doubtful value, particularly if flutter occurs. Both Costagliola and MacLaren assumed a definite spring stiffness associated with the valve and, while this is the case with most discharge valves and with disc and ring-plate type suction valves, the normal suction valve in small high-speed compressors is a thin metal reed set loosely over the suction port, not being clamped at either end. This type of construction, which has been arrived at for a number of sound practical reasons, will, of course, tend to aggravate blowback losses.

Leakage past the piston is another loss which has not received very exhaustive experimental treatment. A certain amount of work with reference to internal combustion engines has been published including papers by Williams and Young (25) 1939; Dykes (7) 1947 and Aue (2) 1954, but the vastly different speeds, clearances and fluids make this work of only slight application to refrigeration. Gosney (11) refers to losses of this type, estimated more or less incidentally during his tests. Unpublished work by Brown (R.C.S.T.) discloses a satisfactory method of measuring this leakage and shows its dependence on suction enthaply and oil condition. From the foregoing tests it can be concluded that, under normal operating conditions, loss of efficiency due to piston leakage is slight.

Leakage due to imperfect seating of the valves is difficult to estimate under working conditions and experimental results are not available. A paper by Higham, (13) 1942, is concerned with this problem from the practical point of view. Compressor manufacturers usually check the static seating of individual valves and re-build faulty assemblies.

Examination of published work concerning refrigeration compressors reveals that concern about the individual factors affecting volumetric efficiency is of comparatively recent origin and that careful experimental examination of the processes taking place within the cylinder is an even more recent development, stemming principally from the researches of Brown (4) Gosney (11)and MacLaren (17). The comprehensive paper in which Lorentzen (16a) displayed results from a wide variety of compressors was of considerable reference value and the development of electronic indicating techniques since its publication suggests that a similar paper summarising the present state of knowledge and technique would be of value.

CHAPTER 2

OBJECT AND SCOPE OF PRESENT WORK

The foregoing review has indicated the need for a detailed investigation of the various losses which affect the efficiency of modern compressors. A certain amount of work has already been done in this field, (Refs. 5, 4, 10, 16.), but the losses investigated proved, in general, too small under normal operating conditions to explain the relatively low volumetric efficiencies accepted in the Refrigeration Industry. For example, the throttling losses during the early part of induction, which can be accurately estimated by the method of Costagliola are small compared to the losses due to underpressure and valve inertia at bottom dead centre. These latter losses occur at a point in the cycle remote from the valve opening and are difficult to compute accurately, particularly if any oscillation, transient or otherwise, of the valve occurs. It is desirable, too, that investigations into the behaviour of suction reeds of small high-speed compressors should consider the practical case in which the reed is not clamped over the port but has several thousandths of an inch freedom of movement at the "hinge" end. As this type of constraint increases the degrees of freedom of the valve and causes the spring stiffness to act in a non-linear manner upon the valve's reaching the limit stop it is apparent that clear experimental results are required in support of any theoretical approach to the problem.

The research programme for the elucidation of these losses developed along the following lines:

1. A Preliminary Investigation was undertaken to determine the approximate amount of blowback taking place. By electro-

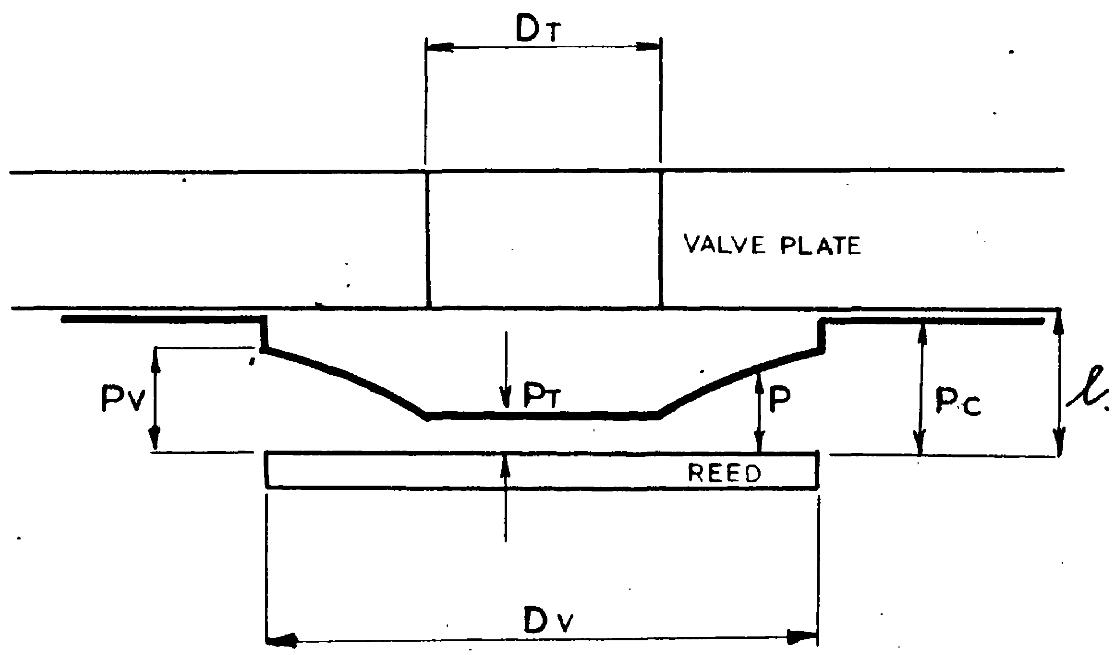
magnetic means the suction could be closed at any position and the resulting changes in performance could be measured.

2. Attention was then directed to the development of techniques by which the losses occurring with the piston near bottom dead centre can be measured with minimum trouble and sufficient accuracy. This work was mainly electronic, and it is dealt with separately in Appendix III.

3. The direct measurement of blowback under various operating conditions was next undertaken. This was done by indicating the gas velocity in the suction port using a constant temperature hot-wire anemometer and comparing the volume induced per cycle with the reversed flow quantity.

4. A theory was developed by which the losses studied experimentally could be related to the conditions under which the compressor was operating during the experiment.

5. Valve designs were considered, to eliminate, as far as possible, the losses encountered in this work. As this is more a development than a research problem it has been treated in Appendix VI.



ASSUMED PRESSURE DISTRIBUTION DURING REVERSED FLOW

FIG. 1

CHAPTER 3

THEORETICAL ESTIMATION OF BLOWBACK PAST SUCTION VALVE

3. 1 Assumptions made in theoretical analysis.

It is assumed that :

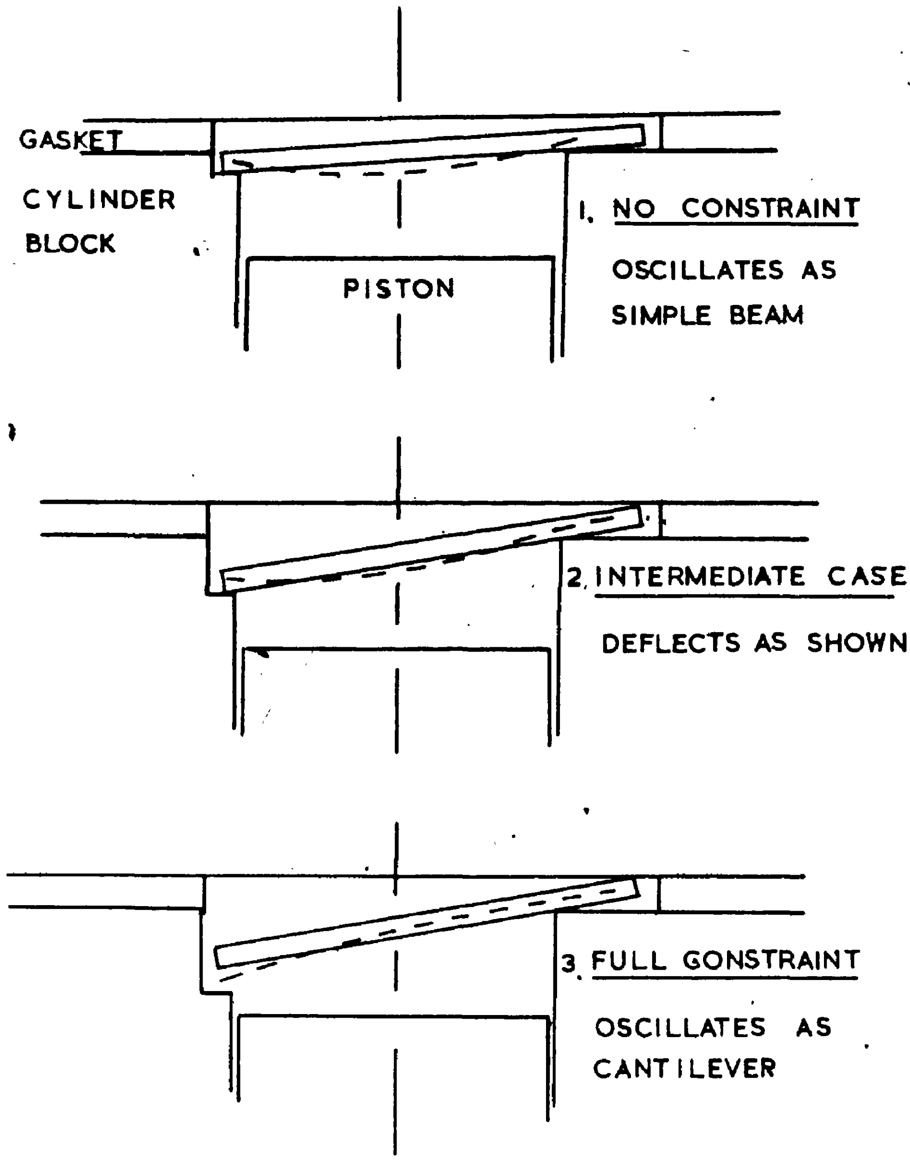
(1) Valve is a flat disc without spring stiffness.

(2) As blowback occurs with relatively low piston velocity, flow is incompressible.

(3) Losses in the passage between valve and plate can be neglected.

(4) The pressure distribution on the valve is as shown in Fig. (1). i.e. There is no recovery of pressure at the port entrance.

(5) The velocities in the cylinder are small compared with the velocity between valve and plate.



EFFECT OF PERMITTED LIFT ON VALVE BEHAVIOUR

FIG. 2

3. 2 General derivation.

From the pressure distribution assumed in Fig. (1) the following relationships can be deduced :

$$\frac{P_v}{\omega} + \frac{V_v^2}{2g} = \frac{P_T}{\omega} + \frac{V_T^2}{2g} \quad \text{—— (1)}$$

$$\frac{V_T}{V_v} = \frac{D_v}{D_T} = R \text{ (say)} \quad \text{—— (2)}$$

$$\frac{P_c - P_v}{\omega} = \frac{V_v^2}{2g} \quad \text{—— (3)}$$

The total force on the valve is,

$$\frac{\pi}{4} D_v^2 P_c - \frac{\pi}{4} D_v^2 P_T - \int_{r_T}^{r_v} (P - P_T)\, 2\pi r\, dr$$

Using the relationships (1), (2) and (3) this reduces to,

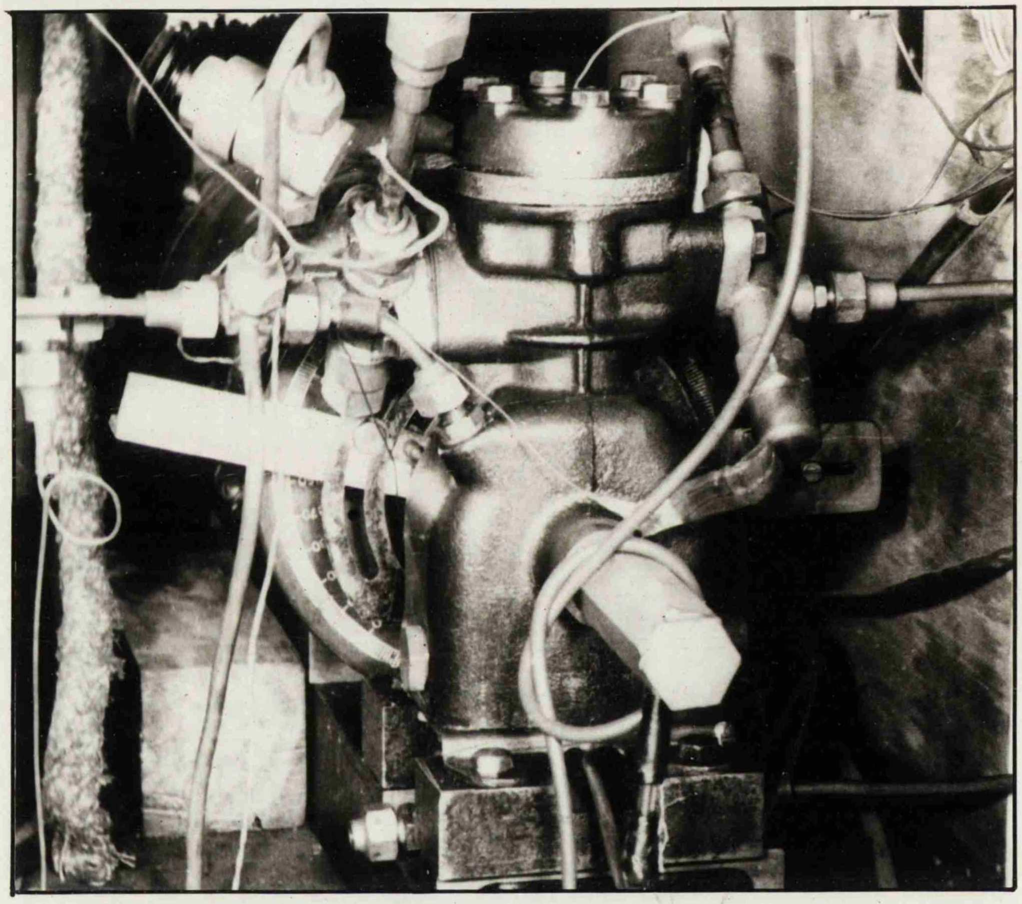
$$\pi r_v^2 \left[\frac{\omega V_v^2}{2g} + P_T + \frac{\omega}{2g}(V_T^2 - V_v^2) \right]$$

$$- \pi r_v^2 P_T - \frac{\pi \omega V_T^2}{g}\left[\tfrac{1}{2}(r_v^2 - r_T^2) - r_T^2 \log_e R \right]$$

which equals

$$\underline{\frac{\pi \omega V_T^2 r_T^2}{2g}\left[1 + 2 \log_e R \right]} \quad \text{—— (4)}$$

As is to be expected, the force on the valve is proportional to the square of the velocity through it, for any given set of conditions.



A - COMPRESSOR

FIG.3

It is now possible to derive the critical velocity required to commence closure of the valve, and, as the flow is assumed to be incompressible, this can be related to a definite piston speed and position after bottom dead centre.

For critical conditions

$$W_v = \frac{\pi \omega V_T^2 r_T^2}{2g}\left[1 + 2\log_e R\right]$$

$$\therefore V_T = \sqrt{\frac{2g W_v}{\pi \omega r_T^2 \left[1 + 2\log_e R\right]}} \qquad \text{——— (5)}$$

also $$V_T = \frac{V_p r_p^2}{2 r_T . l.} \qquad \text{——— (6)}$$

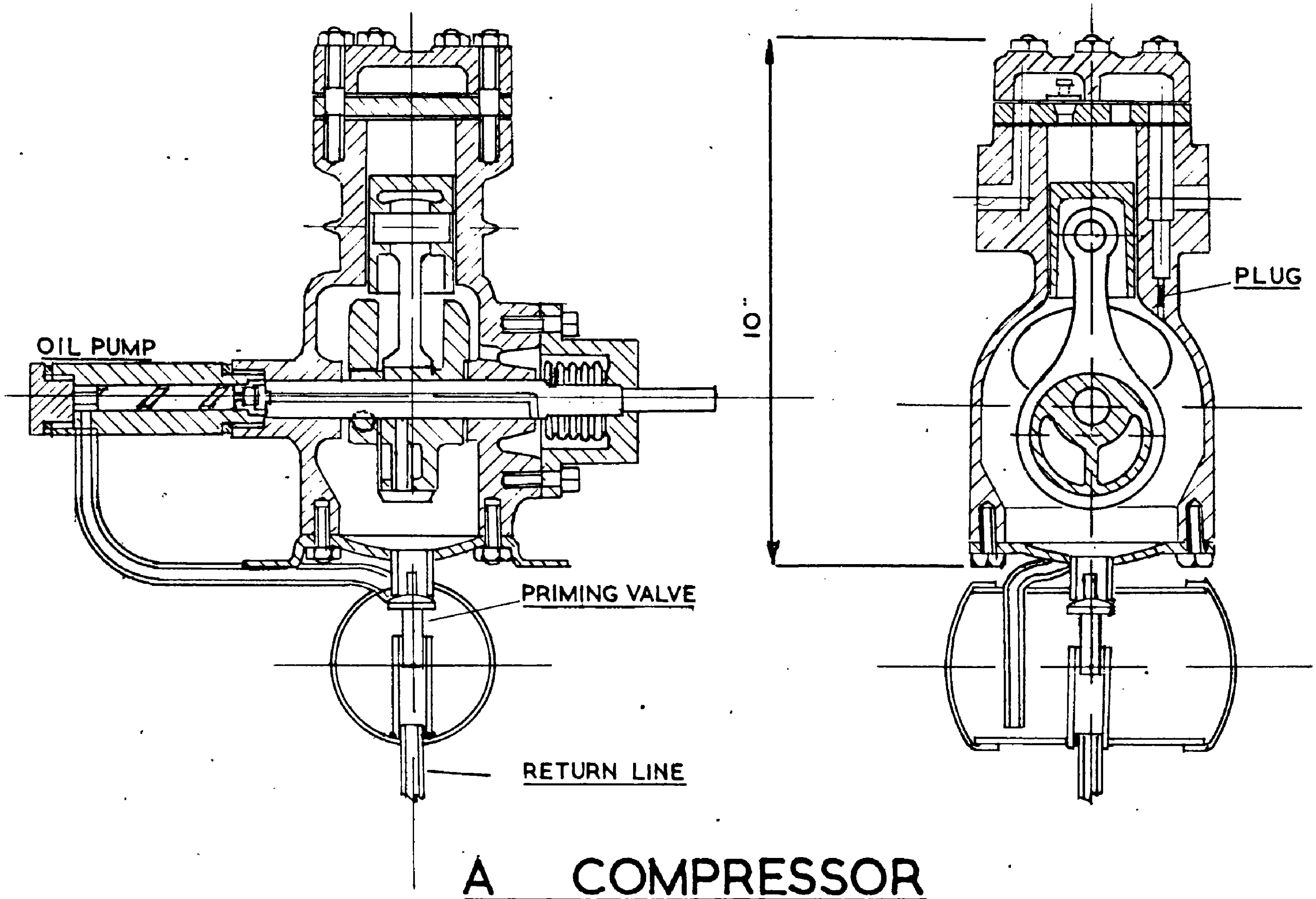
The corresponding piston velocity is given by

$$\underline{V_p = \frac{2l}{r_p^2}\sqrt{\frac{2g W_v}{\pi \omega (1 + 2\log_e R)}}} \qquad \text{——— (7)}$$

Closure of the valve cannot commence till the piston attains this velocity. This lag is due to the dead weight of the valve and will be called the "Dead Weight Loss." If the throttling loss is less than the dead weight loss, no error will be involved in assuming that all the loss at this part of the stroke is due to the dead weight of the valve.

It should be noted that, if the above formula is applied to a beam type reed valve, W_v should be replaced by an equivalent weight to allow for the support given at the "hinge" end.

FIG. 4



A COMPRESSOR

Now it is possible to write down the differential equation governing valve motion under these conditions. Force on valve due to gas flow can be written

$$\frac{\pi \omega r_T^2}{2g}\left[1 + 2\log_e R\right] V_T^2$$

$$= \frac{\pi \omega r_p^4}{8g\ell^2}\left[1 + 2\log_e R\right] V_p^2 \qquad \text{———} \qquad (8)$$

but,

$$V_p \doteqdot V_m \sin\theta$$

therefore accelerating force can be written as

$$\frac{\pi}{8}\,\frac{\omega r_p^4}{g}\left[1 + 2\log_e R\right]\frac{V_m^2 \sin^2\theta}{\ell^2} - W_v \qquad \text{———} \qquad (9)$$

also, Accelerating Force equals $\dfrac{W_v}{g} \cdot \dfrac{d^2\ell}{dt^2}$

therefore,

$$\underline{\frac{d^2\ell}{dt^2} = \frac{\pi}{8}\cdot\frac{\omega r_p^4}{W_v}\left[1 + 2\log_e R\right] V_m^2 \frac{\sin^2\theta}{\ell^2} - g} \qquad (10)$$

This equation applies to valves lying parallel to the valve plate, supported at their centre of gravity and having acceleration induced normal to the valve plate. The conventional suction reed for a small high speed compressor is better represented as a beam, simply supported at one end, having an accelerating force applied near the other end. In this case it is a better approximation to assume an equivalent weight for the valve of $\frac{W_v}{2}$ The equivalent inertia now approximates to $\frac{W}{3g}$

Using above values, the equation of motion, which now applies only to reed valves of the type described, can be written

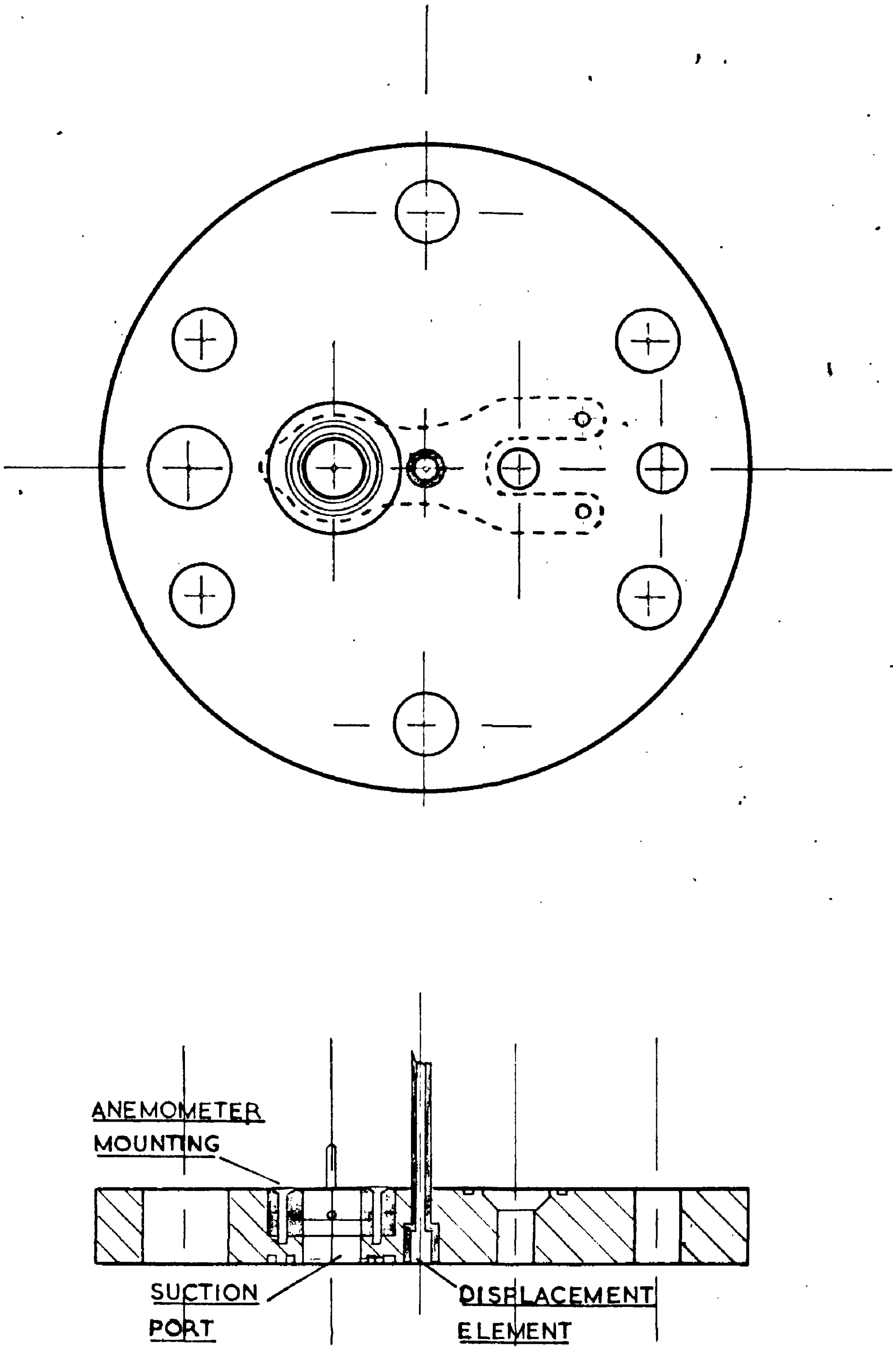
$$\frac{d^2\ell}{dt^2} = \frac{3\pi}{8}\cdot\frac{\omega r_p^4}{W_v}\left[1 + 2\log_e R\right] V_m^2 \frac{\sin^2\theta}{\ell^2} - \frac{3g}{2} \qquad (11)$$

This equation is, in effect,

$$\underline{\frac{d^2\ell}{dt^2} = K\,\frac{\sin^2 nt}{\ell^2} - \frac{3g}{2}} \qquad \text{———} \qquad (12)$$

3. 3 Method of solution.

The differential equation written above is a non-linear type which



A COMPRESSOR
VALVE PLATE DETAILS

FIG. 5

FULL SIZE

does not have a known general solution. This difficulty was encountered by Costagliola and MacLaren in their attempts to obtain a more comprehensive analysis of the problem. However, in the present case, a re-iterative method can be employed to solve any particular numerical example. The method is as follows.

A first approximate curve of $\frac{d^2\ell}{dt^2}$ is drawn from consideration of the varying force which would be applied to the valve if held in the open position.

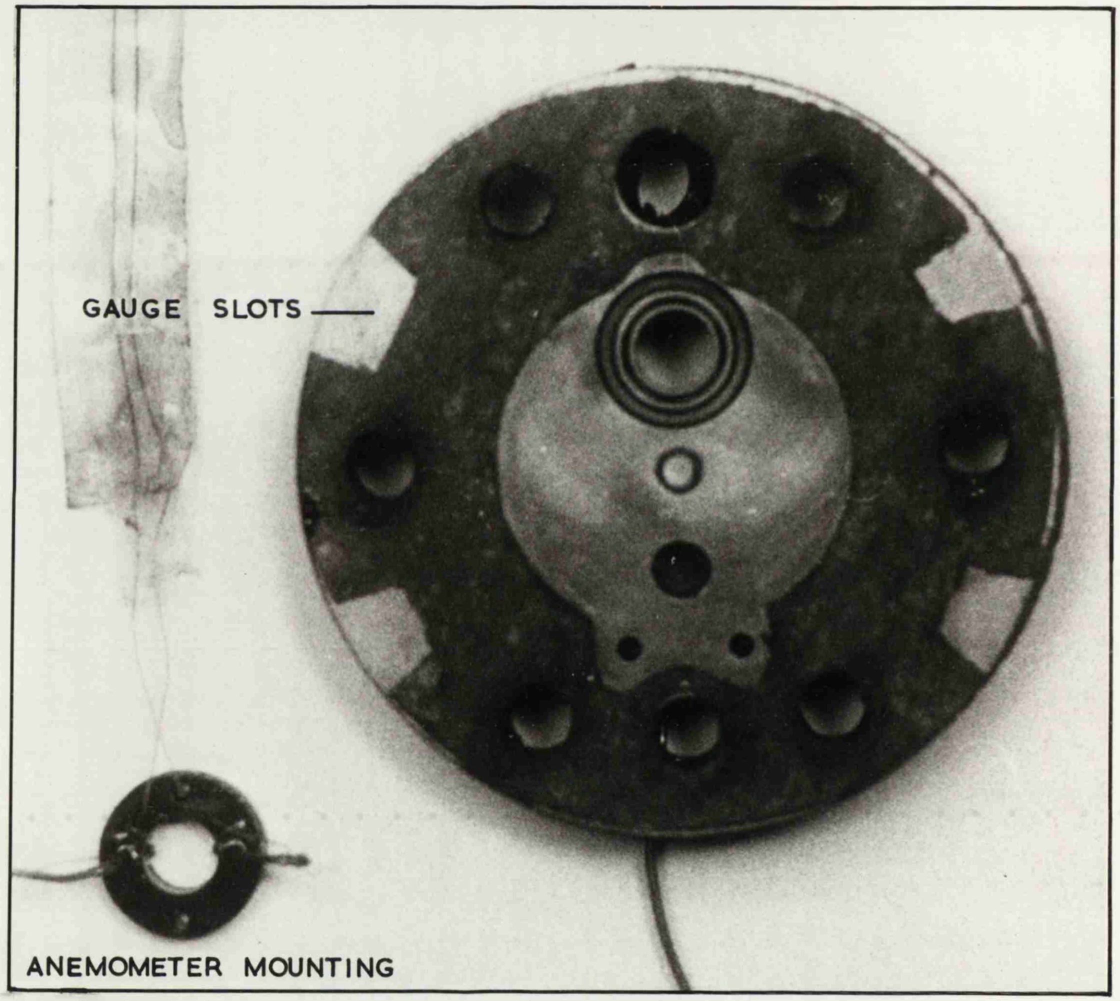
Double integration of this first $\frac{d^2\ell}{dt^2}$ curve produces a first approximate curve of lift with respect to time. This lift curve is used to produce a second approximate $\frac{d^2\ell}{dt^2}$ curve from which a more accurate lift curve can be deduced. Fortunately, this method of solution converges rapidly and can be used to produce results to the degree of accuracy required.

A specimen work sheet is included among the calculations contained in Appendix II.

3. 4 Effect of Flutter.

The foregoing analysis does not include the effects of valve flutter which can be expected to have a profound effect on blowback quantity. However, by neglecting the small effect of variations in valve opening on cylinder pressure at bottom dead centre, the method is rendered independent of a tedious and inaccurate step by step solution from the point at which the valve lifted. This simplification is valid because, as a rough approximation, it can be assumed that the calculated dead weight loss will include the primary effects of valve throttling.

