THE UNIVERSITY OF GLASGOW.

AN INVESTIGATION OF THE BEHAVIOUR OF AUTOMATIC VALVES IN SMALL HIGH-SPEED GAS COMPRESSORS AND WATER PUMPS.

A Thesis submitted for the
Degree of Doctor of Philosophy.

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ABSTRACT.

Automatic Valves for Gas Compressors.

The purpose of the investigation was to study the performance of the automatic valve of the reed type in common use for porting small high speed reciprocating gas compressors. The effect of valve behaviour on volumetric efficiency was the prime concern.

The theoretical approach was similar to that of Costagliola, but simplification of the equations reduced the work of analysis and avoided graphical solutions. However, the analysis of Costagliola assumed that there was no valve flutter. It is now shown, both theoretically and experimentally that this may be present.

New experimental techniques were developed including a portable recorder which indicated the very light valve without attachments to it, and measured the pressure drop across the valve during operation. The loss of volumetric efficiency due to suction valve throttling was computed.

The effect on actual volumetric efficiency of the valve variables, spring stiffness, valve weight and permitted valve lift were examined experimentally, together with the compressor variables, pressure ratio and speed, when pumping (a) air and (b) Freon 12.

Observations were made in connection with valve failure, and also on the relative performance of a ported compressor of the same dimensions.

Automatic Valves for Water Pumps.

Much information is available on automatic valves for water pumps, but there has been no study of a small pump. Little interest has been shown in the subject during the last 25 years, and the older techniques to study the valves in large low speed pumps were not suitable for the speeds now in common use.

An inductive pick-up unit was developed which gave simultaneous records of the movement of both valves under combinations of pump and valve variables. It may be analytically deduced that the valve movement is a displaced sine wave and it was found that the travel normally approximated to this. However, for a number of reasons, very abnormal valve travel occasionally occurred.

Graphical records of 120 tests show the effect of valve shape, valve mass, valve spring stiffness, and dissimilar pairing of valves on maximum valve lift, valve coefficient of discharge and pump performance. The maximum valve lifts obtained from the formula due to Krauss exceeded those actually obtained by almost 100%. The curves for the valve discharge coefficient were similar to those of Krauss in form but revealed a size effect not previously observed. Observations on valve noise substantiated the later values of the noise limit published by Berg. Dissimilar pairing of valves and springs showed that, on occasion, improvement in pump performance could be obtained.

NOMENCLATURE - GAS COMPRESSORS.

Symbol.	
A	area of flow in partly open valve.
A_o	area of flow in fully open valve.
A_p	area of piston.
a	area of valve face.
C	accoustic velocity or electrical capacity.
C_D	pressure drag coefficient $= \frac{\text{drag force}}{\text{pressure head}}$
c_d	coefficient of discharge.
c_p	specific heat at constant pressure.
c_v	" " " " volume.
c	fractional clearance or (suffix) clearance.
D	characteristic length.
d	diameter or (suffix) discharge.
E	Young's modulus or internal energy.
Fr	Froude number.
G	parameter $= \frac{c_d A_o C}{\omega A_p s}$
g	gravitational constant.
h	valve lift
I	moment of inertia.
i	(suffix) inlet.
J	parameter $= \frac{C_D a p_i}{k h_o}$ or Mechanical Equivalent Heat.
K	compressibility.
K_p	electrical permittivity.
k	spring constant.
L	a function of γ and θ
l	length of connecting rod
M	Mach number.

Symbol.

m	mass.
N	a function of γ and θ
o	(suffix) orifice or open.
p	pressure or (suffix) piston.
Q	volume flow rate.
R	gas constant.
Re	Reynolds' number.
r	pressure ratio or crank radius.
s	piston stroke.
T	temperature.
t	time.
V	volume.
v	velocity or (suffix) valve.
w	mass flow rate.
Z	valve resistance coefficient $= \frac{\text{pressure head}}{\text{velocity head}}$
z	piston position from valve plate.
α	$= \frac{A}{A_o}$
γ	= isentropic index.
δ	$= \frac{1}{2} + \frac{\text{clearance volume}}{\text{stroke volume}}$
η_v	volumetric efficiency.
φ	$= \frac{p}{p_i}$
ρ	density.
θ	crankangle after T.D.C.
ω_n	natural frequency of valve reed.
ω	angular speed of crank.
Δp_v	pressure difference across valve assembly.

NOMENCLATURE - WATER PUMPS.

Symbol.

a	area of valve disc.
a_1	throat area (excluding central spigot).
a_1'	throat area (excluding central spigot and ribs).
a_2	radial flow area at perimeter of valve disc.
c	valve discharge coefficient at constant flow.
c'	" " " when operating.
c_B'	" " " based on disc area (Berg).
c_k'	" " " " " throat area (Krauss).
d	diameter of valve disc.
F_{max}	valve spring force at h_{max}.
g	gravitational constant.
h	valve lift.
h_{max}	maximum valve lift.
k	valve spring stiffness.
l	perimeter of disc valve.
l_1	valve throat perimeter.
N	pump speed.
P	pressure load per unit area.
P_o	pressure load per unit area below valve disc.
P_{max}	valve loading due to valve weight and spring force.
Q_v	theoretical average flow rate through valve.
Q_c	actual flow rate from pump.
W	weight of valve in air.
W_w	" " " " water.
W'	" " spring in air.
W_w'	" " " " water.

Symbol.

x	ratio of a_2 to a_1.
λ	noise limit coefficient.
ρ	density of water.

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AUTOMATIC VALVES FOR GAS COMPRESSORS.

CHAPTER I.

DISCUSSION AND REVIEW OF PREVIOUS INVESTIGATIONS.

1. 1 Introduction.

The pumping efficiency, the actual volumetric efficiency and mechanical efficiency of small high speed gas compressors are often surprisingly low. While the factors contributing to low efficiencies are readily enumerated, their relative importance is not fully known.

The simplicity of the automatic valve has made it universally used for porting all types of these compressors. Little is known of the influence of the valve on compressor performance, particularly at the high rotational speeds now used to provide the necessary compact, directly driven, machine. The few investigations on the subject are not related and no review of them has been made previously.

In small compressors flexing reed valves of the cantilever, beam or disc types are usually employed. The backing spring designs also vary greatly. The desirable qualities of the automatic valve, the description of practical valve assemblies and their development are discussed by Kohler (16) and Reif (25), in German and by Watson (34) and Quertier (22) and Rowledge (26) in English.

1. 2 Losses in reciprocating compressors.

An important criterion of performance is volumetric efficiency. The economy of size and power input of an air compressor is related to it as is also the refrigerating effect and coefficient of performance of a refrigerating circuit.

The principal factors affecting volumetric efficiency are (a) pressure ratio, (b) clearance volume, (c) heat transfer (d) gas leakage, (e) valve performance, and some of these complex factors are interdependent.

The early attempts to study compressor performance resulted in the publication of empirical formulae to fit test results. Gill (8) analysed the results of volumetric efficiency tests conducted by Scheel (27) on compressors pumping dry natural gas, wet natural gas and air. In the refrigeration field an empirical formula was developed by Voorhees (32), based on the early work (1890) of Denton and of the York Manufacturing Company (1904), (37), for large slow speed ammonia compressors. Denton did not claim great accuracy for his work and Oldham (21) declared the formula "fundamentally at fault". Voorhees subsequently (1931) withdrew it.

Reid and Ambrosius (23) could not correlate their own results for an ammonia compressor with those from other investigations prior to 1931. For such a comparison to be profitable it is necessary to separate the factors involved, and by considering each, assess its relative importance. This approach has been adopted in two investigations. Fuchs, Hoffmann and Plank (5) conducted tests (1940) on a six cylinder ammonia compressor. The losses due to throttling at the valves were included with heat transfer and friction losses. The approach was more fully developed (1949) by Lorentzen (18). The partial losses in volumetric efficiency were computed for six modern multicylinder refrigerant compressors. The wide scope of this extensive experimental programme did not permit a detailed study of valve losses.

The delivery pressure for an air compressor is fixed by the duty required of it, and the suction pressure is usually atmospheric. In a refrigerating circuit the suction pressure is determined by the required evaporating temperature, and the delivery pressure by the condensing conditions available. The pressure ratio, r, is, therefore, usually defined. Any increase in r will, in general, result in an increase of all the partial losses except the valve loss.

The primary effect of compressor clearance volume is to reduce volumetric efficiency. The indicated ideal volumetric efficiency can be readily expressed in the form

$$\eta_v = 1 - c\left(r^{\frac{1}{\gamma}} - 1\right)$$

This is the maximum volumetric efficiency for given values of pressure ratio and clearance volume, but the effect of heat transfer, gas leakage and valve performance is neglected.

The heat transferred during the cycle has been studied during six investigations - by Wirth (36)1932; Smith (28) 1934; Giffen and Newley (7) 1940; Lorentzen (18) 1949; Brown (2) 1951; and Gosney (9) 1953. The magnitude and complexity of heat transfer effects justify the efforts made to study them. They particularly affect the volumetric efficiency of a compressor pumping a vapour near its saturation temperature; the early investigations were conducted on relatively large slow speed ammonia machines, the last two on small high-speed Freon 12 machines. The tests by Brown were carried out on a compressor identical to that used in the present investigation, and it was shown that an increase from 0.5 dry to 80°F. superheat suction vapour resulted in 20% improvement in the indicated and 30% in the actual volumetric efficiency. The indicator diagram, is therefore, an unreliable guide to the volumetric efficiency of the compressor. This fact was appreciated, in general terms, as early as 1904 (37) and is discussed in more detail by Giffen and Newley. The indicator card is an asset, of course, in assessing other factors detrimental to compressor performance. A set of 14 indicator cards showing faulty compressor performance was collected by Lenhart, and published by Voss (33). An interesting feature is that in 9 cases, poor operation can be attributed, partly or wholly, to faulty valve performance.

The gas leakage loss depends largely on the design of piston clearances and valves. Fuchs, Hoffmann and Plank (5) obtained values from static and running trials using air. Gosney (9) observed that static leakage with refrigerant (Freon 12) was greater than leakage during operation. It is known that an investigation on piston leakage is in progress.

The effect of oil on any one of the previous factors is not necessarily negligible and the valve noise and valve life are also affected by it. The quantity of oil necessary at the valves is a matter of compromise between conflicting requirements. A paper by Higham (11) deals with this problem. The heat transfer rate will be affected due to the thermal resistance of the oil on the cylinder wall, and to the miscibility of the oil with a refrigerant. The leakage losses are dependent on the quantity, nature and temperature of the oil present. Fuchs, Hoffmann and Plank (5) showed the effect of oil level and oil temperature on leakage. The effect of oil level alone on this loss was $\pm$ 15%.

The effect of compressor speed on volumetric efficiency is difficult to predict since it affects heat transfer, gas leakage and valve performance. Heat transfer and leakage losses will be more serious at low speeds, but will be reduced as a percentage loss at higher speeds, since a greater mass of gas is then pumped per unit time. At high speeds valve performance may deteriorate. There may be an optimum speed therefore, at which the sum of these losses is a minimum. There are a few test results available, for various large ammonia compressors, in papers by Reed and Ambrosius (23), Jenks (14), Fuchs, Hoffmann and Plank (5), and Giffen and Newley (7). In general, speed has surprisingly little effect on actual volumetric efficiency.

1. 3 Valve Losses.

Loss of compressor performance due to the valves is caused by (a) the irreversibility of the throttling process through the valve, (b) gas leakage due to "blow-by" during closure and (c) gas leakage due to imperfect sealing. Aerodynamic, thermodynamic and dynamic aspects will, therefore, be involved. The valve, aptly described by Willey (35) as an "aerodynamic horror", disturbs the gas flow and the energy expended to overcome the resistance to flow will eventually be dissipated, due to viscosity, as a rise in gas temperature, decreasing both the pumping and volumetric efficiency of the compressor. This resistance is a function of the gas density, viscosity, velocity of sound in the gas, gravity, gas velocity and valve dimensions. Only the last two factors can be materially modified to reduce resistance for a particular gas being pumped and their effect may be examined by either static tests of the valve assembly or by compressor testing.

There have been four investigations to study the gas compressor automatic valve; Lazendorfer (17), 1931; Fuchs, Hoffmann and Schuler (6), 1941; Hanson (10), 1945; and

Costagliola (3), 1949.

Lazendorfer tested a ring plate valve under steady air flow conditions and also as a discharge valve in an air compressor under normal operating conditions, to find whether static test results were relevant to actual conditions. The comparison of flow through the valve calculated using the coefficient of discharge found for each valve lift by static test did not compare closely with the discharge through the valve as deduced from the change in cylinder pressure and volume.

Fuchs, Hoffmann and Schuler examined the losses through a number of ring plate valves. A series of static tests were conducted with water to find the effect of lift and fluid velocity on the throttle loss. This method simplified the flow measurement technique and a relatively small quantity of water had to be pumped to obtain velocities to give the same Reynold's number as that when a refrigerant was flowing at the mean velocities met with in practice. It was deduced from the results obtained that to avoid increase of the flow resistance coefficient, (Z), low lift was desirable; the necessary flow area to be obtained by fitting a larger number of valves.

Hanson published the results of wind tunnel tests with large (4X) scale wooden models of plate-valves. Modifications reduced the loss through the valve assembly considerably, but like all similar static tests, there were difficulties of correlation with actual conditions.

The gas compressor valve may not cause recognised flow patterns at any lift since the gas velocity through it continuously varies, and only 1/70 Sec. or less is available to pass the complete charge. However, recent work by Stanitz (29), with poppet valves, indicates that statically determined coefficients may be applied to dynamic conditions without great error.

Concerned with the performance of "feather" type reed valves for use in the multivalve arrangement of a free piston gas generator Costagliola made an important contribution to the subject by computing the valve losses mathematically. Three dimensionless parameters were evolved and the important criterion was found to be that involving flow area through the valves and piston speed. Valve dynamics only became important if flow areas were already adequate and for optimum dynamical conditions, it was shown that the valve should have no weight and a light spring giving infinite natural frequency to the valve system. This ideal could not be approached since impact stresses with a light valve were theoretically high.

1. 4 Valve Flutter.

That flutter occurs with this type of valve has long been suspected. It may be shown that a reversal of pressure difference across the valve can occur due to radial flow even under steady flow conditions. This is small compared to the reversed pressure difference which may be observed on occasion from indicator cards, due to pressure surges caused by the initial large pressure difference required, with high speed compressors, to overcome the effect of valve seat area and of valve inertia. Even when the valve breaks away from its seat the cylinder pressure may continue to fall until the flow rate of the gas is appreciable. The pressure fluctuations thus initiated may be so violent that the cylinder pressure may subsequently be greater than the pressure in the suction pipe in the case of the suction valve and less than the pressure in the receiver in the case of the discharge valve. Such phenomena led Bishop (3) to suggest that the inertia of the gas associated with the valve passages should be studied, the cylinder and valve passage being considered as a Helmholtz resonator. By making a number of simplifying assumptions he was able to calculate the discharge portion of an indicator diagram and obtain reasonable agreement with an actual card. The pressure fluctuations were so large that it was concluded that severe valve flutter must result.

Pressure wave action, which may appear in the suction or discharge line with a resultant wave of rarefaction in the cylinder, was studied in an air compressor by Voissel (31). The effect of various lengths and curvatures of pipe line was investigated and good agreement obtained with Sommerfeld's theory. Recently Willey (35) showed indicator cards where considerable wave action was present during discharge from an air compressor. When air was passed through a discharge valve a high pitched rattle was heard together with an audible note in the discharge pipe. It was deduced that the pipe resonance was enhanced, or perhaps initiated by the flow interruption due to valve flutter, since the pipe resonance ceased if the valve was secured. An unusual suction line (Fig. 1c) could be obtained with a resultant increase in volumetric efficiency. From the evidence available the pressure wave action is less troublesome during the suction process than during the discharge process, probably due to the lower piston velocity at commencement of suction. Gosney (9) showed that pressure wave action was unlikely in small compressors pumping Freon 12.

CHAPTER 2.

AIM AND SCOPE OF PRESENT WORK. (GAS COMPRESSORS).

The valve performance of a small high speed reciprocating gas compressor is examined following the general theme of the mathematical analysis by Costagliola, but evolving somewhat simpler equations. The primary object is to further knowledge on the partial losses contributing to the large difference which is known to exist between the theoretical indicated and actual volumetric efficiency of such a compressor. The study is mainly concerned with the action of a cantilever reed type suction valve and the losses due to it. Certain coefficients required for the theoretical analysis are determined by static flow tests.

Since the problem is so complex various assumptions must be made to obtain any analytical solution. A substantial experimental programme is therefore also required to find the actual effect of the principal variables involved. New techniques were necessary to reduce the experimental errors usually associated with compressor testing. The compressor performance is examined, with varying permitted valve lift, valve spring stiffness, valve weight, compressor speed and pressure ratio. Since compressors of this type are widely used to pump air and also in very large numbers in small domestic type refrigerating units, the test series are carried out using (a) air and (b) the dense refrigerant Freon 12.

CHAPTER 3.

THEORETICAL ANALYSIS OF THE PERFORMANCE OF THE SUCTION VALVE OF A SMALL HIGH SPEED GAS COMPRESSOR.

The usual approach to a study of the mechanically operated valve of an engine is to treat the valve assembly as an orifice and so obtain the mass flow in terms of pressure difference across the valve. In such an application the variable flow area, or valve lift, is a known function of the crankangle. In the case of the automatic valve the valve lift is an unknown function of the unknown pressure difference and is therefore also variable with compressor speed, valve weight, spring stiffness, fluid properties, etc. Hence the very simplicity of the automatic valve makes for a complex analysis, since a further relationship is required between the valve lift and pressure difference across the valve.

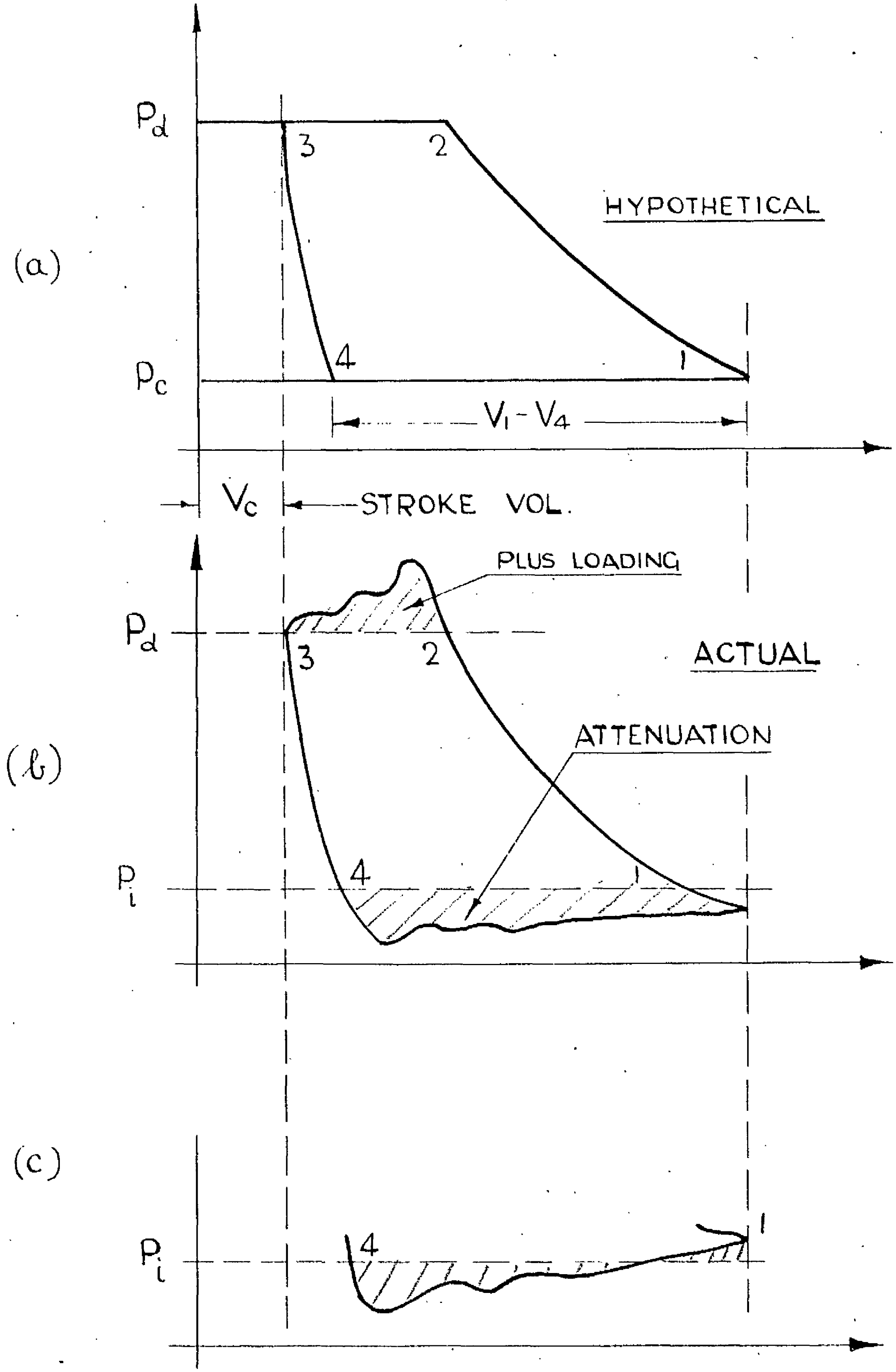
To obtain an analytical solution the following assumptions were made:-

1. Processes during the cycle were adiabatic except during the suction phase.
2. There was no gas leakage past the piston or closed valves.
3. The valve spring force was directly proportional to the valve lift.
4. The valve drag coefficient and coefficient of discharge were constant and the values were those determined by static test.
5. There was no damping of the valve movement by oil.
6. The inlet manifold was sufficiently large that the pressure remained constant there.

3. 1 Valve Opening and Pressure Drop from Flow Considerations.

From the standard formula for one dimensional subcritical flow through an orifice, and referring to Fig. 1.

$$W = c_d A\sqrt{\frac{2g\gamma}{RT_i(\gamma-1)}}\cdot P_i\cdot\sqrt{\left(\frac{P}{P_i}\right)^{2/\gamma}-\left(\frac{P}{P_i}\right)^{\frac{\gamma+1}{\gamma}}} \qquad (1)$$



HYPOTHETICAL AND ACTUAL P.V. DIAGRAMS

FIG. No. 1

If $p_i - p$ is small compared to p_i this may be reduced to

$$W = c_d A \sqrt{\frac{2g}{RT_i}} \cdot p_i \cdot \sqrt{1 - \frac{p}{p_i}} \qquad (2)$$

From the subsequent experimental data it may be shown that the calculated value of w at the worst combination of compressor and valve operating conditions is only in error by 6% (high) and under normal operating conditions the error is 1% (high). Hence a considerable simplification is effected by this reduction and the error incurred is not greater than the experimental error in the determination of c_d.

since $$C = \sqrt{g \gamma R T_i} \;,\; \frac{p}{p_i} = \varphi \;,\; A = A_o \alpha$$

then $$W = \frac{c_d A C}{V_i} \sqrt{\frac{2}{\gamma}} \cdot \alpha \cdot \sqrt{1 - \varphi} \qquad (3)$$

By consideration of the general energy equation for a gas and its differentiation with respect to time a further expression is obtained for w.

Neglecting heat transfer effects during suction, then, External work done = change of energy in the system.

$$\int_{V_4}^{V} \frac{PdV}{J} = -c_v m T + c_v m_4 T_4 + c_p (m - m_4) T_i \qquad (4)$$

where the last term on R.H.S. denotes the enthalpy addition to the closed system due to the flow energy of the induced charge.

Since $c_p = c_v + \frac{R}{J}$, $pV = RT$, $V = A_p z$

$$(m - m_4) P_i V_i - \int_{z_4}^{z} P A_p dz = J c_v \left\{ mT - m_4 T_4 - (m - m_4) T_i \right\} \qquad (5)$$

since $\frac{Jc_v}{R} = \frac{1}{\gamma - 1}$

$$\gamma m P_i V_i - (\gamma-1) m_4 T_i - (\gamma-1)\int_{z_4}^{z} P A_p dz = p A_p z - m_4 R T_4 - m_4 R T_i \tag{6}$$

dividing by $\gamma P_i A_p$

$$\frac{m V_i}{A_p} - \frac{\gamma - 1}{\gamma}\int_{z_4}^{z} \varphi \, dz = \varphi \frac{z}{\gamma} + \text{constants} \tag{7}$$

differentiating w.r.t. time

$$\frac{dm}{dt}\frac{V_i}{A_p} = \frac{\varphi}{\gamma}\frac{dz}{dt} + \frac{z}{\gamma}\frac{d\varphi}{dt} + \varphi\frac{dz}{dt} - \frac{\varphi}{\gamma}\frac{dz}{dt} \tag{8}$$

since $\theta = \omega t$

$$\frac{dm}{dt} = \frac{\omega A_p \varphi dz}{V_i \, d\theta} + \frac{\omega A_p z \, d\varphi}{V_i \, \gamma \, d\theta} \tag{9}$$

$$\frac{d\varphi}{d\theta} = \frac{w}{\omega}\frac{V_i \gamma}{A_p z} - \frac{\gamma}{z}\varphi\frac{dz}{d\theta} \tag{10}$$

since $w = \frac{dm}{dt}$, then (3) = (9) and w is eliminated.

$$\frac{d\varphi}{d\theta} = \frac{c_d A_o C \gamma}{\omega A_p z}\sqrt{\frac{z}{\gamma}} \, . \, \alpha \, . \, \sqrt{1-\varphi} - \frac{\gamma}{z}\varphi\frac{dz}{d\theta} \tag{11}$$

$$\frac{d\varphi}{d\theta} = \left(\frac{s\sqrt{8\gamma}}{2z}\right)\left(\frac{c_d A_o C}{\omega A_p s}\right) . \, \alpha \, . \, \sqrt{1-\varphi} - \left(\frac{\gamma}{z}\frac{dz}{d\theta}\right)\varphi \tag{12}$$

$$\varphi' = (L)(G) \cdot \alpha \cdot \sqrt{1-\varphi} - (N)\varphi \tag{13}$$

where L and N are known functions of θ, dependent on compressor dimensions. G is a known dimensionless parameter depending on operating conditions: if multiplied by a factor $\sqrt{\frac{8}{\gamma - 1}}$ it is equal to the flow parameter B used by Costagliola. This is the first relationship between the unknown valve lift ratio α, the unknown pressure ratio φ, and the crankangle θ.