To produce conditions in which flutter can occur, some form of mass-elastic system is required. When the valve is not rigidly clamped



A - COMPRESSOR VALVE PLATE

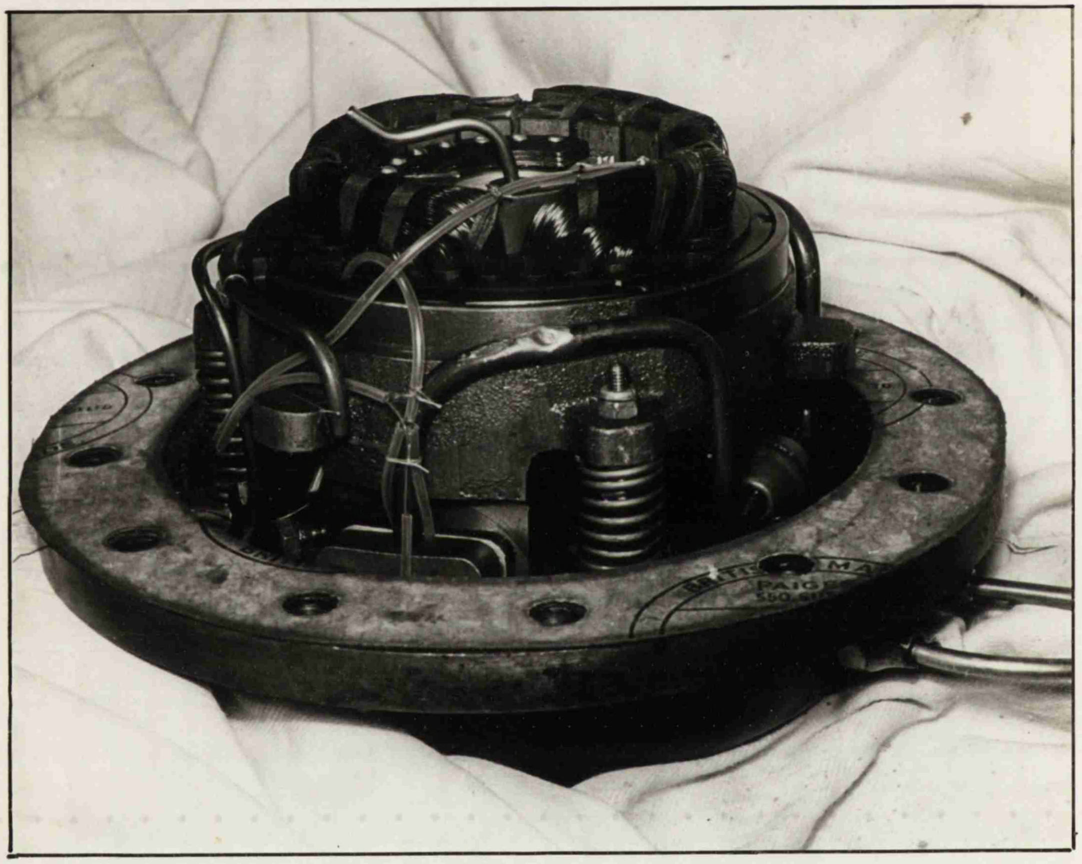
FIG.6

at the hinge end, these conditions do not obtain till the valve strikes the limit stop. After the valve strikes this stop it is free to vibrate as a simply supported beam under non-linear forcing. Such vibrations are likely to be of relatively small amplitude because of the stiffness of the valve in this mode and because of the oil damping applied at each end. As the limit stop is lowered, however, a condition is reached in which the unflexed valve is not able to open as far as the limit stop on account of constraint applied at the hinge end. The operation of this constraint can readily be understood with reference to Fig. (2) which illustrates the condition. In this case the valve will oscillate as a cantilever and the amplitude of oscillation will be relatively large because of the reduced stiffness and damping. To allow for the correction of theoretically predicted results it has been assumed that when oscillating in this mode the valve built up an amplitude sufficient for it to reach the limit stop in its extreme position. It is not possible to predict the amplitude which the valve reed will attain when vibrating as a simply supported beam, but the conclusion that it is small is supported by experimental evidence. The reduction of damping as valve lift increases suggests that the amplitude of this vibration will reach a maximum just before the valve begins to vibrate as a cantilever. In practical cases, of course, the valves are arranged to reach the limit stop without constraint effects, but in experimental investigations where the permitted lift is a variable these effects may be encountered.

Flutter of the valve affects blowback in two main ways.

(1) It provides the valve reed with kinetic energy which may help or hinder the process of closing.

(2) It alters the effective value of lift from that provided for by the limit stop.



H - COMPRESSOR

FIG. 7

The second of these effects is the greater in importance because the relationship between valve lift and gas force, being of the type Force $\propto \frac{1}{x^2}$, where x is the distance of the valve from its seat, produces large closing forces at the points of the oscillation where the reed is near the seat. This will almost invariably cause the valve to close earlier than predicted by simple theory, thus masking the first effect. Experimental evidence indicates that, on occasion, valve closure can be retarded by flutter but this is rare and must be regarded as a freak of timing.

The theoretical allowance for the effect of flutter on blowback is made by assuming an effective lift equal to the minimum distance between reed and seat. In most cases the effect of beam type oscillation is neglected.

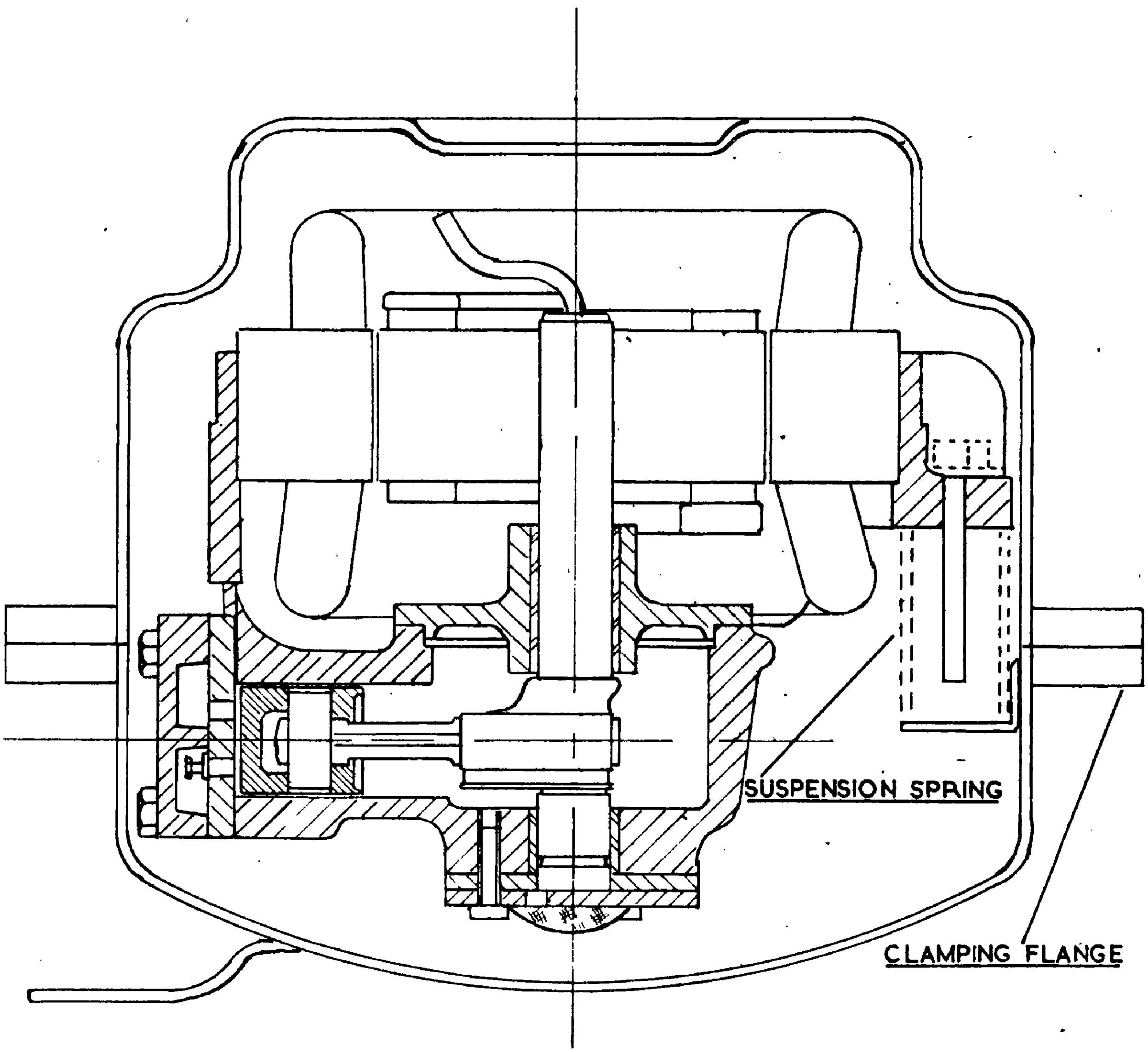
3. 5 Theoretical Implications.

Before discussing the implications of the theoretical analysis it is fitting that its validity should be examined in view of the following assumptions, previously made in order to produce a workable theory.

(1) That the valve is a flat disc without spring stiffness.

This assumption was made in order to provide a symmetrical pressure distribution so that pressure force could be obtained by simple integration. The actual shape of the suction reed used in the experimental compressors is such that pressure forces slightly greater than the theoretical can be expected. This will lead to theoretical blowback quantities being slightly above the actual values. The ratio, R is taken as the relationship between port diameter and a characteristic dimension of the reed valve, equivalent to the diameter of the disc.

The assumption, that the valve is without spring stiffness, holds, for the type of valve considered, when the valve has lifted from the limit



H COMPRESSOR

$\frac{1}{2}$ FULL SIZE

FIG 8

stop. Before that point, the elasticity of the reed may affect the blowback.

(2) That, as blowback occurs with relatively low piston velocity, flow is incompressible.

In view of the fact that the acoustic velocity in Freon is of the order of 500 feet/second and that maximum piston velocities in the compressors considered are well below 10 feet/second, this assumption is seen to be valid.

(3) That losses in the passage between valve and plate can be neglected.

This assumption must be considered with the fourth assumption.

(4) That there is no recovery of pressure at the port entrance.

These rather sweeping assumptions are grouped together because they introduce errors of opposite sense which thus tend to cancel. The validity of assumptions (3) and (4) must depend to some extent on compressor dimensions and speed, but in the practical region investigated they appear justified.

(5) That velocities in the cylinder are small compared with the velocity between valve and plate.

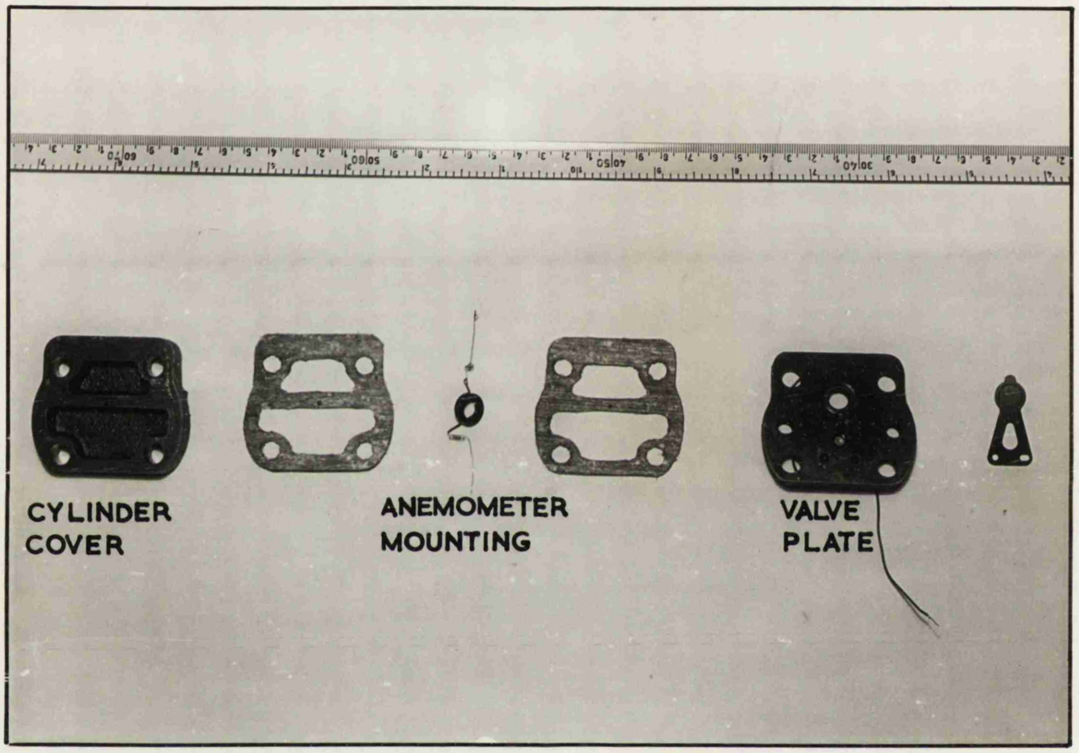
This is obviously true from consideration of the flow areas.

It is also assumed in the derivations that

$$V_p = V_m \sin\theta$$

This is, of course, only approximately true and has been introduced for convenience. If desired, the higher harmonics of piston movement may be included in this expression, though, in view of earlier assumptions and approximations, the value of such a step is questionable.

The theoretical results deduced from these assumptions indicate



H - COMPRESSOR

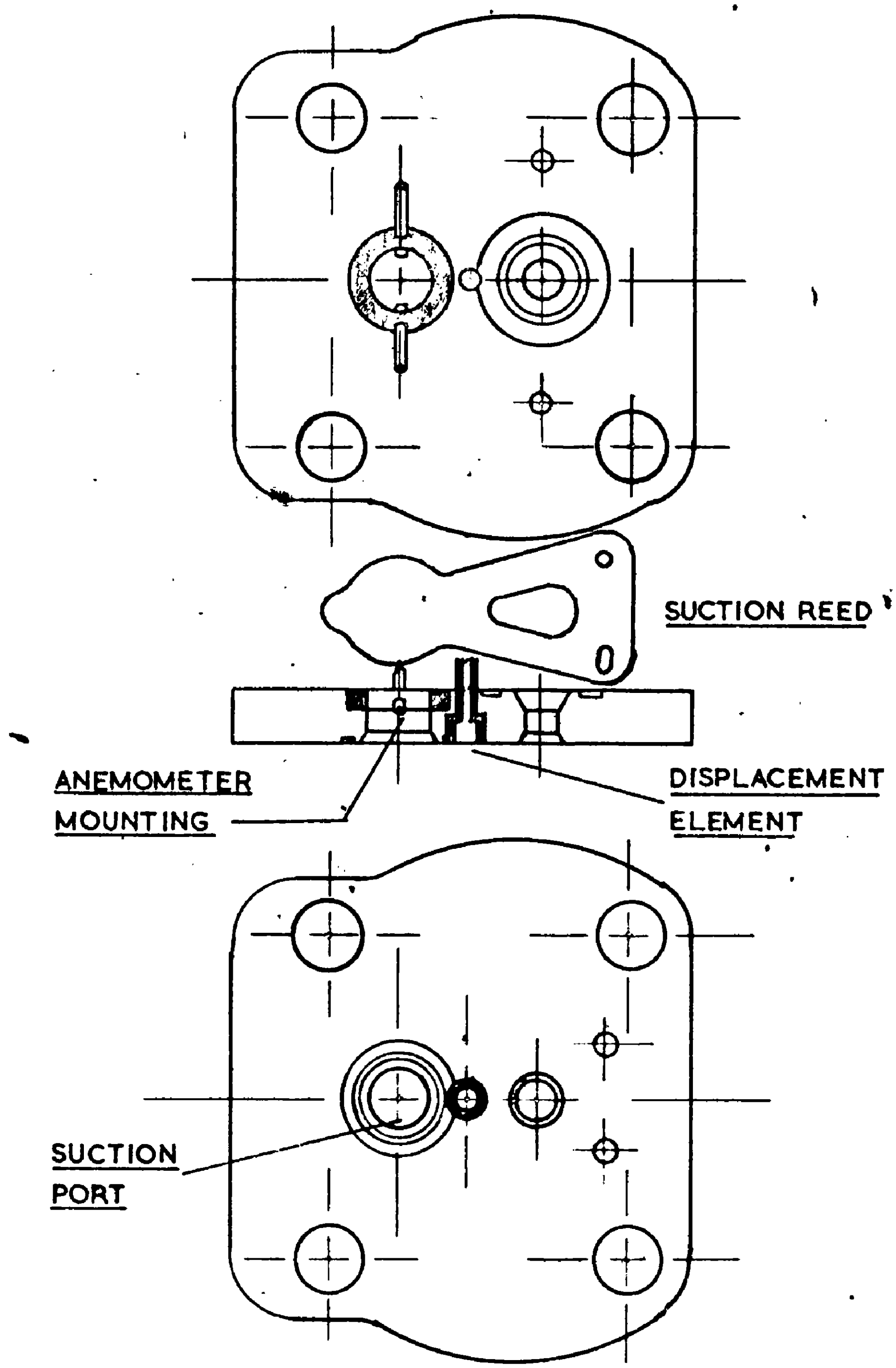
VALVE PLATE

FIG.9

the manner in which blowback depends on compressor dimensions and working fluid.

The dead weight loss, which can be calculated from the critical piston velocity, is seen to depend directly on the valve lift. As the valve throttling losses can be assumed to vary inversely with lift it at once becomes apparent that there is distinct possibility of a maximum value occurring in the volumetric efficiency against permitted lift curve. Although it is impossible to make a direct solution of the differential equation of valve motion, thus producing figures of blowback likely to take place while the valve is closing, it is reasonable to deduce the manner in which such blowback varies by considering it to vary in some manner inverse with respect to the accelerations of the valve. This shows that the blowback during closing also varies inversely with lift. Theoretical blowback curves produced from graphical solutions of the dynamic valve equation can be compared with experimental results in Figs. 31, 32 and 33, and in Fig. 29.

The dependence of blowback on speed is clearly illustrated in the derived equations. It can be seen that the dead weight loss may account for the whole charge if the compressor speed is slow enough. This was indicated by MacLaren in his recent paper to the Institute of Refrigeration (17b). It should be pointed out that this remark applies only to unclamped suction reeds. As compressor speed is increased the dead weight loss drops sharply becoming assymptotic to zero at infinite speed. The dynamic blowback thus rises sharply from zero at the point where the dead weight loss ceases to account for the whole of the induced charge and then decreases as the speed is increased. Owing to the inertia of the valve, however, this loss does not drop so sharply as the dead weight loss and thus the



H COMPRESSOR

VALVE PLATE DETAILS

FULL SIZE

FIG. 10

dynamic loss becomes proportionately more important as a component of blowback as speed increases. If valve throttling becomes severe, both blowback losses may be reduced though, of course, there is no gain in overall performance. This variation of blowback with speed indicates that the unclamped reed is at its best where speeds are comparatively high.

Valve weight is also seen to affect both dead weight and dynamic blowback losses, though not to the same extent as lift. Costagliola (6) points out that stress considerations limit the amount by which valve weights can be reduced.

The dependence of blowback on vapour density suggests that unclamped reeds are unsuitable for light gases and vapours. Conversely, blowback losses are likely to be much reduced when pumping dense vapours. This consideration is of considerable importance in the design of valves for use with Halogenated Refrigerants.

The effect of valve dimensions on blowback is covered by the term

$$(1 + 2 \log_e R)$$

in the theoretical expressions. This indicates the advantage of large values of R. Practical considerations of valve weight and minimum port diameter, however, restrict the permissible variation of this parameter.

A theoretical examination of the factors affecting blowback has been made and a comparison between experimental and predicted results is made later in this thesis. (Fig. 29).

The rigorous approach of Costagliola and MacLaren has not been followed as it cannot be applied with sufficient accuracy to events at the point of the cycle where blowback occurs. Instead, a simple theory, based on pressure distribution during reversed flow, has been applied to the case of unclamped suction valves.

CHAPTER 4

EXPERIMENTAL MEASUREMENT TECHNIQUES.

4. 1 General Apparatus.

Three compressors were used during the tests. These are referred to as the A Compressor, the H Compressor and the E Compressor.

The A Compressor (Figs. 3 & 4) is an 1/8 H.P. open type, single cylinder machine of 1.1/2 inch bore by 1 inch stroke, driven by a variable speed D.C. motor. A machine of this design was used by MacLaren (17) in his experimental work. The A Compressor used in the present series of tests was modified by blocking the equalising port between crankcase and suction so that piston leakage could be measured if desired.

In order to free the experimental results from the scatter produced by splash lubrication, a separate oil sump was constructed below the crank-case and the oil pumped up from there to be introduced along the centre line of a specially drilled crankshaft.

This compressor was designed to run at 600 R.P.M. in its original application but during the tests a speed range of 400 to 1200 R.P.M. was employed.

The Valve Plate (Figs. 5 & 6) contains single suction and discharge ports and is ground to a surface finish of 4 to 5 micro inches.

The discharge valve is a spring loaded beam type and the suction reed is a cantilever located on dowel pins. In the majority of tests carried out, the suction reed was not firmly clamped in place but rested loosely on the pins according to normal practice. However, when valves of different thicknesses were being tested, care was taken to allow each valve the same freedom of vertical movement by building up the hinge ends

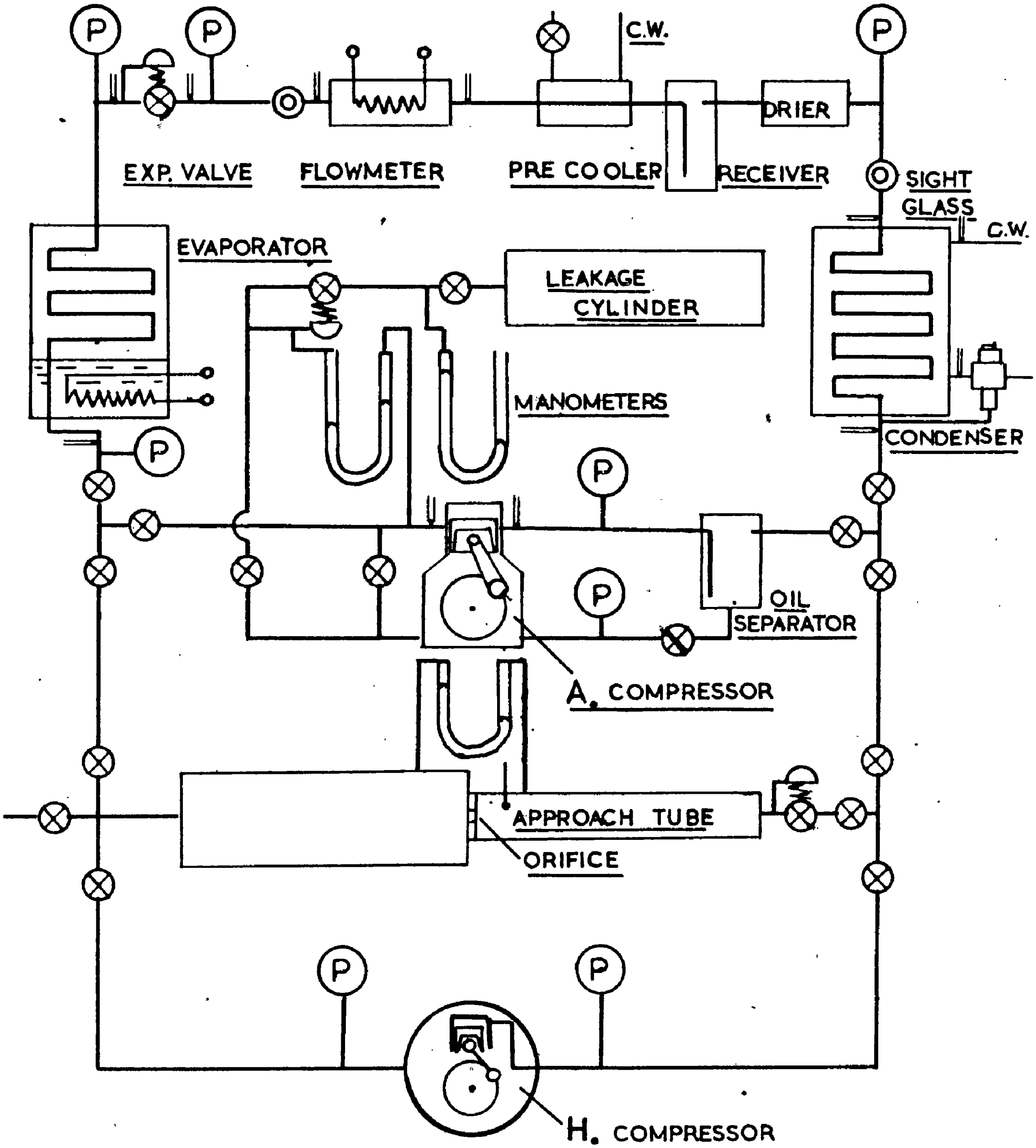


FIG. 11

COMPRESSOR TESTING CIRCUITS

to equal the thickness of the stiffest valve used.

The H Compressor (Figs. 7 & 8) is a totally enclosed 1/8 H.P. single cylinder machine of American design driven at 1450 R.P.M. by a single phase A.C. motor. The compressor has a bore of 1.125" and a stroke of 0.717". The valve plate (Figs. 9 & 10) is very similar to that of the A Compressor having single suction and discharge ports masked by conventional suction and discharge reed valves.

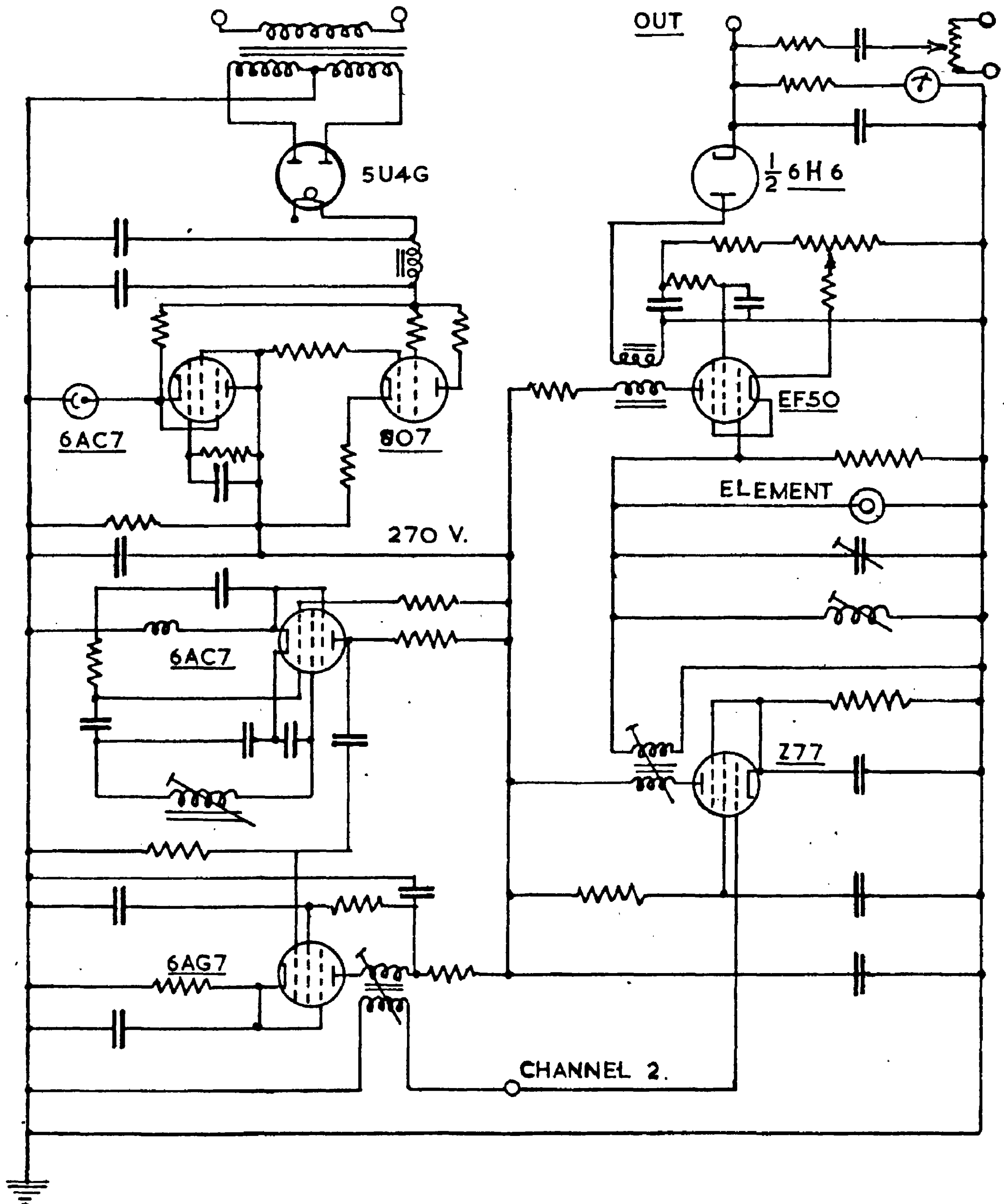
The E Compressor is a twin cylinder experimental compressor which is described more fully in Appendix (VI).

The Compressor Testing Circuit (Fig. 11), in which the H and E Compressors are interchangeable, is arranged so that compressors can be employed either in a refrigeration circuit or in a non-condensing circuit where flow quantity is measured by means of a calibrated orifice. The quantity flowing in the refrigeration circuit is measured by a meter of the type described by Smith (22). The orifices used in the non-condensing circuit were specially constructed and calibrated for this purpose.

4. 2 Valve Lifter Apparatus.

As a preliminary to direct measurement of blowback losses it was decided to operate the suction valve of the A Compressor in a positive manner so that the approximate size of these losses could be deduced from changes in the volumetric efficiency produced by altering the instant of valve closure. For this experiment air was used as the working fluid. The compressor speed was controlled by varying the armature current of the D.C. driving motor. Fine control was obtained by means of a friction brake on the flywheel.

The discharge pressure was read on a Bourdon type gauge and controlled by an expansion valve.



PICK-UP ELEMENT AMPLIFIER.

DESIGNED BY J. BROWN.

FIG. 12

The flow of air was measured by means of a water manometer across a sharp edged orifice situated at the end of a large diameter approach tube to minimise the approach velocity.

A small capacitor element was fitted in the cylinder to measure valve movement. This element is a modification of the type used by MacLaren in his investigation into valve behaviour and the signal from it was amplified in the unit designed by Brown (Fig. 12) for use with capacitative pressure pick-ups. This displacement pick-up consists of an element 0.125" dia. sealed into the valve plate and insulated from it by synthetic resin. The lead from the element is taken outside the compressor by means of a hole drilled through the cylinder cover web, thus producing a gas seal without the use of glands. (Fig. 13).

A Capacitative type Pressure element (Fig. 14) was fitted into the cylinder head to indicate cylinder pressure.

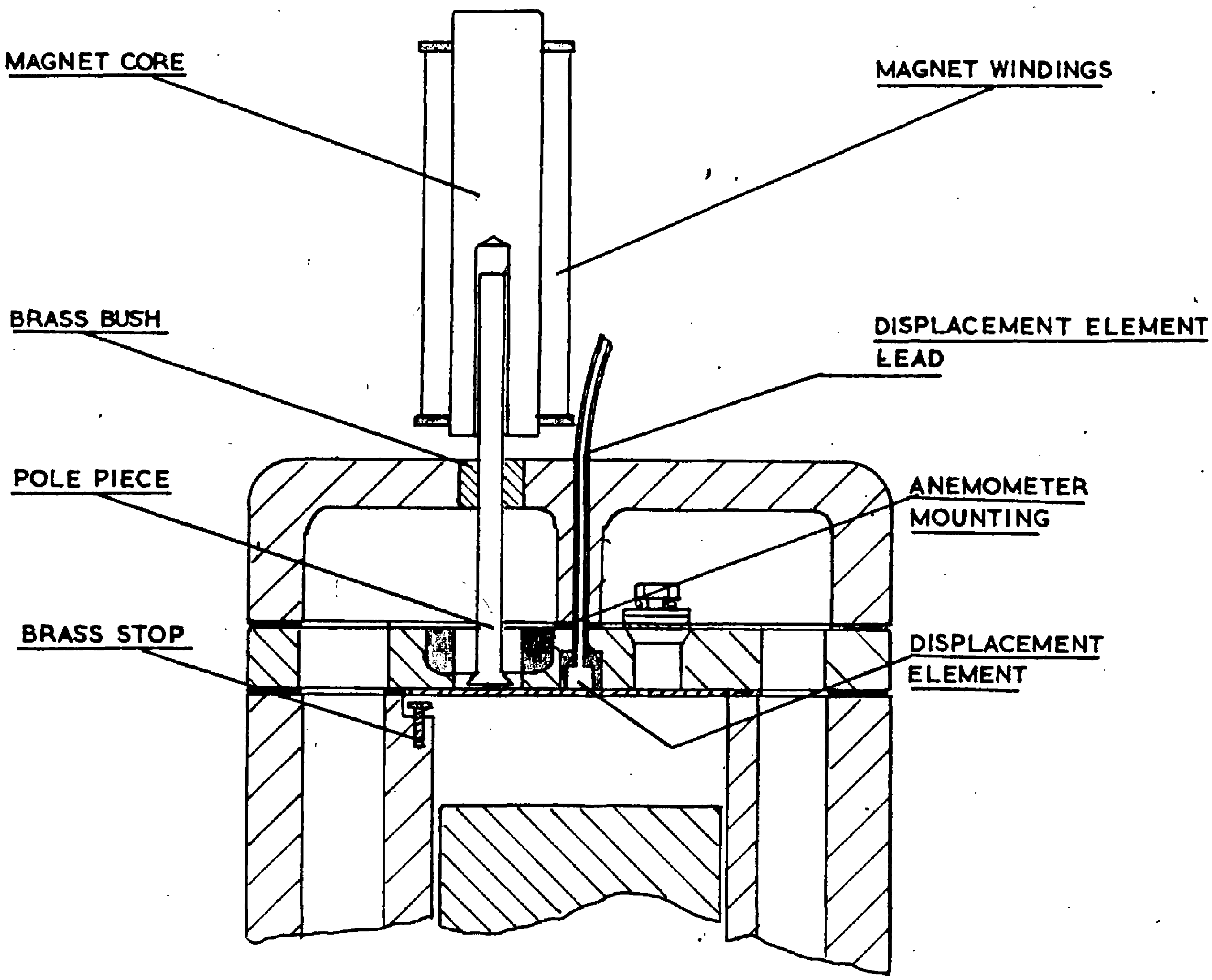
The valve lifter consists of a mild steel magnetic pole sweated into a brass plug in the cylinder head.