3. 2 Valve opening and Pressure Drop from Dynamic Considerations.

The following relationship between φ, α and the crank angle θ is that given by Costagliola (3).

Force due to fluid drag on the valve = $C_D \, a (p_i - p)$

Through the small valve lift the spring stiffness may be assumed constant. There is no spring load when the valve is closed. Hence

$$\frac{m_v}{g}\frac{d^2h}{dt^2} = C_D\, a(p_i - p) - kh \tag{14}$$

$$\frac{m_v}{g}\,\omega^2 h_o \frac{d^2\alpha}{d\theta^2} = C_D\, a\, p_i(1-\varphi) - kh_o\alpha \tag{15}$$

$$\frac{m_v}{gk}\,\omega^2\frac{d^2\alpha}{d\theta^2} = \left(\frac{C_D\, a\, p_i}{kh_o}\right)(1-\varphi) - \alpha \tag{16}$$

since $\omega_n = \sqrt{\frac{gk}{m_v}}$ and if $q = \frac{\omega}{\omega_n}$, $J = \frac{C_D\, a\, p_i}{kh_o}$

$$q^2\frac{d^2\alpha}{d\theta^2} - J(1-\varphi) + \alpha = 0 \tag{17}$$

where q and J are known dimensionless parameters dependent on compressor characteristics and operating conditions. When evaluating J, Costagliola used $C_D = 1.3$. This is the velocity drag coefficient of a circular disc suspended in a broad stream. The pressure drag coefficient for the overall valve assembly was shown by experiment (Appendix III) to be about 0.2 when based on the whole valve face, or 0.535 when based on the port area.

3. 3 Relationship Between L, N and θ.

The relationship between z and θ is deduced in the usual way from the compressor dimensions. If z is the distance from the piston crown to the valve plate

$$z = r\left(1 - \cos\theta + \frac{r}{\ell}\frac{\sin^2\theta}{2}\right) \quad + \text{clearance} \tag{18}$$

$$= r\left(1 - \cos\theta + \frac{r}{4\ell} - \frac{r}{4\ell}\cos 2\theta\right) \quad + \text{clearance} \tag{19}$$

$$\frac{2z}{s} = \left(2\delta + \frac{r}{4\ell} - \cos\theta - \frac{r}{4\ell}\cos 2\theta\right) \tag{20}$$

where $\delta = \frac{1}{2} + \frac{\text{clearance}}{\text{stroke}}$

$$\frac{2dz}{s\,d\theta} = \left(\sin\theta + \frac{r}{2\ell}\sin 2\theta\right) \tag{21}$$

A mean value of the fractional clearance throughout the experimental work was 5% hence $\delta = 0{\cdot}55$ and since $\frac{r}{\ell} = 0.142857$ for the compressor examined,

$$\frac{2z}{s} = \left(1{\cdot}1357 - \cos\theta - 0{\cdot}0357\cos 2\theta\right) \tag{22}$$

$$\frac{2dz}{s\,d\theta} = \left(\sin\theta + 0{\cdot}071428\sin 2\theta\right) \tag{23}$$

Hence $$L = \frac{\sqrt{8\gamma}}{(1{\cdot}1357 - \cos\theta - 0{\cdot}0357\cos 2\theta)},\; N = \frac{\gamma\left(\sin\theta + 0{\cdot}071428\sin 2\theta\right)}{(1{\cdot}1357 - \cos\theta - 0{\cdot}0357\cos 2\theta)} \tag{24}$$

The graphs of L and N to a base of θ are shown to a small scale in Fig. 2.

3. 4 Solution of equations.

The valve lift and pressure drop across the valve, as expressed by the unknown variables α and φ respectively, are given by the simulatneous equations

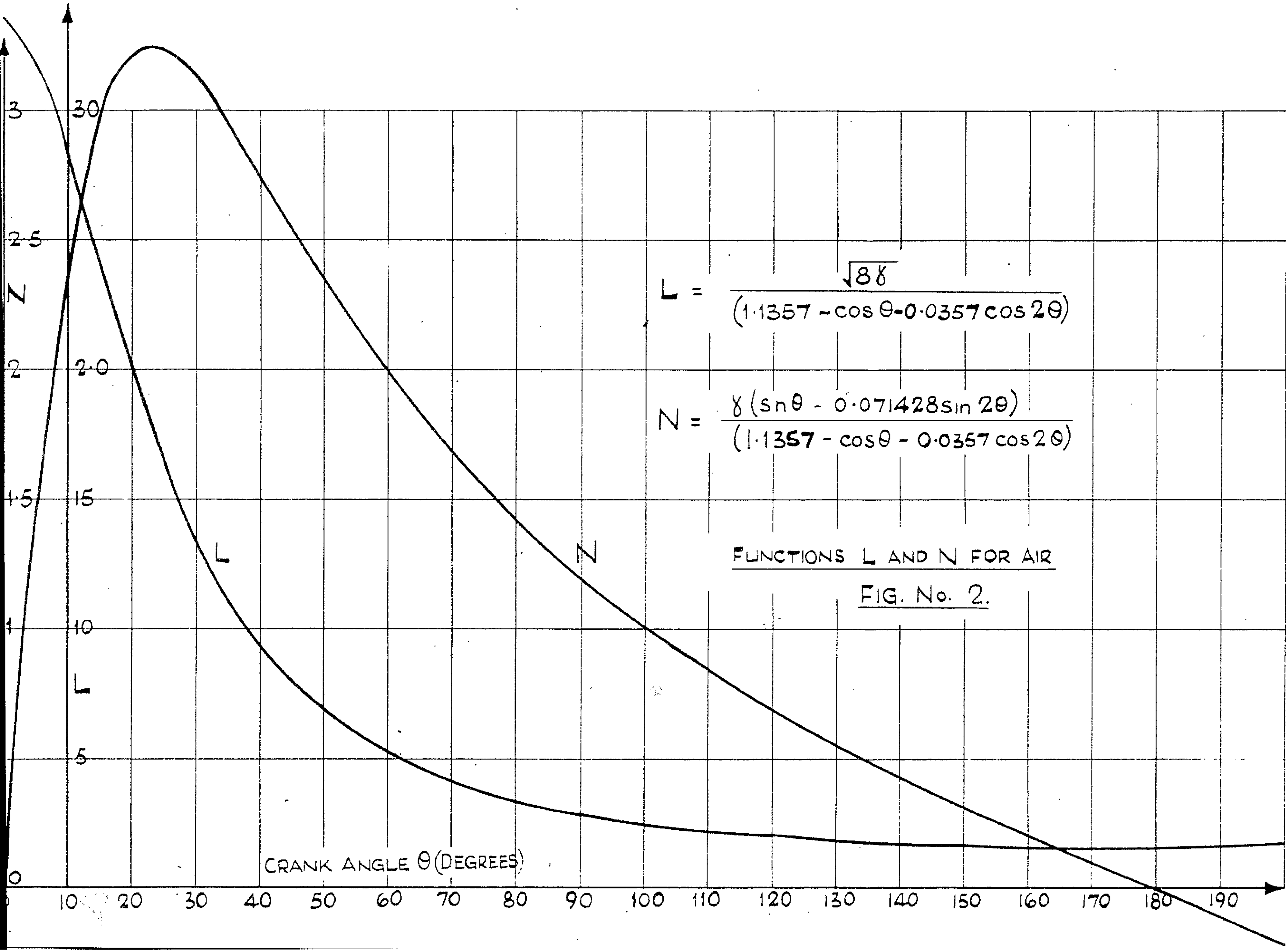
$$\varphi' = LG\alpha\sqrt{1-\varphi} - N\varphi \tag{13}$$

$$q^2\alpha'' = J(1-\varphi) - \alpha \tag{17}$$

where L and N are known functions of θ, dependent on compressor proportions and gas properties, and G.J and q are known parameters dependent on compressor dimensions, operating conditions and gas properties. The range of these parameters for the compressor used in the experimental work is shown in Fig. 3.

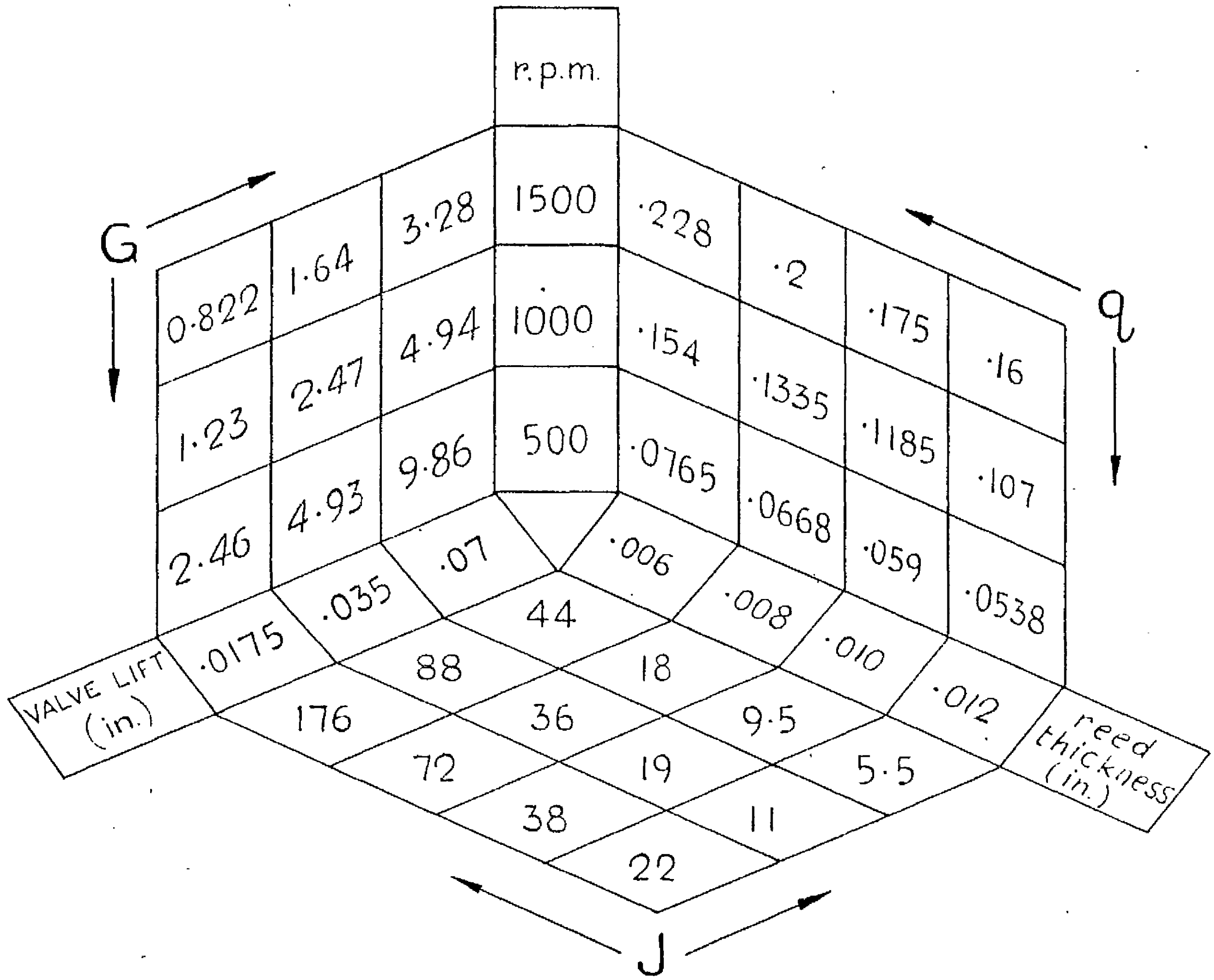
The whole suction process may be broken down into a number of stages.

Stave I. The valve is opening, and both equations hold. The initial value of θ (θ_4, Fig. 1.) is obtained from the isentropic re-expansion of the clearance gas. Initially $\alpha = 0$ and $\varphi = 1$. The stage normally finishes when $\alpha = 1$, i.e. the valve meets the stop. The valve may come to rest before reaching the stop.



FUNCTIONS L AND N FOR AIR

FIG. No. 2.



DIMENSIONLESS PARAMETERS G.J.q. FOR SMALL COMPRESSOR (AIR)

FIG. Nº 3.

Stage II. If the valve remains open against the stop only eq. 13 holds, till the pressure falls to a value given by $\varphi = 1 - \frac{1}{J}$ when the valve starts to close. Under certain conditions the valve may leave the stop and return to it several times before the final closing process, i.e. valve flutter may take place.

Stage III. The valve is closing and both equations hold. Initially $\alpha = 1$ and $\varphi = 1 - \frac{1}{J}$. The stage finishes when <u>either</u> $\alpha = 0$ or $\varphi = 1$.

Stage IV. The final stage continues till both $\alpha = 0$ and $\varphi = 1$. This is a stage of interest in a study of the loss due to gas leakage by blow-by after piston reversal at B.D.C.

The expression developed by Costagliola equivalent to equation (13) had no direct solution. In an attempt to remedy this, the present simpler form was deduced. However, there is still no algebraic solution and it was finally concluded that the difficulties were inherent to such an expression which, based on thermodynamic considerations, results in a non-linear equation.

Since direct solutions could not be obtained a step-by-step process by Milne's (74) methods of numerical calculus were sought. The behaviour near $\varphi = 1$ of the term $\sqrt{1-\varphi}$ caused difficulties and neither a starting process by Taylor's series nor by starter formulae was successful.

Equation 17 is a form of the second order differential equation lacking the first derivative which is found in many vibration problems. Since $\varphi_0 = 1$ at $\theta = \theta_0$ (= θ_4 Fig. 1.) at the start of opening, then the solution* is

$$\alpha = -J \int_{\psi=\theta_0}^{\psi=\theta} \varphi' \left[1 - \cos \frac{\theta - \psi}{q} \right] \psi' \qquad (25)$$

* see reference 71, page 403.

where ψ is a variable of integration which vanishes when the limits are substituted. If it is assumed meantime that remains constant during opening then at $\theta = \theta_{stop}$.

$$\alpha_{stop} = 1 = \mathcal{J}\varphi_q' \int_{\psi=\theta_o=\theta_{seat}}^{\psi=\theta_{stop}} \frac{1}{q}\left[1 - \cos\frac{\theta-\psi}{q}\right]\psi' \qquad (26)$$

$$\text{The integral} = \frac{\theta_{seat} - \theta_{stop}}{q} - \sin\frac{\theta_{seat} - \theta_{stop}}{q} = H \qquad (27)$$

which is plotted to a small scale in Fig. 4.

Initially $\varphi_o = 1$ at $\alpha_o = 0$ and from eq. 13 $\varphi_o' = -N$ which may be evaluated from eq. 24.

Hence H seat to stop $= \frac{1}{\mathcal{J}q N_o}$ from which the first approximation of the interval θ seat - θ stop. is obtained and is shown in Fig. 5.

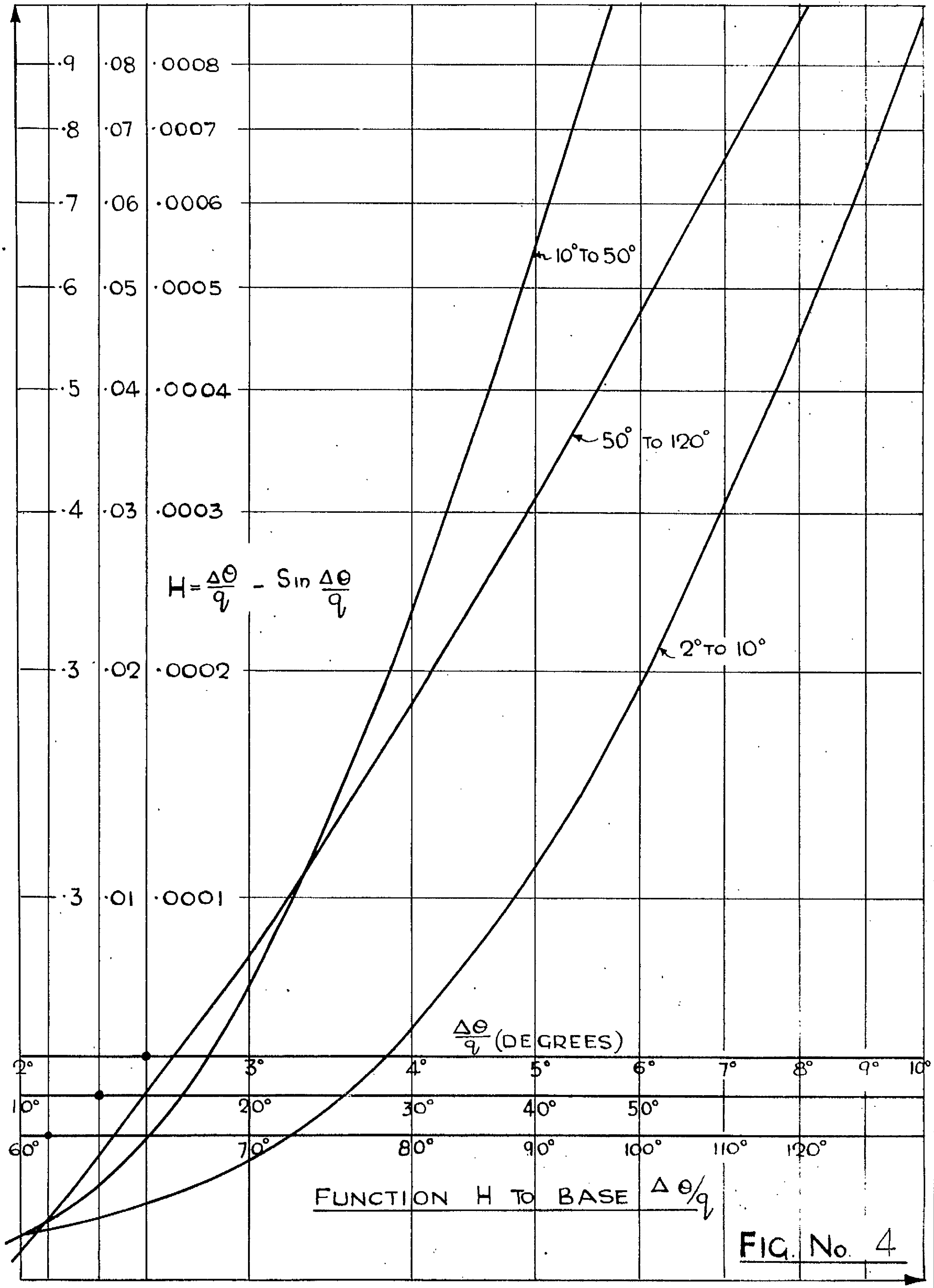
The subsequent more precise determination of this interval showed that the first approximation was quite good, although somewhat too large in every case investigated. In the shortest time of valve opening the first approximation was 2.28° compared to the more precise value of 2°; under conditions for the longest time the first approximation of 12.1° compared to the precise value of 10.9°.

Similarly the first approximation of the valve motion is given by

$$\alpha = -q\mathcal{J}\varphi_o' H = \text{constant} \times H \qquad (29)$$

and the approximate valve displacement curves during opening, which may be shown to approximate to cubic parabolae, are obtained.

By arranging eq. 25 several but similar methods of precise step-by-step determination of α and φ are possible. While none was found to be neater than that of Costagliola, the present



FUNCTION H TO BASE $\Delta\theta/q$

FIG. No. 4

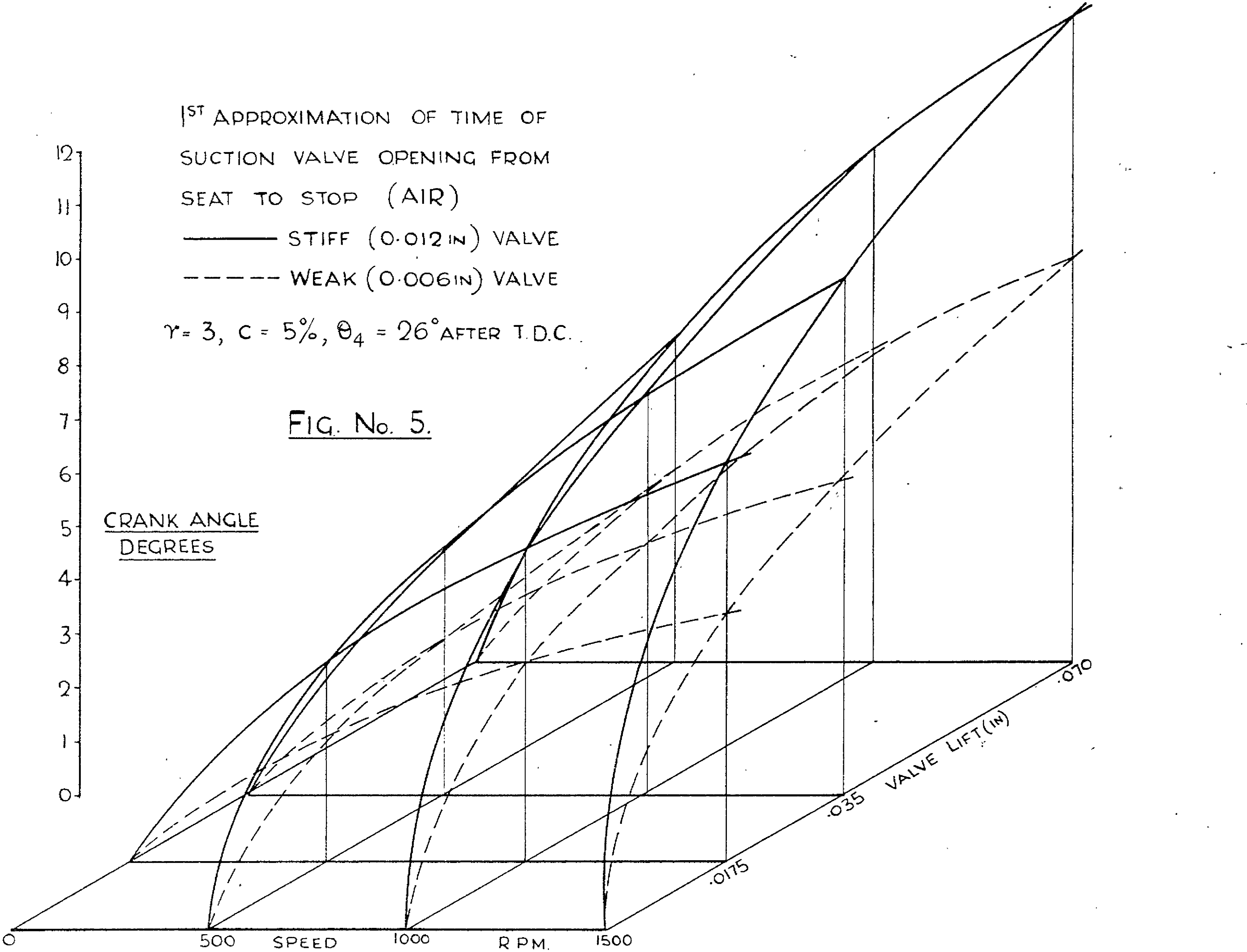

1ST APPROXIMATION OF TIME OF
SUCTION VALVE OPENING FROM
SEAT TO STOP (AIR)
——— STIFF (0·012 IN) VALVE
- - - - - WEAK (0·006 IN) VALVE
γ = 3, C = 5%, θ4 = 26° AFTER T.D.C.
CRANK ANGLE
DEGREES
0
1
2
3
4
5
6
7
8
9
10
11
12
0
500
SPEED
1000
R.P.M.
1500
VALVE LIFT (IN)
.0175
.035
.070

FIG. No. 5.

simplification of the equations enabled his graphical methods to be superceded and solutions obtained by calculating machine. While the process is still laborious, values of α and φ correct to 0.1% could be obtained in a reasonable time after operating experience. Solutions converged rapidly during valve opening and four steps appeared sufficient for the determination of the α and φ curves.

If $\frac{\theta_{stop} - \theta_{seat}}{4} = \Delta\theta$ with θ_{stop} determined by the first approximation in Fig. 13, and suffices 0, 1, 2, 3, 4 indicate equal increments during opening, then the values of H_1, H_2, H_3, H_4, are $H_1 = \frac{\Delta\theta}{q} - \sin\frac{\Delta\theta}{q}$, $H_2 = \frac{2\Delta\theta}{q} - \sin\frac{2\Delta\theta}{q}$ etc.,

and $\Delta H_1 = H_2 - H_1$, $\Delta H_2 = H_3 - H_1$, $\Delta H_3 = H_4 - H_1$

For step (1) $\quad \alpha_1 = -q^J \varphi_0' H_1 = +q^J N_0 H_1 \qquad (30)$

With this value of α, φ_1' is obtained from eq. 13 with appropriate values of L and N and the mean value of φ' over the interval $\Delta\theta$ is

$$\varphi'_{mean} = \frac{1}{2}\left[\varphi_0' + \varphi_1'\right]$$

$$\text{and} \quad \varphi_1 = 1 + \varphi'_{mean}\,\Delta\theta \qquad (31)$$

So by similar steps to step (4) whence

$$\alpha_4 = -q^J\left\{\varphi_3' H_1 + \varphi_2'\Delta H_1 + \varphi_1'\Delta H_2 + \varphi_0'\Delta H_3\right\}$$

$$\text{and} \quad \varphi_4 = \varphi_3 + \varphi'_{mean}\,\Delta\theta \qquad (32)$$

Since the first approximation of the time taken for the valve to open ($\Delta\theta$) was always too large the value of α_4 from the "exact" solution was always greater than unity. With the revised value of $\Delta\theta$, the whole process was again carried through, experience enabling a guess of $\Delta\theta$ to be made so that α_4 was correct to within 0.1% after this second computation.

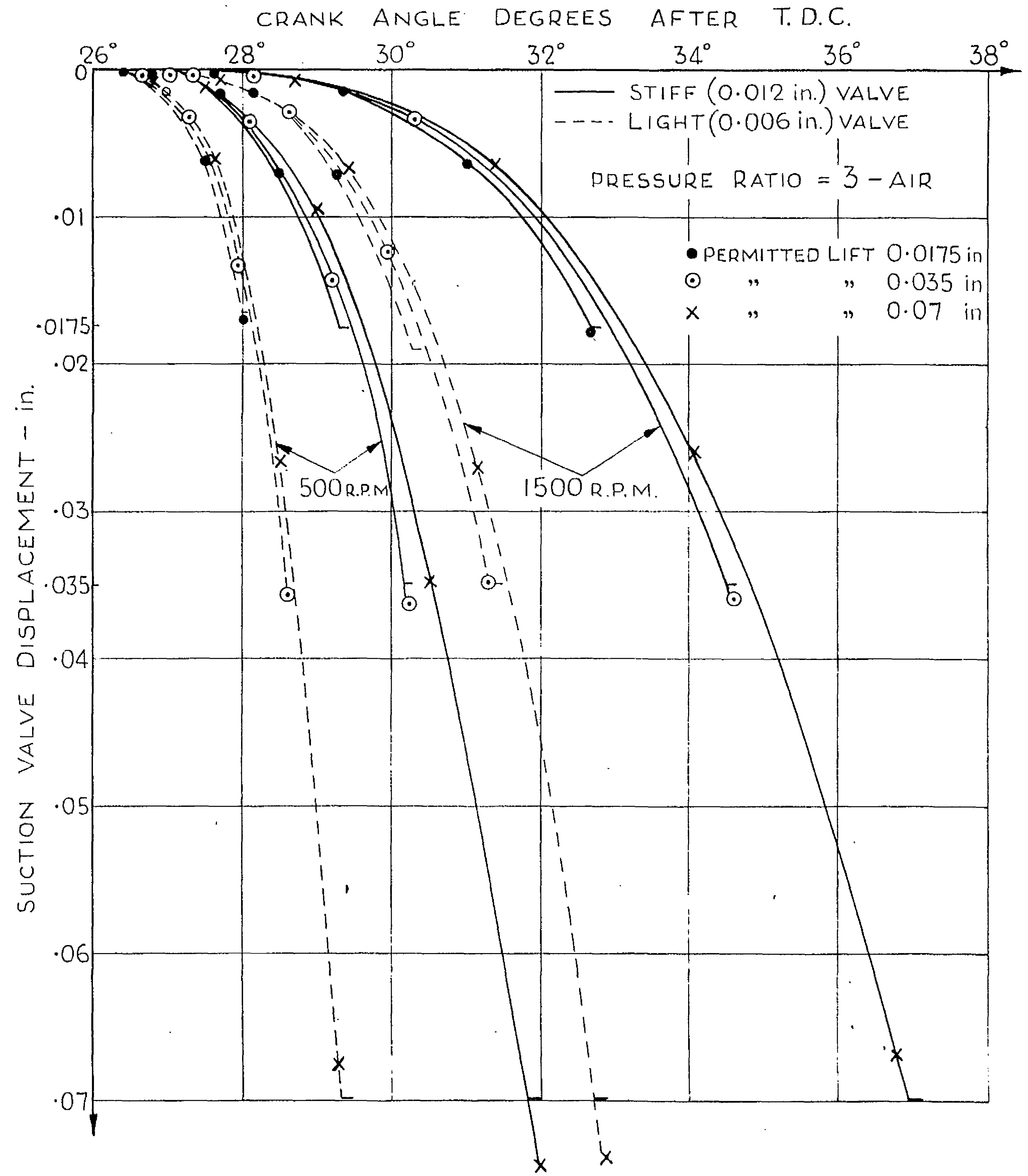
The upper and lower extreme values of compressor speed and valve spring stiffness at three permitted valve lifts were examined. The curves of valve lift during opening and the curves for pressure drop across the valve for these twelve cases are shown in Fig. 6 and Fig. 7.

3. 5 <u>Discussion of Results</u>.

The initial part of the suction loop showed that the fall in pressure in the cylinder was a function of the compressor speed and of the valve spring stiffness, but was independent of permitted valve lift. As the pressure difference across the valve decreased from the maximum reached, the curves at any one speed and valve spring stiffness always diverged slightly for different permitted lifts, a phenomena which was not readily explained.

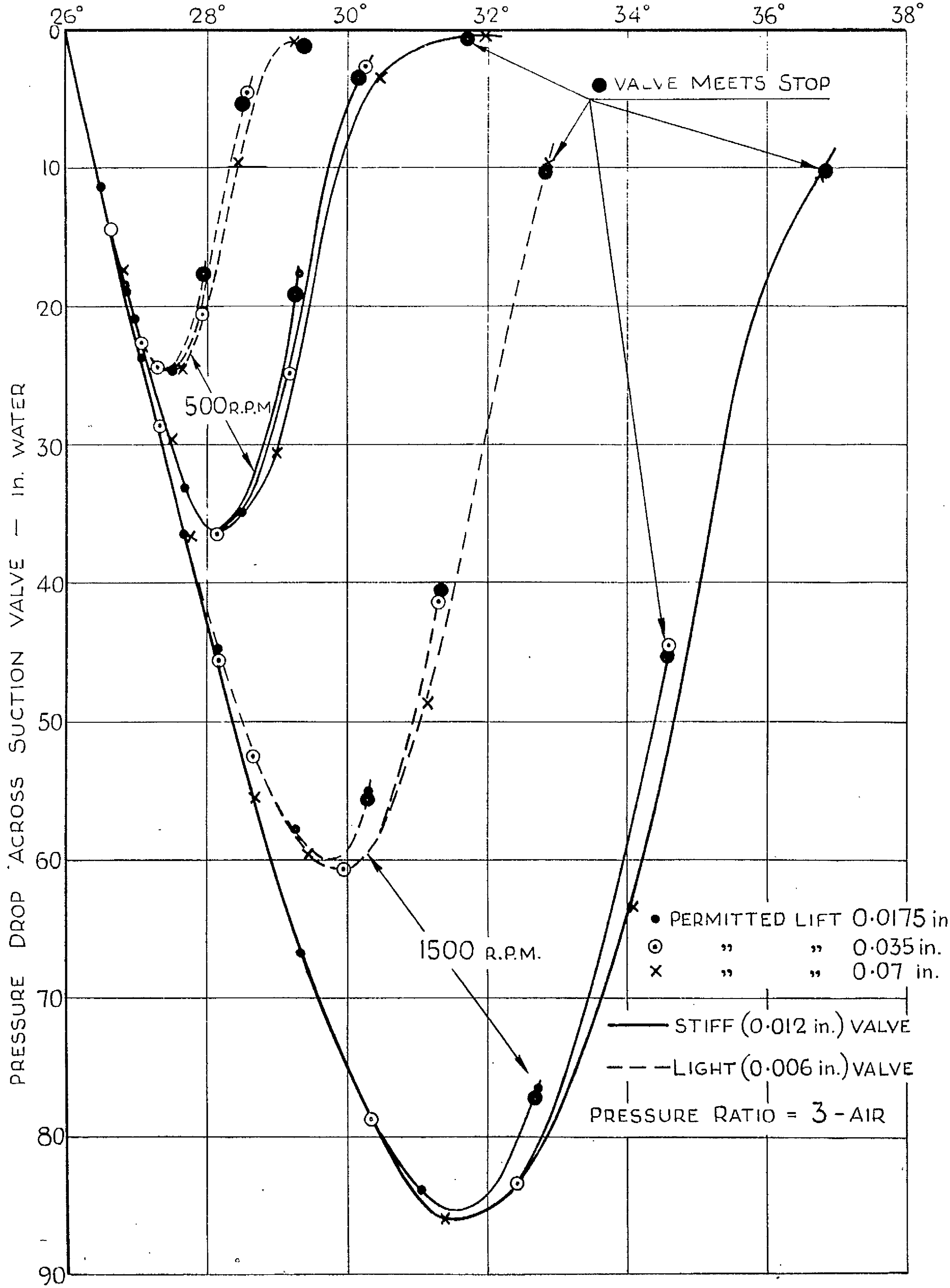
Within the range examined the pressure difference across the valve as it neared the stop was rapidly decreasing for each permitted valve lift except the smallest. This result has far reaching consequences.

In the cases of minimum permitted lift the valve will remain against the stop since the pressure difference across the valve will increase due to increasing piston velocity. At higher permitted lifts, the valve will probably leave the stop as the pressure difference is decreasing. The subsequent decrease in valve flow area together with increased piston velocity will result in an increase in φ and a return to the stop. Hence a flutter of the valve will be initiated. In the event of the valve failing to reach the stop on opening, at very high permitted



THEORETICAL SUCTION VALVE DISPLACEMENT DURING OPENING

FIG. Nº 6.



THEORETICAL PRESSURE DROP ACROSS SUCTION VALVE DURING OPENING PHASE

FIG. Nº 7.

lift, it will tend to close from the point of rest and subsequently reopen to a high lift, or to the stop, due to increasing piston velocity. That this occurs has been shown by recent experiments with a ring type valve without a limiting stop where the discontinuous curve of valve displacement on opening was observed by the technique described later. It was also observed with the cantilever reed valves when there was no valve lift limit (see Chapter 4. 5).

When the valve reaches the stop and stays there the subsequent analysis for stage II is relatively simple. Eq. 17 lapses and eq. 13, with $\alpha = 1$ holds until the decreasing piston velocity results in a decrease of pressure difference across the valve to a value given by

$$\varphi = 1 - \frac{1}{J} \qquad (33)$$

at which the valve begins to close. During closure (Stage III) equation 13 holds together with

$$\alpha = 1 - J\int_{\psi=\theta_{\text{leaving stop}}}^{\psi=\theta} \varphi'\left[1 - \cos\frac{\theta-\psi}{q}\right]\psi' \qquad (34)$$

However these circumstances arose only if the permitted valve lift was somewhat less than 0.0175 in. The later experimental work shows that the actual volumetric efficiency was constant over a wide range of valve lift, but the low lift (0.0175 in.) lay below this range for both air and Freon 12 and suction valve throttling then adversely affected volumetric efficiency. This case, therefore, is not of practical interest for the compressor examined.

The value of φ obtained from eq. 33 gives the pressure difference which must exist across the valve, in inches of water, if, on reaching the stop, the valve is to remain there.

Permitted Valve Lift (in.)	0.0175	0.035	0.07
Light Spring (0.006 in. thick valve reed).	2.3	3.23	6.93
Stiff Spring (0.012 in. thick valve reed).	18.5	37	74

Since the pressure difference across the valve is always decreasing as it approaches the stop and since there will be some rebound due to impact at the stop, it is evident that valve flutter will occur in the majority of cases examined. As shown later the flow conditions are such that, once initiated, this flutter is readily sustained. The valve movement and valve pressure loss may still be computed by eq. 13 and 34, or by 13 alone in the (unlikely) event of the increasing piston velocity eventually holding the valve on the stop. However, the values of α and φ vary so rapidly during valve flutter it was found that very small increments of crankangle (a fraction of a degree) would be required in the step-by-step process and the labour of computing over 150 degrees becomes impracticable. Further, if valve blow-by losses are important the main interest centres on the end point when the valve has just returned to its seat, and the errors have continued to accumulate to that point.

Hence the analysis of the valve opening phase has explained some of the phenomena associated with the behaviour of a vallve of this type. However, within the range of valve lifts of practical interest, valve flutter occurred and the mathematical equations require an impracticable amount of time for their complete solution.

CHAPTER 4.

EXPERIMENTAL MEASUREMENT OF VALVE MOVEMENT AND VALVE PRESSURE LOSS.

The movement of a cantilever type valve reed during the cycle in a small high speed compressor was recorded with various valve and compressor variables. On account of the lightness of the valve in a small compressor, the high rotational speed and the possibility of high frequency oscillations due to flutter and impact, mechanical indicators are not suitable and a new experimental technique was developed.

The cyclical pressure drop across the suction valve while in operation was measured. This provided a "light spring" diagram of the actual attenuation during suction and the loss of volumetric efficiency due to this effect was computed. The effect of valve and compressor variables on the throttle loss was observed.

4. 1 Apparatus.

(a) The Gas Compressor. The compressor had a single acting, single cylinder, 1½ in. bore, 1 in. stroke. The cylinder head consisted of a valve plate and a top cover. The valve plate carried the cantilever type valve reed. The suction valve port through this plate was 3/8 in. long and 11/32 in. diameter. The exhaust valve consisted of a bridge holding a beam type valve. The clearance volume was found by measurement of all the clearance spaces, and the fractional clearance varied from 4.64% to 4.91% depending on the thickness of the valve reed in use.

The compressor was belt driven by a 1 B.H.P. compound wound D.C. Motor. Speed was measured by a stroboscope focused on the compressor pulley and directly coupled to the mains frequency.

The compressor speeds chosen were all simple fractions of this.

(b) The Valve Reed. The cantilever type valve reed (Fig. 9) is typical of that in common use in small gas compressors. To provide a range of valve weights and stiffness, sets of valves were stamped from steel plate thickness, 6, 8, 9, 10, 12 x 10^{-3} in. This was the widest practical range. A lighter valve would be liable to failure by collapse into the valve port when the high and low side pressures acted across it; a heavier and therefore stiffer valve was unlikely to reach the desired valve lifts at low compressor speeds. The Swedish manufacturers of the plate provided a chemical analysis:- Carbon 0.95 to 1.1%, Manganese 0,25 to 0.4%, Silicon 0.25 to 0.35%, Phosphorous 0.023% max., Sulphur 0.075% max.

The physical properties of the material were determined in preliminary tests. The Vickers Pyramid Number was about 600. The Ultimate Tensile Strength, averaged from several test pieces, was 107 tons/in.^{2}, this high figure being due to the strain hardening from the cold rolling process in the manufacture of the plate: the makers claim an U.T.S. 120 tons/in.^{2}. The average surface finish of the valve reeds, measured by Talysurf recorder varied from 2 to 7 micro inches. From tests with ten specimens the fatigue strength was found to be 53 tons/in.^{2} on a basis of 16 x 10^6 reversals. The natural frequency of the valve reeds was measured by electromagnetic oscillator and also calculated. The spring stiffness was calculated from the natural frequency and also measured.

Valve reed thickness, in. x 10^{-3}	6	8	9	10	12
Fundamental frequency, ~/sec. (measured)	100	121	136	144	166
Fundamental frequency * (calculated)	109	125	133	141	155
Spring Stiffness lb/in.	0.237	0.563	0.8	1.1	1.9

* The second mode would be at 6.16 x fundamental frequency.

(c) Valve Lift Control. Preliminary tests showed that it was necessary to devise a control of the maximum permitted valve lift which fulfilled the following requirements:-

(i) Varied the valve lift throughout the range required, without stopping the compressor.

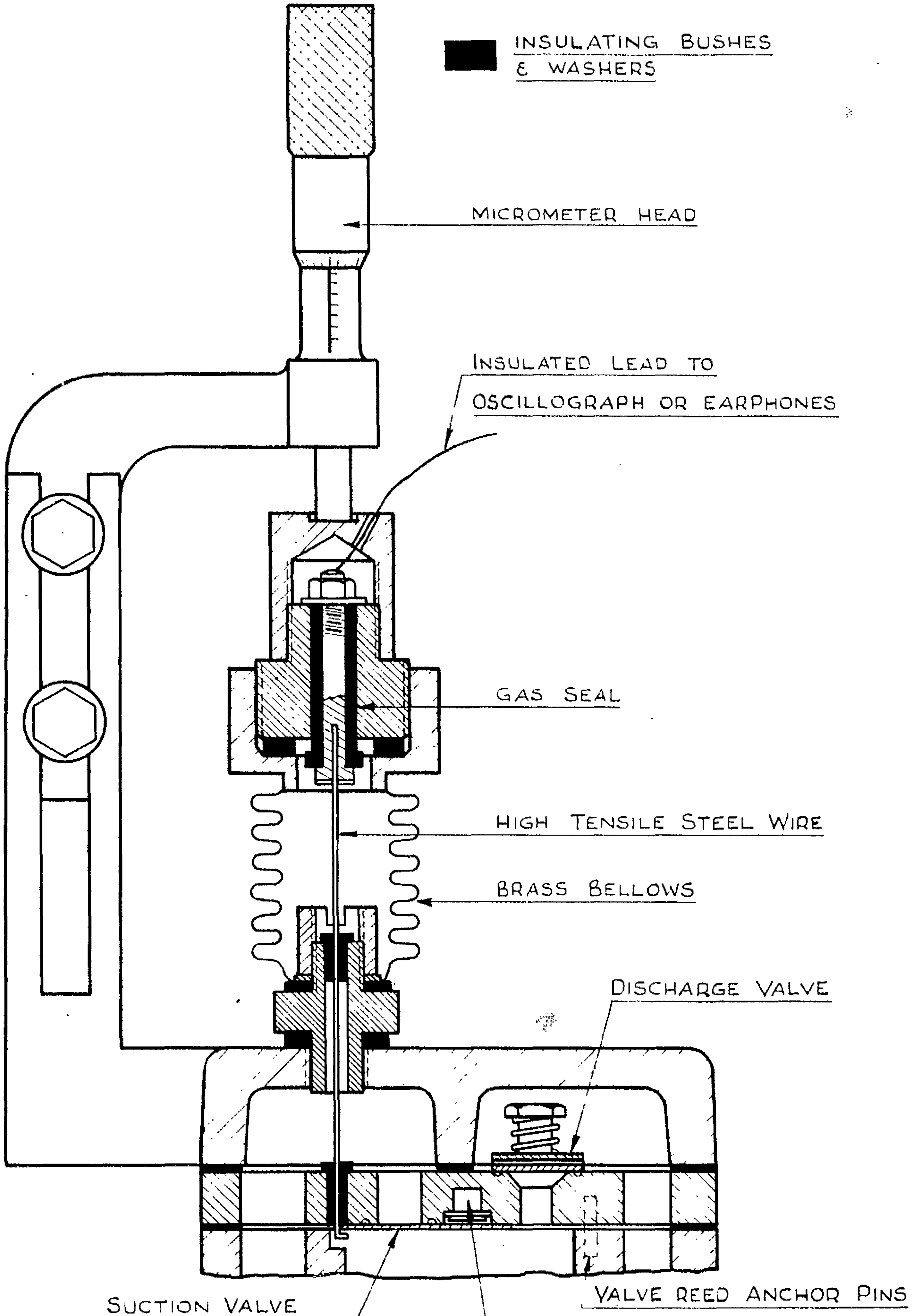
(ii) Measured accurately the valve lift.

(iii) Measured the maximum valve lift, if this lift was less than that permitted by the check.

(iv) Did not alter the very small clearance volume.

(v) Was absolutely gas tight.

These requirements were met by the device shown in Fig. 8. The hard steel wire 0.042 in. dia. was operated by the micrometer head from outside the compressor fulfilling (i) and (ii). To provide for (iii) this wire was electrically insulated from the compressor and so formed a "make and break" contact with the moving earthed valve reed. Increase of the permitted valve lift by the micrometer until no contact was recorded indicated the maximum valve lift, at the tip, for any given operating condition. The recording of the contact was shown visually by the breaking of a trace on the oscilloscope screen. During most of the testing, however, earphones were used to record contact at the maximum lift. An increase in the depth of the milled check in the cylinder wall had to be made to allow the desired range of lifts. The slight increase of clearance volume was partly offset by the volume of that part of the control wire which projected into the cylinder so largely fulfilling requirement (d). The transmission of the micrometer head movement through the cylinder head without any gas leakage (e), was obtained by fitting a brass bellows seal as shown. The electrical lead necessary for the maximum lift indicator was taken through a small gland in the top of the bellows.



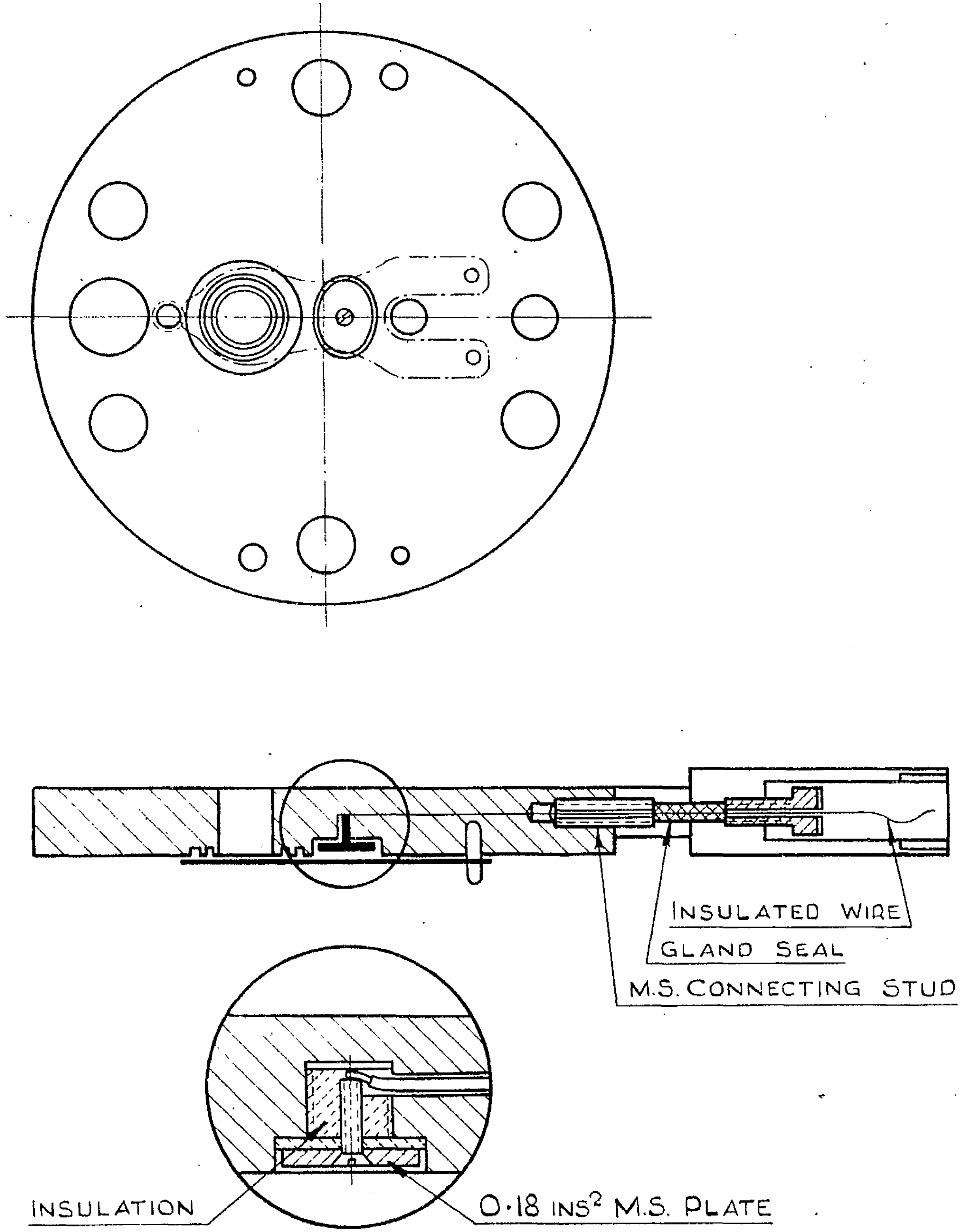
EXTERNAL CONTROL OF SUCTION VALVE LIFT. FIG. Nº 8

SCALE : FULL SIZE

(d) Valve displacement pick-up. Since the valve in a small compressor may weigh only 0.03 oz. no attachment to it is feasible. A mechanical indicator, with its inherent inertia forces would seem precluded although mechanical devices were used in larger compressors by Lazendorfer, to indicate ring plate valves, and by Costagliola to indicate feather valves, the latter (with some difficulty) operating at speeds up to 2000 r.p.m.

The indicator should not affect the small clearance volume and should be easily installed, accurately calibrated, reliable and gas tight. An electrical device meets most of the requirements; the pick-up may be small and a lead from it readily taken out to the bulk of recording equipment sited clear of the compressor. The present choice of physical phenomena which may be used to convert a mechanical movement to a proportional electrical effect is small due to the various limitations imposed by the problem. Either a capacitative or inductive pick-up would suit. The former was selected as it has less bulk, is more robust and of simpler construction, and the working fluids (air and Frenn 12) have good dielectric properties. The pick-up unit is shown in Fig. 9; the moving valve reed formed the earthed plate of a variable condenser and the fixed insulated plate, 0.18 in.^{2} area was bedded in the valve plate. Gas sealing difficulties necessitated a proper gland seal for the lead from the unit. The calibration of the unit is shown in Fig. 53, Appendix IV.