In order to minimise leakage flux, the suction port is enlarged for the greater part of its length. The valve port was restored to its normal dimensions by the introduction of a bush of insulating material. This bush was used to mount the sensing element of a hot-wire anemometer.

To prevent a balanced magnetic force on the valve, the limit stop was replaced by a 10 B.A. brass screw which allowed variations of the permitted valve lift.

The electro-magnet is a coil of 1,500 turns wound on a mild steel core and connected in the anode circuit of the power supply output valves.

The electronic power supply is fully described in Fig. 15 except for the switch which is a simple rotary contactor.



VALVE LIFTER ARRANGEMENT

FIG. 13

FULL SIZE

A phasing signal was produced from a capacitor element on the flywheel.

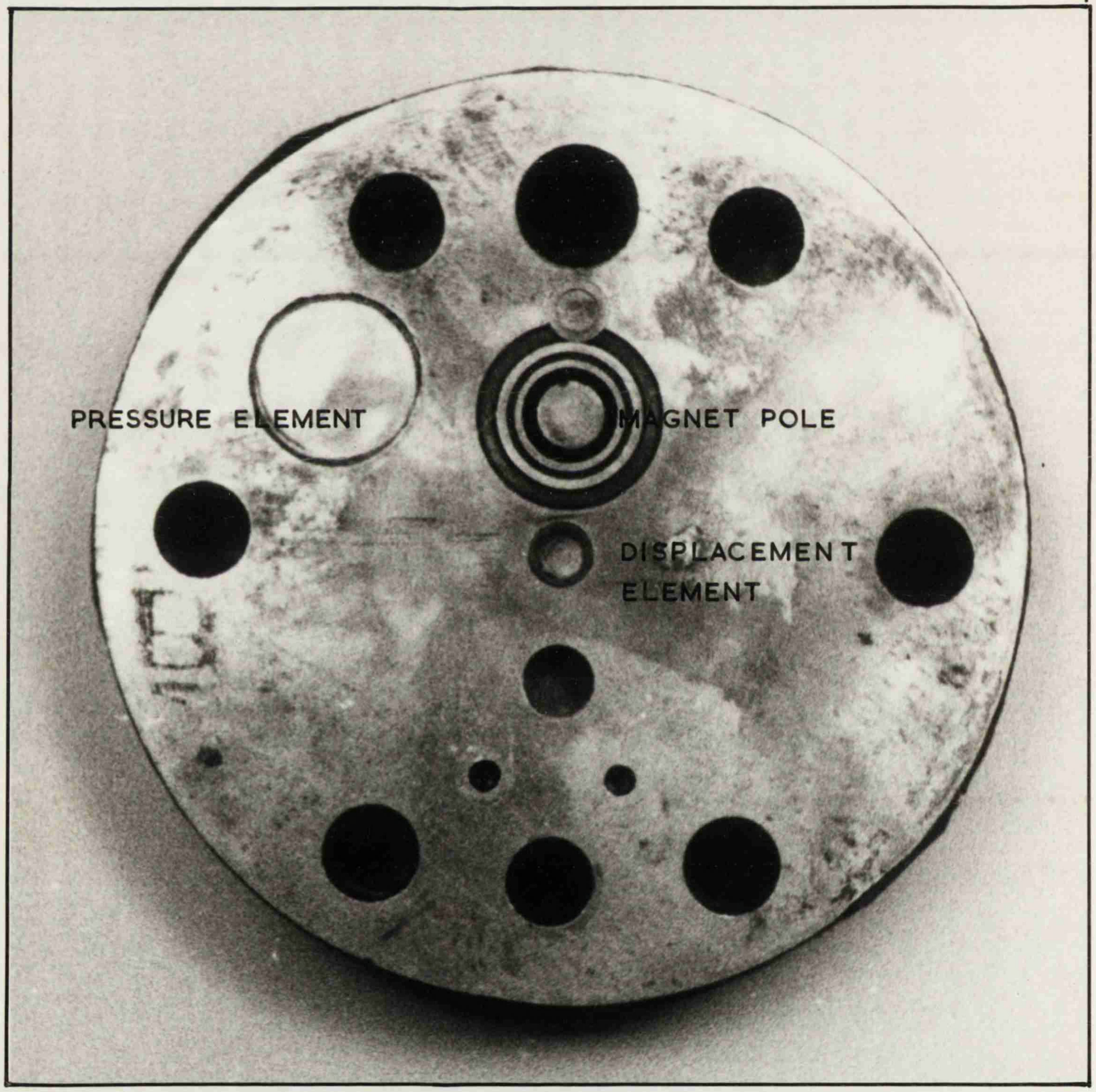
The compressor speed was observed by means of a stroboscope running at mains frequency.

4. 3 Anemometer Equipment.

The main experimental investigation of blowback past the suction valve was carried out by making direct measurements of gas velocity in the suction port.

The instrument used to obtain these readings is an A.C. operated, Constant Temperature Hot-Wire Anemometer, situated in the valve port. The sensitive element of this device is a 0.0005" diameter tungsten wire stretched across the suction port. The wire is mounted on an insulating bush set into the valve plate (Fig. 5) in such a way that the flow pattern is not affected. The wire which forms one arm of a resistive bridge, was heated by a 5 Kc/sec. carrier wave generated by a power oscillator, and the signal produced across the balance points of the bridge used as the input to an amplifier feeding back in series with the oscillator. (Fig. 16). The effect of this system is to maintain the wire at a temperature which is substantially constant, irrespective of the velocity of the suction vapour. The power input to the wire is then a known function of the instantaneous gas velocity, (ref 34) the thermal inertia of the wire having no effect, provided the amplifier maintains the element at a constant temperature. Thus a record of power input during the cycle can be translated to a record of instantaneous gas velocity.

In order to improve the stability of the anemometer system the amplifier contains a stage tuned to 5 Kc/sec. and the circuits are arranged so that, at this frequency, the phase shift through the amplifier is slight.



VALVE LIFTER PLATE

ENLARGED

FIG. 13b

A valve displacement indicator identical to that described in connection with the valve lifter tests was included in the valve plate but no pressure indicators were fitted. Anemometers and displacement elements were fitted to the A and H Compressors, (Figs. 5 & 10) the arrangements being identical except for minor differences due to the relative size of the compressors.

Arrangements for measuring and controlling compressor speed and discharge pressure were the same as for the valve lifter tests.

The quantity flowing in the circuit was again measured by means of a calibrated orifice.

4. 4 Valve Lifter Test Methods.

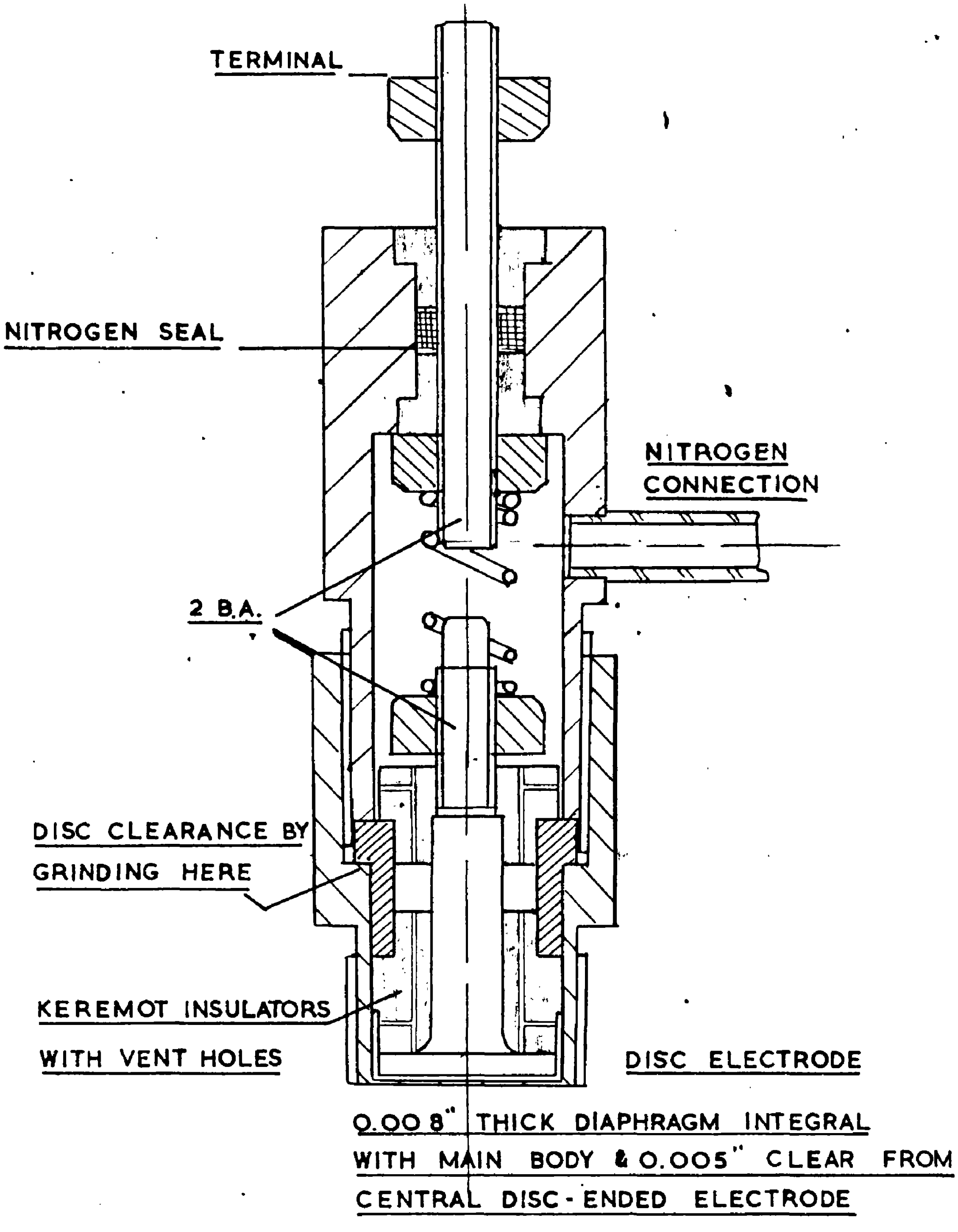
In developing this technique, which is essentially a comparison between the "natural" behaviour of the valve and its behaviour when interfered with in some way, it was desirable to eliminate changes in performance due to other effects such as drift of speed, alteration of cylinder-head temperature or change of clearance volume. In order to do this, tests were run in quick succession after a preliminary warm-up period and speeds were maintained strictly constant by use of the flywheel brake and stroboscope. To obtain the required comparison the compressor was run with the valve controlled magnetically till steady conditions ensued and then the magnet was switched off causing the valve to revert to its normal behaviour. The change in performance was measured by noting the variation in manometer reading when the device was switched off.

If Q is the quantity passing the orifice and H is the head across it in any consistent system of units

$$Q = K\sqrt{H}$$

therefore $$\frac{dQ}{dH} = \frac{K}{2\sqrt{H}}$$

therefore $$\frac{\Delta Q}{Q} = \frac{\Delta H}{2H}$$



PRESSURE SENSITIVE ELEMENT

DESIGNED BY J. BROWN.

FIG. 14

TWICE FULL SIZE

Thus the fractional change in performance due to alteration of valve behaviour can be obtained directly from the manometer readings.

The valve movement and cylinder pressure diagrams associated with various particular conditions were photographed from the oscilloscope if required. A double beam, A.C. Cossor oscilloscope was used in conjunction with a 35 m.m. oscilloscope camera.

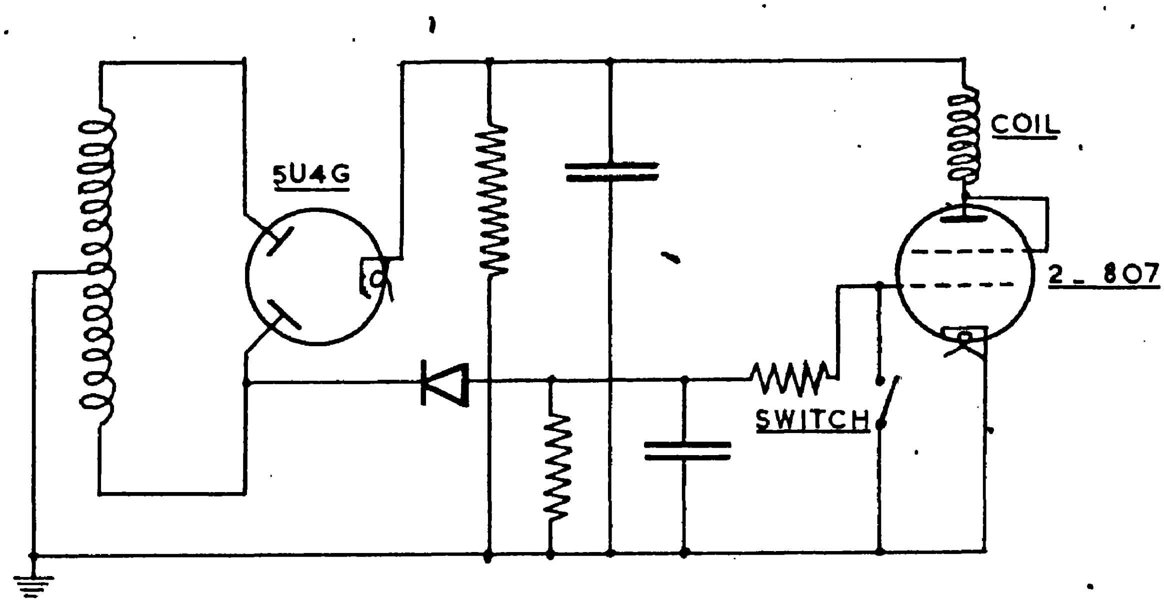
Tests were run to find the positions of the rotary contactor which gave maximum improvement in performance at various speeds and these positions were used to carry out tests at a variety of speeds and pressure ratios. Fig. 18 shows a typical curve of the improvements in performance available at different switch positions.

It is not to be supposed that deduction of blowback by this method approaches the accuracy associated with more direct methods. A summary of the errors which may be involved in estimating blowback from overall performance figures is given in Chapter 5 which deals with the preliminary investigation in detail.

4. 5 Anemometer Test Methods.

As the valve lifter tests cannot be expected to give more than an approximation of the loss due to blowback, the main burden of producing experimental results in support of the theoretical conclusions of Chapter 3 rests on the anemometer method which was used to give a direct indication of the instantaneous gas velocity in the suction port.

The anemometer method was used with both air and Freon 12 but the scope of the Freon tests was restricted by the pressure ratios available with the refrigerant when operating on the non-condensing circuit. The large number of tests required to produce conclusive results indicated that extensive use of the condensing circuit, which had a stabilising time



MAGNETIC VALVE CIRCUIT

FIG. 15

of eight to twelve hours, was impracticable.

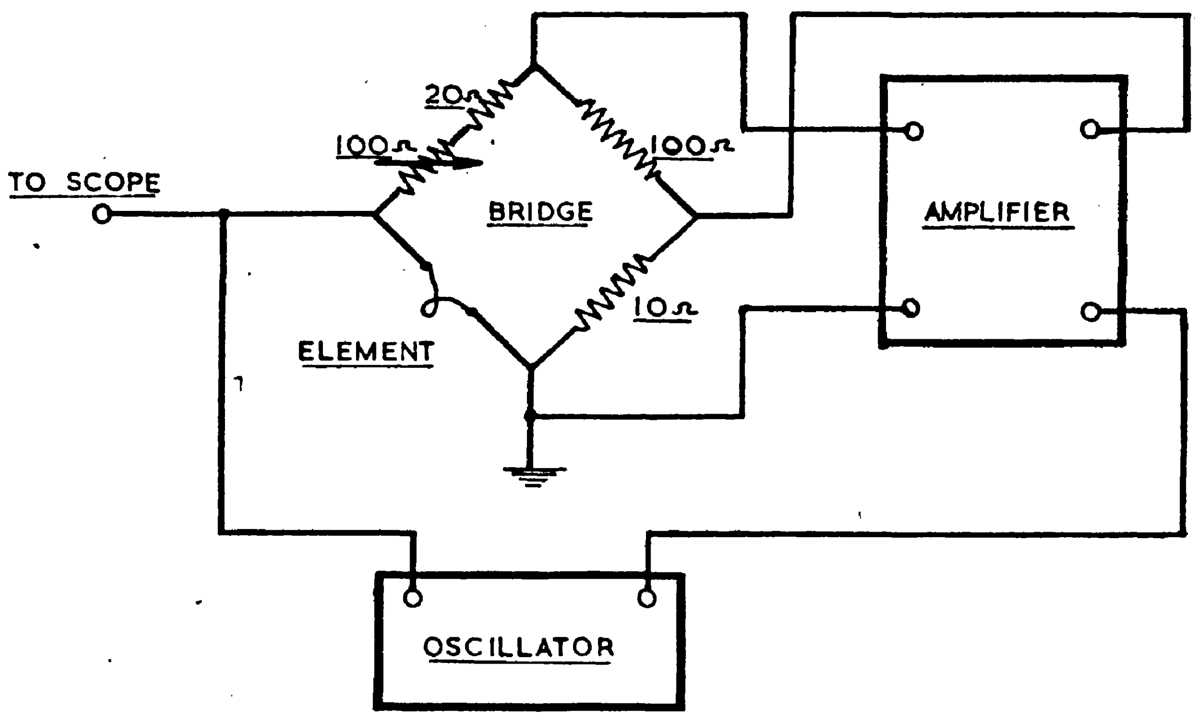
Both the A and H Compressors were indicated using the anemometer but the constant speed motor and the inaccessibility of the H Compressor dictated that a wider range of variables should be applied to the A Compressor.

In variable pressure tests with air the compressor not under test was used to pump air into the system if required.

It was found necessary to run the compressor with a minimum of oil in the crankcase to obtain reproducibility of results. For this reason, in each test, the compressor was allowed to warm up with the oil pump operating and the oil flow was stopped about 20 minutes before test readings were taken. The compressor didnot appear to object to this treatment but the armature current of the driving motor was closely observed for signs of incipient seizure.

Before each test on the A Compressor, the brass valve stop was set to the desired lift using a micrometer depth gauge. The valve plate and cylinder cover were then screwed down until the joint thickness, measured by feelers at four prepared slots in the gasket, was $18. 10^{-3}$ inches.

The compressor was run, with the oscillator and amplifier of the anemometer unit switched on, till steady conditions ensued. The gain of the amplifier was kept at zero during this period to avoid overloading the output stages as the cylinder head temperature rose. The desired constant temperature of the wire was set by adjusting the variable resistor, R1, (Fig. 16) and the oscillator output level altered to produce a mean temperature in this region. Compressor speed and the various circuit pressures were then checked and the approximate manometer readings noted. The gain of the amplifier was then increased till the element began to operate at constant temperature. This condition could be observed by the faithfulness with which the trace recorded the almost instantaneous rise of gas



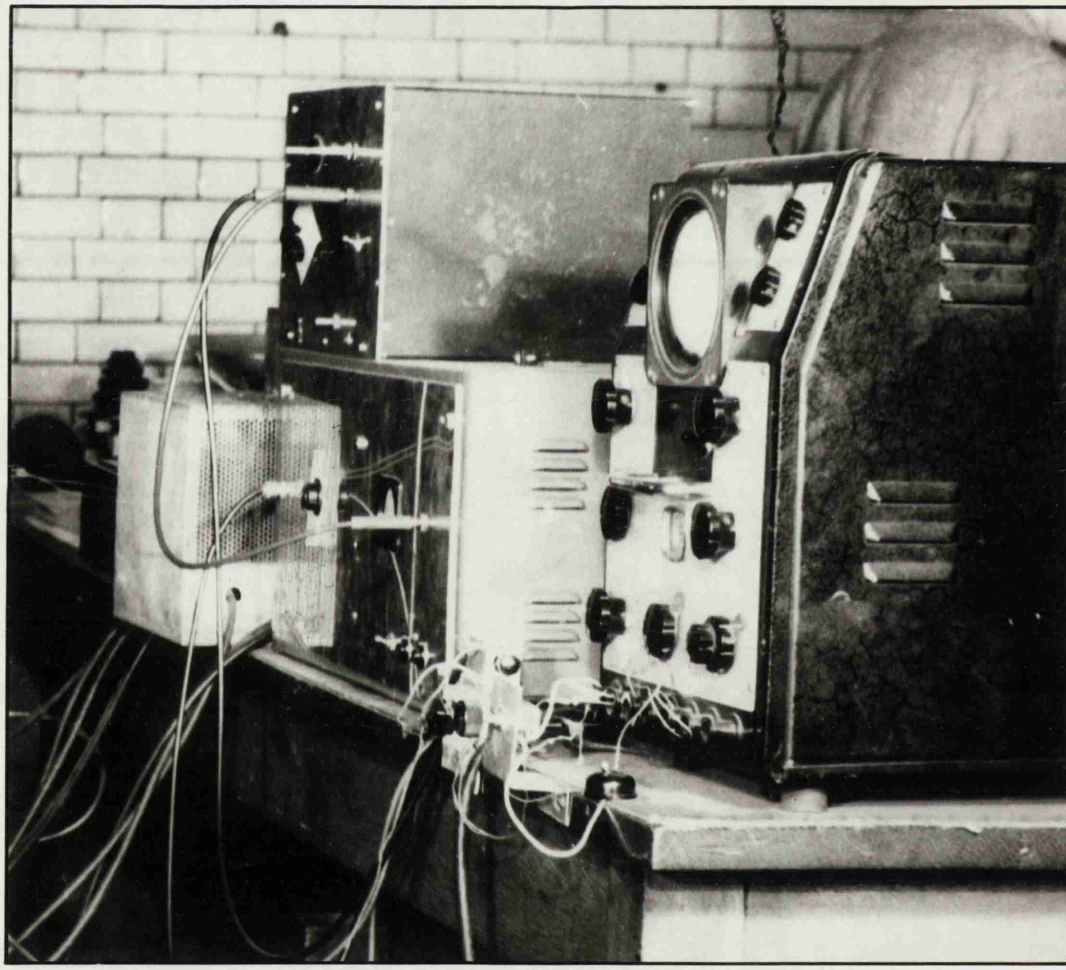
BLOCK DIAGRAM A.C. OPERATED CONSTANT TEMPERATURE HOT WIRE ANEMOMETER

FIG. 16

velocity on opening of the suction valve. However the instrument was not considered to be operating satisfactorily till a condition was reached where increases in gain made no observable change in the recorded velocity pattern. When the instrument was recording correctly the oscilloscope trace was photographed and the orifice manometer readings noted.

The permanent record produced by this method appears on the film as a modulated envelope which is related to the instantaneous gas velocity by King's equation (Ref. 31). This modulated envelope is usually accompanied by a rectified and demodulated trace of valve movement with a phasing blip superimposed at top dead centre. As the valve movement diagram is principally used to observe the phase angles of the various valve motions and was not calibrated in terms of actual displacement, it could be observed on the A2 trace of the oscilloscope, the distortion caused by the coupling of this channel not affecting such measurements.

To obtain direct readings of velocity it is necessary to linearise the trace recorded from the instrument. This can be done by electronic means (Refs. 33 & 39) but it was found that the computing circuits involved were not sufficiently accurate over the range of signal involved. In particular, the behaviour of the required squaring circuits when dealing with very small signals made their value questionable. A note on the behaviour of these circuits is included as Appendix V. Unfortunately the part of the signal which is of greatest interest with reference to blow-back varies from zero to a quantity which is generally small in comparison with the maximum forward velocity, Thus other methods of obtaining a linear velocity trace had to be adopted. The most satisfactory of these was to trace the enlarged photographic records on to specially prepared worksheets (Fig. 42). The amplitude of the trace was then squared using



ELECTRONIC EQUIPMENT

FIG. 17

dividers and a suitable parabolic curve. The resulting curve was effectively demodulated by removing the constant term and then squared again. Thus, assuming King's equation (Refs. 31 & 34),

$$I^2R = A + B\sqrt{V}$$

where I^2R = instantaneous power supplied

V = instantaneous gas velocity

and A & B are constants,

is a good approximation for the constant temperature hot-wire anemometer, a curve has been prepared which is directly proportional to velocity. The scale of this curve can be calculated from the total flow quantity as given by the orifice. Relative flow quantities are obtained by measuring the areas under the linearised curve by planimeter. To ensure consistent treatment of results the primary curve was traced from the edge of the envelope and no attempt was made to compensate for the trace thickness. Fortunately, one effect of using a modulated signal is to produce a rapidly moving spot and consequently, a fine trace. The centre-line of the rectified and demodulated trace of valve movement was taken as being the correct reading.

The types of record obtained by this equipment are shown in Fig. 19. 19A is the signal from the anemometer before linearising, the horizontal axis being time. The quantity marked θ' is a measure of the loss due to re-expansion of clearance volume vapour. θ'' represents the time elapsing between bottom dead centre and reversal of gas in the valve port. This may be due to gas inertia effects or to the effects of throttling at the valve. Other considerations indicate that the latter is the more important effect. Though the instrument was not specifically designed to display these compressor losses, the ease and accuracy with which they can be read

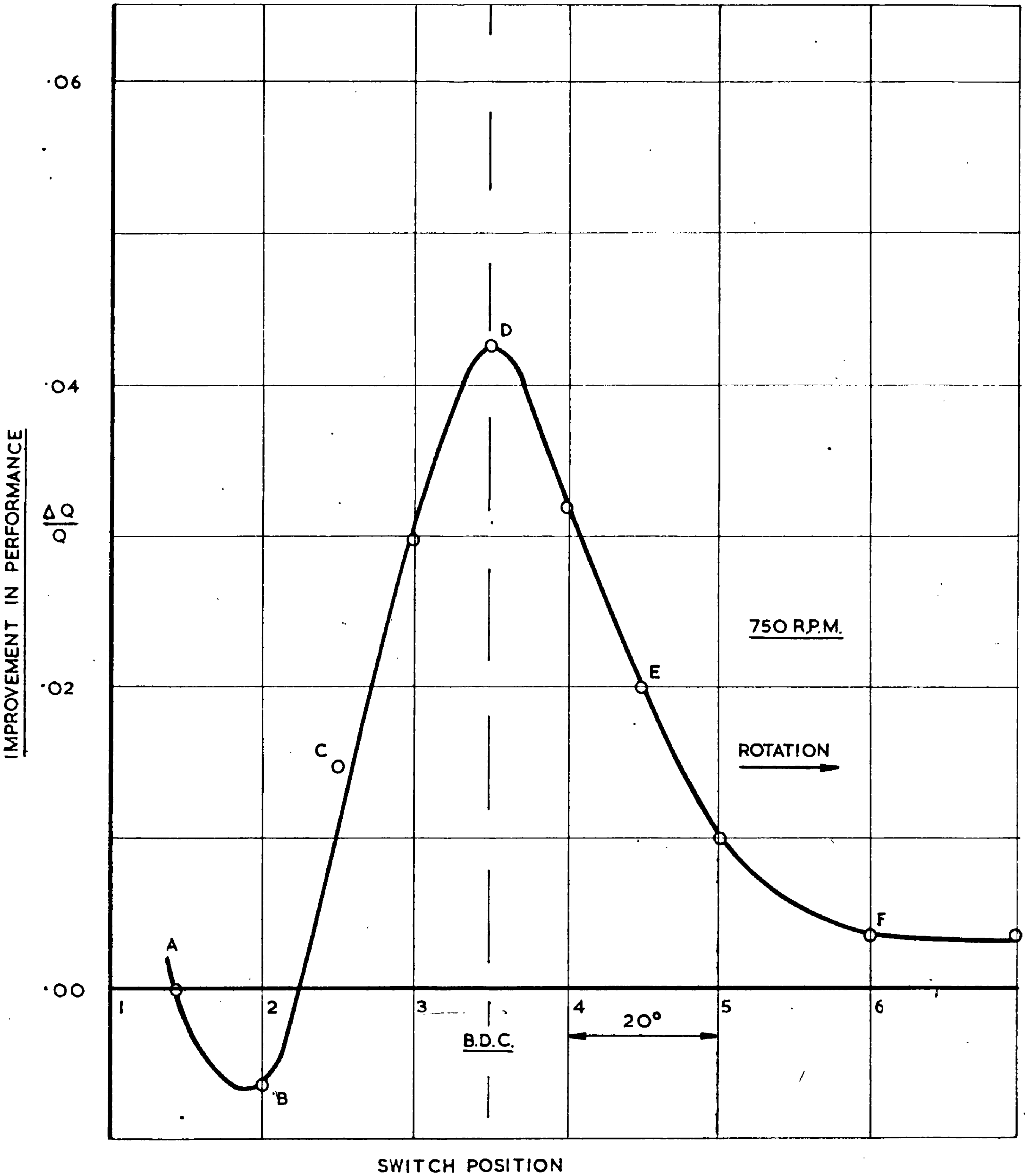

·06
·04
·02
·00
IMPROVEMENT IN PERFORMANCE
ΔQ/Q
A
B
C
D
E
F
1
2
3
4
5
6
B.D.C.
20°
750 R.P.M.
ROTATION
SWITCH POSITION

FIG. 18

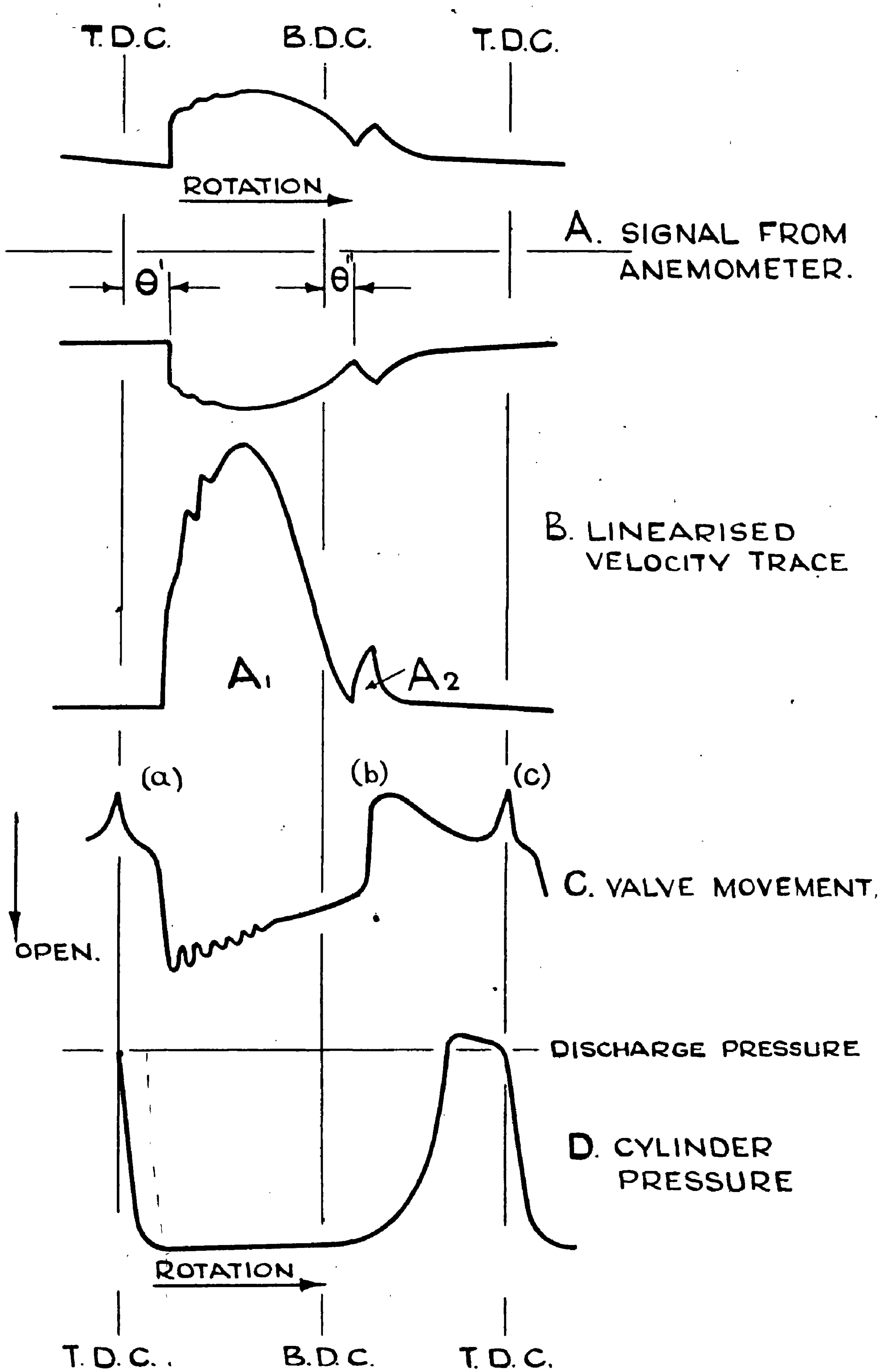

T.D.C.
B.D.C.
T.D.C.
ROTATION
A. SIGNAL FROM ANEMOMETER.
θ'
θ''
B. LINEARISED VELOCITY TRACE
A1
A2
(a)
(b)
(c)
C. VALVE MOVEMENT.
OPEN.
DISCHARGE PRESSURE
D. CYLINDER PRESSURE
ROTATION
T.D.C.
B.D.C.
T.D.C.