(e) Valve Displacement Recorder. (Fig. 10). Since a large range of frequencies of the moving valve, down to static conditions for calibration purposes, would be experienced, some form of "carrier" was desirable. Hence the pick-up unit was



VALVE PLATE & PICK-UP ELEMENT

FIG Nº 9.

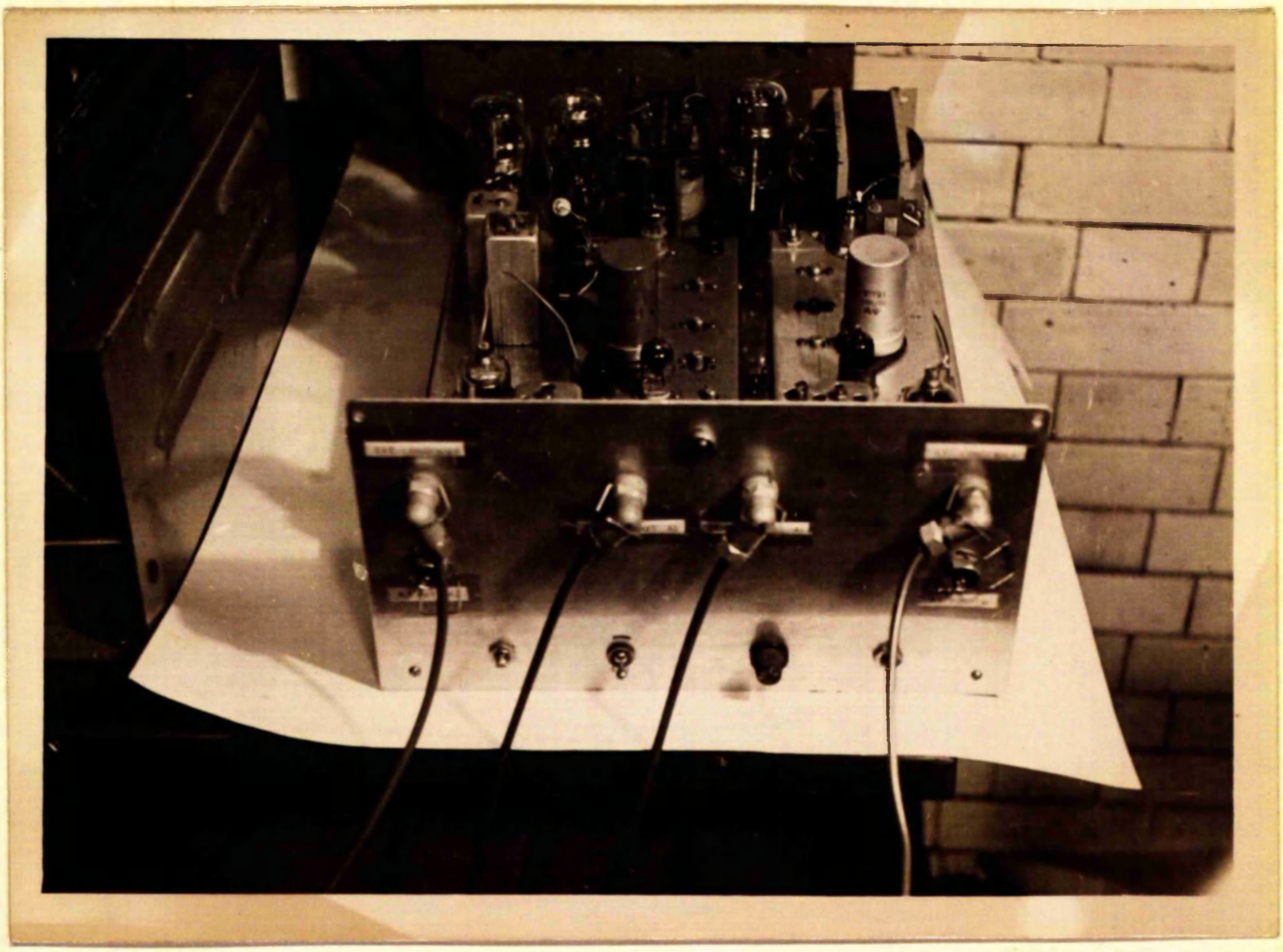
energised by an oscillator, tuned to 500 Kc/s. The original intention was to use a crystal oscillator, but the unit shown in Fig. 12 based on Colpitts' circuit, proved to be very stable.

The pre-circuits for capacitative type pick-up units may be broadly classed as resonant circuits and bridge circuits. The bridge circuit has an inherent advantage of easy tuning (usually the best balance of the bridge), and while both methods would serve the present purpose the bridge method was selected and a two channel unit constructed as shown by the block diagram (Fig.11). The construction was such that the bridge units were detachable from the whole and could be replaced by the inductive bridges used in the water pump valve investigation reported later.

Cathode followers were provided as buffer stages between (a) each bridge and the single oscillator supplying both channels, and (b) between each bridge and the following amplifying stage. These prevent interference between each channel and extraneous disturbance of the bridge voltages.

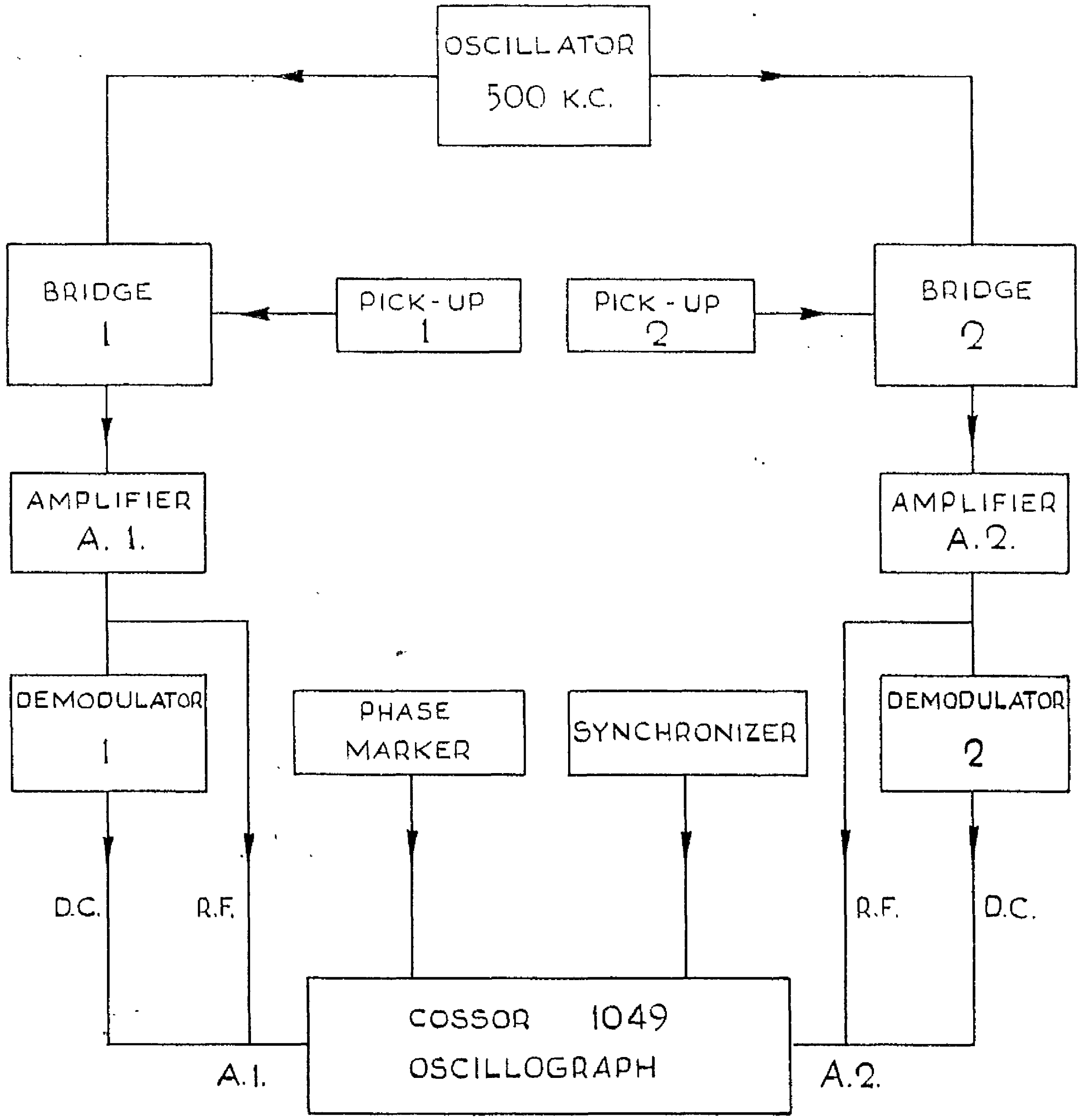
Since one side of the pick-up, the valve reed, was necessarily earthed, a double ended output was inevitable from the bridge. The varying potential difference across the bridge was, therefore, supplied to a differential amplifier and the single output then passed through the cathode follower to the main amplifier. A demodulator stage was incorporated to filter the R.F. carrier and so give a simple line trace. This stage could be readily by-passed to facilitate the balancing of the bridge by viewing the amplitude of the carrier.

Two prototype recorders operated successfully at carrier frequencies of 200 Kc/sec. and 250 Kc/sec. respectively. The final compact two channel unit described here, tuned to 500 Kc/s,



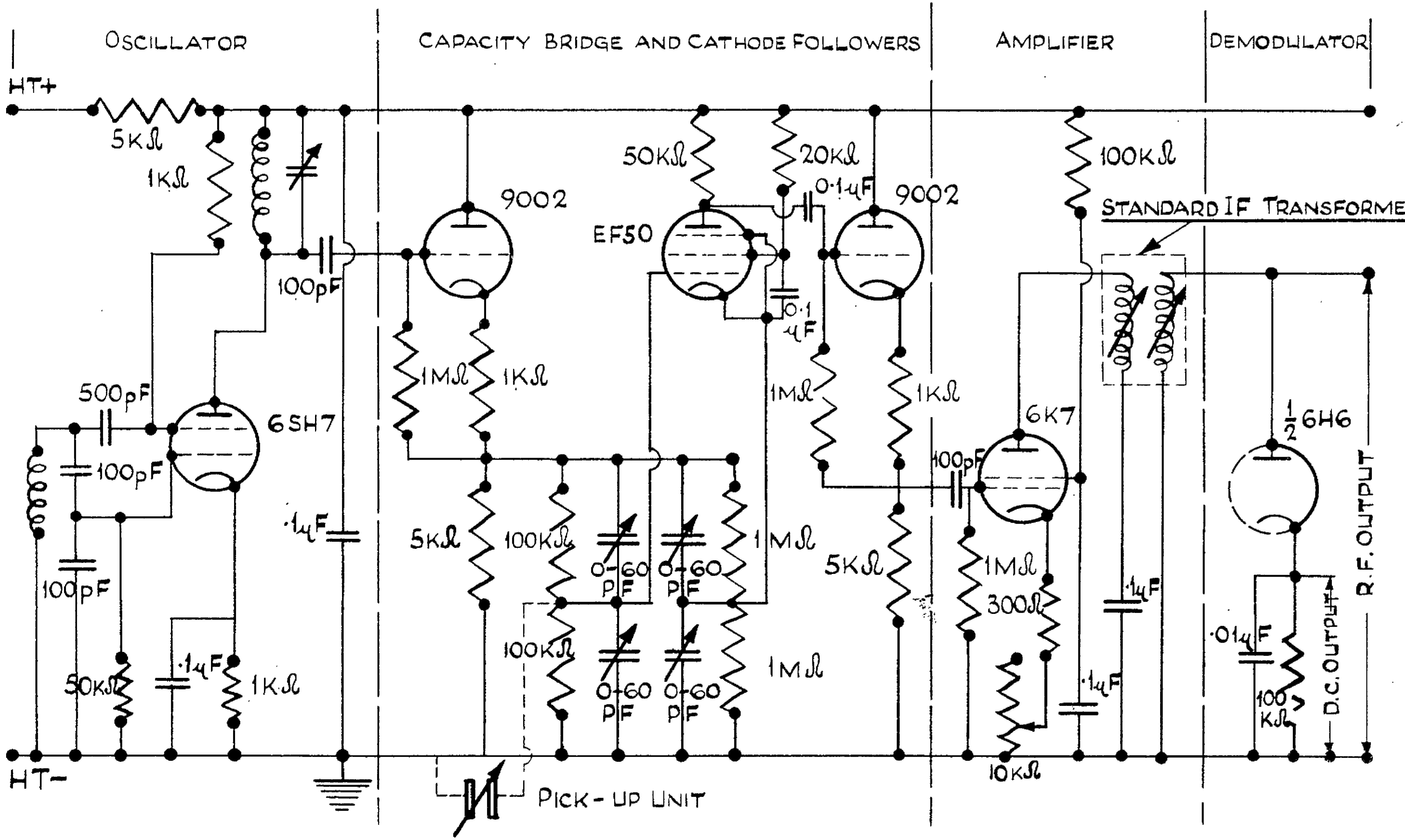
2 CHANNEL VALVE DISPLACEMENT RECORDER

FIG. Nº 10.



BLOCK DIAGRAM OF
2-CHANNEL VALVE DISPLACEMENT RECORDER

FIG. Nº 11.



CIRCUIT DIAGRAM FOR 1 CHANNEL OF VALVE DISPLACEMENT RECORDER

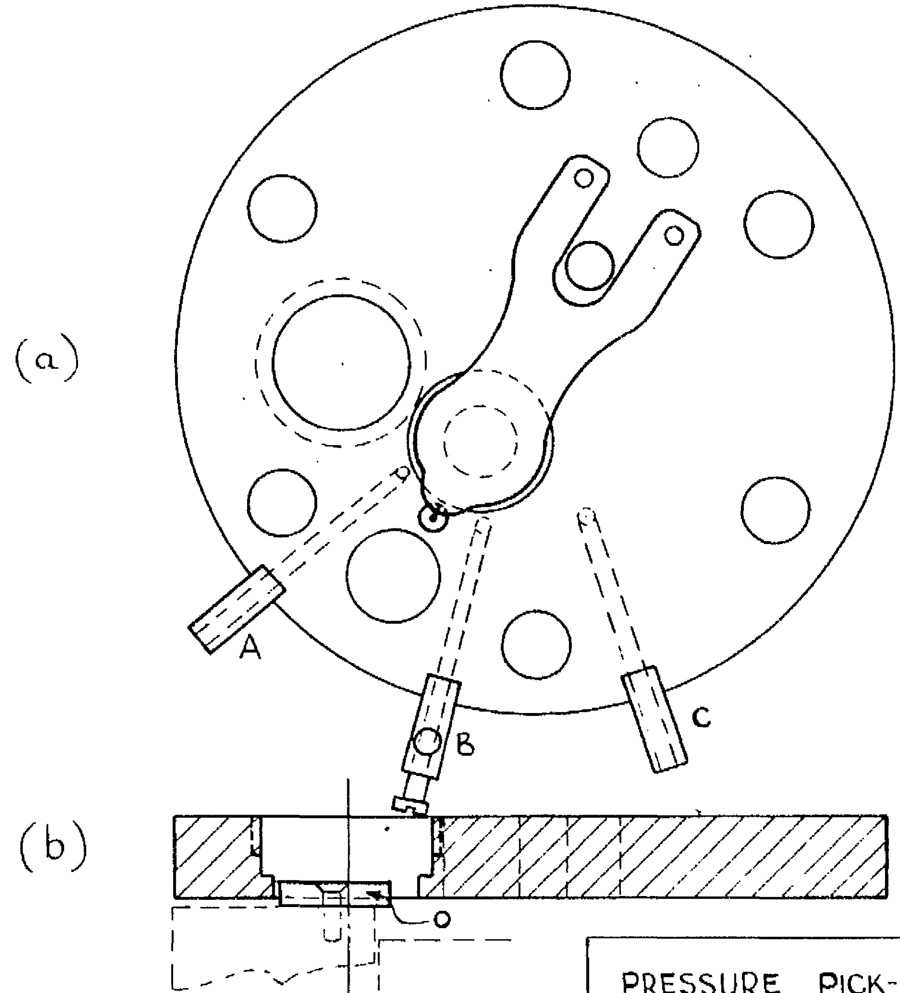
FIG. No. 12.

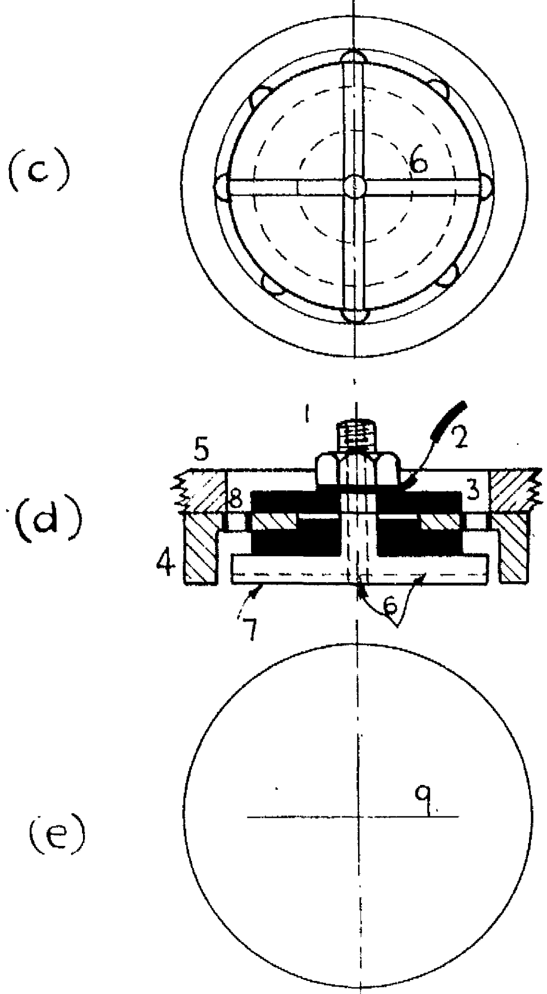
permitted the use of standard R.F. amplifiers. The amplifier gain in each channel was made to suit the different gains of the two channels of the Cossor, model 1049, D.C. double beam oscilloscope. One beam was used to show the amplitude modulated carrier and the other to record phase marking. The traces obtained were photographed by a Cossor camera, model 1425, using 35 m.m. Kodak super XX film.

(f) Differential Pressure Pick-up. The pressure drop across the suction valve was measured by a pressure differential capacitive type pick-up recording through the two channel unit already described.

Capacity type pressure pick-ups have been frequently used but additional problems in this application made difficult the construction of a suitable unit. The back of the diaphragm had to be open to the suction head pressure while the front was exposed to the cylinder pressure. The pick-up had to be sensitive enough to measure accurately a pressure difference of 1 in. water across the suction valve while open, yet sufficiently robust to withstand a pressure difference of 50 lb/in.^{2} or more during delivery, without alteration of calibration. The method of calibration, which for accuracy must be done with the element in situ, also presented problems.

The attractive arrangement would be to have the diaphragm integral with the unit body, and therefore gas tight, and the whole screwed into the valve plate flush with the cylinder side, leaving the clearance volume unaffected. Several units so constructed, from various types of steel, failed to provide an element of sufficient sensitivity. A suitable unit was finally obtained (Fig. 13). A disphragm cut from shim steel 0.002 in.





PRESSURE PICK-UP ELEMENT

A, B, C, PRESSURE TAPPINGS FOR CALIBRATION.

0. PACKING PIECE.

1 ELECTRODE PIN AND FIXED PLATE

2. TAG WASHER AND LEAD.

3. INSULATING BUSHES

4. LOCATING BRIDGE RING

5. LOCKING RING

6. OIL RELIEF GROOVES AND VENTS

7. 0·0005 IN DIAPHRAGM CLEARANCE WITH MICA DIELECTRIC.

8. VENT HOLES IN BRIDGE

9. 0·002 IN SHIM STEEL DIAPHRAGM.

FIG. No 13.

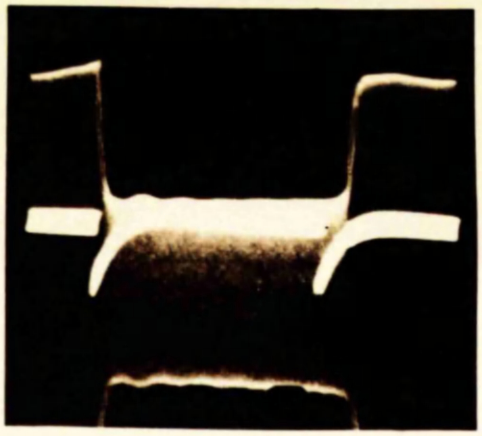
Scale – (a) and (b) Full size

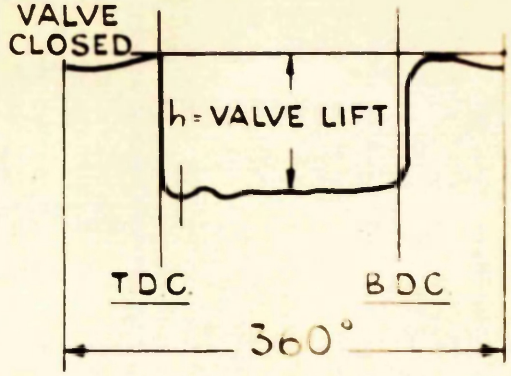
(c), (d) and (e) Twice full size

thick was clamped in the valve plate, thus providing minimum diaphragm thickness with maximum sensitivity. The diaphragm was honed round the edges and both sides alternately lapped on plate glass. Grooves were cut in the face of the fixed insulated plate to provide adequate ventilation of the back of the diaphragm and in the hope that traces of oil would collect there rather than in the gap. This plate, the insulating bushes and locating ring were firmly assembled and the faces of the locating ring and plate trued. A cut of 0.0005 in. then taken from the insulated plate provided the clearance. The face of the locating ring was lapped and then used to lap the step in the valve plate which provided the other locating face for the clamped stretched diaphragm. The dielectric gap was filled by splitting mica till a piece close to 0.0005 in. was obtained. It must be fitted exactly, otherwise the large signal generatedduring discharge would swamp the amplifier, which would not recover quickly enough from overloading to record accurately the desired signal during suction. The assembled unit, which did not exceed the depth of the valve plate (3/8 in.) overlapped the cylinder wall and so allowed a packing disc to be inserted, largely offsetting the increase in clearance volume. The calibration of the unit is shown in Fig. 54, Appendix IV. The fractional clearance was 5.2% with the unit in place, depending on the valve reed in use; i.e. about 0.6% greater than without the unit.

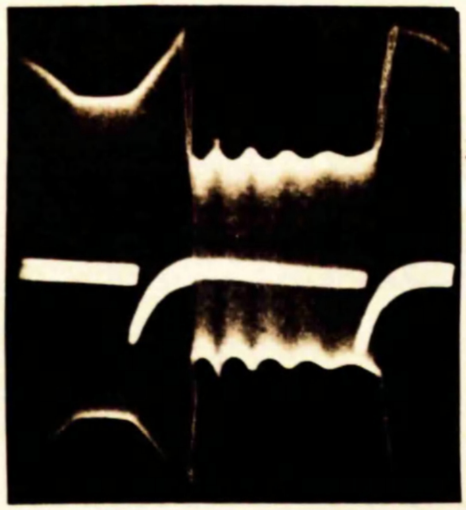
4.2 Interpretation and Measurement of Valve Displacement Oscillograms. Oscillograms of the suction valve displacement to crank angle were recorded for compressor speeds of 500, 1000 and 1500 r.p.m. while pumping air at discharge pressures of 0, 100 and 200 lb/in.^{2} gauge, with valve lifts (at tip) of 0, 0.035 and 0.07 ins. for four valve stiffnesses, i.e. reed thickness 0.006, 0.008, 0.010 and 0.012 in. A set of three of these 108 unrectified diagrams is shown in Fig. 14. The delay in suction valve opening, due to re-expansion of air in the clearance space is marked X in the accompanying sketches. X_1 – X is the time for the valve to open to the stop. The delay in closing after B.D.C. is clearly shown: a slight rebound from the valve seat is just discernable, magnified due to increase of sensitivity of the pick-up near valve closure (see Fig. 52). The large displacement Y which increases with increasing discharge pressure and with decreasing valve reed thickness is due to the punching of the valve reed into the suction port. The large pressure difference across the suction valve during delivery results in a displacement of the reed at the pick-up unit as shown in Fig. 15. This distortion of the valve reed was also observed in the oscillograms (Fig. 15) which recorded the make and break between the valve reed tip and the stop. The vertical displacements in these latter oscillograms have no significance. When the large pressure difference acted across the suction valve during discharge, the valve reed tip was sufficiently bent downwards to make contact with the valve lift check as shown by R, if the permitted lift was small.

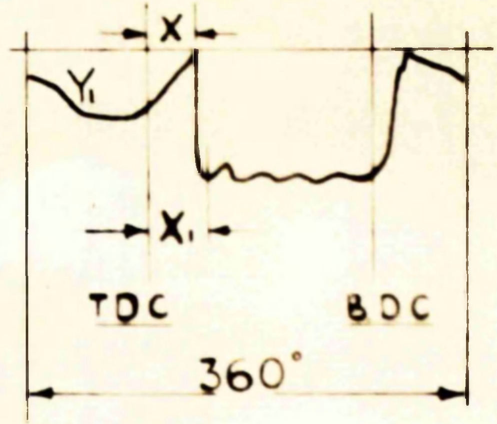
The oscillograms of valve displacement were enlarged and measured. Some of the results obtained are shown in Fig. 16.



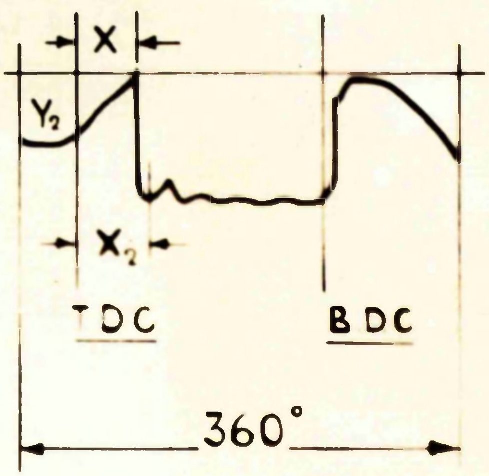


DISCHARGE PRESSURE 0 lb/in²





DISCHARGE PRESSURE 100 lb/in²

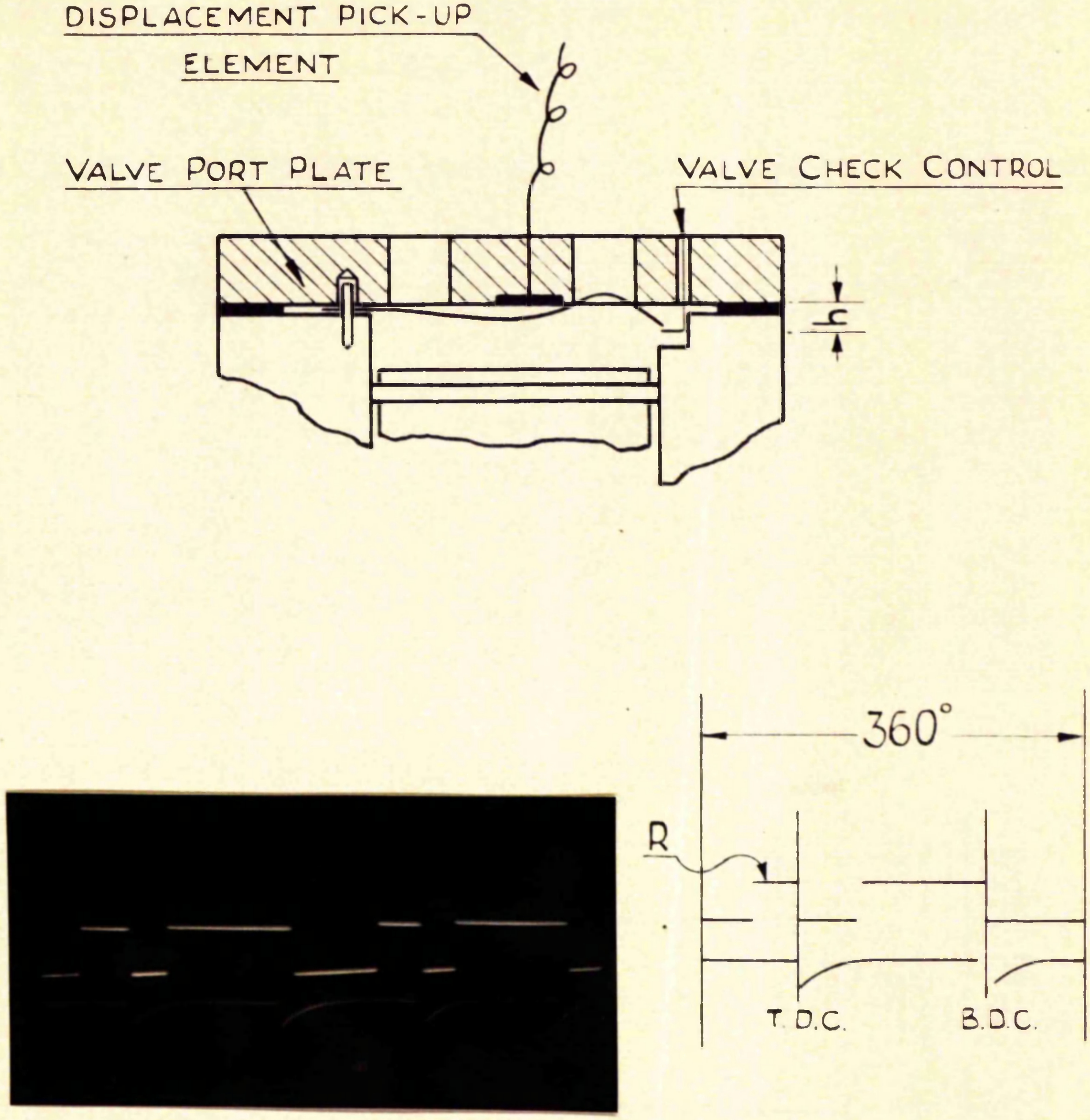


DISCHARGE PRESSURE 200 lb/in²

INTERPRETATION OF VALVE DISPLACEMENT DIAGRAMS

AIR - SPEED 500 RPM h = 0·07 IN - ATMOSPHERE SUCTION

FIG. No 14.



0·010 in. VALVE, h = 0·005 in, 1000 r.p.m. 100 lb./in.² DISCHARGE

DISTORTION OF SUCTION VALVE DURING DISCHARGE

FIG. Nº 15

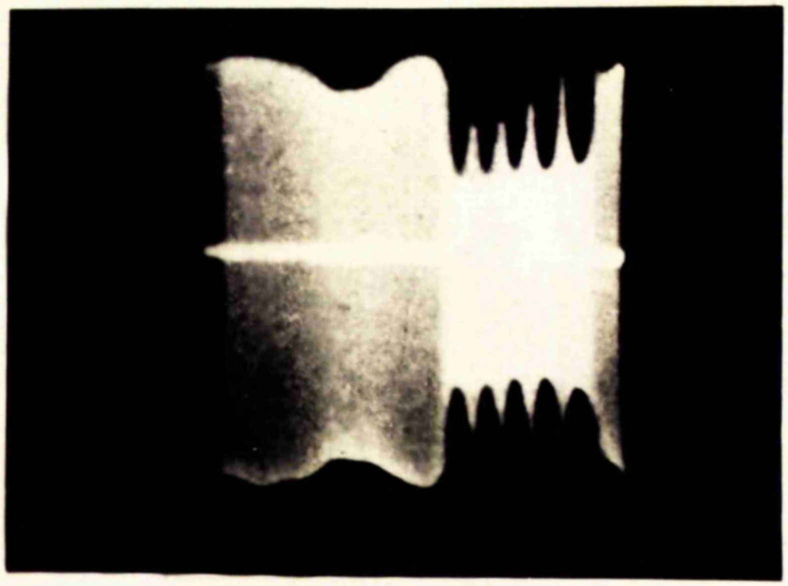
Reed Thickness in.	Discharge Pressure $lb/in^2 g$	Delay in Suction Valve Opening. Crank-Angle Degrees after T.D.C.		Full Open Period of Suction Valve Crank Angle Degrees 1500 R.P.M.		
		1000 R.P.M.	1500 R.P.M.	Lift 0 in.	0·035 in.	0·07 in.
0·006	0	10·8	10·9	206	197	185
0·008		11·7	11·0	186	174	180
0·010		11·8	11·6	202	180	170
0·012		12·0	12·0	202	180	175
0·006	100	43	44	144	134	130
0·008		48	48	156	132	120
0·010		52	52	155	150	145
0·012		54	60	156	145	138
0·006	200	61	65	150	130	130
0·008		66	66·5	-	-	132
0·010		65	70	138	138	156
0·012		69	72	144	132	120

Some Measurements from Suction Valve Displacement Oscillograms - Air

Fig. No. 16.

The inaccuracies involved in the various experimental procedures together with the irregular behaviour of the valve made it difficult to observe small changes in valve action. While the 108 oscillograms gave the first picture of the valve in action and the general trends, quantitative data from them was not very accurate. An improvement in the technique, at great expense for film, would be to drive the film, and so extend the time base. It may be seen, however, that the delay in point of valve opening was only slightly affected by compressor speed, about 3^{o} average, at the highest discharge pressure, where the speed effect is most pronounced due to relatively high piston speed at valve opening. The delay increased noticeably with increase in valve inertia and spring stiffness. The period during which the valve may be considered fully open decreased with increase of valve stiffness and increase of valve lift but there was a wide scatter of individual results due to the effect of oil and valve flutter.

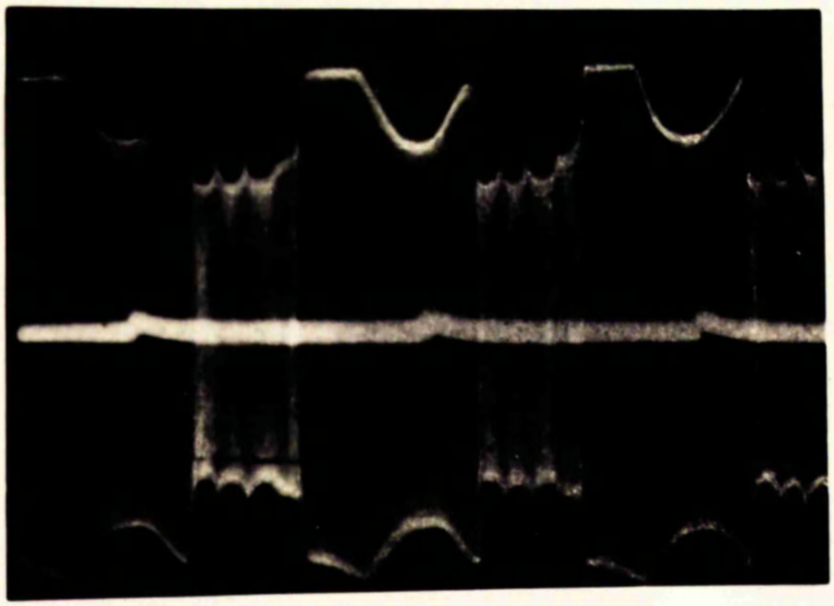
4.3 Valve Flutter. When the valve lift stop was in operation the flutter was usually of the small amplitude shown in Fig. 14, and the frequency was close to that of the natural frequency of the valve reed as a simple cantilever. It was found that the amplitude was severely damped, particularly with the lighter valves, due to the oil on the stop. As shown in Fig. 18 the flutter could interfere with the commencement of the closing process. Normally the flutter tended to die away, but exceptions were not uncommon. Fig. 17 shows the build up of flutter during the time the valve was open. It is possible that self excited flutter of the valve takes place; that there is an alternating force which sustains the flutter, the necessary



BUILD-UP OF VALVE FLUTTER

0·012 IN. VALVE, h = 0·08 in, 1000 r.p.m. DISCHARGE 200 lb/in^2

FIG. No. 17



INTERFERENCE BETWEEN VALVE FLUTTER AND CLOSING PROCESS

0·006 IN. VALVE, h = 0·035 IN., 1500 r.p.m, DISCHARGE 200 lb/in^2

FIG No 18

energy being supplied by the flowing fluid, even if the fluid flow is steady. Consider the valve vibrating about an average position. In this average position the outflow from the suction valve into the cylinder is equal to the inflow into the compressor head cover. If the valve is further open than this position the outflow is greater than the inflow, i.e. the pressure in the head cover is reduced. If the valve is further closed than the average position the outflow is less than the inflow, i.e. the pressure in the head cover is increased. Therefore, when the valve is just past the average position and opening, pressure is diminishing, so that ½ cycle later, when the valve reaches the average position but closing, the pressure in the head cover is a minimum. Similarly, when the valve passes the average position during its opening phase the pressure is a maximum. Hence the pressure energy does work on the flutter. The energy so supplied may be considered as a negative damping force. This force will be partially offset by the positive damping due to the viscosity of the flowing fluid, thus preventing an exponentially increasing amplitude of flutter. The frequency of such a flutter will be, for all practical purposes, the natural frequency of the valve system. Since this type of flutter is not dependent on dynamic considerations it should be possible to reproduce it by static test. However, during the static flow tests (Appendix III) it was observed that flutter could neither be initiated or maintained by a steady gas flow. In the compressor the greater the permitted valve lift the more severe was the flutter, due to the decrease in the damping effect of the valve stop. With the heavy valve the amplitude was often equal to the whole valve lift.

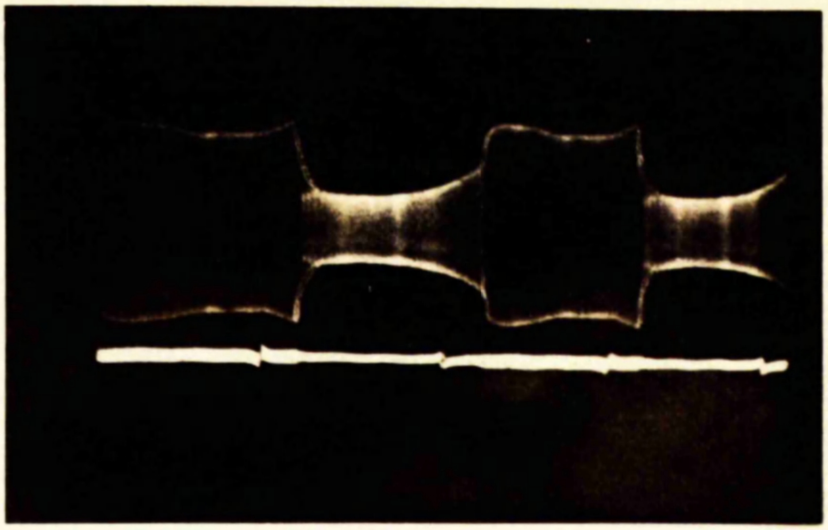
4. 4 Valve Action with Freon 12. Valve displacement oscillograms were obtained by the same technique when pumping Freon 12 with this compressor in a refrigerating circuit. The range of pressure ratios available was necessarily much reduced. The same compressor speed range was used at two pressure ratios with fixed evaporating conditions (5°F.) with three valve reed thicknesses, 0.006, 0.009 and 0.012 in. at three valve lifts. Specimen oscillograms are shown in Fig. 19.

While the general form of the oscillograms is similar to those obtained when pumping air, a feature not previously observed was obtained at high compressor speeds. Close to T.D.C. when the valve reed was distorted, it temporarily returned to the valve plate (point D, Fig. 19) leaving again just after T.D.C. This was due to the piston striking the reed, the pressure of the clearance gas being sufficient again to distort the reed during re-expansion. Thus the clearance volume, while adequate to accommodate the flat reed, could not accommodate the distorted reed, with consequent severe punishment to the already highly stressed valve.

The valve flutter was seldom greater than in the examples shown, but there were a few cases of build up of flutter similar to Fig. 17. In the tests with air a greatly increased amplitude was obtained by throttling the suction. Perhaps at lower evaporating temperatures the flutter would increase in severity.

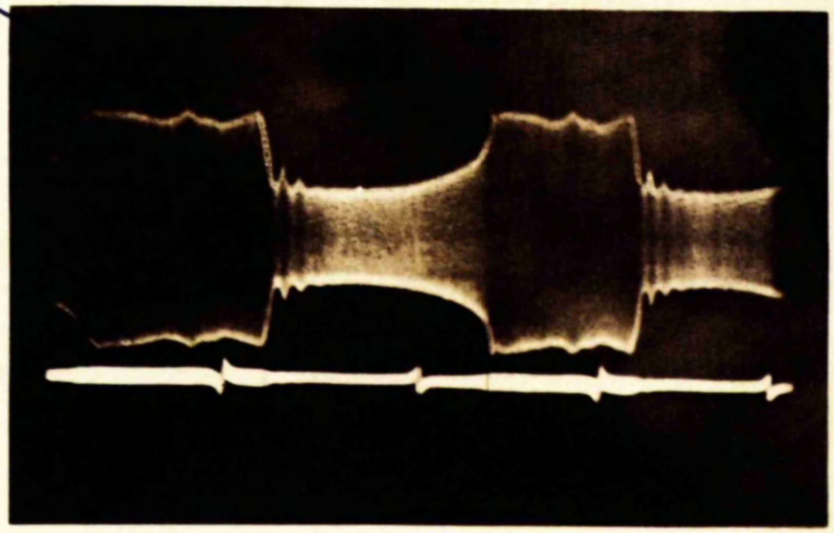
Some measurements from the oscillograms are shown in Fig. 20. These show that (i) the discharge valve was effectively sealing so that the pressure ratio did not materially affect closing after B.D.C., (ii) the delay in suction valve opening (and in closing) was again only slightly affected by compressor speed

750 r.p.m.



D

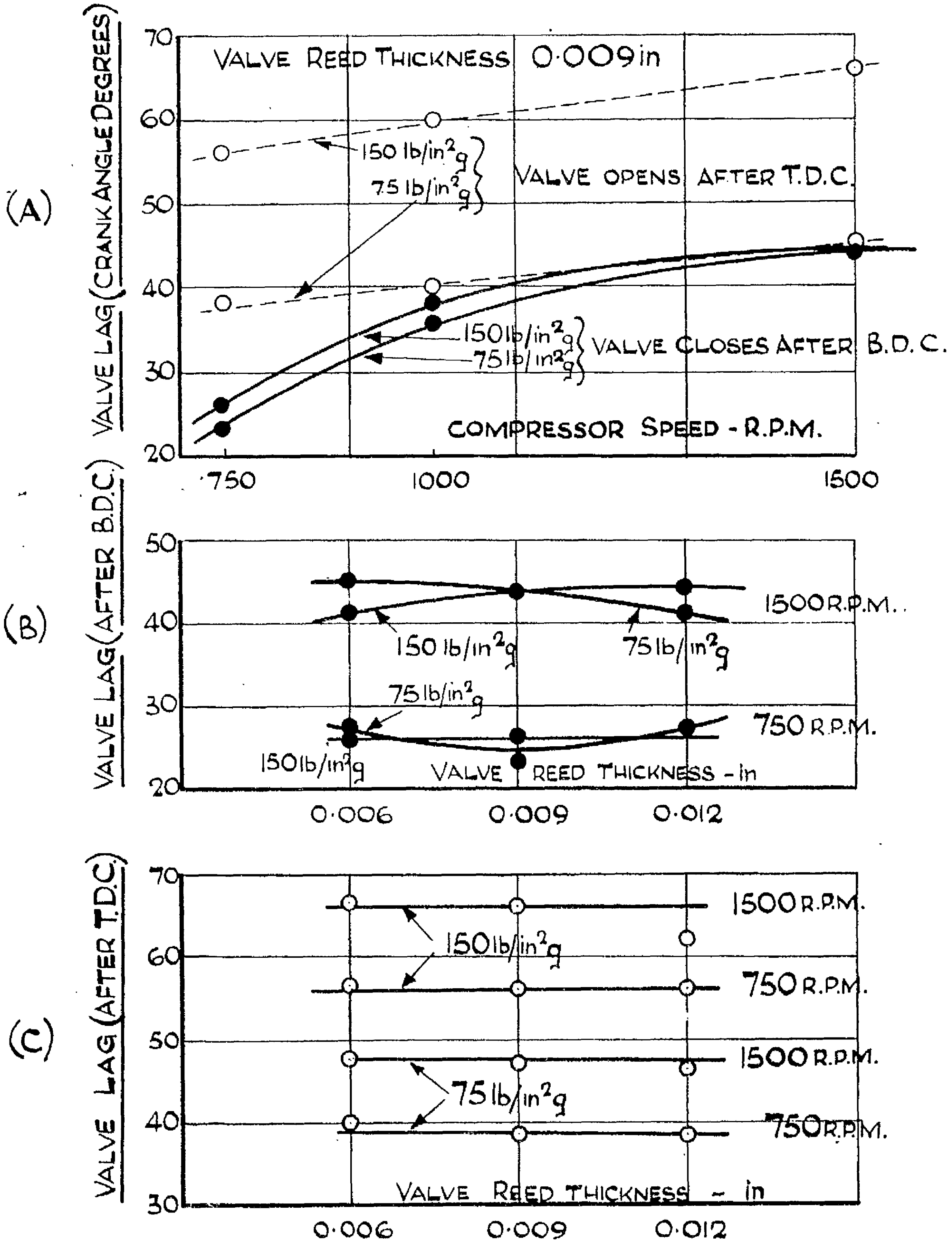
1500 r.p.m.



SPECIMEN VALVE DISPLACEMENT OSCILLOGRAMS

FREON 12. – 0·012 VALVE, h = 0·015 IN, r = 3·43

FIG. No 19.



VALVE LIFT 0·035 in., SUCTION PRESSURE 11·8 lb/in²g 60°F, FREON 12

SOME MEASUREMENTS FROM SUCTION VALVE DISPLACEMENT OSCILLOGRAMS.

FIG. No. 20.

(Fig. A), about 6^{o} change from 1000 to 1500 r.p.m. compared to about 3^{o} with air, (iii) the action was virtually independent of the dynamics on the valve. This last is at variance with the figures using air; perhaps the relatively larger mass of Freon 12 which may be assumed associated with the mass of the valve reed tended to reduce the effect of the wide range of valve weights and spring stiffness used.

4.5 Direct Observation of Valve Action. The valve action was also directly observed by stroboscope. A ported type compressor (see Appendix II) was fitted with a suction valve reed in lieu of a discharge valve and the valve plate clamped in position without a cylinder head. With no discharge pressure the valve behaviour approximated to the suction valve action in a conventional arrangement.

In the first test the compressor was driven up to 3000 r.p.m. without a valve stop. During the opening of the valve there was a small but distinct flutter or lag in the movement at about half the amplitude, from which the valve quickly recovered, opening thereafter to the full amplitude of the valve opening with subsequent flutter.

Compressor Speed - r.p.m.	800	1000	1200	1600	2000	3000
Unrestricted Lift - in.	.06	.08	.1	.14	.15	.16
Free Vibrations (flutter)	4	3	2	1	1	1

Depending on the pressure ratio used there would be a number of compressor speeds at which the flutter synchronised with the valve closing process. At slightly higher compressor speeds the

the closing process was interfered with, probably to the detriment of performance due to increased blow-by. At compressor speeds slightly less than these critical values the closing action could be initiated smartly due to commencement of a flutter.

When the valve reed tip was checked the action was similar but the amplitude of the flutter was reduced: the damping effect due to the oil film between the reed tip and the stop could be clearly observed. For this same reason rebound from the valve seat, which could be observed, was always small.

4. 6 Valve Pressure Loss. The 35 m.m. film records of the rectified oscilloscope trace of valve pressure drop to crankangle (Fig. 21), for 72 cases were enlarged some 15 times on a Hilger Enlarger and then traced. From the calibration curves (Appendix IV) and a stroke to crankangle chart these tracings were converted to the conventional P.V. diagrams (Fig. 22) and the areas measured by planimeter.

The loss of volumetric efficiency due to attenuation may be computed as follows:-

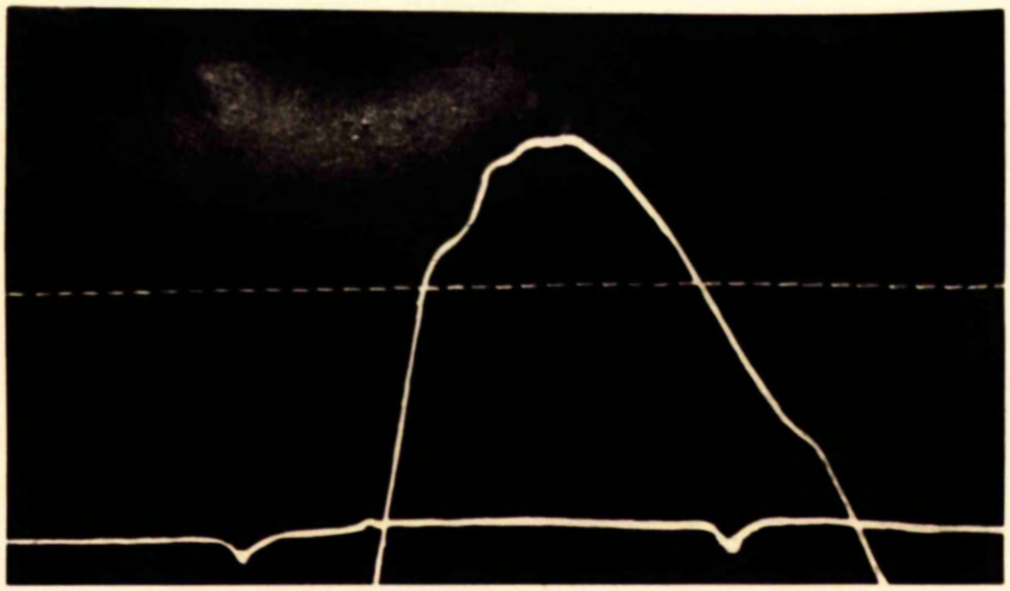
Neglecting heat transfer effects during suction and assuming that the weight induced is discharged to the receiver.

$$dE + \frac{PdV}{J} = 0$$

Referring to Fig. 1.

$$c_v(m_1T_1 - m_4T_4) + \int_{V_4}^{V_1} \frac{PdV}{J} - c_p(m_1 - m_4)T_i = 0 \qquad (35)$$

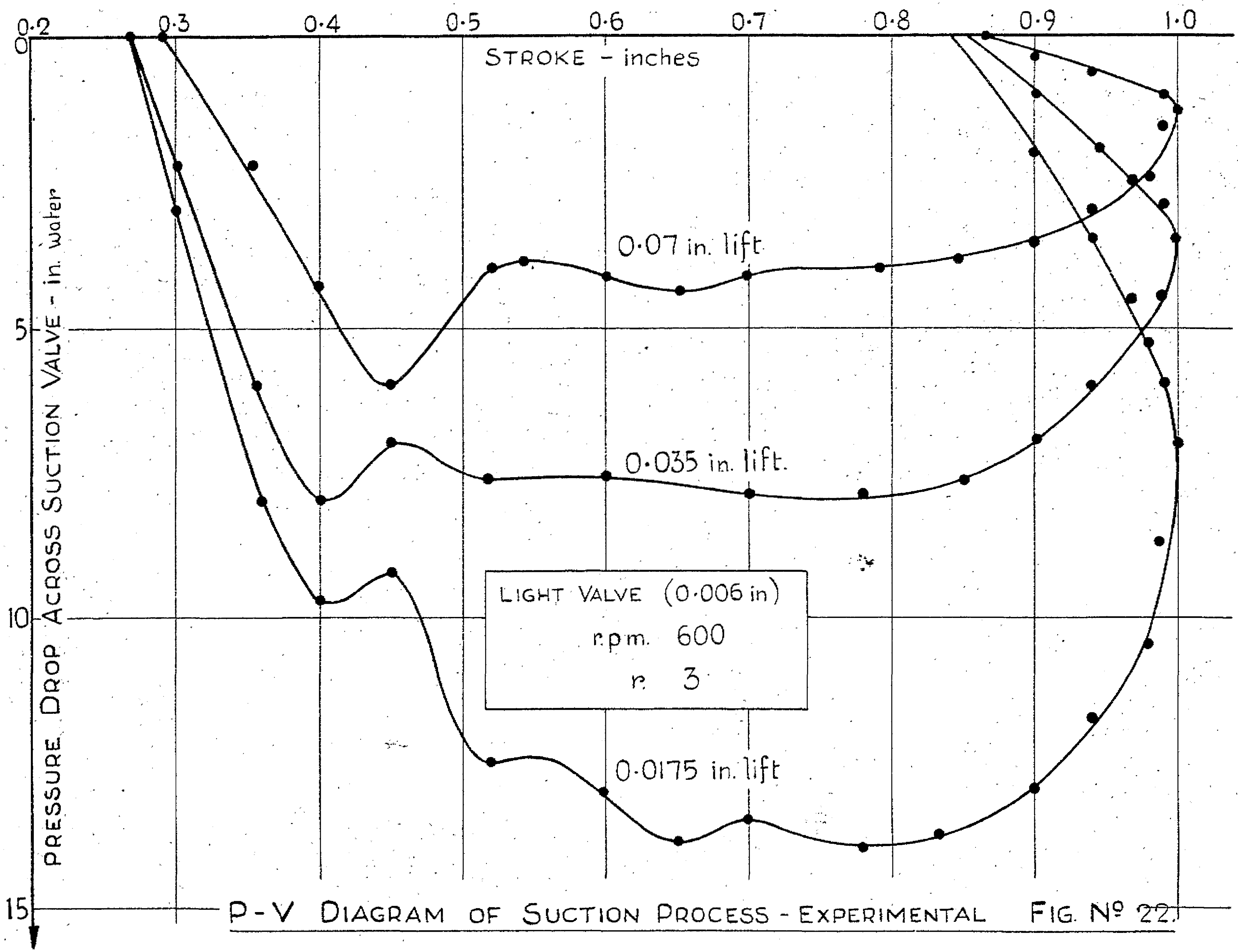
the last term on L.H.S. denoting the enthalpy addition to the closed system due to the flow energy of the induced charge. Depending on the form of the suction line the term $\int_{V_4}^{V_1} \frac{PdV}{J}$ may be part positive and part negative.