FIG. 19

from such a diagram contrasts sharply with the procedures and calibrations required to estimate them from, for example, a pressure/time diagram.

19B is the velocity trace linearised according to King's equation. The area A1 represents the volume of gas induced during the suction stroke. The area A2 represents the quantity of gas blown back past the suction valve before it closes. By allowing for re-expansion and throttling losses the blowback quantity can be calculated on a percentage basis. 19C is a record of valve movement and phase obtained directly from the displacement element. This trace is distorted vertically due to the coupling of the A2 trace of the oscilloscope but the phasing is not affected. When required, an accurate displacement diagram was obtained using the A1 trace which had a better frequency response. 19D is a curve of cylinder pressure recorded from a capacitative pressure pickup. This is included for comparison only and such diagrams were not generally used in the anemometer tests.

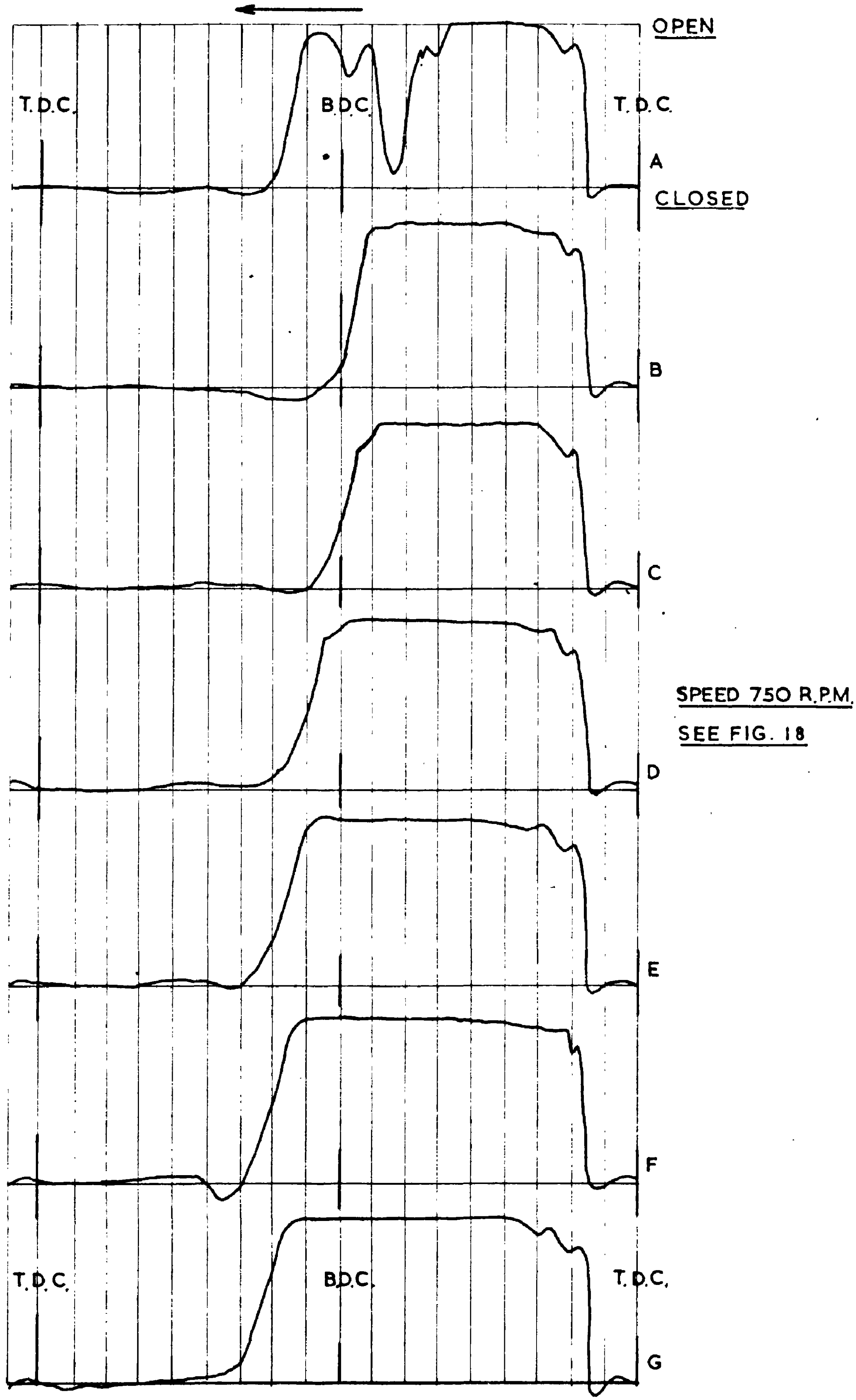


FIG.20 VALVE MOVEMENT DIAGRAMS

CHAPTER 5

EXPERIMENTAL INVESTIGATION USING ELECTROMAGNETIC VALVE CONTROL

This preliminary investigation was carried out in order to discover if blowback quantities were sufficiently large to warrant more detailed investigation.

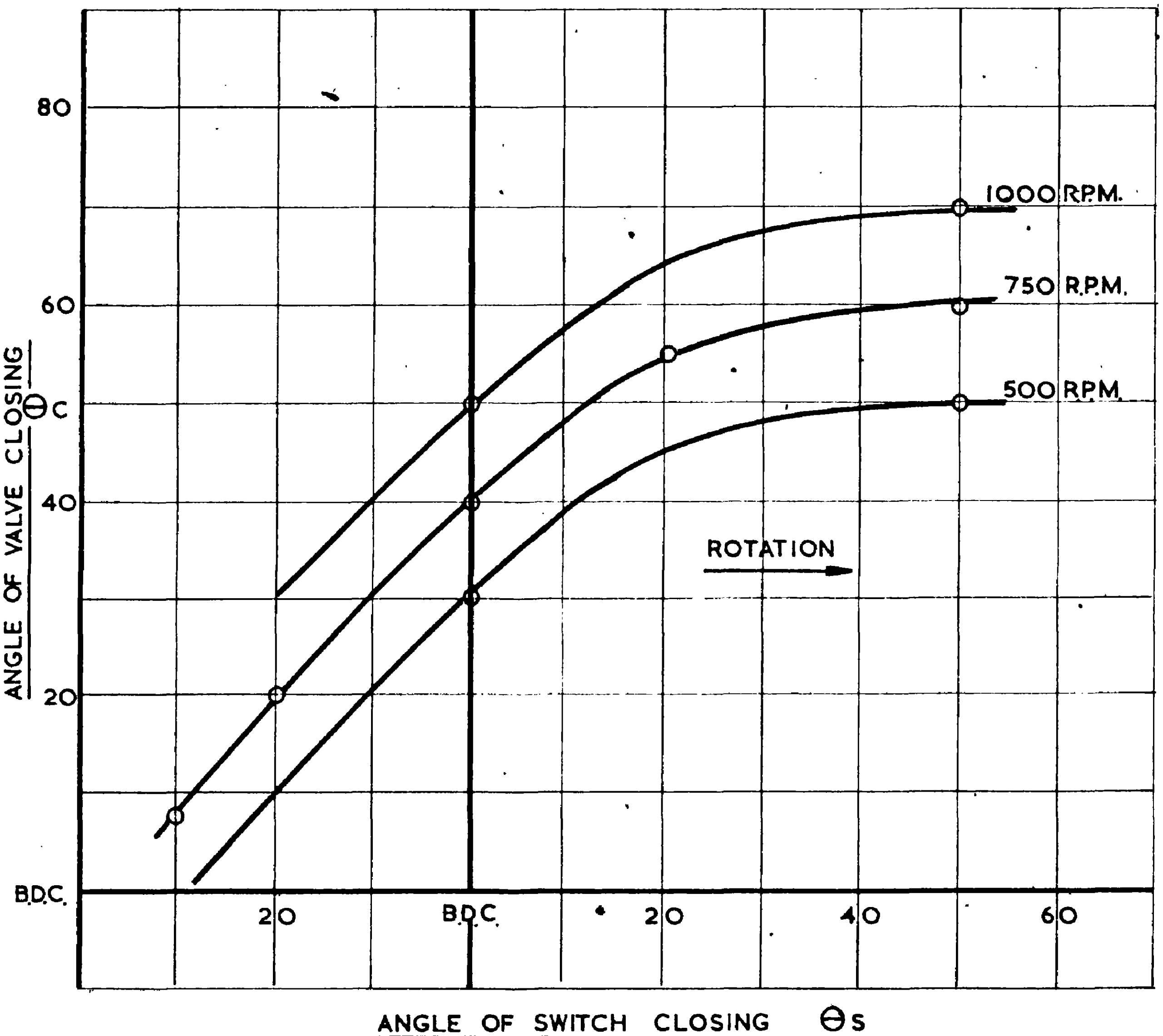
5. 1 Scope.

The experiments were carried out using the valve lifter apparatus and techniques described in Chapter 4. The valve lifter method was applied to the suction reed of the A Compressor with air as the working fluid.

The first tests were made in order to find the positions of the rotary switch which gave maximum improvement in performance. These phasing tests were run at speeds of 500, 750 and 1,000 r.p.m. Photographs of the valve movement during these tests were taken from the oscilloscope screen, (Fig. 29). When the optimum switch positions had been established, tests were run at speeds varying from 400 to 1,200 r.p.m. in order to find the greatest improvements in performance available at various speeds. A test was also run at 600 r.p.m. with discharge pressures varying from 20 lb/in^2 to 100 lb/in^2. Photographs of valve movement diagrams and cylinder pressure traces weretaken for representative tests in each series.

5. 2 Experimental Results.

The relationship between improvement in performance and position of the rotary switch when causing this improvement is graphed in Figure 18. The valve movement diagrams, (Fig. 29) were used to construct curves which showed, for various speeds, the actual phase angles at which the valve closed (θc) against the angles at which the switch was set to contact, (θs). These curves are shown in Figure 21. The curves of Figure 21 were then used to



EFFECT OF MAGNETIC ACTION ON VALVE CLOSURE

FIG. 21

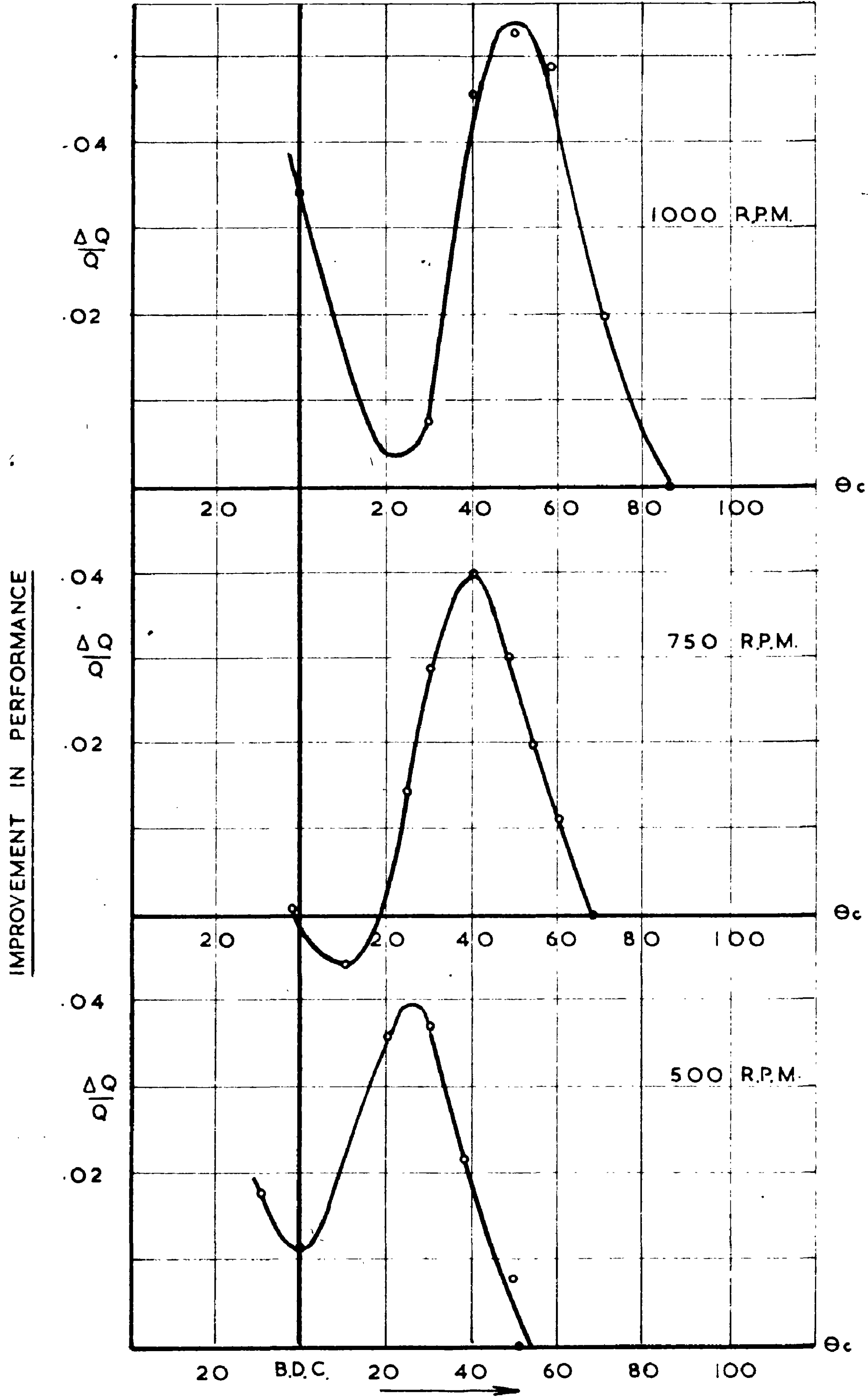
plot improvements in performance obtained by use of the switch at various speeds against the actual angle of valve closure, (θc), resulting from the action of the magnet. As θc cannot be estimated with great accuracy, some scatter has been introduced to the curves which are shown in Figure 22.

An attempt was made to estimate the manner in which blowback varied with speed and discharge pressure but the results did not give any clear support to the theory of Chapter 3. The reasons for this failure are considered below.

5. 3 Discussion of Results.

General Remarks.

All estimations of blowback made by this method depend on the tacit assumption that, if improvements in performance can be made by advancing the closure of the suction valve, then these improvements are due to reduction of blowback. If it is further assumed that the maximum improvement in performance occurs when the valve closes at a position which eliminates the blowback without affecting the throttling loss then the improvements produced are a measure of the blowback loss. Unfortunately the second assumption is not entirely valid, there being no doubt that the operation of the valve lifter has a considerable effect on the throttling loss. This explains the difference between curves of $\frac{\Delta Q}{Q}$ produced by this method and the predicted blowback curves. However it is safe to assume that operation of the magnet will increase the throttling loss, giving rise to $\frac{\Delta Q}{Q}$ figures which are actually less than the blowback occurring. By this reasoning it was assumed that blowback was at least as great as the maximum $\frac{\Delta Q}{Q}$ obtained by use of the magnetic valve lifter. Later direct experiments confirmed this. The performance changes were noted on a water manometer as changes of liquid level in one limb $(\frac{\Delta H}{2})$. This quantity varied during the tests



EFFECT OF CLOSING ANGLE ON PERFORMANCE

FIG. 22

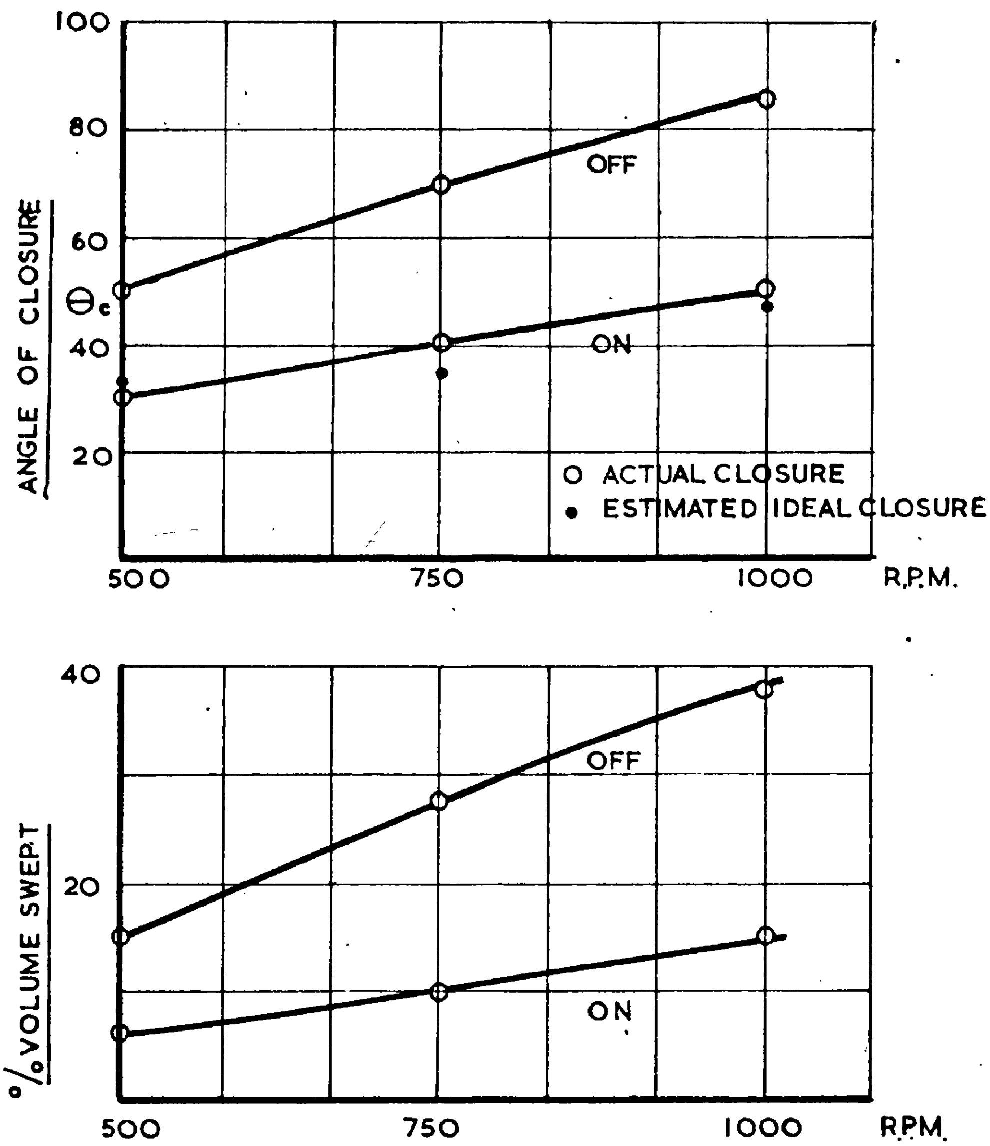
from about one quarter of an inch to two inches. Thus, assuming that the scale could be read to 0.05" with accuracy, the error involved may range from 20% to 2.5%. As this range is comparable with the variation of blowback expected in the tests it can readily be seen that the apparatus can provide no more than a rough guide to the quantity of blowback occurring. Errors due to variation of the orifice coefficient variation of compressor speed and other causes are slight compared to the inaccuracy attributable to this cause. In practice the scatter of experimental results appeared to be less than 20%.

Phasing Tests.

These tests were run with varying speed and switch position. Figure 18 is a typical curve of performance improvement against switch position. At switch positions around 60° before B.D.C. there is some improvement in performance because, though the magnet closes the valve long before the ideal closing position, the throttling of gas into the cylinder is so severe as to re-open the valve which closes for a second time nearer the ideal position with consequent enhanced performance. This effect is not of great importance, as the angle at which the valve closes for the second time depends on compressor speed, valve natural frequency, oil damping and other factors. The valve movement diagram for this type of operation can be seen in Figure 20A.

For switch position 2, 30° before B.D.C., there is a slight <u>decrease</u> in performance; this is due to throttling of the gas flowing into the cylinder which is not alleviated by re-opening of the valve. The valve movement diagram can be seen in Figures 20B.

As the switch position approaches B.D.C. the performance improves due to reduced throttling and, the performance reaches a maximum when the valve



EFFECT OF MAGNET ON CLOSURE

FIG. 23

closes at a point where the combined loss due to the throttling and blowback effects is a minimum. This should occur when the gas flow through the suction port is just about to reverse and is the ideal closing point. The difference between the actual and the ideal closing point is assumed to be a measure of blowback. Figures 20C and 20D show the valve approaching ideal closure. The pressure diagram showed that throttling was slight at the ideal closing position.

Beyond the ideal closing position, throttling is assumed to be slight and blowback is present as a function of the time the valve remains open during reversed flow. A valve diagram for this condition is seen in Figure 20E.

When the switch fires at position 6 the closure of the valve is not materially affected. The slight improvement in performance observed is probably due to residual magnetism effects. Figure 20F gives this valve diagram. The natural valve diagram is given in Figure 20G.

Figures 22A, 22B and 22C show the performance improvement to a base of actual closing angle at various speeds. This shows the ideal closing angle to be dependent on speed. The values to the left of the minimum value in each curve should be treated with reserve as the time of closure is not entirely predictable in this region.

Figure 21 shows the angle at which the valve closes under action of the magnet, graphed against the angle at which the magnet was switched on. The curves tend to a closing angle slightly before the natural closing time and this is probably due to the fact that when the coil is operating there is enough residual magnetismpresent to accelerate the final stages of closure even if the switch is firing beyond the natural closing point.

Figure 23A relates the angle of closure of the valve to compressor speed. It is interesting to compare the measured ideal closing positions (the lower curve) with the estimated values which were obtained from the normal valve diagrams as the point at which the valve appeared to start closing under the action of cylinder pressure. Figure 23E expresses the same results in terms of stroke volume.

The improvement in performance obtainable by this method varied between five and six percent. Improvements in performance were consistently greater at the higher speeds where theory suggests that blowback should be decreasing. This indicates that the valve lifter technique introduces throttling losses which are not insignificant, particularly at low speeds where the "natural" throttling effect is least pronounced.

It must also be remembered that the magnetic pole alters the cross section of the port in a way which is presumably detrimental to efficiency. However, the magnetic method of altering valve behaviour which was selected as being the least likely to have an adverse effect on compressor efficiency, has several important advantages. It allows the timing of the valve to be altered, while the machine is running, by a rotating contactor external to the compressor. Thus, performance readings can be taken in quick succession ensuring that test conditions do not alter during the experiment. As the maximum improvement in performance obtained by altering valve timing is of the order of five percent it is vital that other effects influencing the performance be controlled. The magnetic method has the advantage that the two performance figures, which are to be subtracted, are obtained at the same time with the same valve reed and gasket without disturbing the head, at the same temperature, and with the same amount of oil present. Each test is run with the valve controlled magnetically and then the magnet is switched off causing the valve to revert to its normal behaviour.

The main disadvantage of the method is that overall performance figures have to be used to estimate the effect of a single loss which is merely a fraction of the total loss associated with the compressor. This disadvantage, of course, is not peculiar to the magnetic valve lifter technique but will afflict all techniques which do not face up to the problem of making a direct estimation of the individual losses. A method of making direct measurements of blowback past the suction valve was devised and is described in the next Chapter.

CHAPTER 6

EXPERIMENTAL INVESTIGATION OF FACTORS AFFECTING BLOWBACK PAST THE SUCTION VALVE, USING HOT-WIRE ANEMOMETER.

The experiments described in this Chapter were undertaken in order to provide a direct experimental check on the results which can be predicted using the theoretical approach of Chapter 3.

6. 1 Scope of Tests.

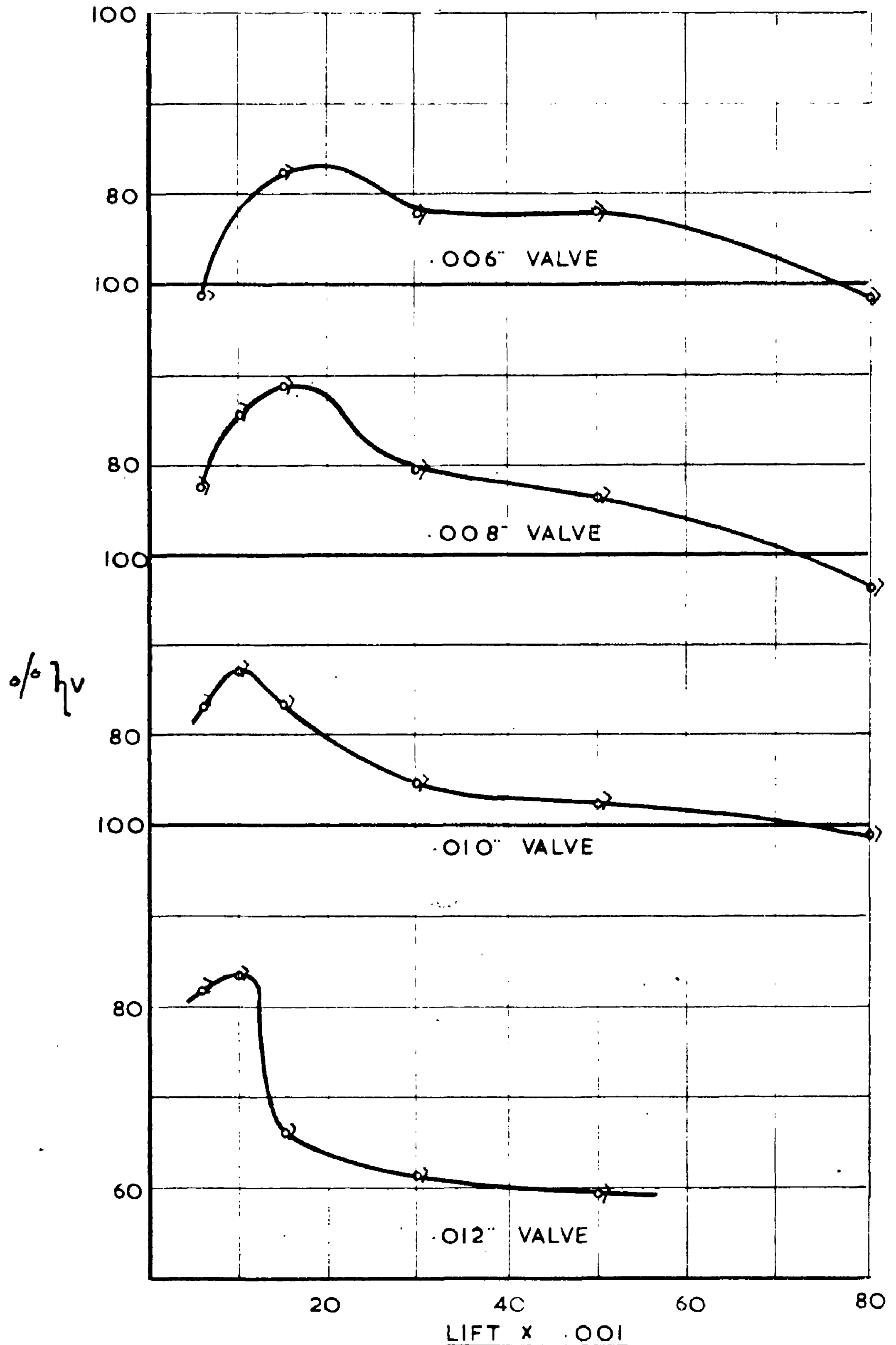
The anemometer equipment described in Chapter 4 was used to provide diagrams of instantaneous gas velocity in the suction ports of the compressors investigated. When required, additional information was obtained by indicating the movement of the valves under examination using a capacitative type pick-up.

The capacity of the compressor was estimated by means of a calibrated orifice.

The variation of volumetric efficiency of the A Compressor with valve lift was noted at speeds of 500, 750 and 1000 r.p.m. using various thicknesses of valve. The records of efficiency variation obtained in this way were interpreted in the light of information obtained from the anemometer records as the variation of blowback with lift under similar conditions. Air was used for these tests.

The effect of compressor speed on blowback was also examined using both Air and Freon 12 as the working fluids.

It was soon observed from the anemometer records that the reversal of gas flow into the cylinder often took place some time after bottom dead centre. This was assumed to be due mainly to the effects of throttling of the inlet process and, assuming the supercharging effect of the inlet gas velocity to be slight, curves were drawn to show the effect of the valve



EFFECT OF LIFT ON VOLUMETRIC EFFICIENCY AT 500 R.P.M.

FIG 24

lift on the throttling loss with air as the working fluid.

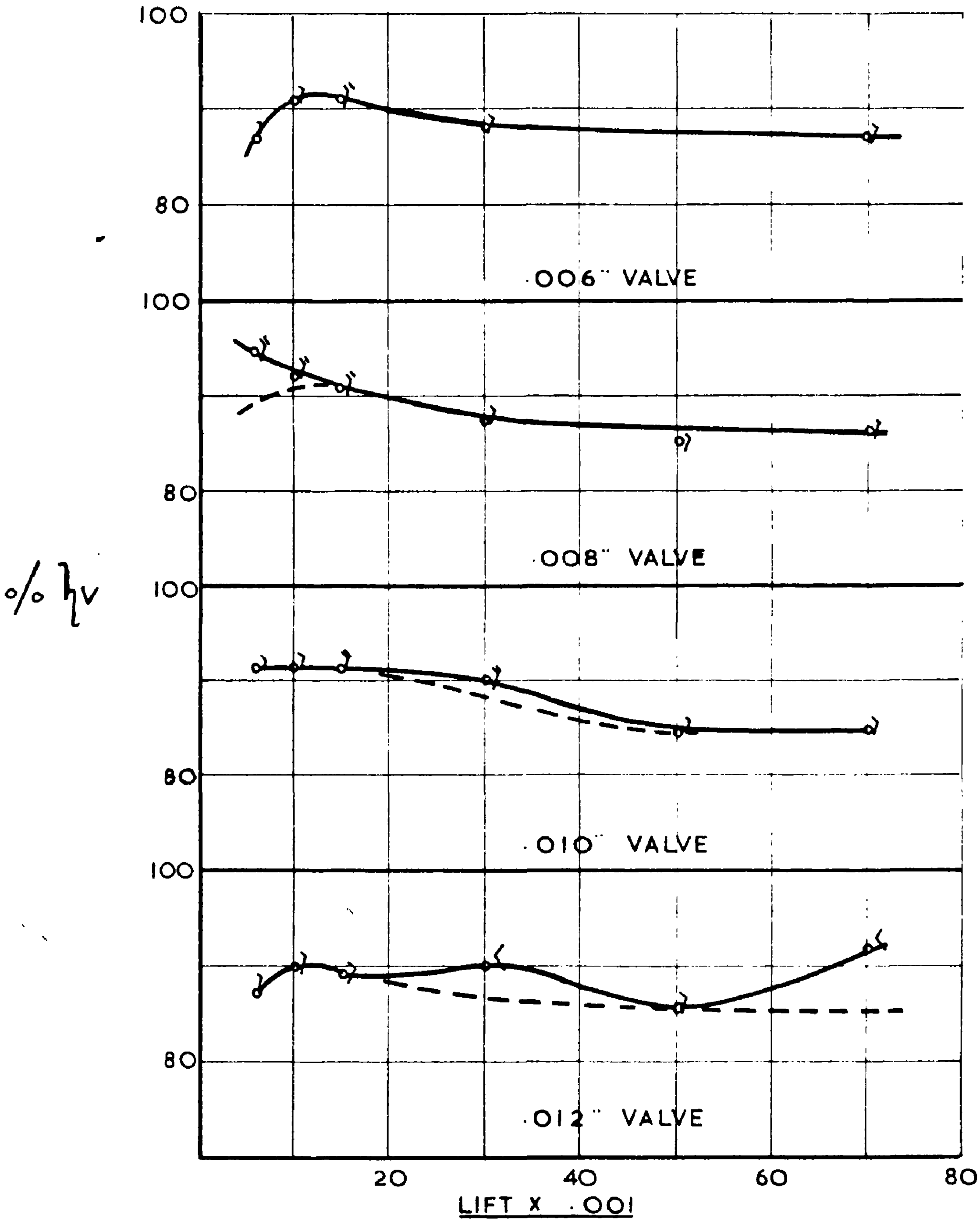
As the theory of Chapter 3 indicated that there would be a variation of blowback quantity with fluid density, a series of tests was carried out using a constant pressure ratio of three and varying the absolute suction pressure between wide limits. Air was again used as the working fluid in the majority of these tests though a few comparable tests were run using the dense refrigerant Freon 12 in a highly superheated condition.

In order to conform to general practice the suction reeds were not clamped firmly over the suction port but rested loosely on their locating pins. To observe the effect of clamping the reed in place, as recommended by MacLaren, the end freedom of the valve was progressively reduced by building up the thickness of the "hinge" part of the reed till it was effectively clamped on compression of the gasket. The effect of this clamping on blowback and valve movement was noted.

Tests were also run using the H Compressor, indicating both instantaneous gas velocity and valve movement. Owing to the single phase A.C. motor employed and the relative inaccessibility of the cylinder head these tests were restricted to observing the effects of suction density and pressure ratio on blowback and valve behaviour.

6. 2 Experimental Results.

In many cases the results obtained in this section of the work can be understood more fully if the manner in which the suction reed behaves is known. Chapter 3 contains a description of the various ways in which such a reed can behave, illustrated by figure 2. The various types of behaviour have a material influence on the overall performance of the compressor and on the blowback quantities. For this reason symbols have been devised so that the type of valve behaviour pertaining to any particular



EFFECT OF LIFT ON VOLUMETRIC EFFICIENCY AT 750 R.P.M.

FIG. 25

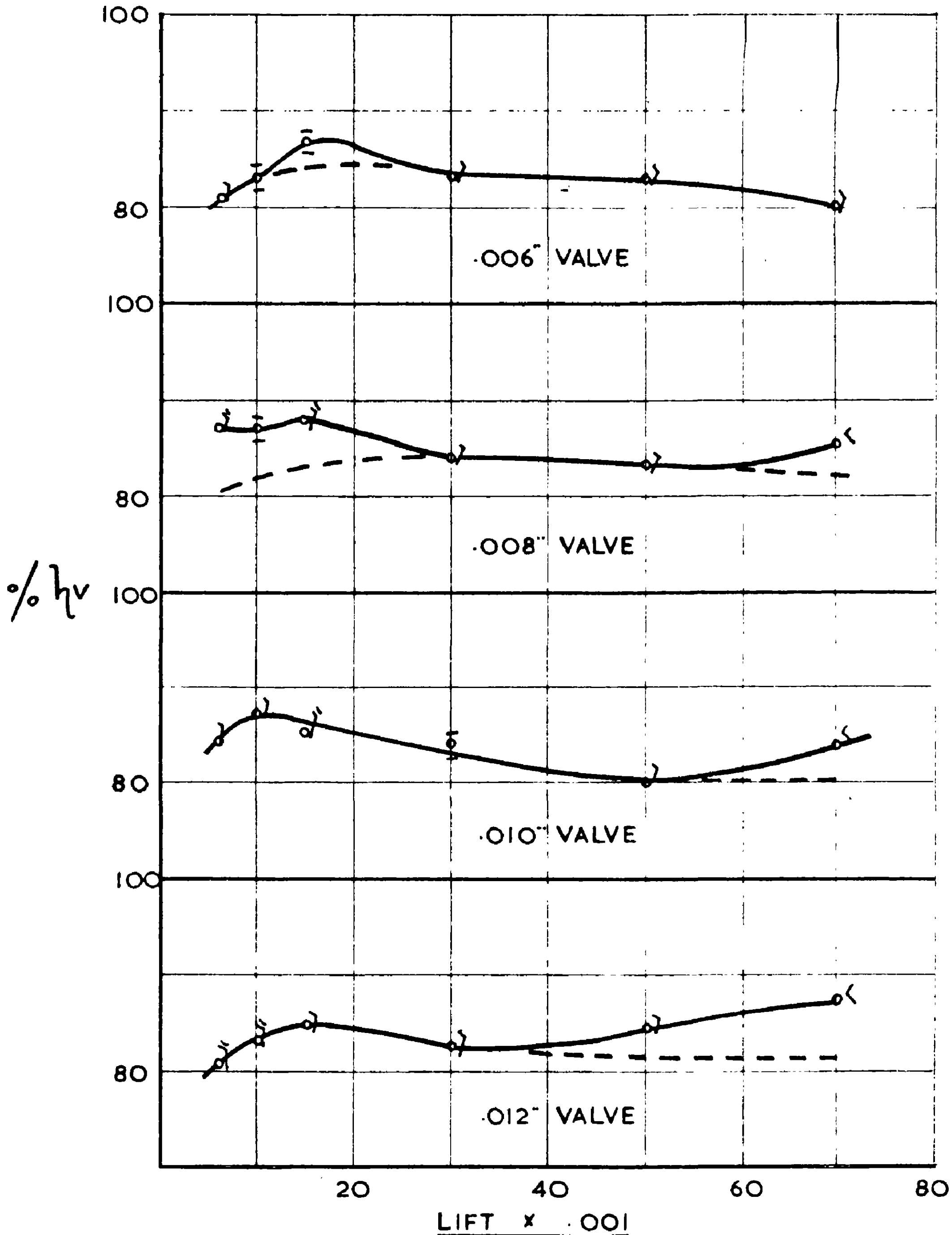
experimental point may be illustrated.

On opening and striking the limit stop the suction valve is almost invariably excited into an oscillation of some form. If the oscillation is a damped transient which disappears before the angle at which the valve begins to close then the experimental point has been distinguished by the sign > . If oscillation is obviously transient but persists during the period which the valve is open, the experimental point is marked ≫ . Generally speaking it can be assumed that such oscillations will have slight effect on compressor performance. If the oscillation is maintained at constant amplitude during the period in which the valve remains open, the experimental points are marked = . This type of oscillation can be expected to affect the performance of the compressor. In the final case, where the oscillation is divergent and of large amplitude during the valve's open period, the experimental points are marked < . This type of oscillation has a profound effect on compressor performance.

6. 2a Effect of Valve Lift on Volumetric Efficiency.

The results of these tests which were run with valves of varying thickness at speeds of 500, 750 and 1000 r.p.m. are shown in Figs. 24, 25 and 26. Figure 24 shows the results obtained at a speed of 500 r.p.m. At this speed there was very little tendency for the valve to break into oscillation. This uniformity of valve behaviour results in reduced scatter of the results.

In each case there is a marked rise in volumetric efficiency as the valve lift is reduced to about ·010 inches. Below that lift the efficiency falls rapidly due to throttling. This improvement in performance is due to the great reduction of blowback at low lift and was commented on by MacLaren. In the figures produced by MacLaren the performance improvement



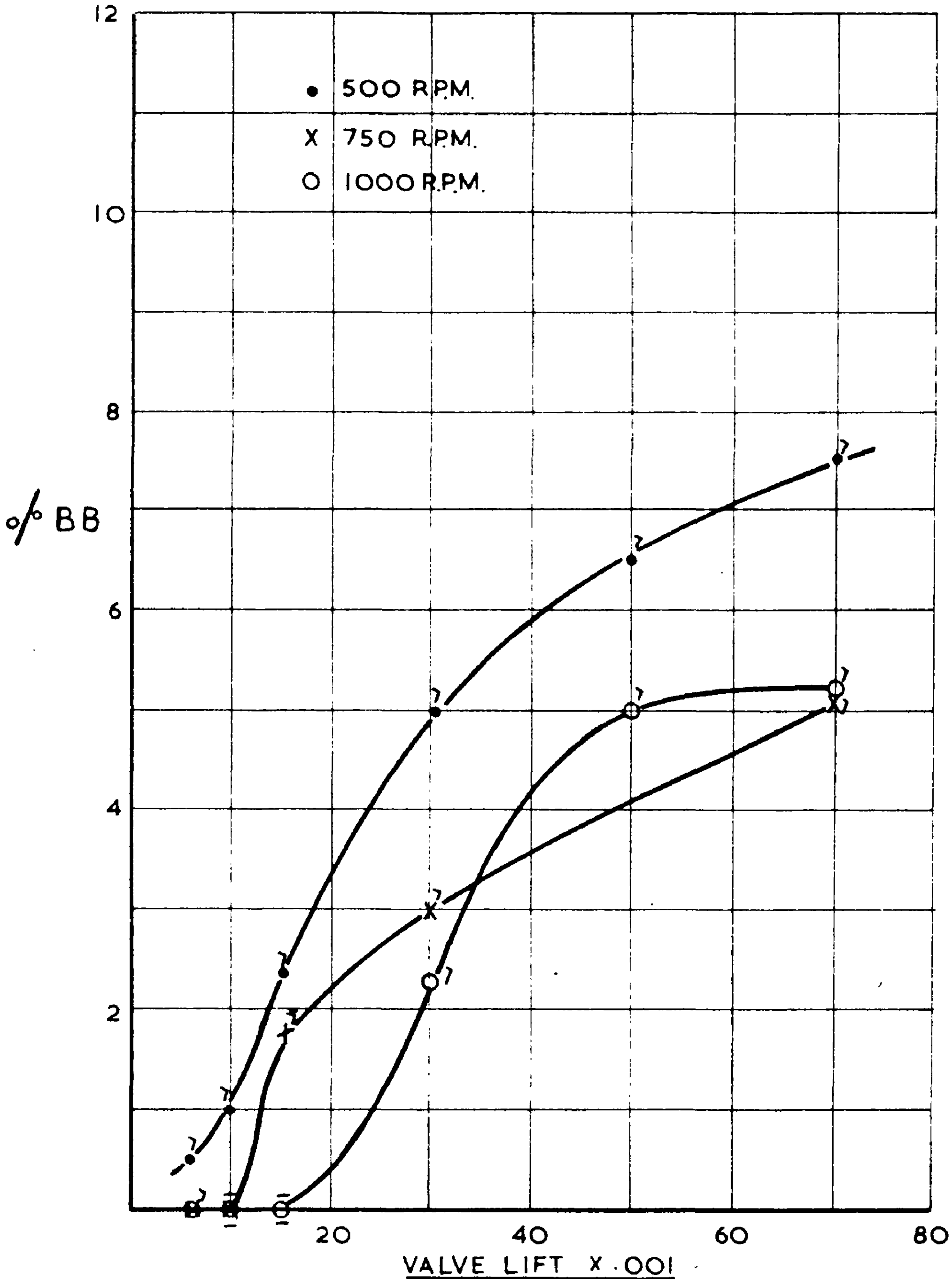
EFFECT OF LIFT ON VOLUMETRIC EFFICIENCY AT 1000 R.P.M.

FIG. 26

was not so marked especially in the case of the heavier valves, which have greater stiffness. This is presumably due to the manner in which these valves were clamped. In the present series of experiments where the valves are loosely set on the locating pins, it is to be expected that blowback will be greater, particularly with the heavier valves. This is clearly shown in the results of Figure 24 where the performance of the heavy valve is seen to be much inferior at high lifts. It is noteworthy that the maximum efficiency attained with each valve is roughly the same though, again, the lighter valves appear slightly better and attain their peak efficiencies, as would be expected, at higher lifts.

Figure 25 shows similar results at 750 r.p.m. though, in this case the curves are affected by the results of valve oscillation. The actual curves obtained have been drawn in full and the shape which it is presumed the curves would have taken but for the oscillations is shown dotted. It should be noted that the usual effect of oscillation is an improvement in performance owing to reduction of blowback, though this is normally offset by increased valve noise and greater power consumption. Oscillation usually causes an increase of throttling loss and in a modern high speed compressor, where blowback is slight, this has been known to result in impaired volumetric efficiency.

Figure 26, which shows the results for a speed of 1000 r.p.m. is similar to the preceeding diagrams, though again affected by oscillation. It should be noted that the difference in performance between heavy and light valves and the amount of improvement available at the best value of lift are both much less at the higher speeds. This suggests that the blowback is more significant at low speeds. This impression is confirmed by later results and agrees with the theory of Chapter 3.



EFFECT OF VALVE LIFT ON

% OF INDUCED VOLUME BLOWN BACK

FIG. 27 ·006 VALVE

6. 2b Percentage blowback as affected by permitted lift.

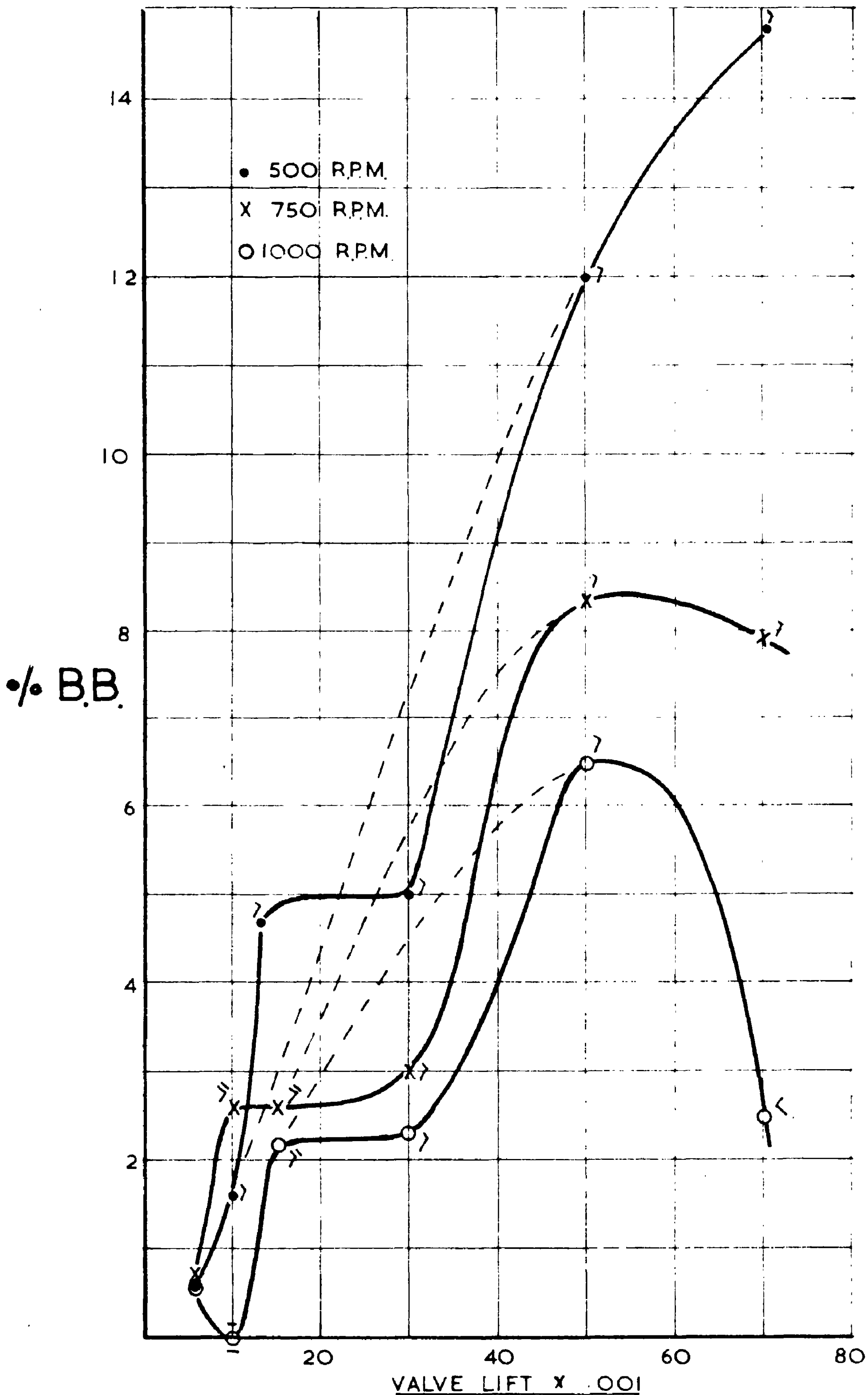
In these tests, which were run with valves varying in thickness from 6 to 10 thousandths of an inch, the lift was varied from ·006" to ·070" by means of the threaded limit stop. Tests were again run at 500, 750 and 1000 r.p.m. The discharge and suction pressures were held constant and blowback was estimated from the areas under the linearised velocity diagram, as a fraction of the volume of gas induced. When the other losses associated with the particular conditions under investigation are known, it is possible to convert the fractional blowback into a percentage loss of volumetric efficiency. This was not generally done when graphing blowback results because of the amount of scatter introduced from sources not connected with the actual blowback measurements. The equation used to obtain the absolute blowback quantities was -

$$\text{volume induced (inches}^2) - \text{volume blowback (inches}^2)$$

$$\equiv \text{Piston leakage (ft.}^3\text{/sec.)} + \text{Flow quantity (ft.}^3\text{/sec.)}$$

However, provided the pressure ratio was kept constant and the index of re-expansion did not change, it was considered that a reasonable estimate of the manner in which blowback varied could be obtained by simple comparison with the induced volume.

Figure 27 shows the results obtained with the ·006" valve. It can be seen that the lower valve lifts have tended to produce oscillation which affects the blowback and destroys the smoothness of the curves. As each result is obtained from careful measurement of a photographed record showing the velocity trace during the test it has been assumed that all results are valid and no attempt has been made to shape the blowback curves to conform to any theory or pattern. It is interesting to note that this valve was free from serious oscillation at high lifts where the blowback



EFFECT OF VALVE LIFT ON

% OF INDUCED VOLUME BLOWN BACK

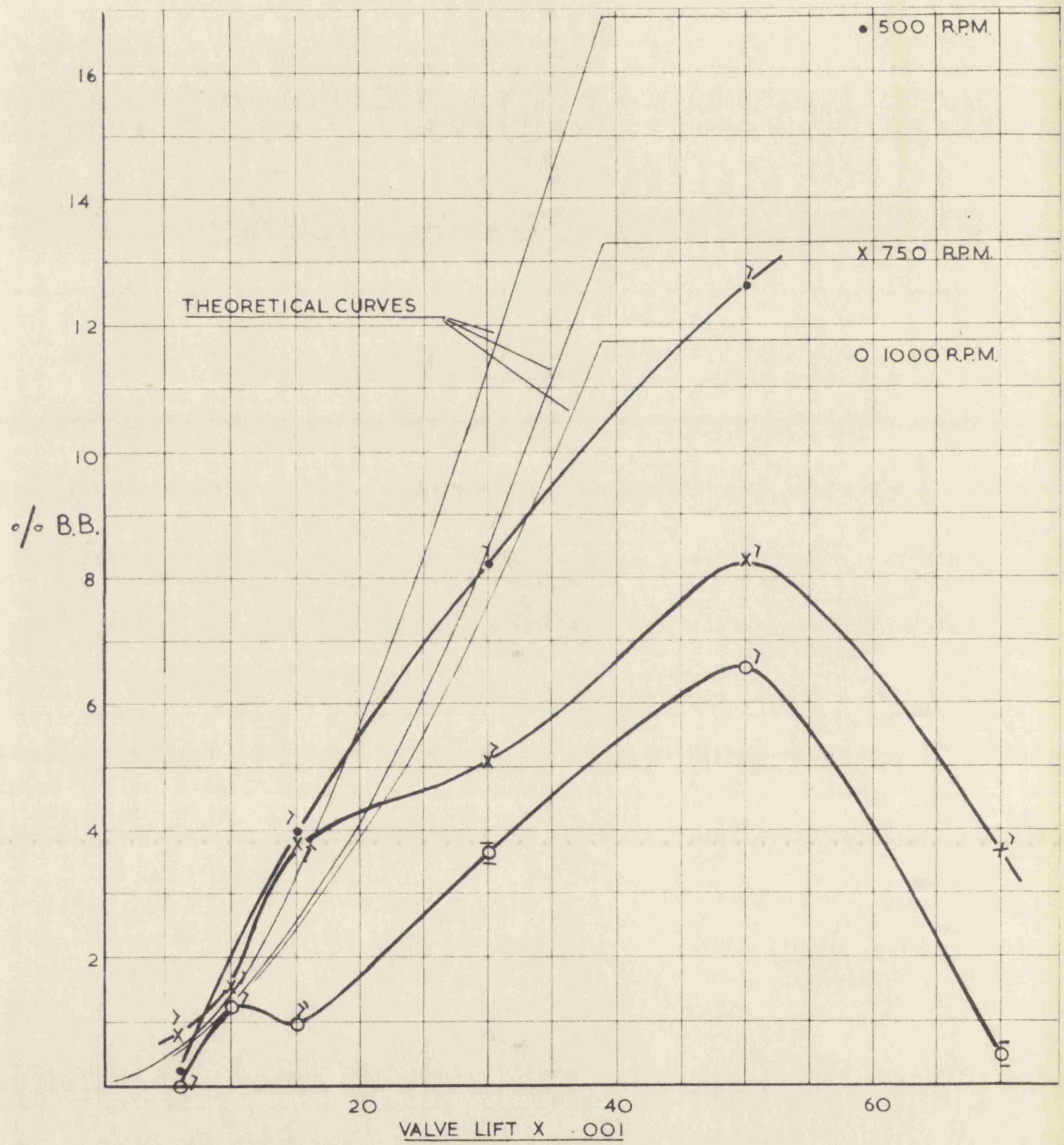
FIG. 28 · .008" VALVE

results at 750 and 1000 r.p.m. are nearly equal to each other and considerably less than the 500 r.p.m. results. This is in agreement with the theoretical curves of Fig, 33. Generally speaking, in the absence of oscillation the results behave in a predictable and reproducible manner giving smooth curves. The onset of oscillation, which can be very much affected by oil quantity and temperature, is an arbitrary factor affecting the results in a manner which is not readily predictable.

The blowback is seen to increase with lift as would be expected but tends towards a maximum. This limitation of blowback quantity is due to the manner of fixing the valve which places an end constraint on it at lifts over .040". Owing to the flexibility of the valve this constraint is more gradually applied than simple theory would suggest.

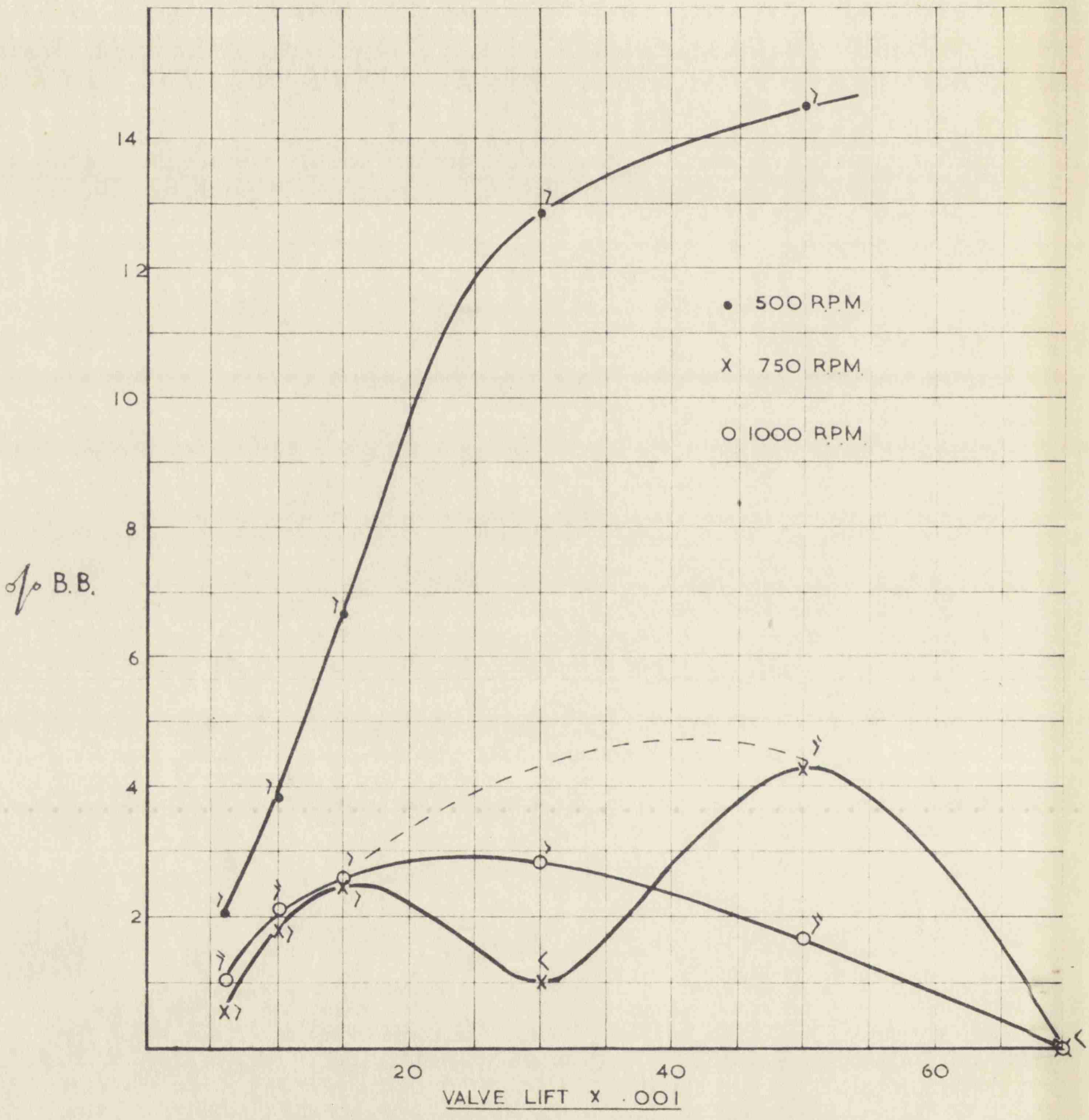
The flexibility of the valve has the effect of increasing the blowback above the theoretical at low lifts where the valve deflects as shown in Fig. 2(1). As the valve passes through the behaviour illustrated in Fig. 2(2) to the eventual cantilever type deflection illustrated in Fig. 2(3) the blowback gradually falls below the theoretical values associated with the setting of the limit stop and attains its limiting value when the stop is no longer reached by the valve. Of course, if oscillation occurs at these high lifts the blowback may be very considerably reduced. It is to be expected that the results conforming most closely to theory will be those where oscillation is slight and where the valve is least flexible.

Figure 28 shows the manner in which blowback varies with lift for a valve of .008" thickness. The graphs, which are again affected by oscillation at the lower lifts, have been drawn through the experimental points without making any attempt to draw smoothed curves. However, the possible shape of the curves if unaffected by oscillation is indicated by the dotted line.



EFFECT OF VALVE LIFT ON
% OF INDUCED VOLUME BLOWN BACK

FIG. 29 ·010" VALVE



EFFECT OF VALVE LIFT ON % OF INDUCED VOLUME BLOWN BACK

FIG. 30 .012 VALVE

The drooping characteristic resulting from lift limitation is again apparent as is the effect of speed on blowback. At 1000 r.p.m. the .070" lift has caused an oscillation of the valve resulting in reduced blowback. As is to be expected, the heavier valve has resulted in blowback quantities greater than those measured with the .006" valve. This is particularly noticeable at the higher lifts.

Figure 29 contains the blowback results obtained at various lifts for the .010 valve. Superimposed on this graph are theoretical blowback curves for a valve of this thickness, at speeds of 500, 750 and 1000 r.p.m. These curves are drawn neglecting the effects of oscillation and assuming the valve to be infinitely stiff.