PRESSURE DIFFERENCE ACROSS SUCTION VALVE/CRANK ANGLE

LIGHT VALVE (0·006in) h = 0·0175in, 1250 r.p.m., GAIN × 10

FIG. No. 21.



P-V DIAGRAM OF SUCTION PROCESS - EXPERIMENTAL FIG. Nº 22.

Since $c_p = c_v + \frac{R}{J}$, $PV = mRT$ then

$$Jc_v\left(m_1T_1 - m_4T_4 - (m_1-m_4)T_i\right) = (m_1-m_4)P_iV_i - \int_{V_4}^{V_1} P\,dV$$

Since $p_i = p_1 = P_4$, $\frac{Jc_v}{R} = \frac{1}{\gamma-1}$ then

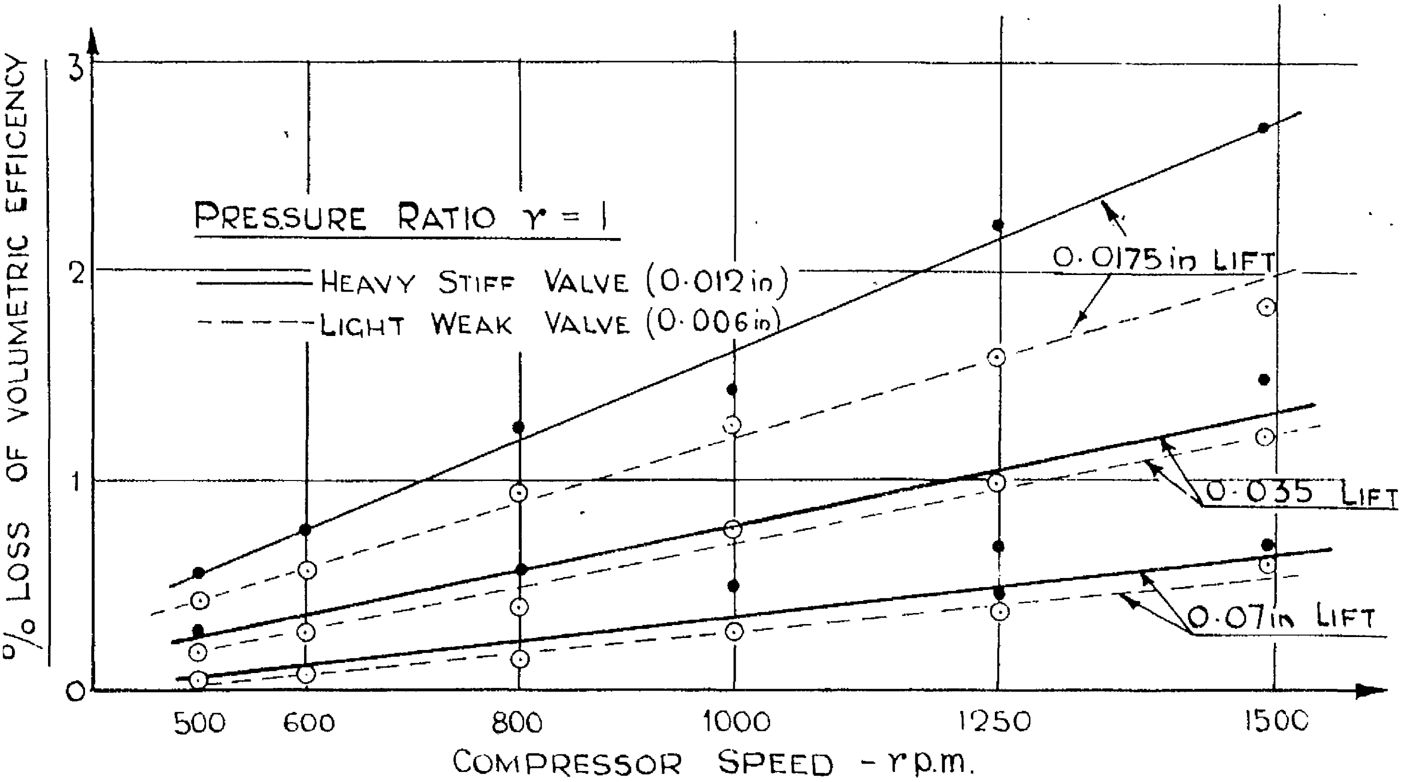
$$\frac{V_1-V_4}{\gamma} - (m_1-m_4)V_1 = \frac{\gamma-1}{\gamma}\int_{V_4}^{V_1}\left(1-\frac{p}{p_i}\right) - \frac{\gamma-1}{\gamma}\left(V_1-V_4\right)$$

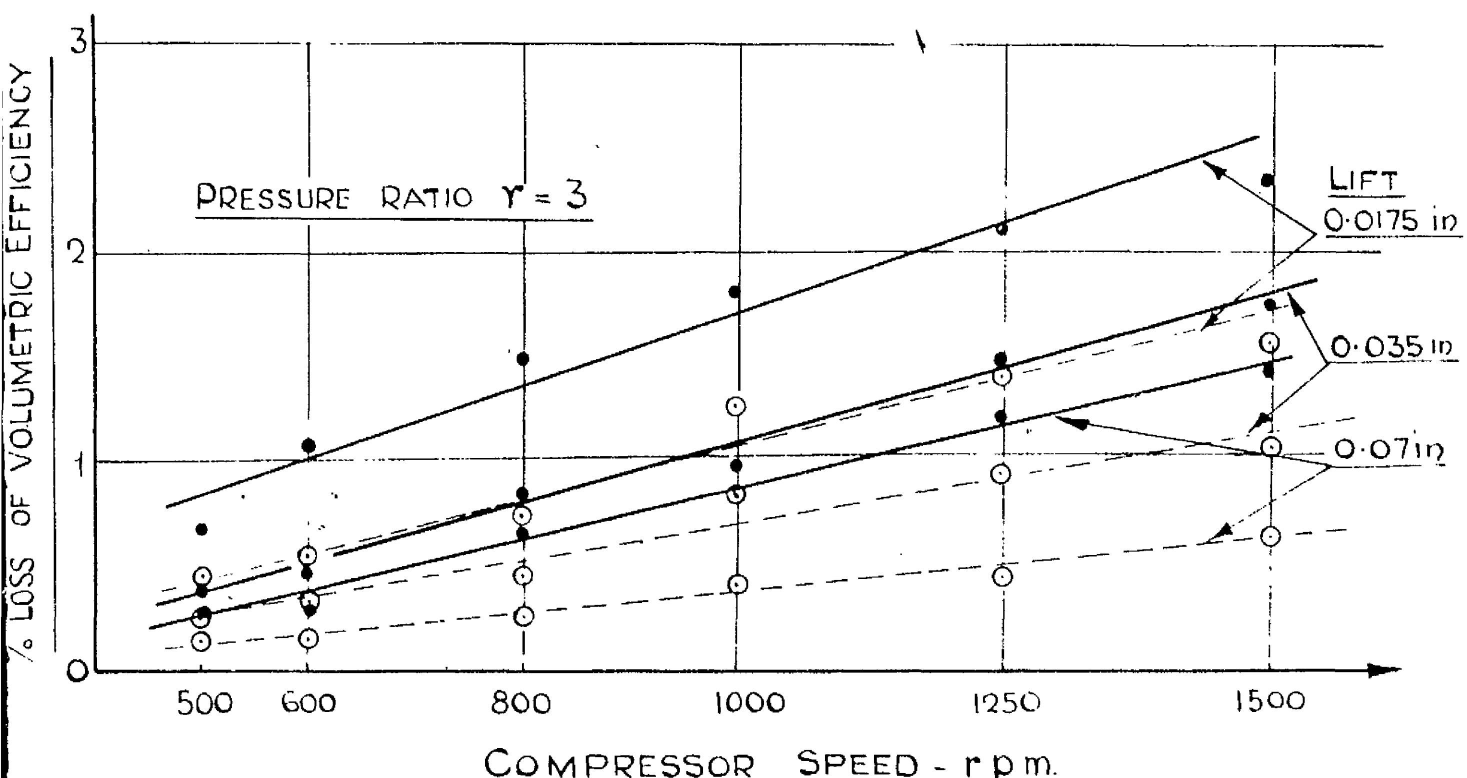
$$\frac{V_1-V_4}{V_s} - \frac{(m_1-m_4)V_i}{V_s} = \frac{\gamma-1}{\gamma V_s p_i}\int_{V_4}^{V_1}\Delta p_v\,dV \qquad (36)$$

i.e. η_v indicated $-$ η_v actual $= \frac{\gamma-1}{\gamma V_s p_i}$ times area of suction loop

By eq. 36 the loss of volumetric efficiency due to suction valve throttling was computed and the results are shown in Fig. 23.

The outstanding feature was the small effect of throttling loss on volumetric efficiency; even when the air velocity through the restricted area of the valve was as high as 500 ft/sec. the loss of volumetric efficiency thereby was only 2½%. At more normal operating conditions the loss did not exceed 1%, so that there was no severe deterioration of performance at high compressor speeds or low valve lifts because of the choking effect of the suction valve. The leakage effects and heat transfer effects, relatively large when the machine is small, probably decrease with increased compressor speed and the optimum compressor speed may, therefore, be surprisingly high since the valve throttling loss is so small. At a pressure ratio of unity there was little difference between the losses due to the light and heavy valves. This may have been due to the low piston velocity at opening, making valve dynamics unimportant, and to the valve being open for a long period of the stroke reducing the effect of differing valve characteristics at opening and closing. The appreciably superior performance of the light valve at very low lift was probably due to its greater ability to flex with the large pressure drop, providing a somewhat greater flow





EFFECT OF COMPRESSOR SPEED, VALVE LIFT AND SPRING STIFFNESS ON VOLUMETRIC EFFICIENCY - FROM EXPERIMENTAL THROTTLE LOSS

FIG. No. 23

area than the nominal.

At the higher pressure ratio the difference between the valves was more marked. The higher piston speed at opening now made apparent the superior dynamical properties of the light valve at this phase, but at any given speed and lift the difference between the loss of volumetric efficiency did not exceed 0.75%.

The leakage loss due to blow-by during closure is not included in equation 36. During closure the stiffer spring of the heavy valve may favour its performance by reducing blow-by, offsetting its initial disadvantage that it reaches the stop later, if at all, and leaves earlier, resulting in a lower "effective" lift. Although it had double the mass it had spring stiffness eight times greater. Hence overall the stiff valve may be equal to or even superior to the light valve, since the throttling effect on volumetric efficiency is so small. This is later shown to be the case.

In the analysis of valve performance (Chapter 3) it was apparent that to calculate the points at which the valve closes and the pressure across the valve is finally zero was very laborious if valve flutter was present. These two points may be obtained from the oscillograms (Fig. 14 and Fig. 21), but the inaccuracies due to the small time base and experimental errors are considerable. However, if the experimental techniques used were further developed with the aim of studying losses due to valve blow-by, the approach would be more fruitful than the laborious theoretical analysis with its inherent accumulated errors due to the step integration process.

Consideration of Fig. 6 and Fig. 22 shows that there is a large difference between the theoretical and actual point at which the valve can begin to open. The loss of volumetric efficiency for this cause is about 20%, mainly due to the action of the discharge valve. A pressure of about 15 lb/in.^{2} was necessary to operate the discharge valve, i.e. the effective pressure ratio was two, when the nominal pressure ratio was one. While the loss due to this decreases with increasing pressure ratio, the effect is still serious in the range of pressure ratios used in refrigerant compressors. All previous investigations have assumed that the discharge valve could have little effect on volumetric efficiency, but it is apparent that the loss of volumetric efficiency due to its stiff springs may greatly exceed the loss due to suction valve throttling.

CHAPTER 5.

EXPERIMENTAL INVESTIGATION OF THE EFFECT OF VALVE VARIABLES ON AIR AND FREON 12 COMPRESSOR PERFORMANCE.

An experimental study was made of the effect of various variables on the performance of the small compressor.

When pumping air, the valve variables were, lift, 0 - 0.085 in., reed thickness, 0.006 in. and 0.010 in. giving spring stiffness 0.237 and 1.1 lb/in. and natural frequency 100 and 144 ~/sec. The compressor variables were pressure ratio, 1 - 9, and speed, 500 - 1500 r.p.m.

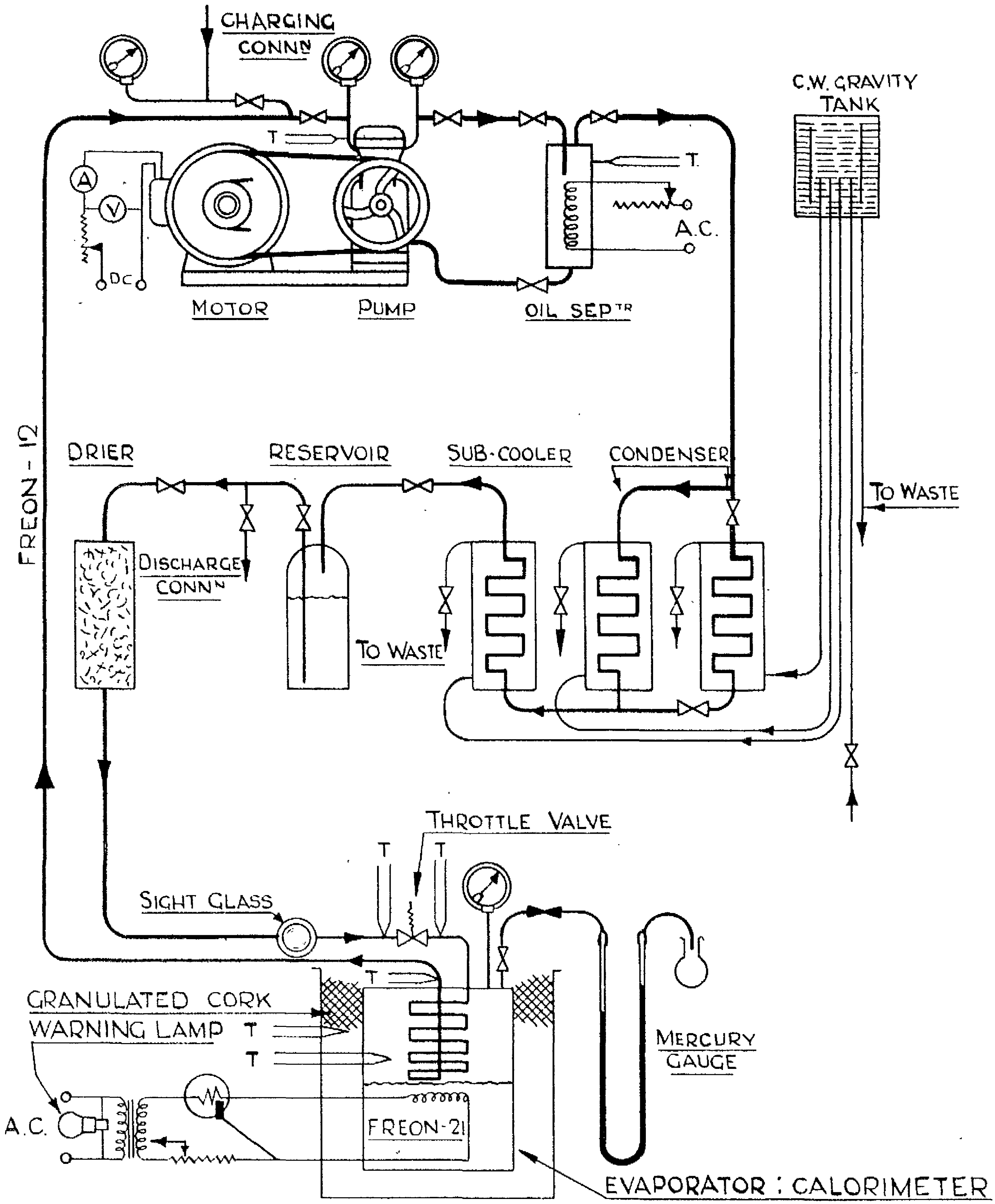
Since a slight improvement in performance with the heavier, 0.010 in., valve was observed in the air compressor tests a still stiffer valve was used, 0.012 in. thick, spring stiffness 1.9 lb/in., natural frequency 166 ~/sec., and comparison of performance between this and the light 0.006 in. valve made, when pumping a dense refrigerant, Freon 12. The speed range and valve lift range were as in the air tests, but the range of pressure ratios available was, of course, now limited.

5. 1 Apparatus. (a) Circuit for compressor tests with Air. The compressor was directly coupled to a torque-mounted electric motor. The air, at atmospheric condition at suction throughout the series was compressed and delivered to an oil separator and receiver. The air was then throttled to the approach pipe of the metering orifice through which it discharged to atmosphere. A vertical water manometer measured the pressure drop across the orifice; a thermocouple measured the air temperature at the compressor inlet flange; pressures were observed on calibrated Bourdon type gauges.

The test procedure finally adopted was to allow the compressor to settle thoroughly at a selected speed and discharge pressure and then measure the air delivered at differing valve lifts, these being controlled by the lift measurement device. The relatively small effect of valve lift on performance did not materially disturb the settled temperatures and pressures. At each discharge pressure the mass flow of air was measured with the valve lift altered by steps, at some minutes interval, from the maximum permitted lift to the position with the valve tip closed and back again to the maximum permitted lift. The lubricating oil was kept at a constant level in the sump, but its presence in varying quantities in various places during operation presented a difficulty in the reduction of the experimental scatter to within the close limits required. The results are presented in graphical form, each curve being the mean of the two obtained as the valve lift was first reduced and then increased again.

(b) Circuit for compressor tests with Freon 12 (Fig. 24).

The Compressor delivered to an oil separator and the separated Freon 12 vapour passed to two condensers in parallel. The cooling water was supplied from a constant head tank and the control of this flow by needle valves regulated the refrigerant high-side pressure. The sub-cooled liquid passed to an accumulator which considerably improved the stability of the circuit. The liquid passed through a silica-gel drier, through a sight glass to a pressure controlled expansion valve and hence to a calorimeter to measure the refrigerant mass flow. This calorimeter also functioned as the evaporator. The suction vapour, superheated to the ambient temperature throughout the present tests returned to the compressor from the evaporator. The required temperatures were measured by calibrated copper constantin thermocouples at



DIAGRAMATIC ARRANGEMENT OF PLANT.

FIG. No 24.

the required points. The cold junction was taped to a suitable thermometer, immersed in a test tube of oil and the whole immersed in a vacuum flask of shaved ice.

The secondary calorimeter was selected as the most suitable for the accurate measurement of relatively small changes of flow. The input to the electrical heater element immersed in the secondary fluid was controlled by a Variac transformer, and slide wire resistor for fine control. The consumption was measured by a substandard wattmeter. Although the calorimeter was sufficiently robust to operate with Freon 12 (72.41 lb/in.^{2} abs. at 60°F.) as the secondary fluid, Freon 21 was used (18.9 lb/in.^{2} abs. at 60°F.) This permitted a mercury manometer to be used as the indicator of temperature of the secondary fluid. No location for either a thermometer or a thermocouple was found which gave as sensitive a register of secondary fluid temperature as did this manometer, which incidentally served as a simple safety device so that no automatic cut-out was necessary in the heater circuit. The calorimeter, insulated with granulated cork, had a low heat leakage and this could be neglected because (a) the expansion valve was kept close to the calorimeter and (b) the suction line vapour temperature and the secondary fluid temperature were kept at ambient temperature throughout the series.

The function of the oil separator was to reduce to a very small fixed amount the oil flowing with the refrigerant through the calorimeter and compressor suction valve. The Freon 12/oil mixture delivered from the compressor entered a cage separating two copper gauze elements. A 125 watt heater evaporated the Freon 12 and heat exchange was effected between the rising vapour and incoming mixture; the whole operated as a simple refractionating column. The upper gauze tended further to separate oil particles remaining in the vapour, while the lower aided the distribution of heat. The denser oil accumulated and could be bled back to the compressor sump, via the drain, at the conclusion of, or if necessary during a test. Input to the heater was controlled by slide wire resistor so that the separated Freon 12 vapour left at 200°F. at any compressor speed, ensuring a similar oil concentration, a small fraction of 1%, in the flow through the calorimeter. The oil quantity passed over by the compressor was negligible since the suction vapour was superheated and the concentration of Freon 12 in it small; less than 8% by weight, at the highest condenser pressure used, at 200°F. Hence the mass flow of Freon 12 through the compressor was virtually that through the calorimeter, and, further, true values of liquid refrigerant enthalpy could be determined by temperature measurement.

The test series using Freon 12 was carried out and, in the light of the results and operating experience, the plant was rebuilt and modified for the present series to improve the control of the many factors influencing performance and so reduce the scatter of results. It was found necessary to operate the plant for about 5 hours before steady conditions were obtained. During this period the various controls had to be adjusted. After commencement of a test, thirty minutes was allowed between each alteration of valve lift to ensure steady conditions again. The duration of each of 30 trials was, therefore, about 10 hours subsequent to the "running-in" period and consistent results could not be obtained in less time.