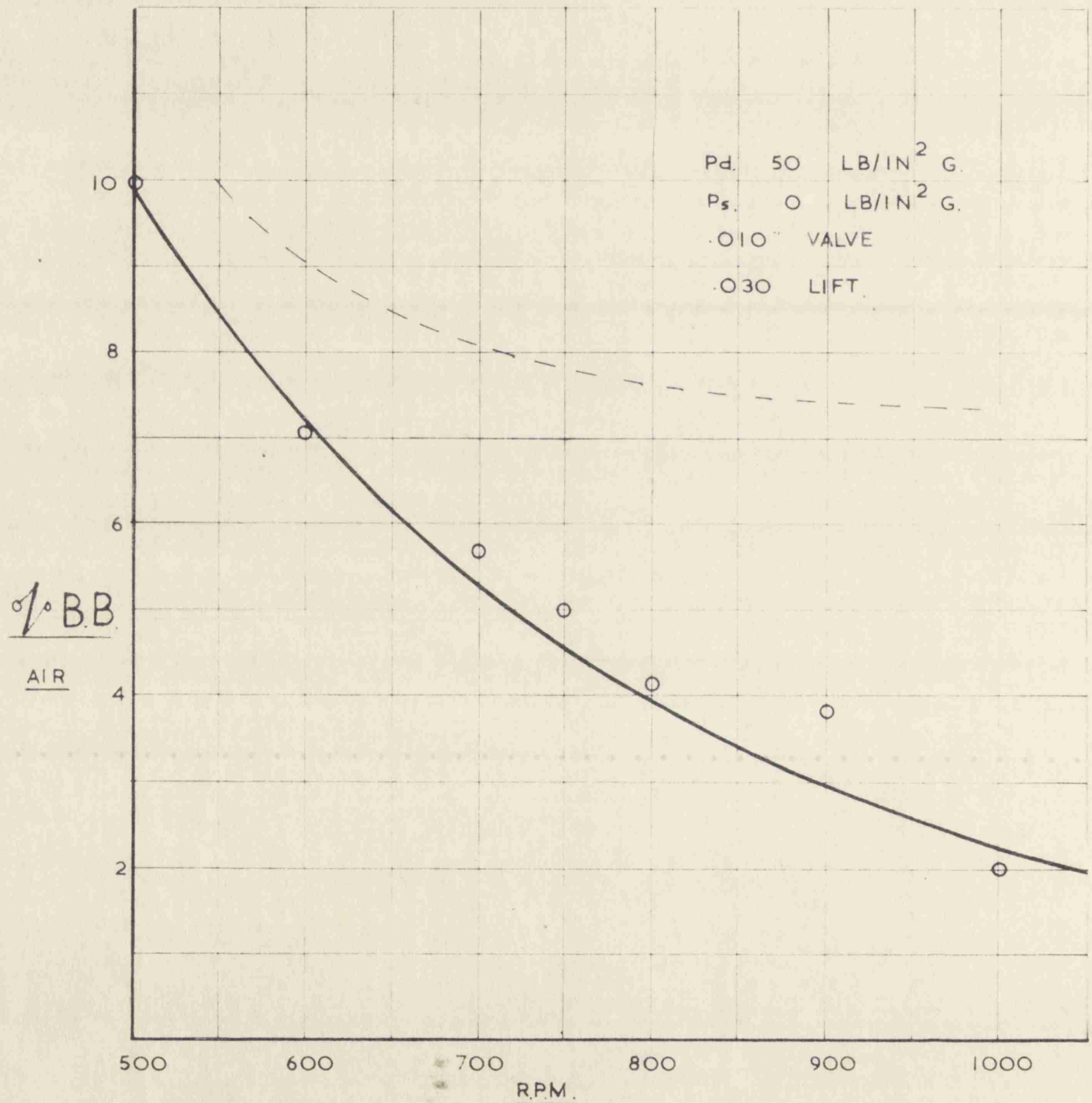
The curves show a form similar to previous experimental results, the 500 r.p.m. curve being unaffected by oscillation and deviating from the theoretical curve owing to its flexibility. The blowback quantities associated with the .070" lift at speeds of 750 and 1000 r.p.m. are reduced owing to the effects of oscillation.

Figure 30 displays similar results for the .012" valve which was the stiffest and heaviest valve tested. At 500 r.p.m. there is a striking correspondence between the experimental curve and the theoretically predicted results for a rigid valve. At the higher speeds, however, the blowback is much reduced owing to oscillation.

6. 2c Effect of Speed on Blowback.

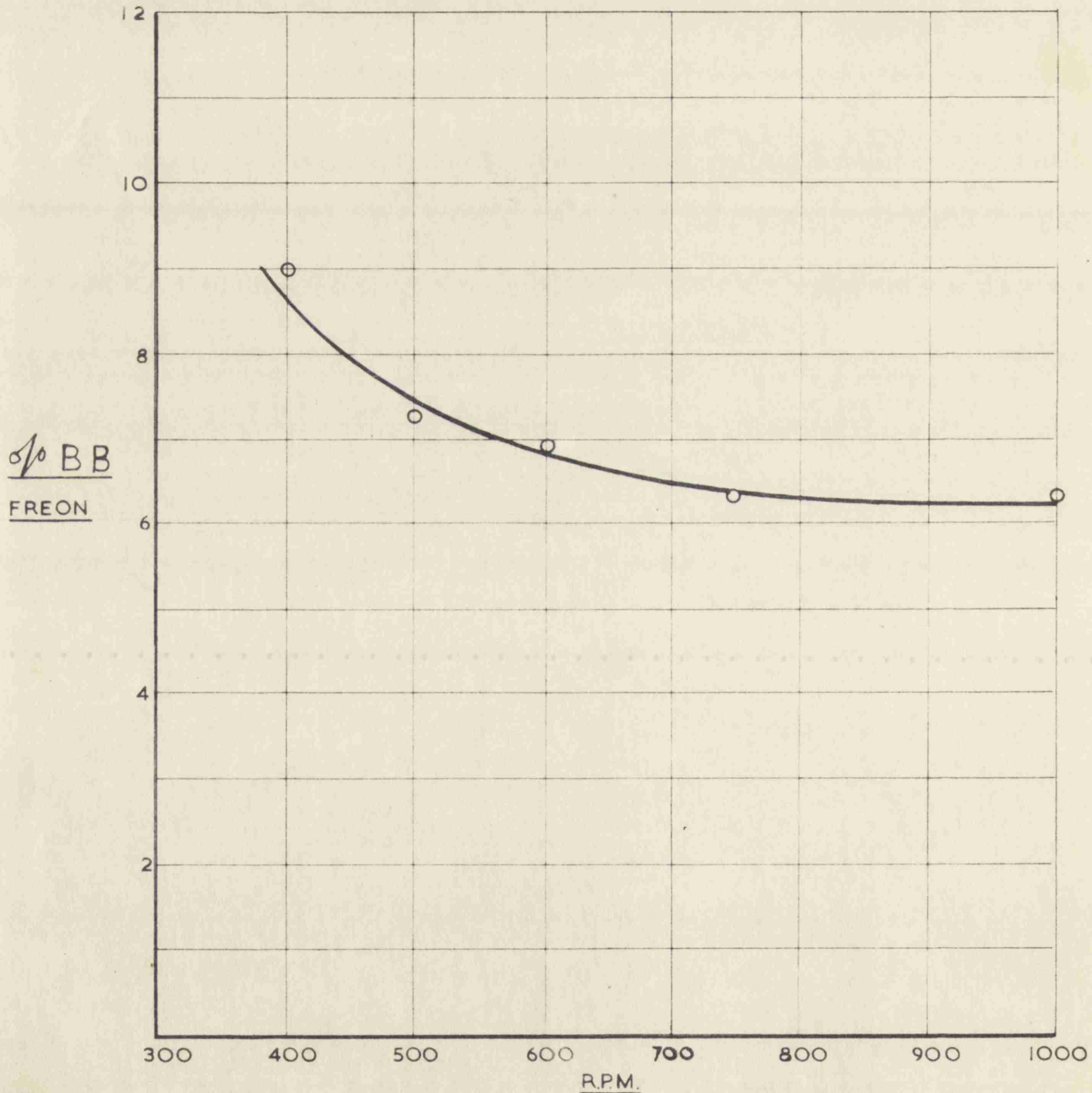
The tendency to operate modern compressors at ever increasing speed lends considerable interest to this investigation which was carried out using the standard .010" valve reed and .030" lift with air and Freon 12 as working fluids.

Figure 31 shows a series of results obtained using air at atmospheric



EFFECT OF SPEED ON o/o OF INDUCED VOLUME BLOWN BACK

FIG. 31



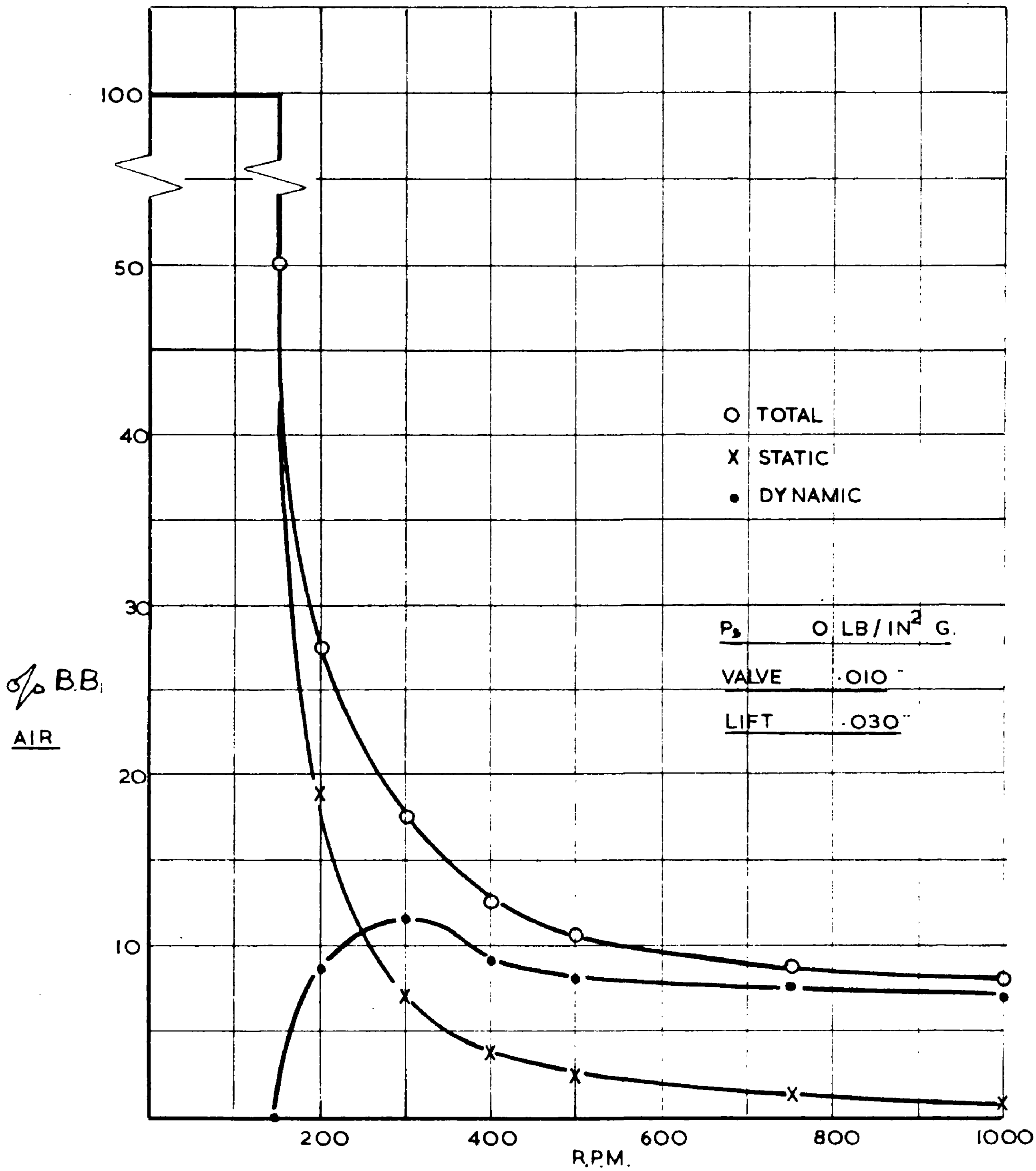
EFFECT OF SPEED ON % OF INDUCED VOLUME BLOWN BACK

FIG. 32

pressure on the suction side, discharging to a pressure of 50 lb./in.^{2}G. The blowback is seen to increase rapidly as speed is reduced as is predicted in the theory of Chapter 3. The reduction of blowback at the higher speeds is greater than indicated by simple theory and it is considered that this is due to the throttling loss increasing with increase of speed. The effect of increased throttling loss is to delay the angle at which gas reversal takes place thus causing valve closure to occur at angles where the piston velocity is high and closure correspondingly rapid.

The upper curve in Fig. 31 is a plot of theoretically derived total blowback included for comparison with the experimental results. The absolute size of the theoretical blowback quantities cannot be regarded as very important because they depend on the characteristic dimension of the valve reed selected as being equivalent to diameter of the disc shaped valve assumed in the simple theory. In the present tests the dimension chosen is the diameter of the disc shaped part of the reed which covers the port. The shape of the curves indicate that the blowback varies as predicted at speeds below about 800 r.p.m. but decreases rapidly as the speed rises above this value.

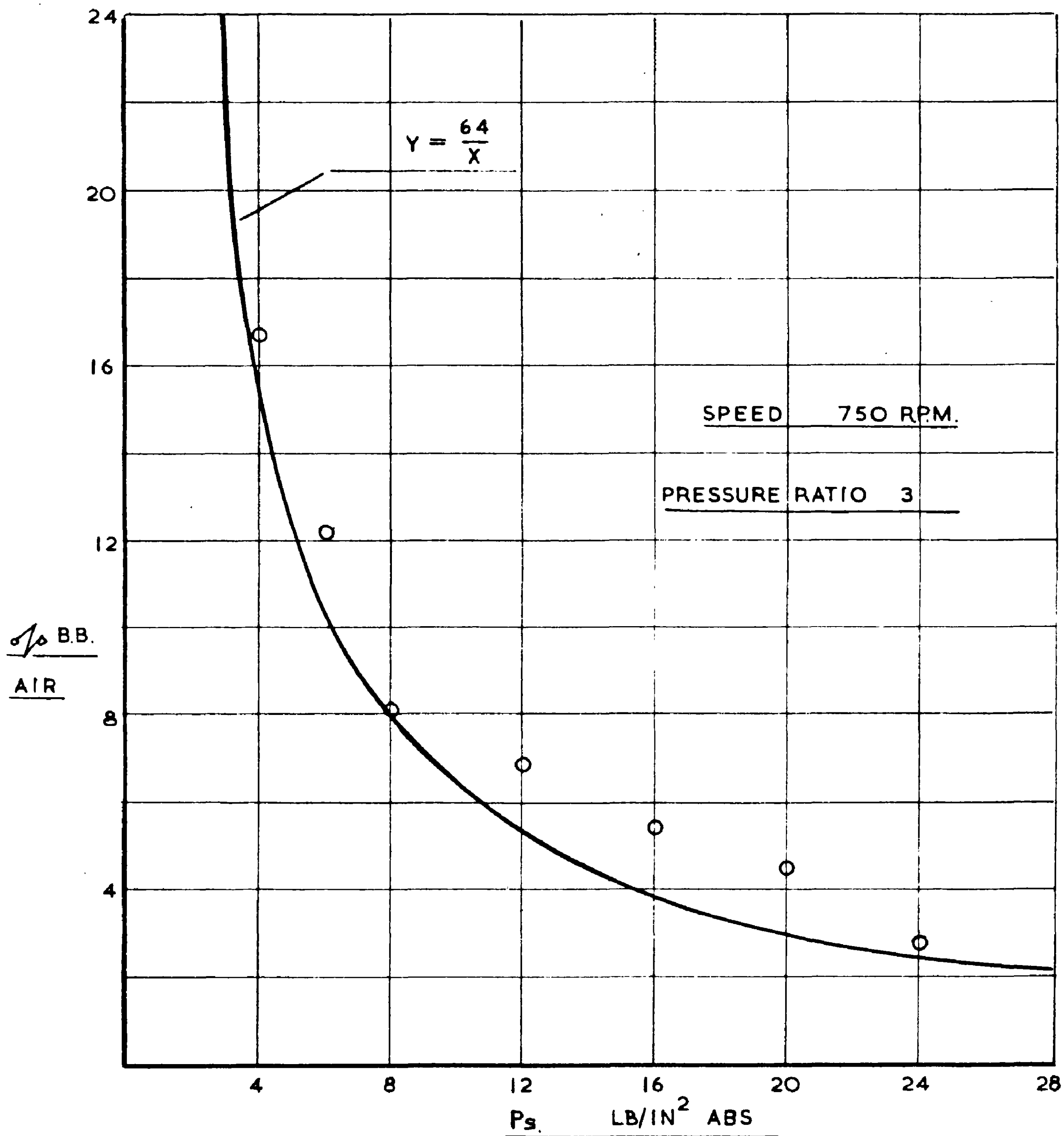
Figure 32 is a curve of blowback quantity using the dense refrigerant Freon 12 and it is seen that the curve is more like the theoretically derived curve than the previous results obtained with air. At first sight this is surprising as the density of the fluid and its low acoustic velocity compared with air would suggest greatly reduced blowback as a result of throttling. However, the relatively slow build up of gas velocity (Fig. 37) on opening of the valve, as compared to the almost instantaneous rise when the fluid is air (Fig. 40), suggests that inertia effects within the Freon may hinder the rise of reversed velocity, while blowback is taking place, to an extent which



CALCULATED EFFECT OF SPEED ON

o/o OF SWEPT VOLUME BLOWN BACK

FIG. 33



EFFECT OF SUCTION PRESSURE ON o/o OF INDUCED VOLUME BLOWN BACK

FIG. 34

delays the closing of the valve. The analysis of this type of behaviour is complex and would require a step by step process beginning at the point where the valve commenced to open.

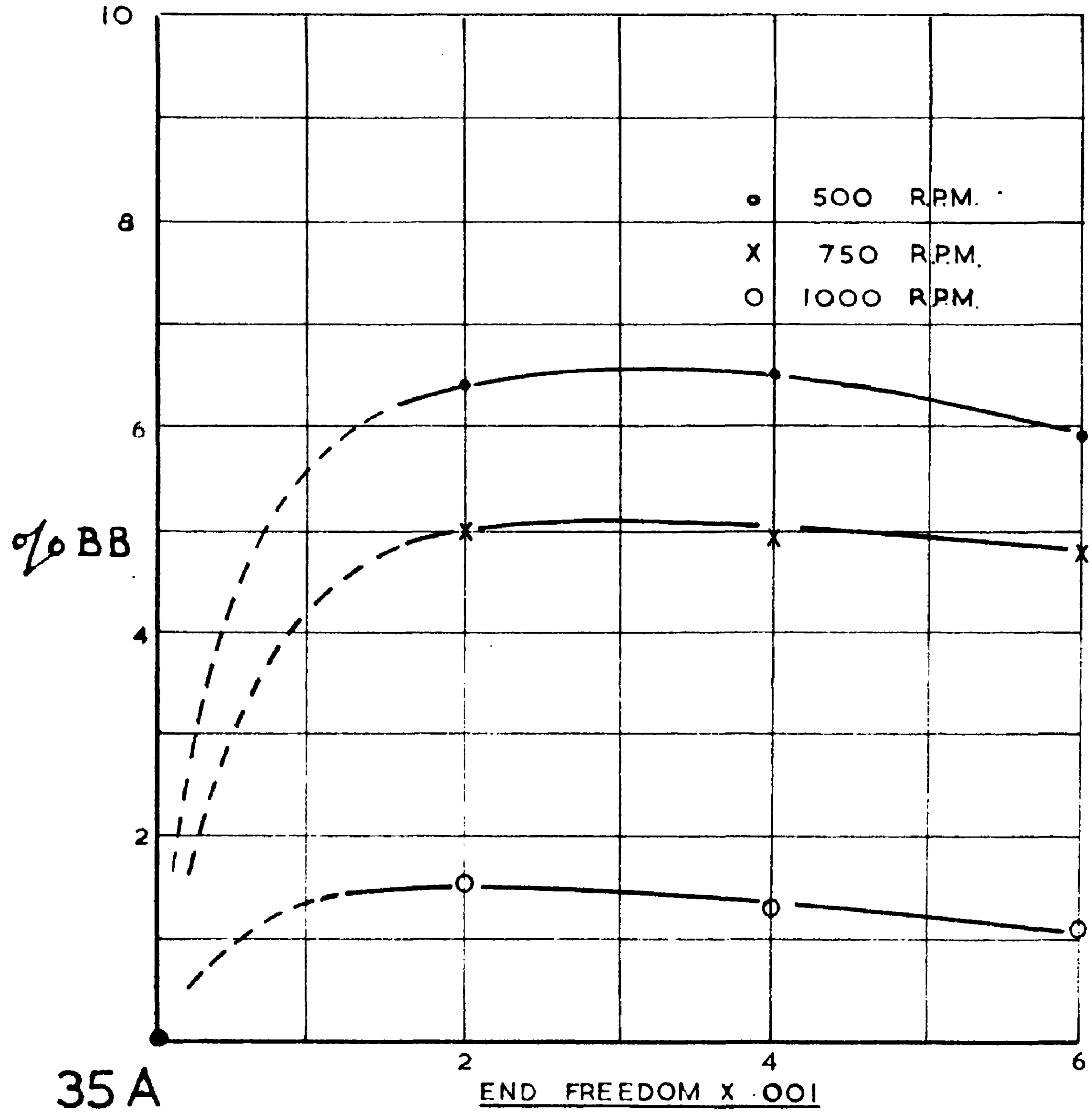
Figure 33 shows theoretically derived curves of Total, Static and Dynamic Blowback calculated for the A Compressor with standard valve and lift when pumping air at atmospheric pressure. In the practical speed range, above 600 r.p.m., the blowback is mainly dynamic but at low speeds blowback past the stationary valve becomes important and, according to calculation would account for the whole of the induced charge at about 150 r.p.m.

6. 2d Effect of Suction Vapour Density on Blowback.

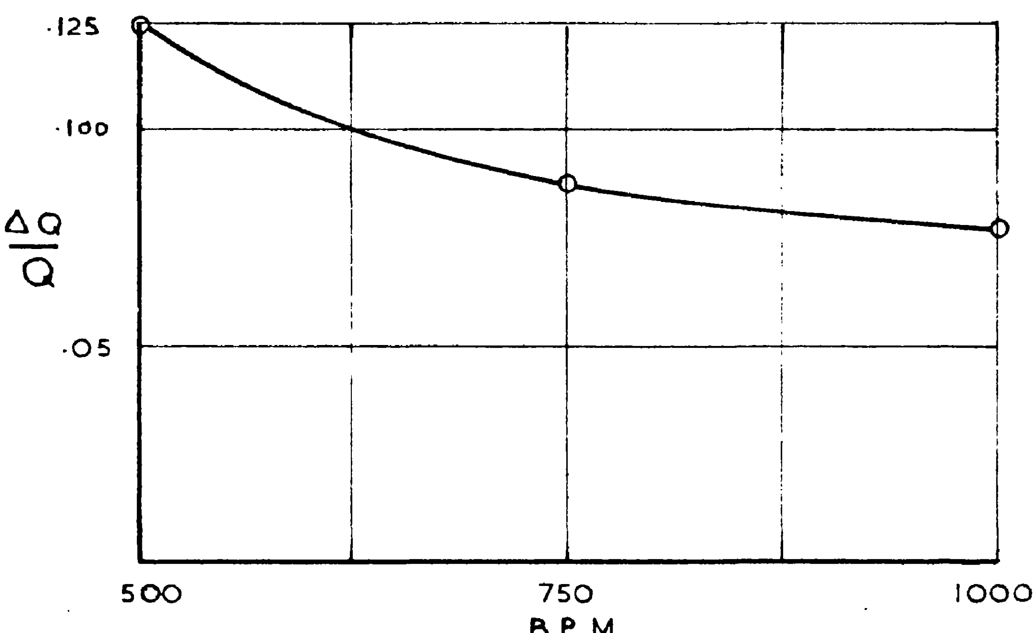
Figure 34 shows blowback quantities measured using air at various suction pressures as the working fluid. The compressor was operated with a constant pressure ratio.

The results show that blowback is very large when the working fluid is of low density. In fact, at extremely low pressures the suction reed appeared to cease to function in any useful way. This is in accordance with the theory of Chapter 3 which assumes the force on the valve reed to depend on the density of the suction vapour. No curve has been fitted to the experimental results of Figure 34 but for purposes of comparison a curve of the form $y = \frac{64}{x}$ has been drawn beside them. This indicates that, in the case of air at least, the amount of blowback is more or less inversely proportional to the density of the suction vapour. As is to be expected there is closer conformity to this rule at the lower densities where a greater proportion of the blowback is static. As is indicated by results using Freon 12 there may be considerable deviation from this type of curve at high densities.

It is interesting to note that, according to simple theory, reducing



35A



35B

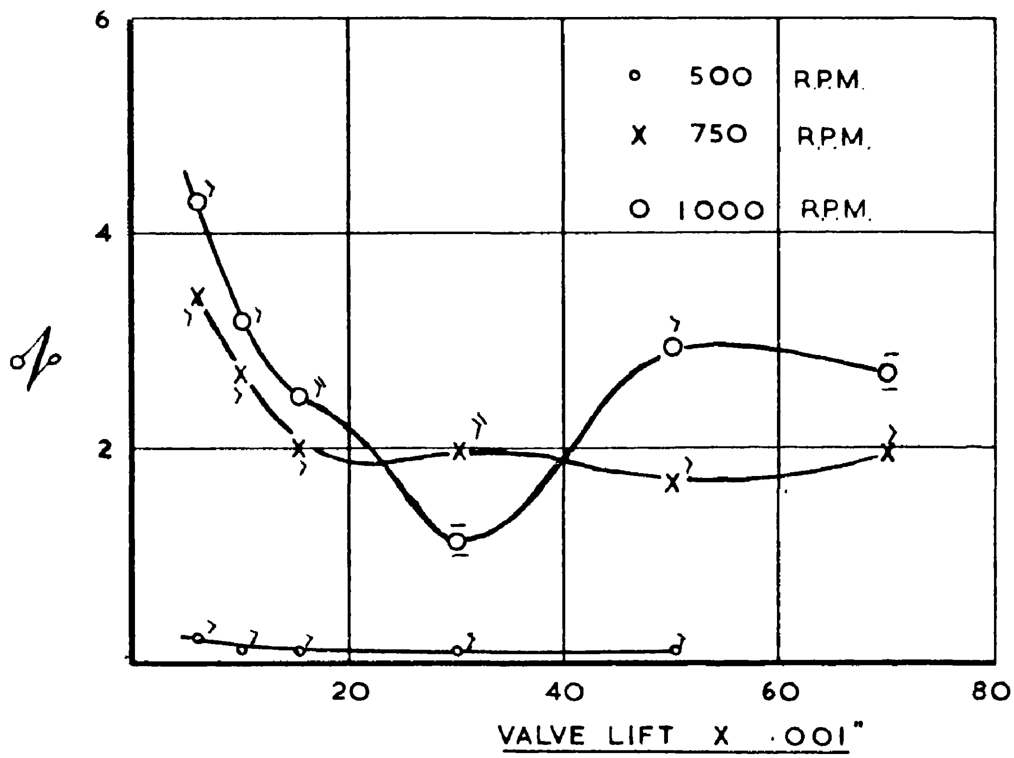
CONSTRAINT RESULTS

the density of the suction vapour has the same effect as increasing the weight of the suction valve. As it did not prove convenient to use more than four types of suction reed it was not possible to provide clear experimental evidence to support this conclusion. But if the theory is valid the horizontal axis of Figure 34 could be considered as being proportional to the reciprocal of valve thickness.

6. 2e Effect of Constraint on Suction Valve.

In order to provide experimental results which bear a direct relation to the types of small refrigeration compressor in use at present it was decided to run the majority of tests with the suction reeds resting loosely on the locating pins as is the normal industrial practice. However, in cases where valves of different thickness were being tested, care was taken to build up the thickness at the hinge end of each valve to a standard size so that test conditions were as uniform as possible.

Unfortunately this experimental technique prevents direct comparison of the results obtained during the present work with the results of Costagliola and MacLaren who worked with spring loaded valves. In order to discover the significance of the various methods of mounting the suction reed in the compressor it was decided to build up the hinge end of the .010" valve to the same thickness as the cylinder head gasket using a small quantity of solder. The cylinder head was then tightened down, checking the compression of the gasket with feelers till the valve was rigidly clamped in place. Tests were run and blowback measured at speeds of 500, 750 and 1000 r.p.m. with the valve in this condition and then the valve was removed from the compressor and the solder at the hinge end filed down to provide .002" of vertical freedom. The tests were repeated with varying amounts of end freedom and the variation of blowback under those conditions is shown in Figure 35.



EFFECT OF VALVE LIFT ON

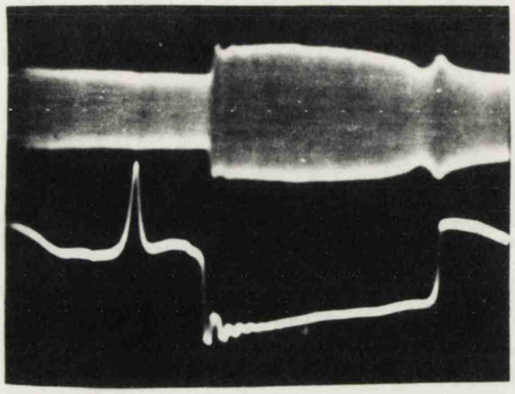
o/o LOSS DUE TO THROTTLING

FIG. 36

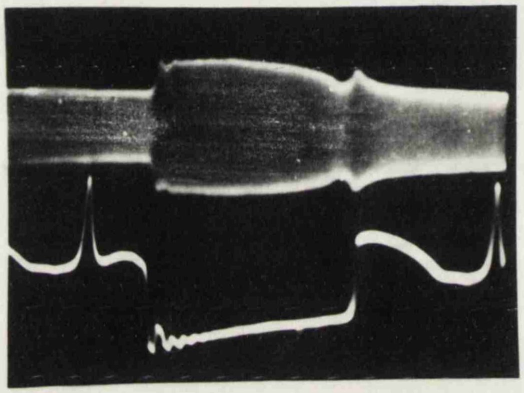
It was found that, when the valve was rigidly clamped in this manner, a violent oscillation was set up and blowback was completely eliminated. The blowback quantities occurring under each condition examined are shown in Figure 35A. Figure 35B shows the average percentage improvement in performance produced at speeds of 500, 750 and 1000 r.p.m. by clamping the valve. It must be emphasised that, though clamping the valve in place is likely to produce oscillation because of the comparatively slight damping of the cantilever arrangement, it is by no means certain to do so and many of MacLaren's results do not seem to have been affected by oscillation. Though elimination of blowback produces an immediate improvement in performance it is not altogether advantageous if accompanied by valve flutter as this increases the power requirements of the compressor and may have an adverse effect on the valve life.

6. 2f Effect of Valve Lift on Throttling Loss.

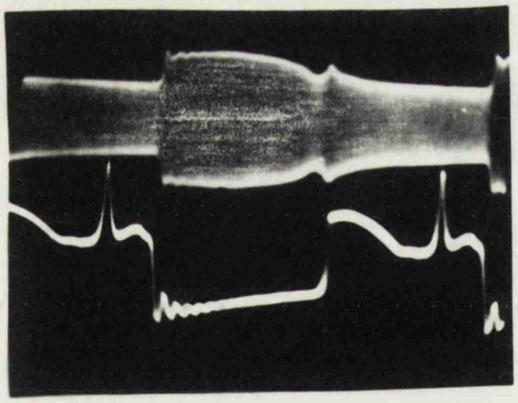
Figure 36 is an example of variation of primary throttling loss with respect to valve lift. The loss was measured from the anemometer diagram as the angle between bottom dead centre and the point of gas reversal. This angle was then expressed in terms of stroke volume to provide a measure of the effect that this throttling loss has on efficiency. The throttling loss can be very much affected by valve oscillation but the example chosen has been selected as being among the least affected in this way. It is most interesting to note that this throttling loss remains relatively constant down to lifts of about .015", suggesting that such lifts could be usefully employed when pumping air. The throttling loss when pumping Freon 12 appeared much greater, varying between 10 and 15%, but in order to concentrate the research on the original blowback investigation this matter was not fully investigated.