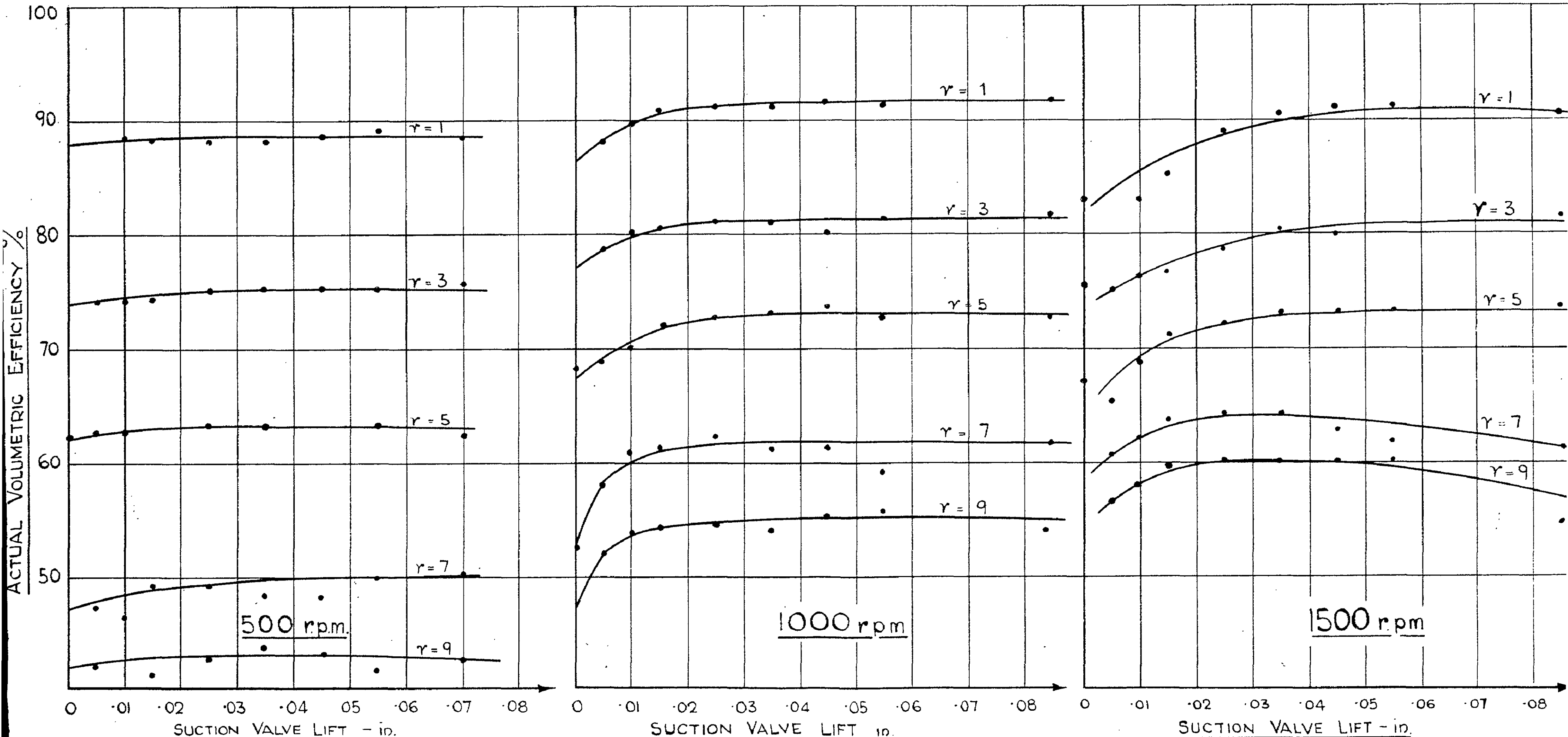
When all modifications for improvement of control had been made, three appreciable causes of error still remained. Firstly, the variations in mains voltage resulted in variation of compressor speed, which was easily adjusted. The effect of the calorimeter was more complex. Secondly, changes in ambient temperature, which not only fluctuated, but varied at different points of the basement laboratory, were sufficient to affect the accuracy required. This was, of course, partly countered by careful lagging. Thirdly, slight variations in water temperature affected the high side pressure. These difficulties were largely met by "running-in" the plant during the evening and carrying out the test between 10 p.m. and 8 a.m. when there was appreciably less disturbance of electrical and water pressures and air and water temperatures.

Throughout the series the evaporating temperature was held at the standard value, 5°F. The suction vapour was superheated to 60°F. Superheating of suction vapour is accepted practice with this vapour. The decrease in theoretical performance is small, due to the low value of the isentropic index, and this is more than offset by the improvement in actual performance due to reduction of heat transfer effects, which are a minimum and constant for this machine at this vapour condition. The condensing temperatures used were 72°F, 84°F, and 120°F. giving pressure ratios 3.43, 4.1 and 6.4, obtained by variation of the cooling water mass flow.

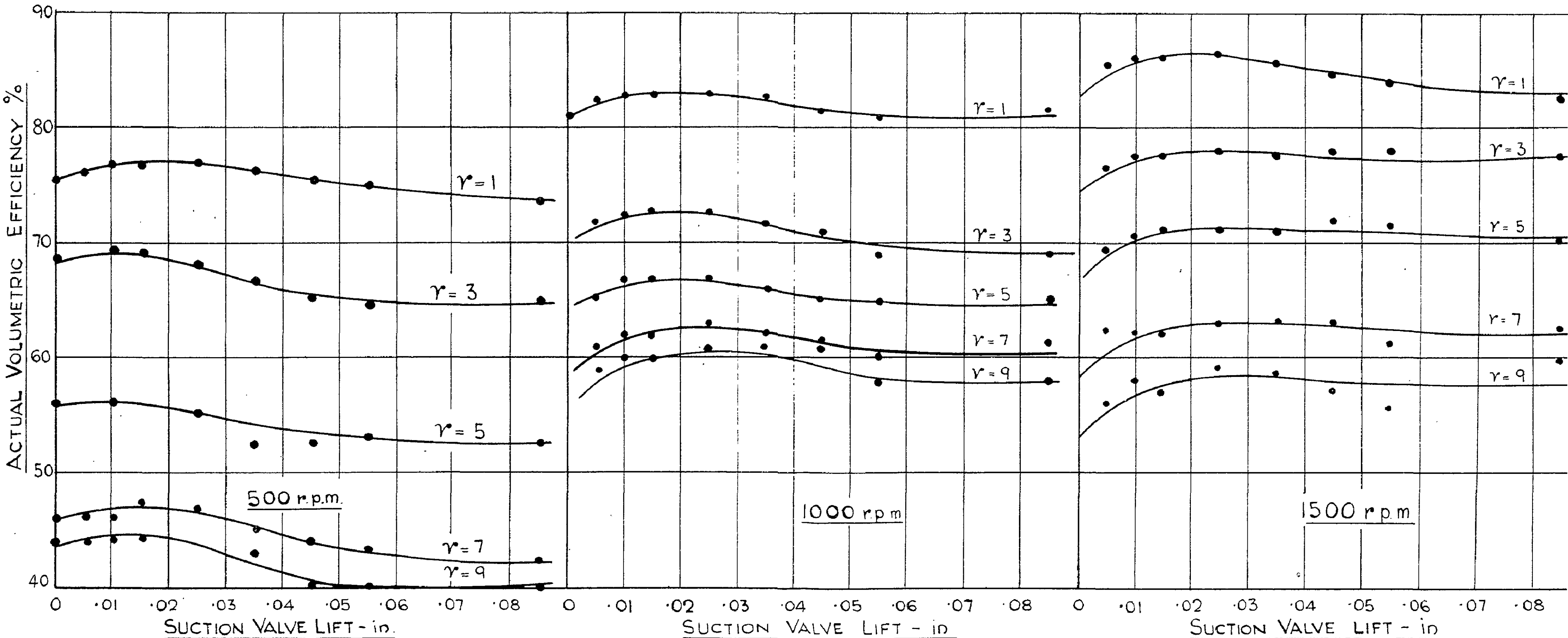
Change of refrigerant flow was reflected by a change in temperature and therefore of pressure, as recorded by the mercury manometer, of the secondary fluid Freon 21. This manometer was open to atmosphere and subject to slight error due to variation of barometric pressure. A chart was constructed from the properties of Freon 21 so that the correct manometer head could be computed for any barometric pressure and ambient temperature. The setting of the manometer head was not varied during any one test and the controlled heat input to the calorimeter maintained the mercury head. It was possible for the small errors accrued by variation of ambient temperature and barometric pressure during a test to be additive, or to cancel each other. To adjust the heat input and hence mercury level, a system of bracketing was used, and, with operating experience, the fluctuation was reduced to a short period. It was essential that the level was exactly obtained and maintained for at least half an hour.

With each weight of valve reed, two test series were conducted, at fixed suction conditions, and at compressor speeds of 500, 1000 and 1500 r.p.m. In the first, the valve lift was varied from 0-0.08 in. at three discharge pressures; in the second, at particular valve lifts, the discharge pressure was varied from 75 - 177 $lb/in.^2$ gauge. Large scale charts were constructed for the various properties of Freon 12 required for accurate calculation of the results.

5. 2 Discussion of Results. (a) Effect of Valve Lift on Actual Volumetric Efficiency - Air. There was no reduction in volumetric efficiency at 500 r.p.m. with reduction in valve lift (Fig. 25). At speeds of 1000 r.p.m. and 1500 r.p.m. a slow reduction occurred if lift was reduced below about 0.025 in. and 0.40 in. respectively; that is, at mean gas speeds through the suction



AIR - COMPRESSOR TEST RESULTS - 0·010 in VALVE REED - FIG. No. 25



Air - Compressor Test Results - 0·006 in Valve Reed - Fig. No. 26

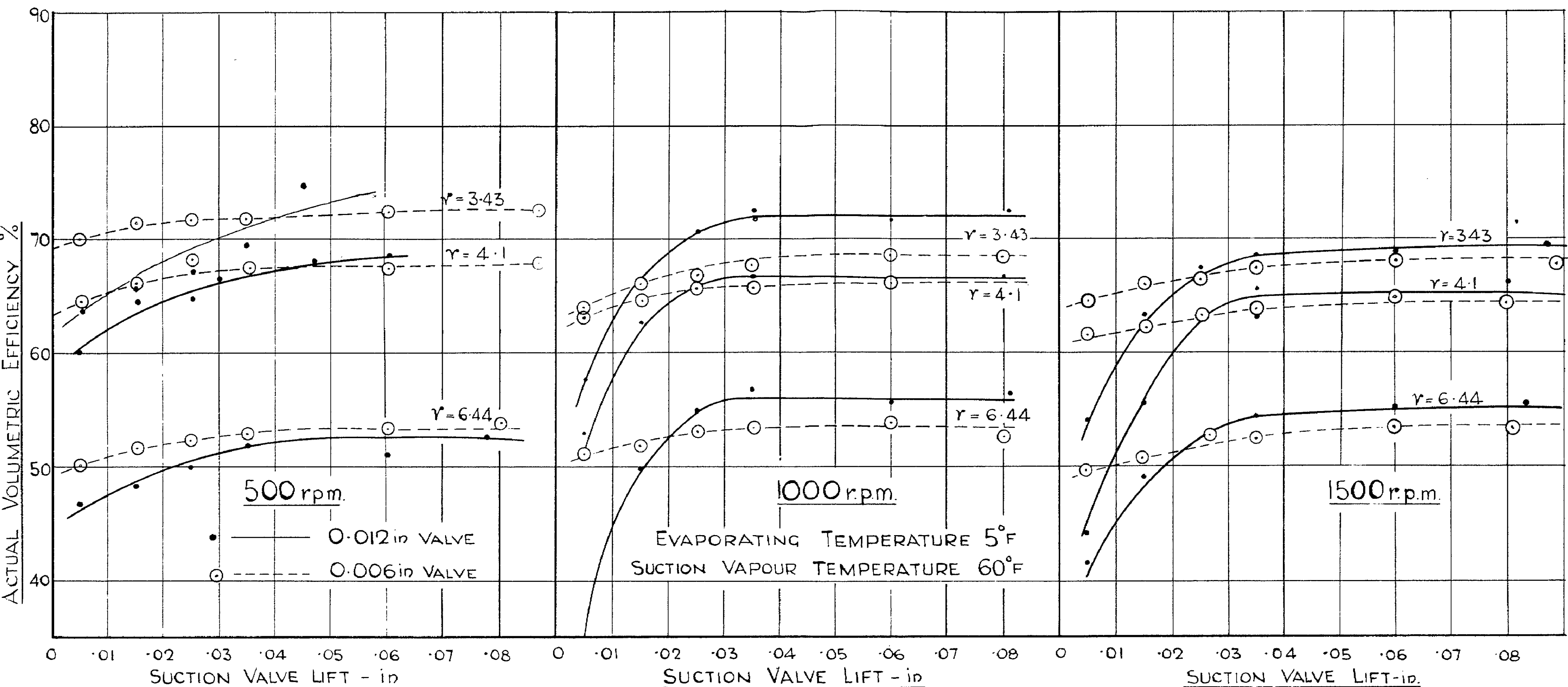
valve in excess of about 230 ft/sec. In the case of the light valve, Fig. 26, there was actually an increase in volumetric efficiency as the valve lift was reduced, until gas velocities of the same order were reached, after which the expected decrease occurred. That increase of about 2% was definite and readily reproduced, with all other conditions steady, being represented by a change of orifice manometer head of about 0.8 in. water. It was concluded that reduction in throttling loss by increasing valve lift was fully offset by the blow-by during closure and more than offset in the case of the light valve with its weaker spring.

(b) <u>Effect of Valve Lift on Actual Volumetric Efficiency - Freon 12</u>.

With Freon 12 as with air the volumetric efficiency was practically constant over a wide range of valve lift. (Fig. 27). The gas velocity through the restricted valve area exceeded 130 ft/sec. before the throttling loss more than offset the advantages of low lift. Up to this gas speed there was negligible flexing of the stiff valve reed so the figure was reasonably clearly defined.

That the volumetric efficiency would remain constant at gas speeds less than 130 ft./sec. is at variance with common conjecture and required this experimental substantiation. The improved valve performance at lower lifts offset the throttling loss which, as has been seen (Chapter 4), had only a small effect on volumetric efficiency. If the valve lift was unduly reduced until a mean gas speed of 180 ft./sec. was reached only a further 1% loss in volumetric efficiency occurred.

The results at 500 r.p.m. did not conform to these figures.

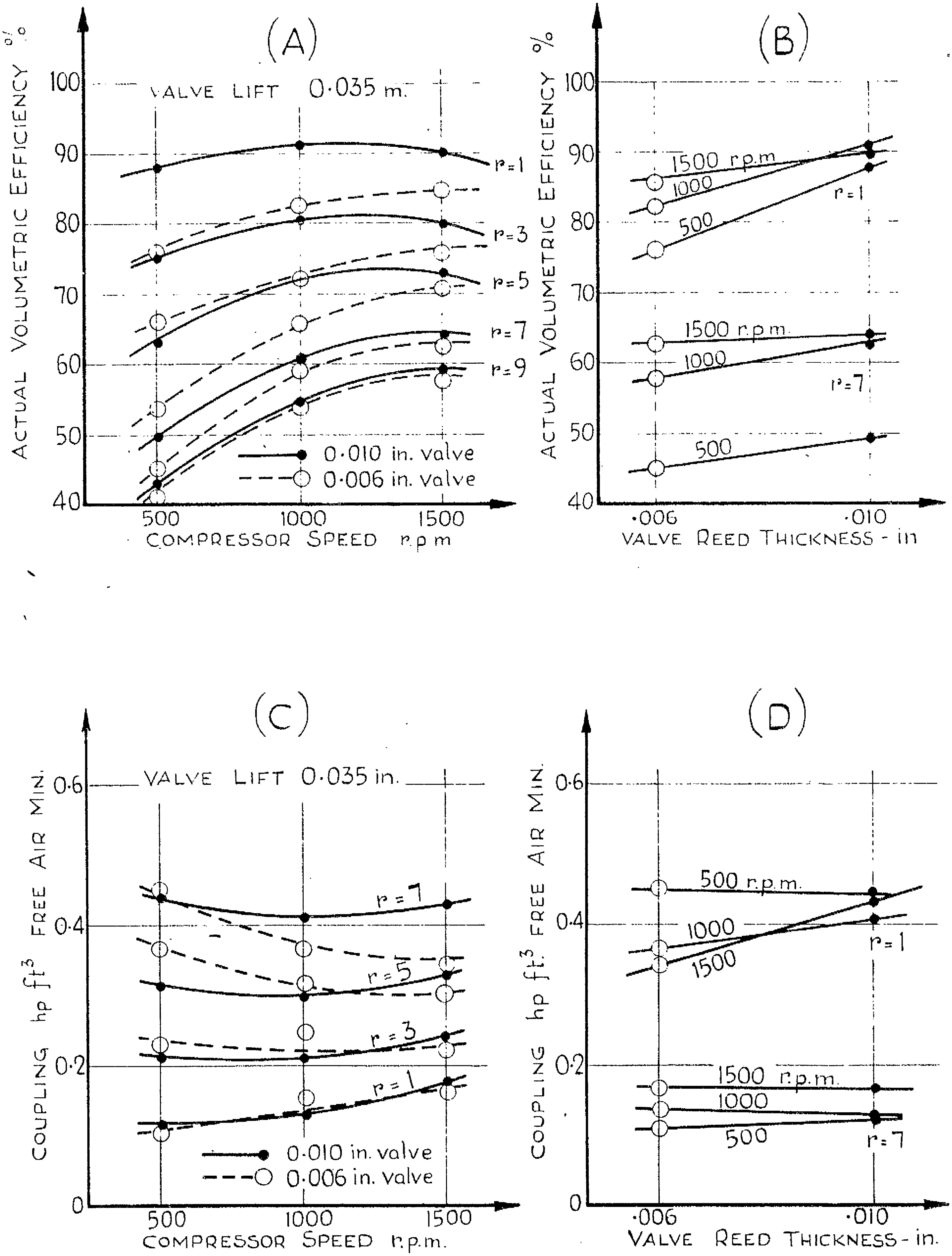


FREON 12 - COMPRESSOR TEST RESULTS FIG. No. 27.

At r = 3.45 the heavy valve did not open fully at high permitted lifts and, therefore, with this valve at low speed the mean flow area was more a function of the compressor speed than of permitted lift.

(c) Effect of Compressor Speed on Actual Volumetric Efficiency -Air.

(Fig. 28 A) This compressor, intended to operate at 600 r.p.m. when the mean piston speed is 100 ft/min. and the mean gas velocity through the suction valve is about 90 ft/sec. showed a marked improvement at all pressure ratios, as speed increased from 500 to 1000 r.p.m. The optimum occurred between 1000 and 1500 r.p.m. At 1500 r.p.m. there was no marked deterioration except at low pressure ratios. Thus the increase of valve throttling loss was more than offset by the reduction of valve blow-by and other losses. At low speeds, particularly at low pressure ratios, the light valve compared least favourably with the heavier valve with stiffer spring. At low compressor speeds the volumetric efficiency was very low. This was due to the manner of clamping the suction valve reed; a method which is in general use. It was not rigidly held on its locating pins, otherwise further even tightening of the cylinder head and valve plate to the cylinder block would not be possible. Hence the compressed valve plate joint had a greater thickness than the reed which was thus free vertically for a few thousands of an inch. This clearance was magnified over the valve port and at low speeds it was possible to blow back the whole induced charge. It is often claimed that this slightly open suction valve assists the breaking of the oil seal at valve opening. While, as has been shown, the oil does affect the valve operation, variation of valve



AIR COMPRESSOR TESTS - FIG. Nº 28.

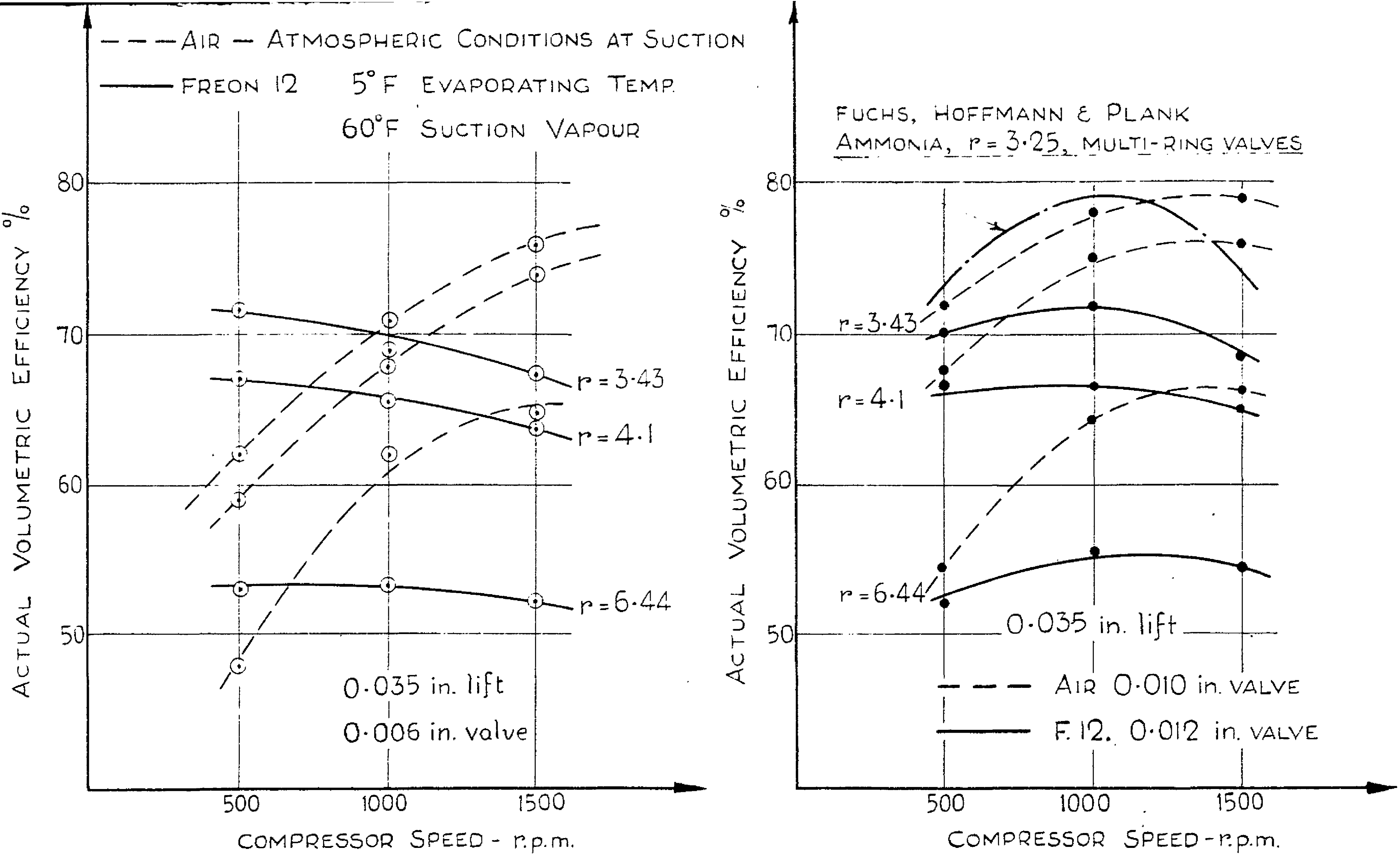
opening characteristics have also been shown to have negligible effect on compressor performance. Hence this appreciable blow-by at low speeds should be reduced by firmly setting the valve to rest over the port, with no spring load.

(d) Effect of Compressor Speed on Volumetric Efficiency - Freon 12.

It was remakrable that this vapour, about 8 times more dense than air could be pumped with a velocity of 130 ft/sec. through the valve before deterioration in performance resulted. At 2½ times the design speed of 600 r.p.m. the volumetric efficiency had only deteriorated by about 2½%.

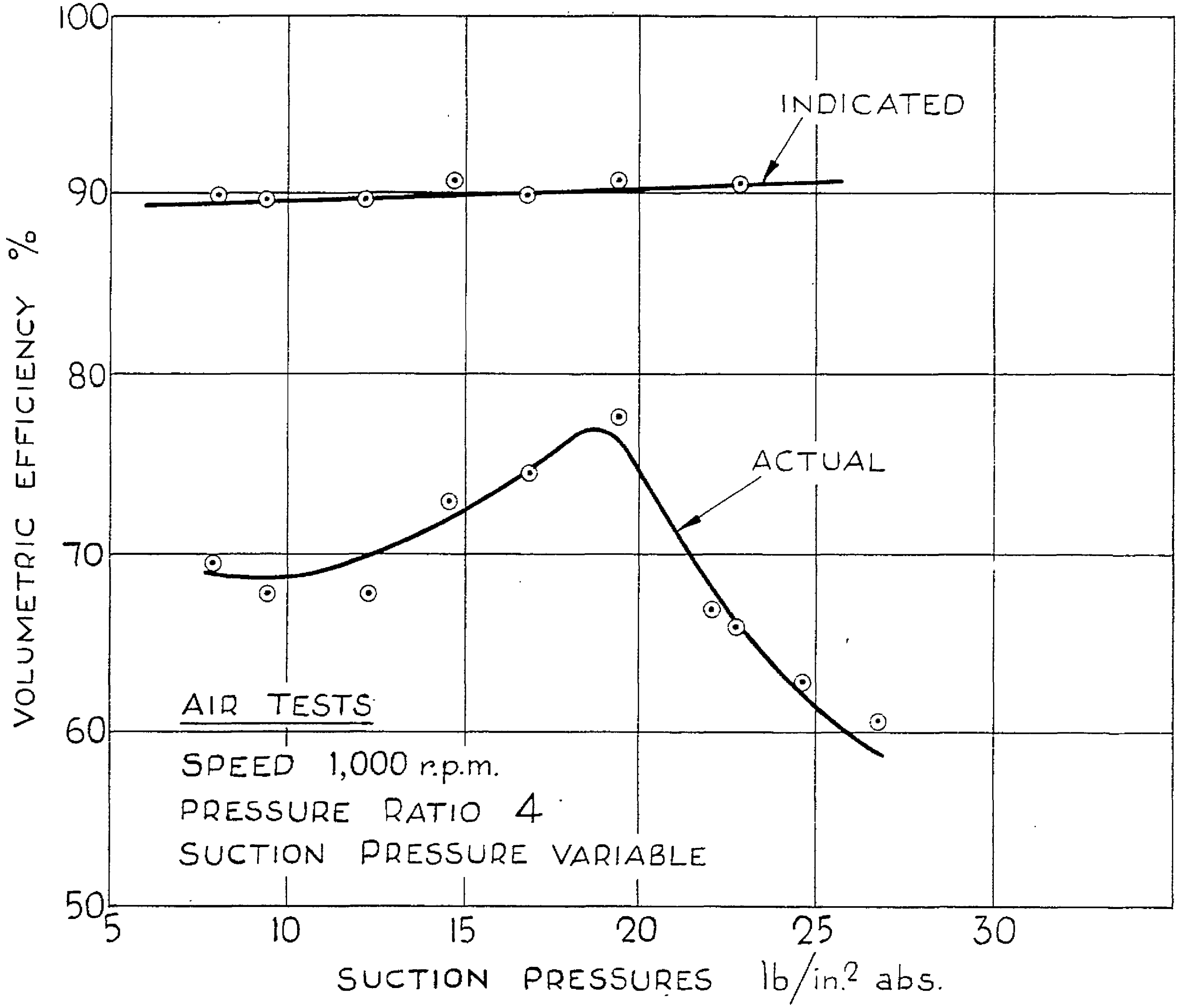
The light valve, the performance of which increased so remarkably with speed when pumping air, did not show any improvement as compressor speed increased; the heavy valve showed a little improvement up to about 1000 r.p.m. perhaps due to its better closing characteristics.