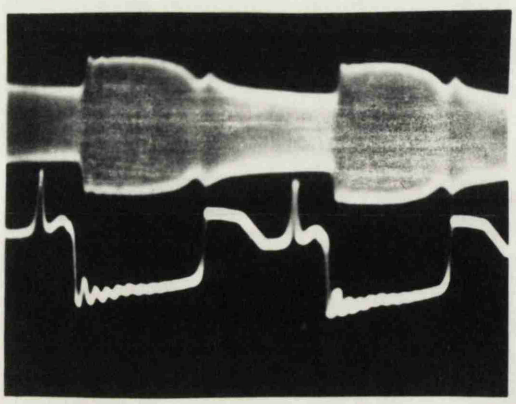
500 RPM



600 RPM



750 RPM



1000 RPM

ANEMOMETER & VALVE DIAGRAMS
FREON TESTS AT VARYING SPEED
·010" VALVE ·030" LIFT

FIG. 37

6. 3 H Compressor Tests.

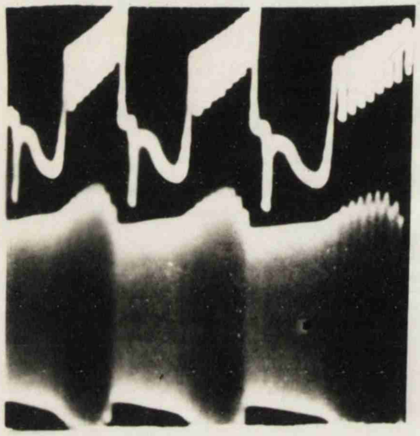
Preliminary tests were made on the H compressor using air as the working fluid. These tests were carried out with atmospheric suction and variable discharge pressure and also with constant pressure ratio and varying suction density. The results were disappointing in that blowback was almost entirely absent owing to the occurrence of large throttling losses. In the rare instances in which blowback was observed it appeared to be the fortuitous result of interference between oscillation and the closing processes of the valve. Figure 38 shows a case of this type.

The magnitude of throttling loss associated with this compressor is interesting and it appears that in this respect, performance could be improved. Another interesting feature of this compressor is the strong tendency for the valve to break into oscillation after a period of normal running. This inconsistent behaviour under apparently identical conditions is probably due to the effects of oil which cannot easily be controlled in a compressor of this type. The occurrence of valve oscillation could, with practice, be detected without the aid of the electronic equipment owing to its introducing a distinctive note to the compressor noise. Often the valve would oscillate in short bursts of under a second in duration in a manner analogous to a quench type oscillator. This was presumably due to build up of oil on the reed resulting in conditions suitable for oscillation. The result of oscillation being to throw off the excess oil. A quiescent period then ensued during which oil could again be deposited on the valve.

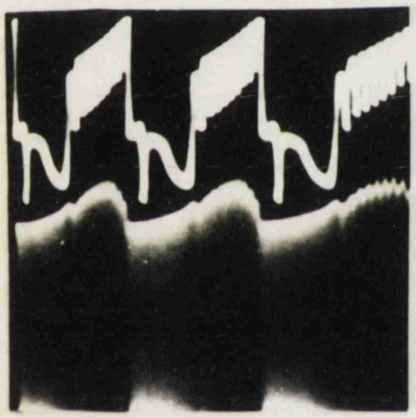
By comparison with the A Compressor tests which used both Freon and Air it can be understood that the valve may be much less prone to oscillation when pumping the dense, oil-soluble refrigerant.

It is interesting to note that oscillation of this valve arrangement

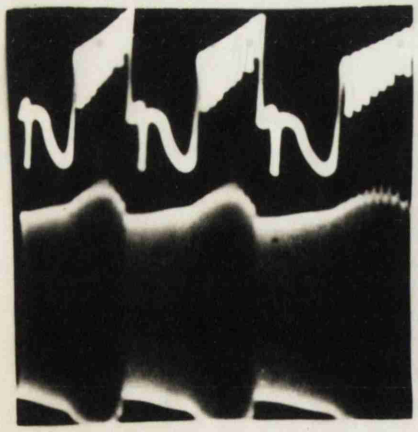
SUCTION PRESSURE



6 LB/IN^2 ABS



8 LB/IN^2 ABS



12 LB/IN^2 ABS

ANEMOMETER & VALVE DIAGRAMS
AIR TEST ON H-COMPRESSOR

FIG. 38

was invariably accompanied by decreased performance as distinct from the A Compressor which usually performed better when its suction reed was vibrating. The reason for the decreased performance of the H Compressor under oscillating conditions is that the throttling loss is increased owing to this effect but the blowback, being slight, does not allow any compensating improvement in performance to be produced at its expense.

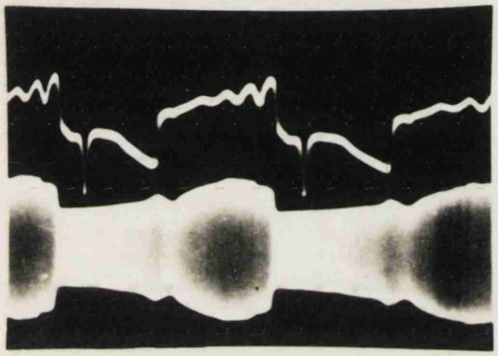
The dimensions and speed of the H Compressor suggest that, according to the theory of Chapter 3, blowback should be small. However it is considered that in the event of oscillation being slight when pumping Freon 12, it would be possible to observe an amount of blowback by means of the anemometer. It was concluded that the experimental methods described in this thesis might indicate some desirable design improvements in the H Compressor valve arrangement but that the general research into valve behaviour which is the object of this thesis would be better carried on the more flexible A Compressor.

Figure 41 shows the variation of primary throttling loss observed from the anemometer records using air at various suction pressures in the H Compressor. The high values obtained may, in fact, be due to the violent oscillation to which the valve was subject during these tests.

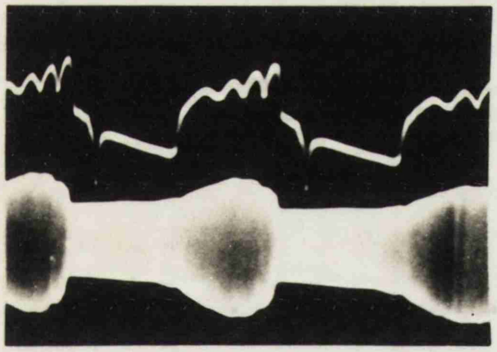
6. 4 Sample Diagrams.

In order to show the type of permanent record which is obtained from the equipment some samples of actual experimental records are presented in Figures 37, 38, 39 and 40.

Figure 37 includes records obtained using superheated Freon 12 as the working fluid at various speeds. The upper symmetrical envelope is the anemometer record of gas velocity in the suction port. This initial result requires to be linearised according to King's equation before amplitudes directly proportional to velocity are produced. The step function type of



950 RPM



1000 RPM

ANEMOMETER & VALVE DIAGRAMS

AIR TESTS ON A-COMPRESSOR

·010" VALVE ·070 LIFT

FIG. 39

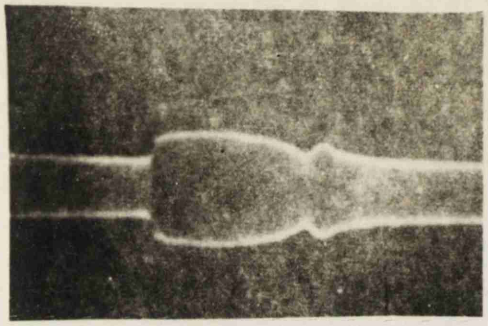
velocity rise indicated in the diagram occurs on opening of the suction valve. The effect of blowback can clearly be seen as a detached velocity peak occurring just before valve closure. The instrument does not record the sudden arrest of gas flow when the valve closes as faithfully as it shows the velocities while the valve is opening. This is because the anemometer is sensitive to turbulence in the gas stream and it continues to give a reading, after the mean velocity in the port has returned to zero, while the turbulence of the gas is dying away. After the double squaring process required by King's equation the effect of this turbulence is seen to be slight.

The demodulated trace shown is an indication of valve movement obtained from the capacitative pick-up. The sharp spike on the trace is introduced to mark top dead centre.

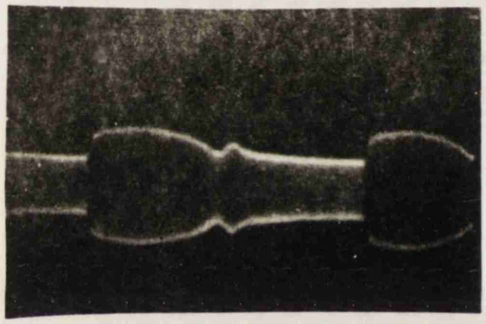
Figure 38 shows results obtained from the H Compressor when pumping air at various suction pressures. The violent oscillation of the valve is very striking and it should be noted that in the 8 lbs./in.^{2} case the oscillation is so timed as to cause some blowback.

Figure 39 shows two records obtained when using air in the A Compressor which was arranged to give very high lift. It should be noted that the valve is now oscillating as a cantilever without touching the stop. This can be deduced from the frequency of oscillation which should be compared with the oscillation of the same valve shown in Figure 37 when the limit stop is arranged to prevent the valve from lifting more than .030". These particular diagrams are included as they demonstrate the profound effect of oscillation on blowback. At 950 r.p.m. the oscillation is phased in such a way that there is little effect on blowback but at 1000 r.p.m. the phasing is such that blowback is eliminated.

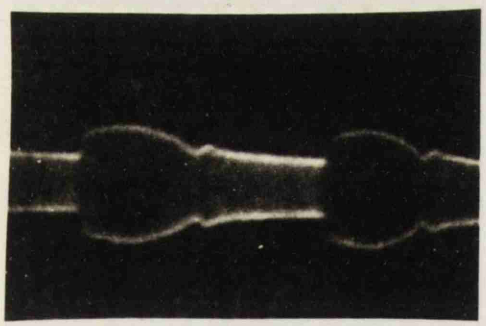
Figure 40 shows anemometer diagrams obtained by using air in the A



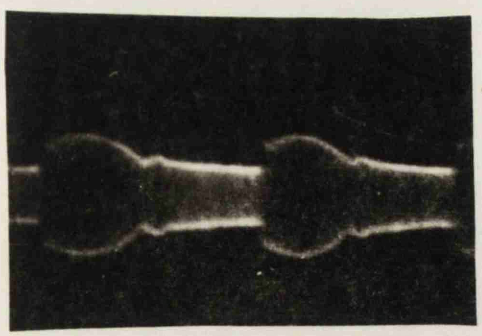
600 RPM



750 RPM



900 RPM



1000 RPM

ANEMOMETER DIAGRAMS
AT VARYING SPEED

AIR TEST ON A · COMPRESSOR
·010" VALVE ·030" LIFT

FIG. 40

Compressor at varying speed and .030" lift. At this lift the valve does not oscillate and the anemometer diagrams show blowback at all speeds.

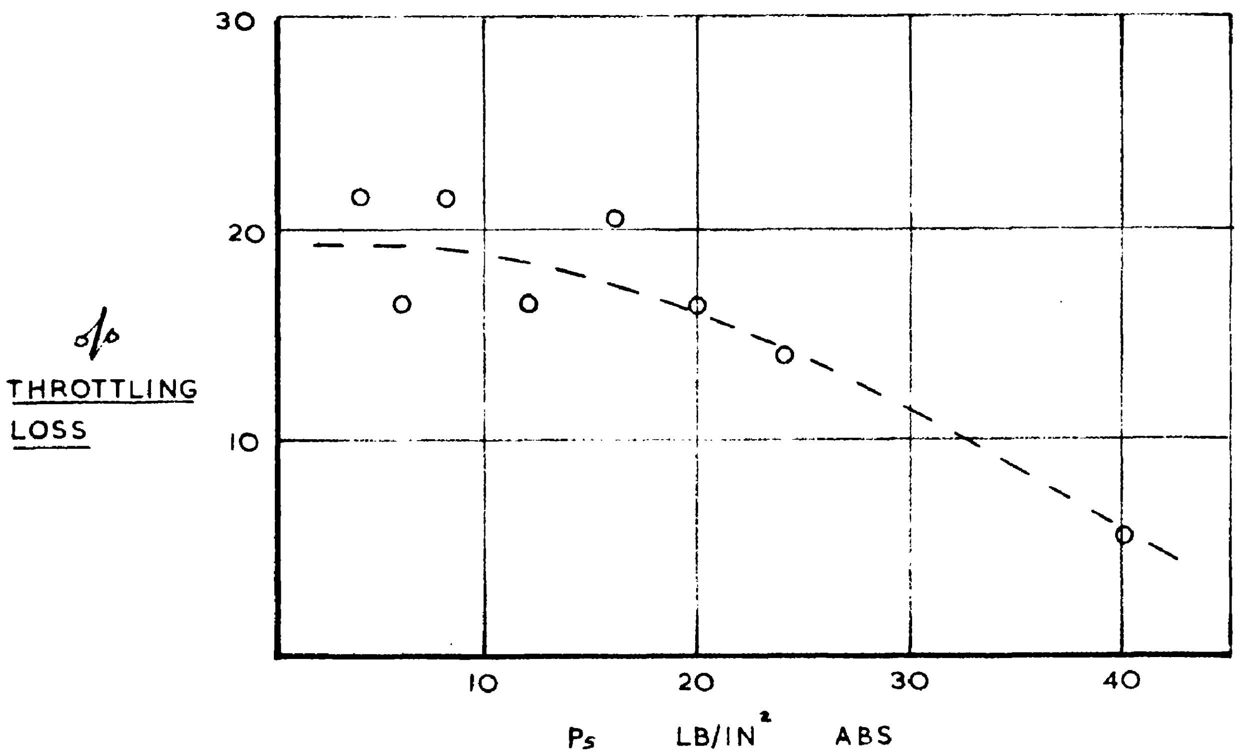
6. 5 General Remarks.

The experimental investigation described in this Chapter has produced results which can be related to the theory of Chapter 3, though it is apparent that the theoretical approach becomes less valid at the higher compressor speeds where throttling loss and gas inertia begin to have serious effects. If required the theory could be modified to allow for inertia and throttling effects though this would make theoretical solutions more complex.

The absolute accuracy of the blowback measurement is not good but care was taken to adopt standard procedure in the reduction of results to ensure a uniformity of treatment which should reduce the scatter of results. In view of the great difficulty in obtaining any measure of blowback it is felt that the adoption of the methods described in this thesis is justified.

The anemometer method is liable to several errors which shall now be considered.

The equation governing the transfer of heat from a heated filament to a transversely flowing fluid as originally defined by King is not strictly correct. (ref 28) However, by operating at constant temperature, some of the error is avoided and it is considered that the equation is of the order of accuracy required. An error is also introduced owing to the variation of gas velocity across the suction port. Owing to the non-linearity of the wire response it is difficult to compensate for this. However if the gas flow is highly turbulent one can assume a mean velocity which is more or less constant over the greater part of the valve port. An examination of the flow paths through the valve passages suggested that highly turbulent flow was likely to take place so it was considered satisfactory to obtain the



HERMETIC COMPRESSOR TEST RESULTS

CONSTANT PRESSURE RATIO 3

FIG. 41

velocity signal from a wire stretched completely across the suction port at a point mid-way through the valve plate.

Another source of error is the different intensity of turbulence experienced at different points of the cycle. The effect of this can be seen in Figs. 37, 38, 39 and 40 where it can be observed that at the point of the gas reversal, where the mean velocity is zero, the anemometer trace has an amplitude greater than its minimum value. Fortunately the magnitude of the error due to such turbulence is much less than appears from the direct readings. The effect of squaring, demodulating and re-squaring the record, in order to linearise it with respect to velocity, is to discriminate against points near the minimum readings. As a result of this it has been found that, on a linearised velocity diagram, the point of gas reversal is rarely above the zero axis by as much as 2% of the maximum velocity.

The decay of turbulence also makes it difficult to decide the point after which the reversed gas velocity becomes negligible. This problem, which becomes important when the amount of blowback is small, is overcome by projecting the initial rate of decrease of reversed velocity to cut the horizontal axis and taking that as a point of zero velocity.

The amount of blowback is assumed to be proportional to the area under the reversed velocity curve and this is measured from an enlargement of the permanent record by means of a planimeter. The average of a considerable number of readings is taken and the result is not considered satisfactory unless the successive planimeter readings are in fair agreement.

By these methods it is calculated that the error due to the operator's treatment of results is kept below 10%.

While the errors described here are by no means negligible it is felt that the technique produces results, which would be difficult to obtain otherwise, with sufficient accuracy to allow variations of blowback quantity to be observed in a way which suggests valid conclusions.

CHAPTER 7

CONCLUSIONS

7. 1 The preliminary tests, carried out on the A Compressor, in which the behaviour of the suction reed was modified by the action of an electro-magnet, indicated that blowback of between 5 and 10% was occurring under the conditions being investigated. Owing to the interdependence of throttling and blowback losses, both of which were affected by the magnetic action, it was not possible to observe how the blowback losses varied under different conditions.

The preliminary tests carried out on the H Compressor using an anemometer and a valve displacement indicator showed violent oscillation of the suction reed and an almost complete lack of blowback except in special cases. The theoretical analysis of the H Compressor also indicated that blowback should be slight. It was considered, however, that the anemometer would be suitable for a practical investigation of the H Compressor performance which might suggest improvements in design.

7. 2 In order to relate this work to the experiments of MacLaren, a study of the effects of clamping the suction reed over the port was made. It was found that clamping the valve reduced the blowback and usually improved the volumetric efficiency of the machine. Clamping the valve also made it very liable to oscillate when in the open position and this increased the throttling loss and the power consumption of the machine as well as producing an audible note which would not be tolerated in a modern hermetic unit.

It was considered that the advantages of clamping might be obtained without oscillation by replacing the single suction reed by two thinner reeds clamped together, the damping due to friction between the reeds acting to reduce the amplitude of oscillation.

7. 3 The anemometer diagrams showed that the gas velocity in the suction port reduced to zero at some distance beyond bottom dead centre and that, in many cases, the velocity rose again till it experienced a second reduction to zero when the valve closed. The second velocity rise was assumed to be due to blowback.

7. 4 Measurement of Blowback was carried out using the Constant Temperature Hot-Wire Anemometer operating in both Air and Freon 12. The results obtained were consistent with the theory of Chapter 3 and indicated the conditions under which blowback is least troublesome when using unclamped valves. The ideal conditions appear to be a combination of light valve, low lift and high speed. It is interesting to note that the empirical approach to design employed in connection with the manufacture of small hermetic compressors has resulted in the retention of unclamped suction reeds for these machines which also embody large suction ports, low lifts, light valves and are intended to run at high speeds.

7. 5 The effect of valve lift on volumetric efficiency was noted using air as the working fluid and a pronounced peak in the curve was noted at low lifts. This was presumed to be due to elimination of blowback at these lifts. It should be noted, however, that overall volumetric efficiency is a very poor guide to the variation of the component losses within the compressor, and the main object of this thesis has been to devise methods of estimating these losses separately.

7. 6 An A.C. Operated Constant Temperature Hot-Wire Anemometer has been used to indicate the instantaneous mean velocity in the suction port of two small compressors. This instrument operated in a reliable manner and produced consistent and useful results using both Freon 12 and Air. It

is considered that there is great scope for an instrument of this type in practice for the measurement of the motion of suction and discharge gases in gas compressors and also in internal combustion engines.

7. 7 A simple theory has been developed from which the blowback past the suction reed can be estimated.

This estimation is based on an assumed pressure distribution in the valve passages during reversed flow and does not require the determination of static drag coefficients.

The results predicted by this theoretical analysis are similar to experimental results though there is some divergence at high speeds owing to throttling and gas inertia effects.

7. 8 An experimental compressor (Appendix) was designed in an attempt to reduce valve and re-expansion losses. The first experimental compressor which was built into a casting similar to the motor housing of the H Compressor had good performance at low pressure ratios but, owing to leakage, its performance at practical refrigeration pressures was unsatisfactory.

In the second experimental compressor which is shown in Fig. 7 it has been possible, by using better manufacturing techniques, to avoid many of the defects of the first model.

APPENDIX I.

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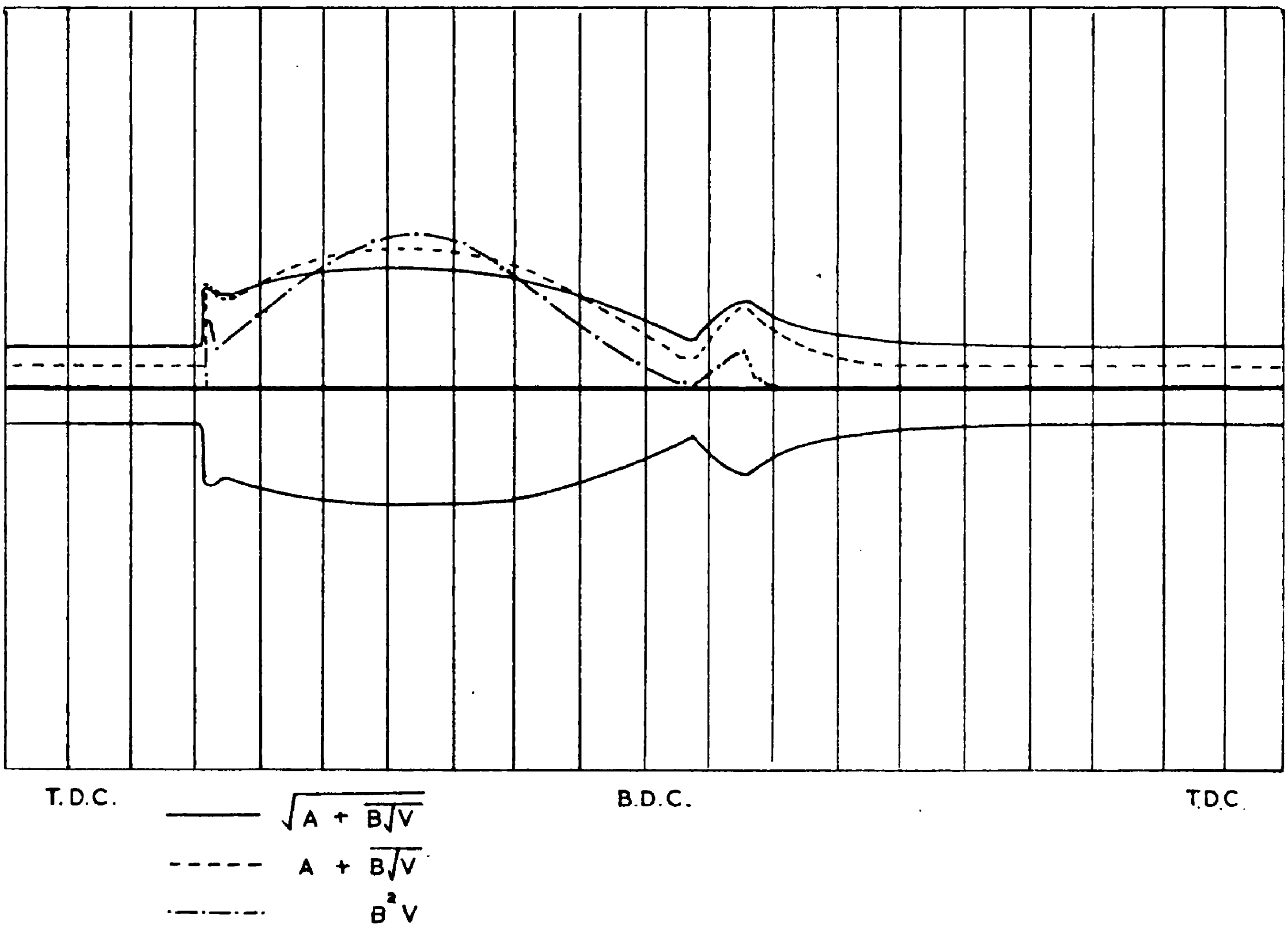
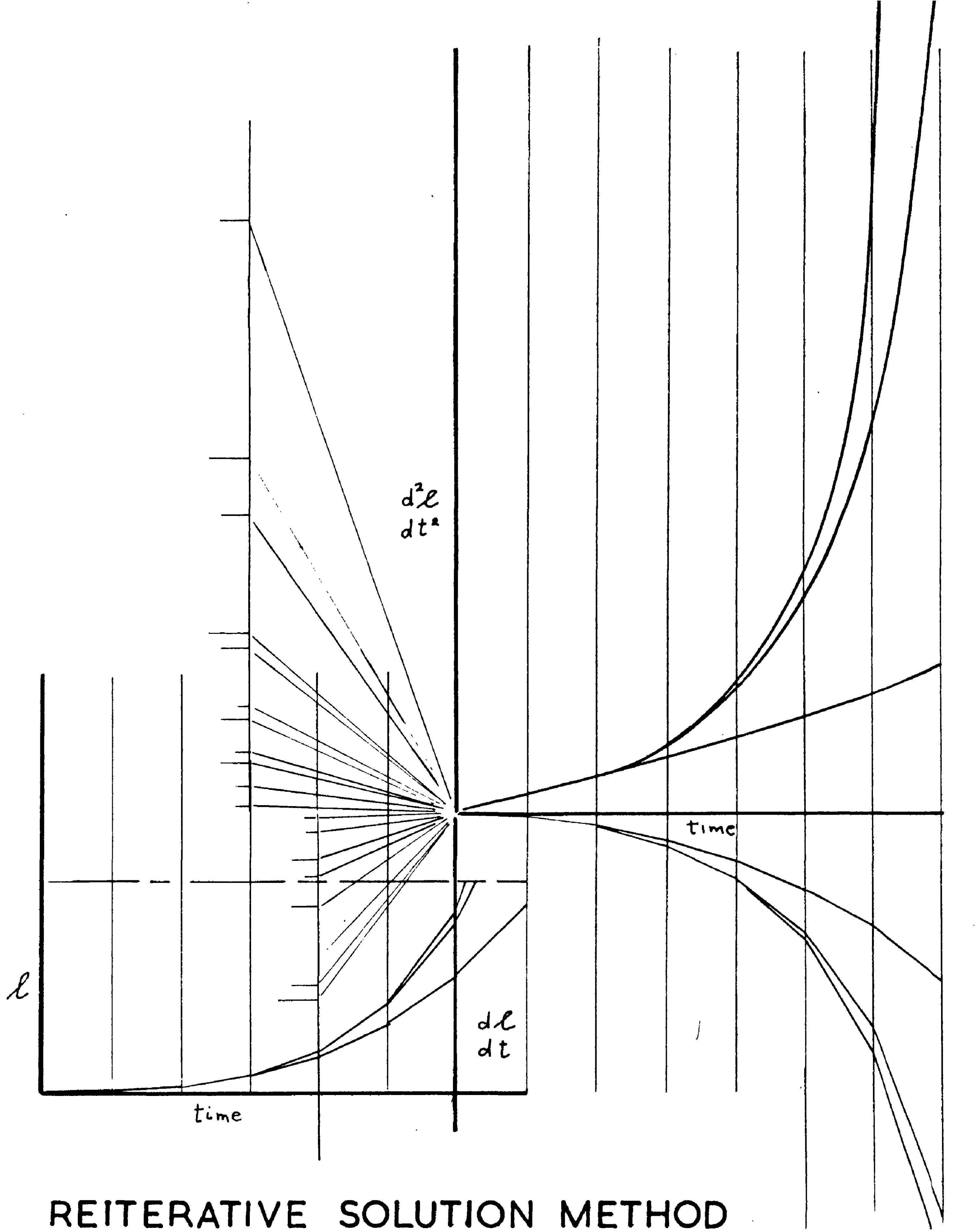


TYPICAL LINEARISATION PROCESS

FIG. 42



REITERATIVE SOLUTION METHOD

FIG. 43

APPENDIX II.

Specimen calculation:

Compressor speed	500 r.p.m.
Working fluid	Air
Valve lift	.030 inches
Approach tube pressure	14.9 lb./in.^{2}
Approach tube temperature	60°F
Orifice head	13 inches water
$\therefore$ Volume of air flowing	.0067 ft.^{3}/sec.
Piston leakage	.0004 ft.^{3}/sec.
Area A1 (c.f. fig. 19b)	1.57 in.^{2}
Area A2	0.12 in.^{2}

A1 - A2 $\equiv$ Total flow + Piston leakage

$\therefore$ 1.45 in.^{2} $\equiv$ .0071 ft.^{3}/sec.

Mean height of diagram = .22 inches

Max. height of diagram = .82 inches

$$\left.\begin{array}{l}\therefore \text{ Mean velocity through port} = 13.5 \text{ ft./sec.} \\ \quad \text{Mean velocity at throat} = 35 \text{ ft./sec.}\end{array}\right\} \text{ Over } 360^\circ$$

Maximum velocity at throat $= 35 \times \frac{.82}{.22} = 130$ ft./sec.

Percentage of induced volume blown back $= \frac{12}{1.57} = \underline{7.64\%}$

APPENDIX III.

Hot Wire Anemometer Theory and Construction.

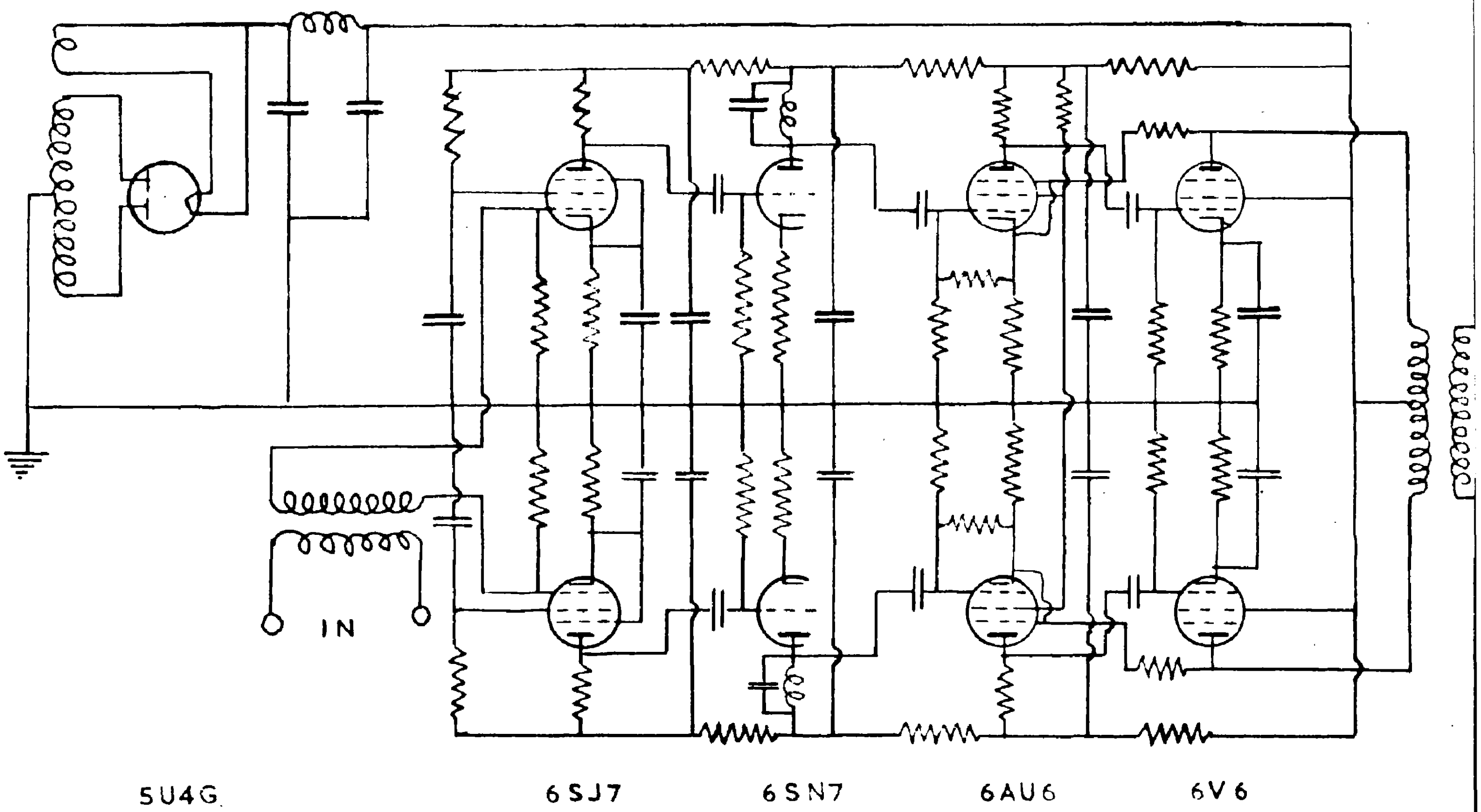
The convenient size and inherent simplicity of the Hot Wire Anemometer makes it a valuable tool for use in investigations which require an estimation of fluid velocity. The fundamental theory required for the application of Hot-Wire instruments was published in 1914 by L.V. King, in a paper to the Royal Society (Ref. 31). Though some of his basic assumptions are now thought to have been incorrect* there is good co-relation between the results of his theory of heat transfer and experimental figures. King deduced that the rate of heat transfer from a heated filament was given by the equation

$$I^2 R_e = J l \left(k + \sqrt{2\pi k \sigma c d V} \right)(\theta_e - \theta_a)$$

Thus, the other factors being known, it is possible to use such a heated filament as a measure of tranverse fluid velocity. Unfortunately, however, it appears that the "constants" of the simplified equation depend, to a certain extent, on such variables as fluid temperature. This effect can be compensated when using the anemometer to measure steady velocities but such compensation becomes difficult when the velocities to be measured are subject to rapid fluctuation.

In the case of a conventional hot-wire anemometer, the voltage across the wire is proportional to $\frac{1}{1 + j2\pi f M}$ where M is the time constant of the wire. M is not strictly constant with respect to temperature. Thus at high frequencies the conventional hot-wire anemometer produces both phase-shift and distortion. In order to overcome this defect, anemometers have been designed which operate by means of an electronic feed-back system, with the wire maintained at a temperature which is very nearly constant. A

* McAdams, "Heat Transmission" p.216.



PUSH-PULL AMPLIFIER FOR

ANEMOMETER

FIG 44

system of this type is shown in Fig. 16. It can be shown that the transresistance of this system is independent of current. Thus at any particular operating temperature the instrument obeys the law

$$I^2 R_e = J \ell (k + c\sqrt{v})(\Theta_e - \Theta_a)$$

over a frequency range which is specified by the time constant of the wire and the characteristics of the amplifier. By designing an amplifier with a response which is much more rapid than that of the simple wire it is possible to produce a stable system which can indicate high frequency velocity fluctuations without appreciable phase shift or distortion.

For a variety of practical reasons which are considered in Appendix IV it was decided to operate the anemometer by means of an A.C. carrier wave.

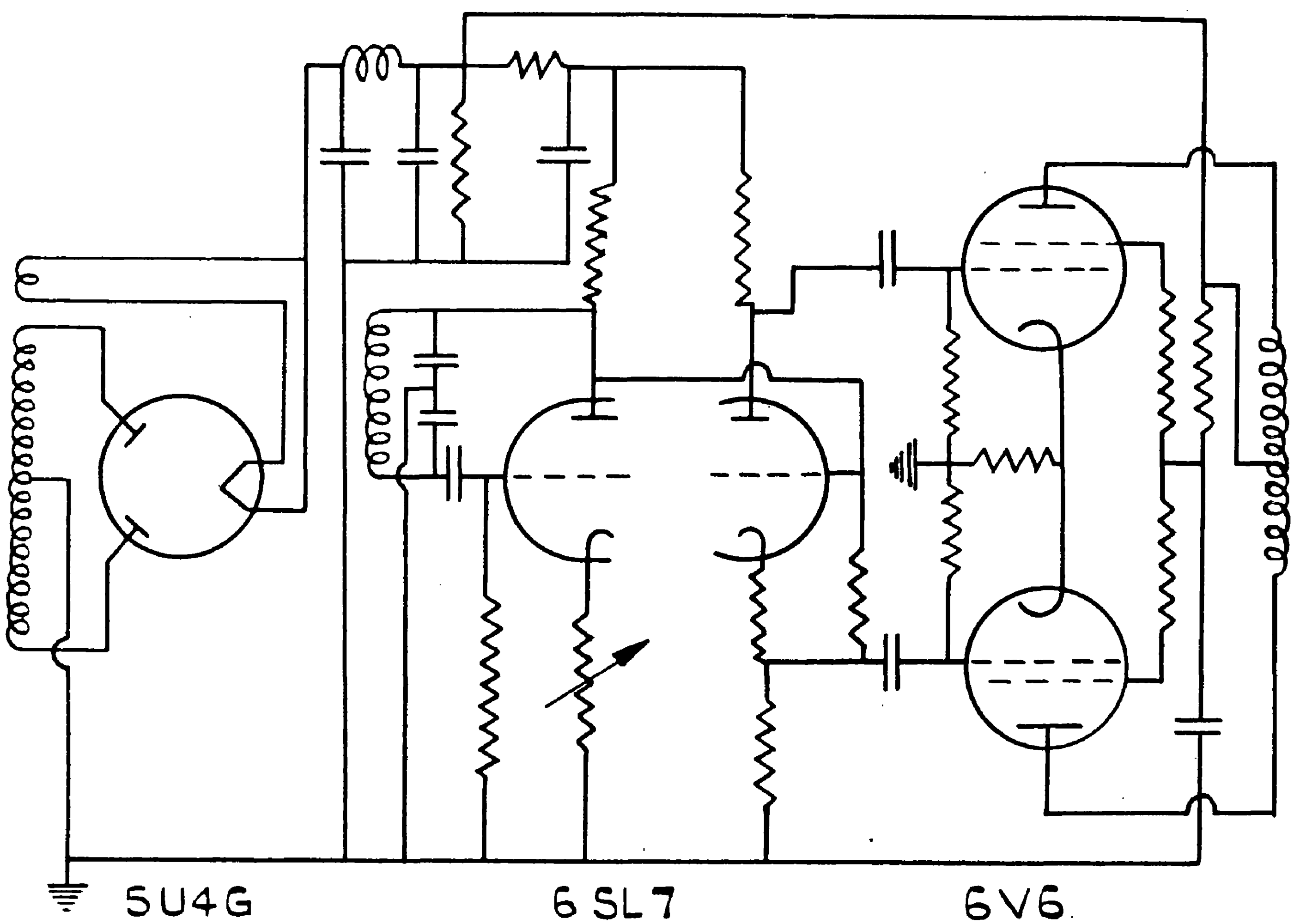
The approximate power required to provide the feedback was calculated using the empirical relationship

$$\frac{hd}{k} = 0.32 + \left(\frac{\rho d V}{\mu}\right)^{0.52} . \, 0.43$$

which, it is assumed, would produce conservative results. The velocity used in evaluating Reynolds number was obtained by multiplying the maximum piston velocity by the ratio of the piston to port areas. The heat transfer coefficients estimated by using the empirical formula in this way were of the order of 300 BTU/ft.^{2}hr.^{o}F. for air and 1500 BTU/ft.^{2}hr.^{o}F for Freon 12.

As the anemometer system must be designed in such a way that the wire is kept below temperatures at which tungsten might be expected to deteriorate, it is possible to estimate the maximum power which is required from the feedback system. It was calculated that operation with Freon 12 would require about 9 watts while the use of air would involve powers of about 1.6 watts. Such powers are readily attainable and, in fact, the system was designed to provide up to 20 watts undistorted output.

The sensitive element of the anemometer consists of a strand of .0005"



POWER OSCILLATOR

FIG. 45.

dia. tungsten wire. The tungsten wire is stretched across the full diameter of the suction port between leads of 16 S.W.G. tinned copper which are mounted in an ebonite ring recessed into the valve plate in such a way that the flow pattern is not affected (Figs. 5 & 10). Mounting the tungsten wire presented an interesting problem as this metal will not "take" solder. Suitable welding equipment was not available so it was decided to copper-plate the .0005" dia. wire where required as suggested by Laurence and Landes (Ref. 32). This plating was accomplished without the use of specialised equipment by folding the parts of the wire to be plated in small pieces of newspaper which were then soaked in copper sulphate solution and laid on a glazed tile. One end of the wire was taped on to the tile and the other remained attached to the small metal reel on which the wire was supplied. Plating current was supplied from a 1.1/2 volt cell connected between the metal reel and the soaked pads. Suitable plating currents were obtained by adjusting the amount of tungsten wire between the reel and the pads. The fineness of the wire made it difficult to handle when disconnected from the reel, so after the appropriate parts of the tungsten had received sufficient copper plating, the plated wire was tinned, connected to the 16 S.W.G. mountings and tensioned before being cut off the reel. The resistance of each anemometer was checked by Avometer before assembly into a compressor. Wires mounted in this way proved very robust. No wires became detached from their mountings during the experiment. All wire failures were due to the human element resulting in burning or breaking of the wire due to gross thermal or mechanical overload. It was possible to lift the ebonite mounting bush by means of the anemometer wire though this is not a recommended procedure.

APPENDIX IV.

Relative Merits of Various Hot Wire Anemometers.

The decision to use an instrument of the hot-wire type to indicate instantaneous gas velocity in the valve ports of a high speed compressor did not by any means specify the exact type of anemometer to be used. The advantages of several types of hot wire anemometer were examined before the variety considered to be most suited to the present application was selected.

The main problem is to design an instrument which will combine a rapid response with the degree of strength required to withstand buffeting by droplets of vapour and oil. These requirements are in mutual contradiction because the response of the simple instrument can be improved only by reducing the thermal inertia of the wire. Such reduction involves decreasing the wire diameter, resulting in impaired strength. It was soon demonstrated that wires of 0.0005" diameter were many times too large to indicate gas velocities in the range of compressor speeds being investigated. As this diameter of wire was the minimum readily available it was decided to improve the response of the system by using an electronic feedback system, as suggested by Ossofsky (Ref. 34), to maintain the wire temperature sensibly constant. It can readily be shown that this improves the response of the system by reducing effects due to the thermal inertia of the wire. It was decided to adopt this method which will be called the Constant Temperature System.

The type of Constant Temperature Hot-Wire Anemometer described by Ossofsky has the advantage of a frequency range extending up to 100 KC/sec. The D.C. signal produced is also fairly easily linearised to produce an output directly proportional to velocity (Ref. 33). However, for the present application it was decided that the complications inherent in the

D.C. system precluded its adoption. The D.C. amplifier required is of specialised and expensive construction. In order to avoid instability, which is always a danger in such feedback systems, the amplifier must have a very flat characteristic to the limit of its range. This called for very careful design and construction which did not appear justified in the present case. In order to provide a D.C. output from the amplifier which is of potential suitable for feeding back to the sensitive element, it is necessary to connect the stages by a series of otherwise unnecessary coupling tubes. The nature of the D.C. system, too, makes it imperative that the input from the sensitive element should be of a differential type so that overall changes of bridge potential are not amplified. This results in additional complication and expense. For these reasons the D.C. system was not adopted.

Ossofsky and others refer to the possibility of A.C. type Constant Temperature Hot-Wire Anemometers. However at the frequency range in which these writers are interested, such an instrument is not a very attractive proposition. Thus when the present work commenced no references to actual instruments of this type were available.

In spite of this, the A.C. instrument offered so many advantages in the present application that it was decided to construct one. The frequency range of an A.C. instrument is necessarily limited by the carrier frequency chosen. However the fundamental frequencies involved in this case were so low that the 5 KC/sec. carrier which was adopted was equivalent to the 200th harmonic in the worst cases. This was considered to be completely satisfactory despite the high harmonic content of the signals produced and the 100% velocity fluctuation expected.

In order to produce a system which is theoretically stable when the hot wire is both above and below its equilibrium temperature a separate oscillator is required. Owing to the thermal shocks which could arise from the presence of liquid droplets in the system it was decided that a separate oscillator was necessary. Provided the wire temperature can be kept just below its equilibrium value it is possible to operate a self excited A.C. type constant temperature hot-wire anemometer. A recent paper by Shepherd (Ref. 36) describes an anemometer of this type operating at about 10 Kc/sec.

The main advantage of the A.C. instrument lies in the simplicity of the electronic equipment required. This consists of standard audio amplifiers and oscillators which are readily obtained or constructed. The resistance capacitance coupling between amplifier stages eliminates potential build-up through the amplifier and the various outputs can be transformer coupled into the hot-wire circuits providing good impedance matching. The use of an A.C. bridge eliminates the need for complicated differential type input stages to the amplifier. In the present application the stability of the complete system was improved by tuning the anode load of one of the amplifier stages to the carrier frequency, thus discriminating against other frequencies at which the system might have tended to oscillate.

The A.C. system adopted provided a robust element with sufficiently good response to give instantaneous readings at all the frequencies encountered. The inclusion, for calibration purposes, of a gain control on the amplifier meant that instability could always be produced at the higher gains available. However those operating the equipment soon became wary of the gain control and adept at preserving the sensitive element.

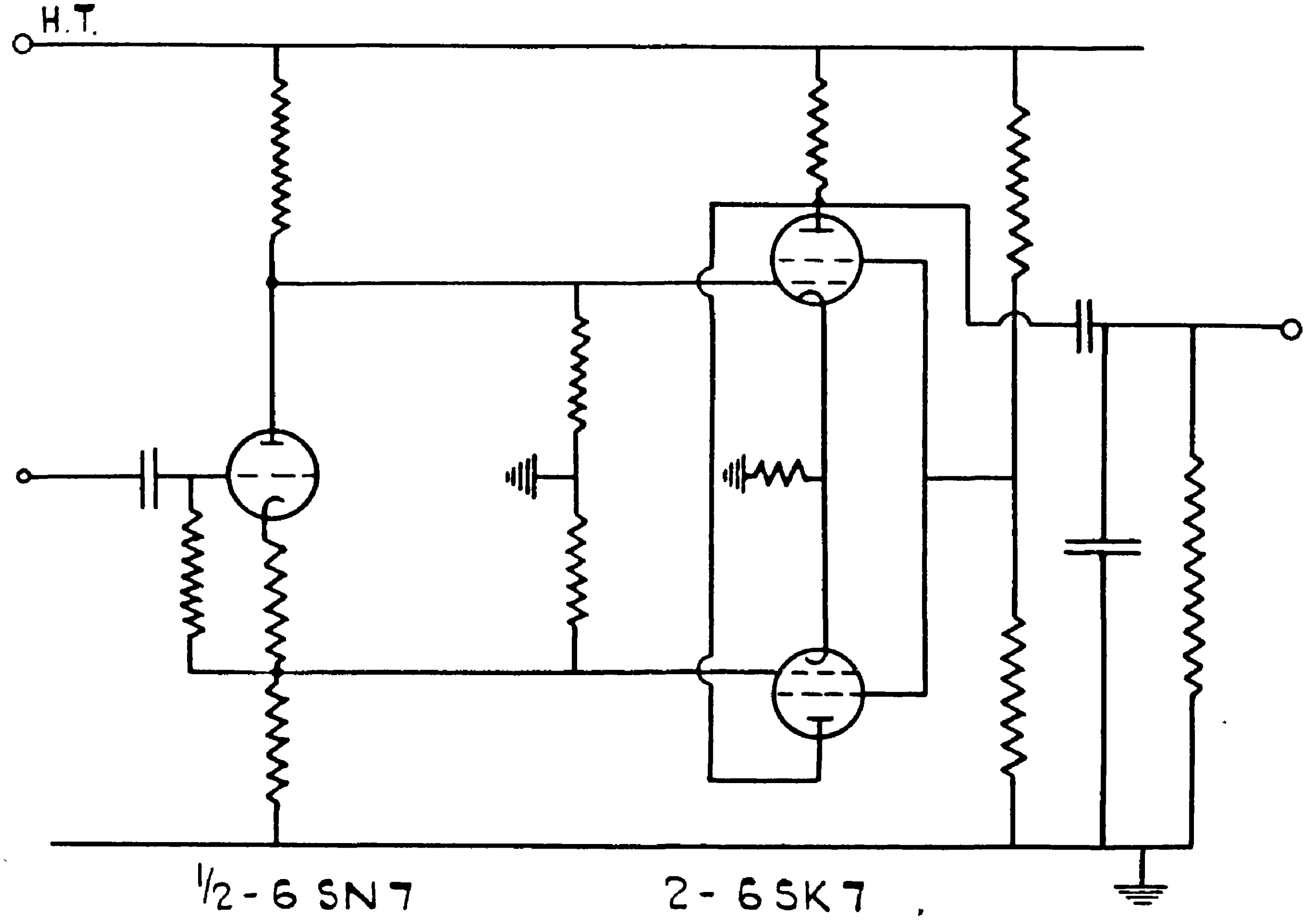
APPENDIX V.

Computing and Linearising Methods.

The signal produced from the hot wire anemometer is not directly proportional to the transverse gas velocity which is being indicated. Thus some means of linearising the output of the instrument must be found before it is possible to deduce flow quantities from the output signal.

Previous workers (Ref. 33) were able to obtain radio valves which displayed a characteristic inverse to that of the sensitive element. The signal from the anemometer was modified by means of the valve and the resulting, more or less linear, response was calibrated against conventional velocity measuring instruments. However, in all previous cases the velocity fluctuation being measured was small compared with the mean velocity of the fluid. In the present case the fluid was expected to reverse its direction in some instances. Also the blowback velocities which were to be measured were small compared to the peak forward velocities. It was considered that satisfactory linearisation could not be obtained over such a velocity range by means of a single standard valve.

The linearising operations required can be regarded as an initial squaring of the signal followed by rectification and demodulation. The demodulated signal is then squared once more to produce a final output which, according to King's equation, is now directly proportional to the velocity. As there are several well known electronic squaring circuits it was decided to attempt linearisation of the signal using such a circuit. Preliminary trials were carried out using a circuit (Fig. A) which took advantage of the square law characteristic displayed by the second harmonic distortion produced in electronic valves. Two valves were connected in such a way that the first harmonics of their output cancelled while the second harmonics were reinforced.



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2ND HARMONIC SQUARER

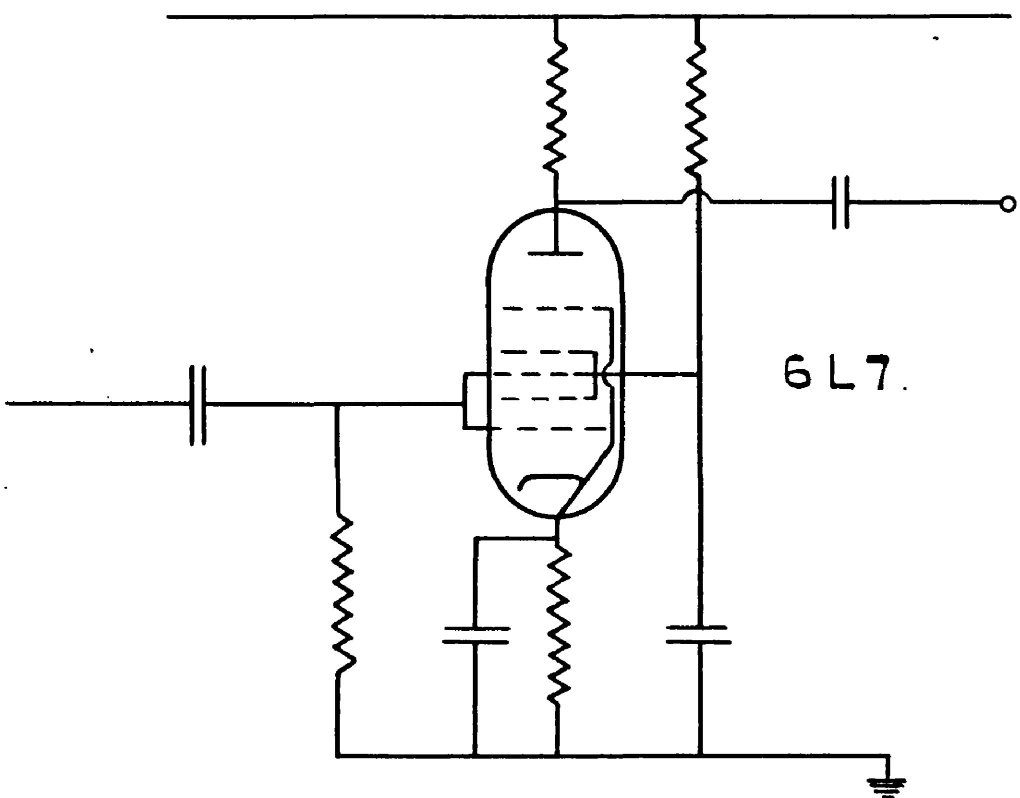


FIG. 46.   PENTAGRID SQUARING CIRCUIT

This circuit operated satisfactorily on steady A.C. signals but no matter how carefully the valves were balanced, the introduction of modulated A.C. signals resulted in the output from the squaring stage being contaminated by unwanted first harmonic. The second harmonic squarer was abandoned as unsatisfactory because of this defect.

The other squaring circuit investigated (Fig. B) used a pentagrid mixer connected as in a multiplying circuit. The circuit was so arranged that the input passed to both the multiplying grids thus becoming squared. This circuit did not change the frequency of the signal supplied to it, but in contrast to the second harmonic squarer, it had a profound effect on the waveform, distorting it according to a square law as would be expected. This circuit was able to deal with modulated signals in a satisfactory manner but it proved impossible to design the squarer to handle signals with accuracy over the range required. As the main experimental work depended on accurate linearisation of the small blowback signals it was regretfully decided to leave electronic squaring methods in favour of a manual method which employed accurately drawn parabolic curves. Though tedious, the manual method was employed throughout the tests to ensure that the required accuracy was obtained.

APPENDIX VI.

Ported Compressor Design.

The large and highly competitive American compressor market has resulted in a virtual elimination of expensive or inefficient designs and has promoted an intensive development and investigation of the relatively few designs which have been able to meet the conflicting demands of cost, efficiency and reliability.

It is of prime importance that the successful compressor should be simple to manufacture and assemble and should contain the cheapest suitable materials manufactured to the minimum necessary accuracy.

The importance of compressor efficiency is greater than appears from consideration of power consumption alone because the size of a given compressor is governed by the thermal and volumetric efficiencies of which it is capable. Other things being equal, a small compressor has a distinct selling advantage over its larger rivals. Inefficient functioning of compressor components too, may result in unwanted noise and lost sales.

Compressor efficiency is largely governed by the type of design chosen. Reciprocating compressors with automatic valves tend to be more efficient than rotary or vane types though the latter are tolerated on account of their cheapness and quietness. Even the reciprocating compressor of high thermal efficiency suffers from severe loss of volumetric efficiency due to re-expansion of gas trapped in the clearance volume and to blow back past the automatic valves during closure.

The ideal compressor should, then, combine the cheapness and simplicity of the vane type compressor with at least the efficiency of the reciprocating compressor without losing any of the reliability of either type. It can be appreciated that such a compressor is not likely to be produced by mere development of the types described above, and, it seems likely that significant improvements in price and performance can not be achieved without the introduction of unorthodox design.

A short list of desirable compressor characteristics can now be drawn up as a basis of future design. The economic aspects of design are considered first because mere mechanical efficiency can always be produced at a price.

1. The new compressor must be of simple construction with few moving parts. Any reduction in the number of accurate components required will produce an immediate lowering of costs. For example, an expensive and unreliable component of the conventional compressor is the valve plate which requires careful machining and assembly. It is obvious too, that the shape of machined components has a decisive influence on production costs. Cylindrical surfaces which can be turned and ground being preferred to irregular shapes which require milling or shaping.

2. To provide high efficiency some sort of piston in cylinder compression process is desirable as this minimises leakage and avoids wasteful constant volume type compression.

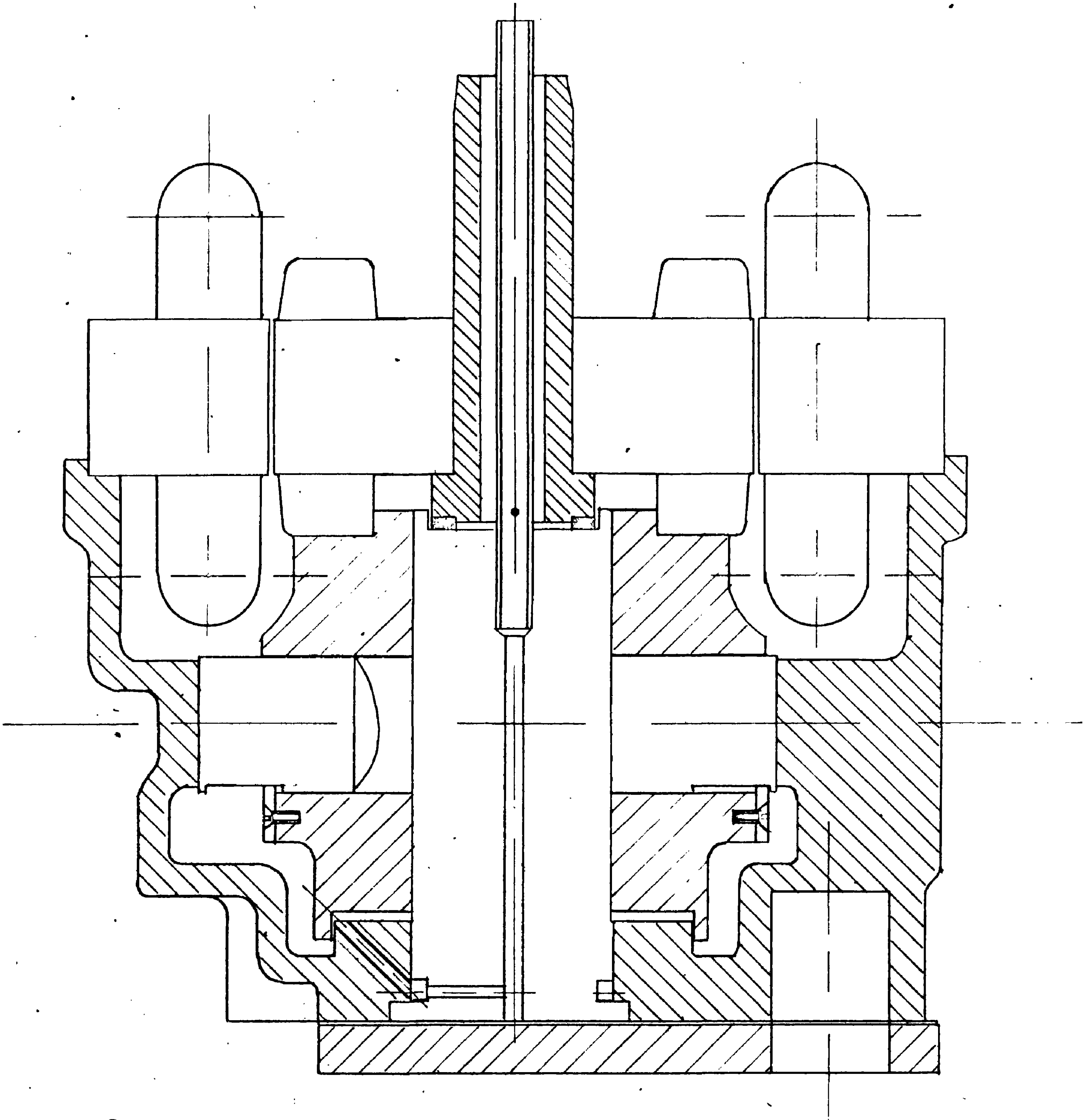
3. The main disadvantage of conventional reciprocating compressors is the clearance volume which must be reduced to improve efficiency.

4. Another loss associated with conventional compressors is due to the automatic valves which allow throttling and blowback to occur. A cheap and efficient mechanical valve would overcome this.

5. The modern hermetic compressor runs very hot and future designs should tend towards good heat transfer from the cylinder block.

6. The configuration of any new compressor should be such as to reduce inertia forces due to unbalanced mass thus allowing high operating speeds.

7. To promote long and reliable service the ideal compressor should have large lightly loaded bearings with a copious supply of oil under pressure.



E - COMPRESSOR —— FULL SIZE

FIG. 47

Past experience has shown that the cheapest and most silent method of supplying oil in a hermetic unit is by built-in centrifugal pump.

Consideration of the above requirements has led to the design of an experimental compressor. This compressor consists of a rotating twin cylinder block whose pistons are held out by centrifugal force against an eccentric track. Valve ports are situated in a non-rotating central cylinder and are opened and closed by the rotation of the block. This compressor is fully described in its patent application No. 18109/56 and a list of its advantages are detailed here.

Advantages of Experimental Compressor. (Fig. 47)

1. The compressor is of simple construction and the number of moving parts has been reduced. In particular the cylinder head, valve plate, automatic valves and connecting rods have been eliminated.

2. Such accurately finished parts as are required are of cylindrical form and thus suited to cheap large scale production.

3. The assembly of the compressor has been much simplified.

4. The new type of valve port arrangement should eliminate the throttling and blowby losses associated with conventional automatic valves.

5. The clearance volume of this compressor can be reduced to as near zero as proves practicable.

6. The port arrangement is of the same order of simplicity as in vane type compressors but the efficient polytropic compression process of the typical reciprocating compressor has been retained.

7. Our of balance forces are produced by the pistons only and the total unbalanced force is the difference of the individual piston forces.

8. The twin cylinders shorten the stroke and give a more even discharge.

9. The rotary cylinder block is better cooled than a normal type.

10. The suction stroke of the compressor is aided by a slight centrifugal effect which should improve volumetric efficiency.

At the time of writing, tests are still being carried out on compressors of this type and performance figures are not yet available. One of the main problems to be faced concerns the silencing of the compressor. Much work is required before the present design approaches the silent operation required of a modern hermetic compressor.

Nomenclature used in connection with valve movement.

D_v	diameter of valve disc
D_t	diameter of valve port
D_p	diameter of piston
r	radius
V_v	gas velocity parallel to valve plate at edge of valve disc
V_t	gas velocity parallel to valve plate at port diameter
V_p	instantaneous piston velocity
V_m	maximum piston velocity
P_c	pressure in cylinder
P_v	pressure between valve and plate at diameter D_v
P_t	pressure between valve and plate at diameter D_t
ω	gas density
W_v	weight of valve
l	distance between valve and valve plate
t	time
Θ	angle between crank position and bottom dead centre
K	constant

See inside back cover for fold out list.

Nomenclature used in connection with anemometer.

V	gas velocity
I	hot wire current
R_a	wire resistance at gas temperature
R_e	wire resistance at equilibrium
R	instantaneous wire resistance
θ_a	gas temperature
θ_e	equilibrium temperature of wire
θ	instantaneous temperature of wire
α	temperature coefficient of wire resistance
c	specific heat of gas
σ	density of gas
k	thermal conductivity of gas
d	diameter of wire
J	Joules' equivalent
f	frequency
M	time constant of wire
h	heat transfer coefficient from surface of wire
μ	viscosity of gas
A, C	constants

See inside back cover for fold out list.