It was surprising that, with the light valve at 500 r.p.m. (Fig. 29 A) the compressor performance with Freon 12 was superior to that with air. Actually, a comparison of the performance of the machine with the two fluids cannot confidently be made, merely on a basis of speed and pressure ratio. There is a suction pressure at which the volumetric efficiency is a maximum for a fixed pressure ratio, (Fig. 30) and a comparison between the two fluids should be made at either (a) the same suction pressure or (b) the optimum suction pressure for each fluid. However, the atmospheric suction pressure of the air compressor and the evaporator pressure of the refrigerant compressor is not likely to fulfil either condition so that the prediction of performance with a vapour from results obtained with air cannot be made.



EFFECT OF COMPRESSOR SPEED & VALVE CHARACTERISTICS ON VOLUMETRIC EFFICIENCY - AIR AND FREON 12 TEST RESULTS

FIG. Nº 29.



VOLUMETRIC EFFICIENCY CURVES by BROWN

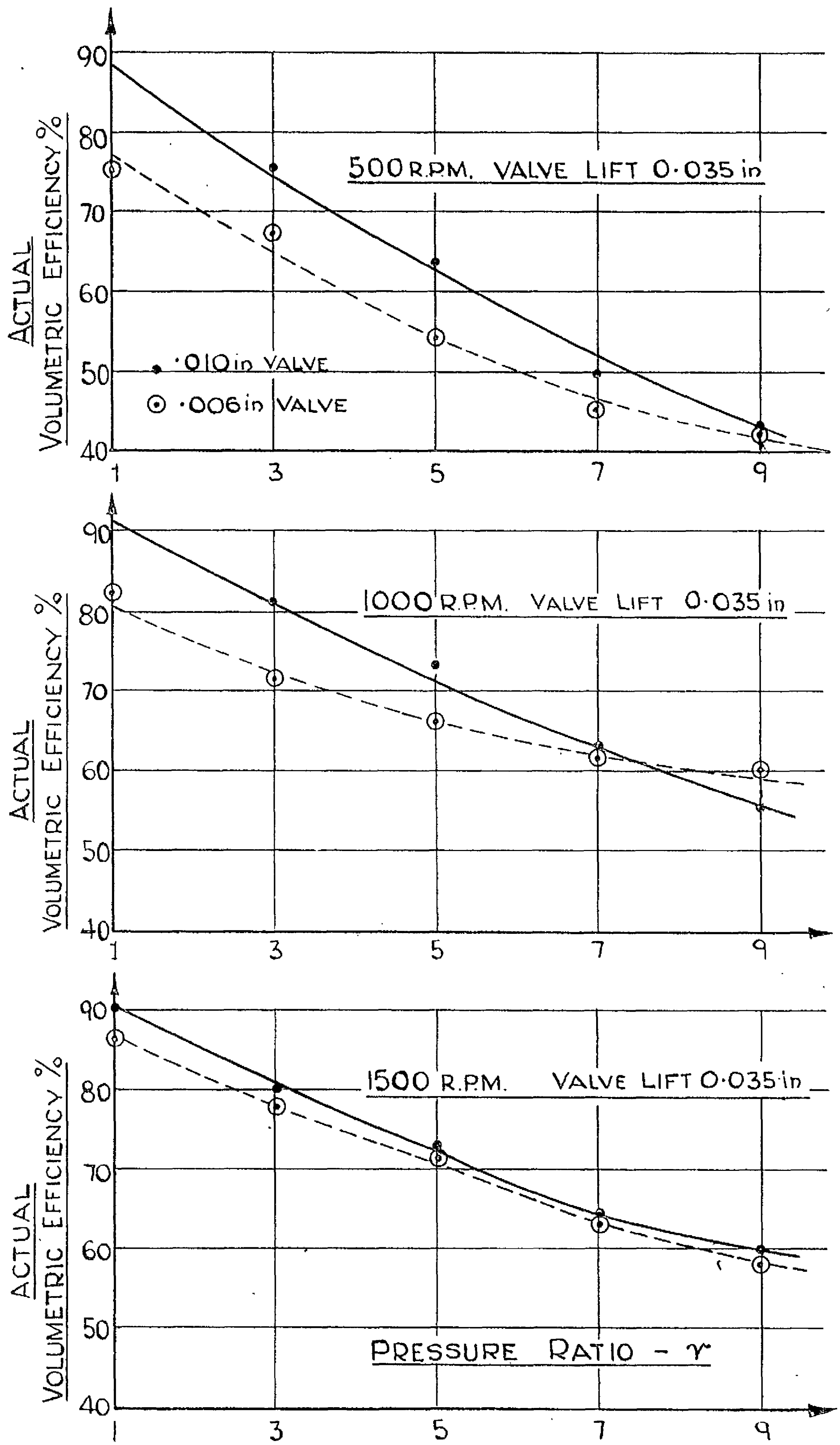
FIG. Nº 30.

(e) Comparison of Valve Weights and Spring Stiffness - Air & Freon 12.

Fig No. 31 shows the comparison of the two valves used in the air tests at the valve lift 0.035 in. The light valve was less efficient throughout, except at high compression ratios when the two were about equal probably due to the blow-by losses being then a small part of the whole loss.

When pumping Freon 12 the faster closing of the stiffer valve again appeared to offset its greater loss at opening and its lower effective flow area, since it reaches the stop later and leaves it earlier. The difference between the two is of the order of only 1%, about the same order as the accuracy of the volumetric efficiency results. Due to the larger volume of the thicker valve, the decrease in clearance volume would result in an increase of theoretical indicated volumetric efficiency of 1.25% at $r = 6.44$. A further test series using a valve reed 0.009 in. thick gave results within the two and so no optimum spring stiffness was determined.

When the difference between the pump performance with each valve is so small, other minor factors assume importance. For example, the leakage past the light valve, due to its greater distortion could account for its apparent slightly inferior performance, rather than its poorer dynamic properties on closing. Further it is not possible to stamp reeds absolutely flat; the greatest distortion was found around the locating holes in the legs. Also, by the Talysurf Recorder, it was possible to show considerable imperfections on valve seats and their edges, although the valve plates were carefully ground. Therefore, the performance of apparently identical parts could vary and the improvement in one set of parts during the running in period could be greater than



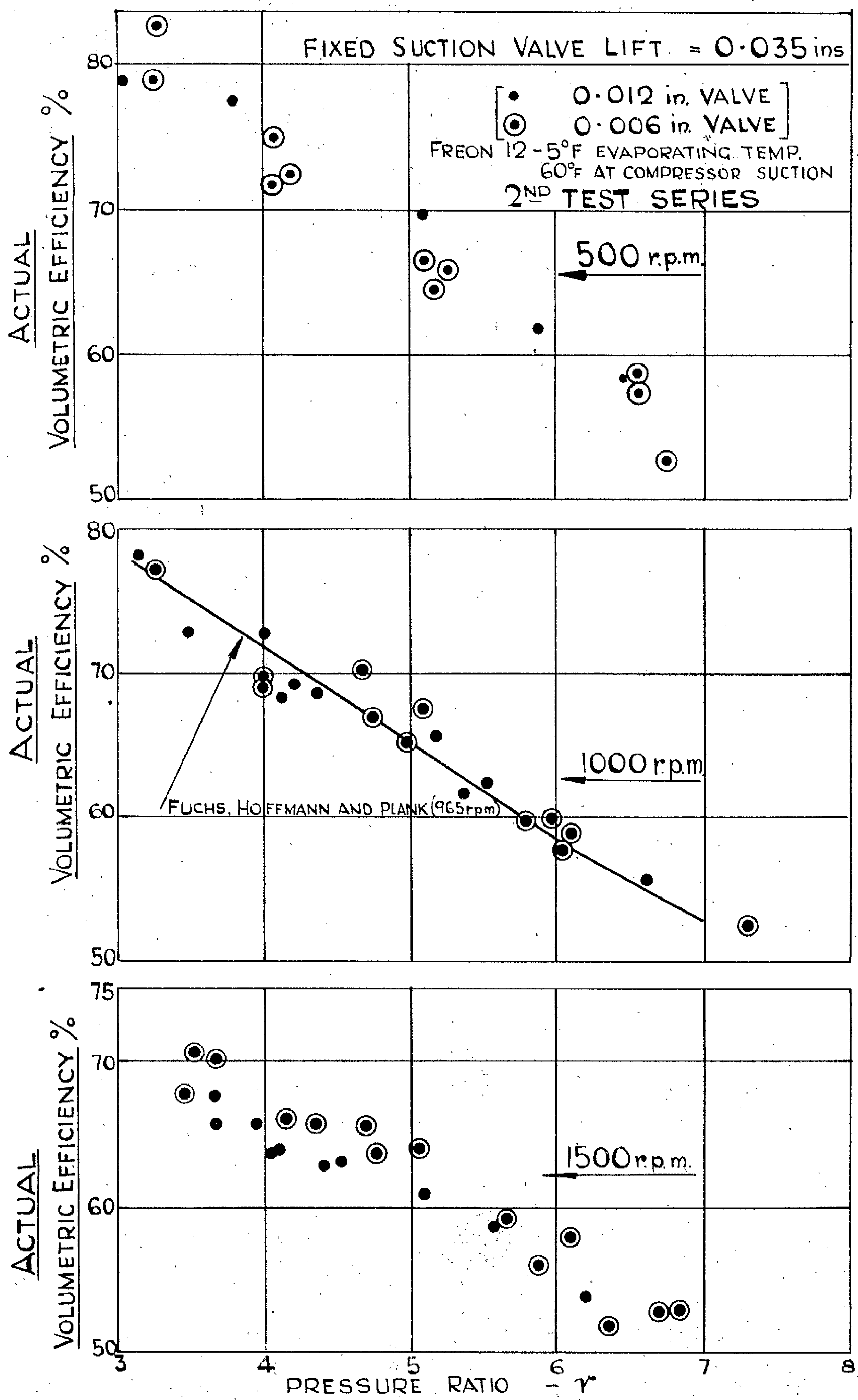
AIR COMPRESSOR TESTS FIG. No. 31.

the difference sought. The valves used were selected for flatness and to give the same leakage under the standard vacuum tests.

In a further attempt to reduce experimental scatter, ancillary equipment not required to measure volumetric efficiency was removed from the Freon 12 plant and a new valve plate and selected valves, at fixed lift, 0.035 in., used in a series in which the evaporator conditions and compressor speed were held constant and the high side pressure allowed to settle to any value. The results (Fig. 32) still showed a scatter in excess of that permissible if definite conclusions were to be made regarding optimum dynamical properties of the valve. Operating experience indicated that close control of ambient temperature by air conditioning would be necessary to improve on the results obtained.

It is interesting to compare these results with those of Fuchs, Hoffmann and Plank (5) obtained from a relatively large six cylinder ammonia compressor (100 m.m. bore x 90 m.m. stroke) (Fig. 32). The two compressors were widely different in every way, except in speed and pressure ratio yet the corresponding volumetric efficiency curves are identical. This emphasises the complexity and interrelation of the partial losses and shows that little information regarding losses can be conveyed by the usual criteria of performance. It also tends to discount a common surmise that a compressor of small size pumping a dense vapour has ipso facto a poor performance.

(f) Isothermal efficiency. This efficiency is sometimes expressed in an inverse form as the coupling horse power per cubic foot of free air delivered per minute, at a given pressure ratio.



FREON 12 COMPRESSOR TEST RESULTS FIG. Nº 32.

The ratio was graphed for all the tests to a base of valve lift, but since the horse power and the volume flow usually vary in the same direction trends were not easily detected nor useful conclusions drawn. Similarly, the coupling horse power per ton of refrigeration does not change significantly with change in valve performance or, for the same reason, with changes in fractional clearance. The power consumption was constant except for an increase at very low lifts. The optimum efficiency therefore occurred about 1000 r.p.m. (Fig. 28 C) the increased friction work at the higher speed being more than offset by the improvement in volumetric efficiency. In such a small compressor, the friction h.p. is high. When pumping air, at 1000 r.p.m., pressure ratio 4 , suction valve lift 0.035 in., the mechanical efficiency was only 57%. Hence small changes in indicated h.p. become very small changes in coupling h.p.

CHAPTER 6.

AUTOMATIC VALVES FOR WATER PUMPS.

INTRODUCTION AND PURPOSE OF PRESENT WORK.

The automatic valve for a gas pump, in the form of a leather clack valve on bellows, has been in use since prehistoric times. As has been shown, it has not been studied to any great extent. The water pump, invented by Hero about 120 B.C. was not understood, in principle, till the 17th century, but since then it has been studied in great detail. There is a considerable quantity of technical literature to be perused even if attention is focused only on water pump valve behaviour. Research on water pump valves has continued almost uninterrupted since 1842. Most of the work was carried out in Germany up to the time of Professor Shrenk's work in 1925. Due to the need of industry during and after the late war for better and higher speed pumping equipment, research into the behaviour of self-acting valves for water pumps was revived and undertaken by Committee 6 of the British Hydromechanics Research Association.

The first object of the present work was to develop a valve displacement indicator which had no attachments to the valve and was unaffected by pump speed. A small water pump was so indicated and a study made of the valve movement under various valve and pump operating conditions.

Since previous investigations had been made on large relatively slow pumps it is intended to examine the performance of a small pump and the effect of valve and pump variables on pump performance.

CHAPTER 7.

APPARATUS.

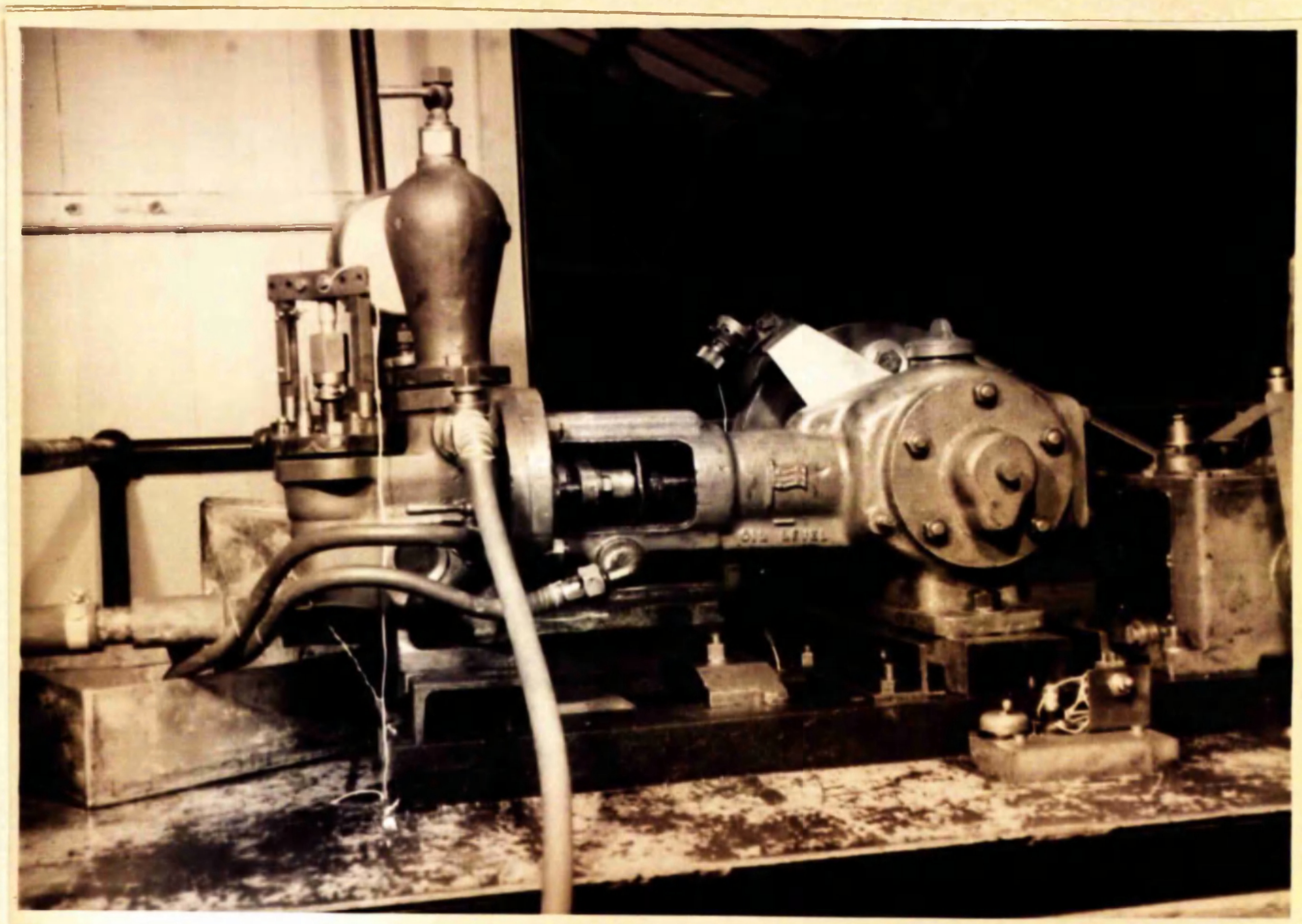
(a) Water Pump. The water pump used was a single acting ram pump, ram diameter 1 in. stroke 2 in. connecting rod length to crank length 5. (Fig. 33). The specifications were: crankshaft speed 360 r.p.m., maximum delivery pressure 150 lb/in.^{2}., maximum suction lift 6 ft., delivery 122 gallons per hour. The pump is normally fitted with similar single disc type suction and discharge valves.

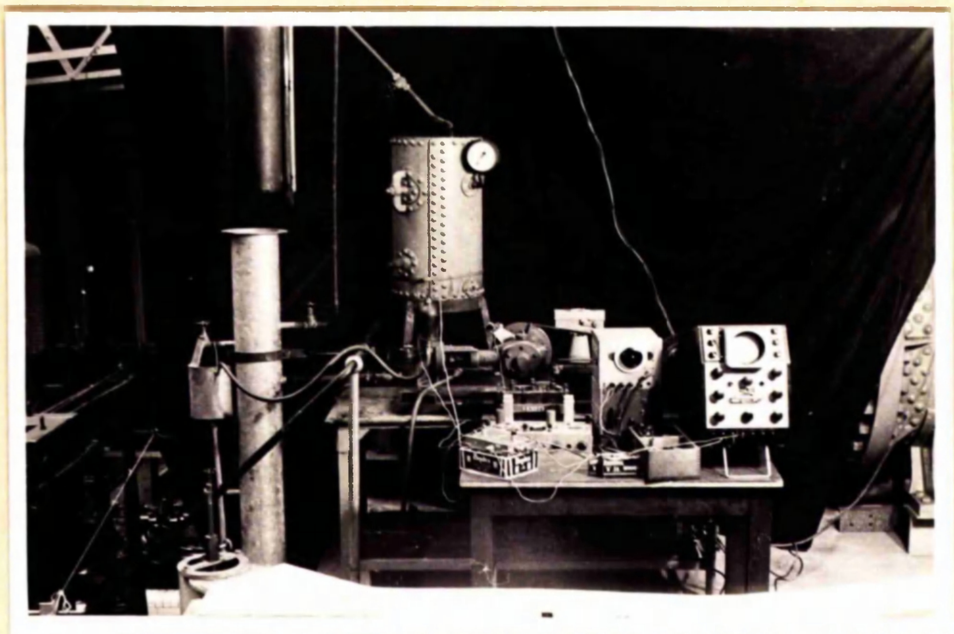
The pump was driven by a variable speed D.C. motor through Vee belts. To obtain optimum control of speed over a wide range, a 13 step Varioratio gearbox was incorporated in the transmission.

To avoid spilling of water when changing valves or springs, drains were fitted at the delivery valve chest and on the suction pipe at pump entrance. A valve was also fitted to the suction air vessel to regulate its air content. To facilitate observation of this air volume a window was fitted and a small electric bulb inserted.

(b) Water Pump Circuit. The suction pipe terminated with a non-return valve at the bottom of a 7 in. diameter suction tank, 12 ft. below pump level. By means of 6 overflow valves at 2 ft. intervals, the water level in the tank, and hence the pump suction head, could be varied by intervals. In later tests the 12 ft. long, 3/4 in. diameter suction pipe was disconnected and water then flowed into the pump under a positive 3 in. head.

To obtain a steady discharge flow from the pump an additional discharge receiver was fitted which had a volume 2490 times the piston displacement. In later tests this receiver was dispensed with, resulting in a more stable behaviour of the pump at high





WATER PUMP AND PLANT LAYOUT. FIG. Nº 33.

speeds. The discharge flow and pressure were controlled by a needle valve regulating the flow into an orifice tank sited above the suction tank. The flow was measured by either of two interchangeable calibrated orifice plates. Various refinements were introduced to damp oscillations of the water level in the orifice tank and to minimise air entrainment in both orifice and suction tanks.

(c) Valve Displacement Recorder. (i) Capacity Pick-up Units. In the measurement of valve displacement in gas compressors a capacity type pick-up was satisfactory in spite of the presence of oil, which has a higher dielectric constant than either air or Freon 12. An attempt was made to measure water pump valve displacement by this technique. The first capacitive unit consisted of a castellated "Perspex" ring which replaced the valve chest cover spigot. A "Perspex" tube joined to the ring led out the lead from a brass annular plate, which fastened to the "Perspex" ring above the valve disc, formed the insulated plate of the element. The whole unit was baked in varnish. The insulation of this unit was infinite in air, but only 50,000 ohms. in water. A second capacity unit was therefore made where the insulating medium was beyond suspicion.

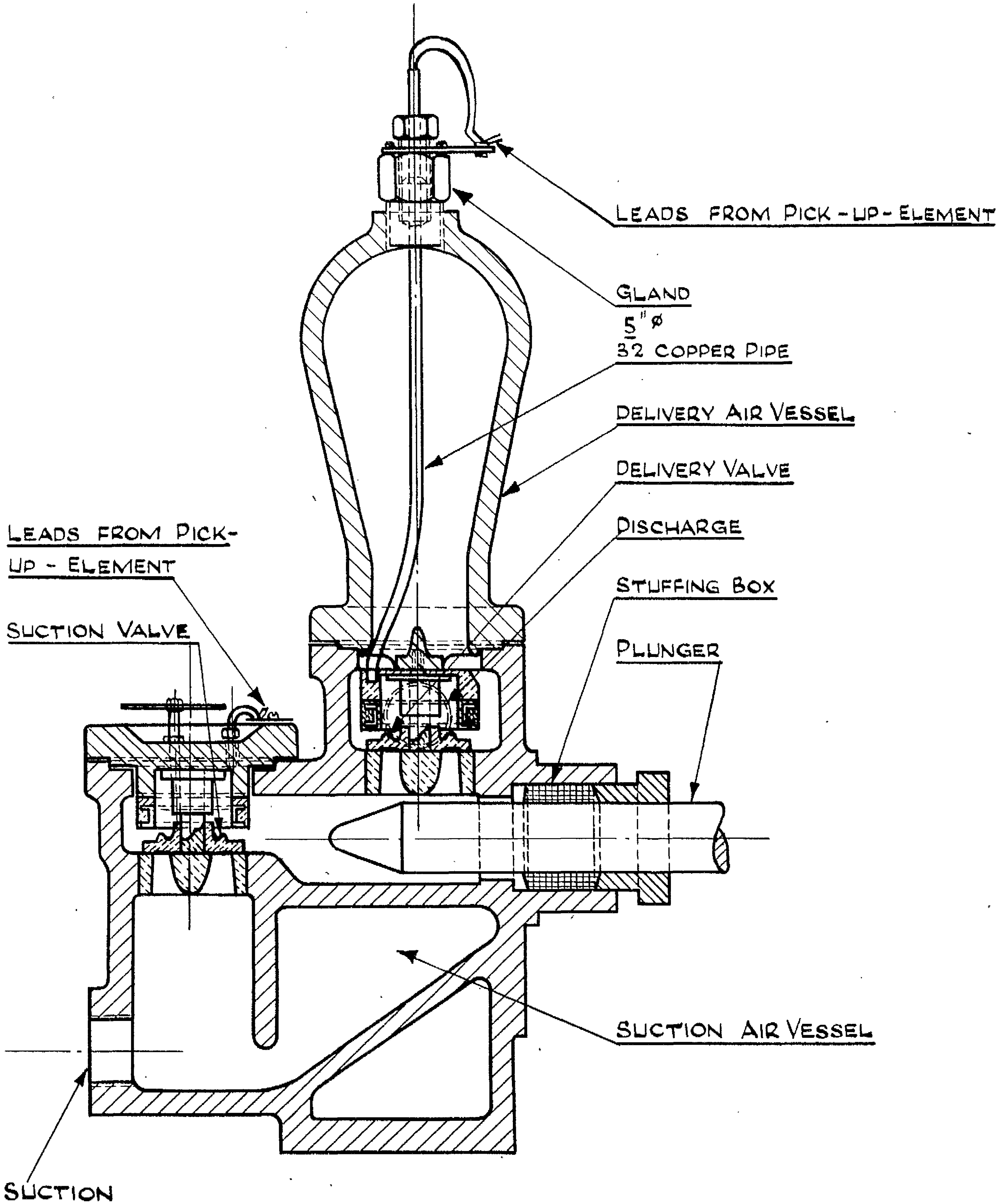
A length of oval section flint glass was sealed at the ends and bent into a ring. Near one end, a glass tube was fused and the whole filled with mercury. The results of static and dynamic tests with this unit showed that the electrical response due to the motion of the water and the presence of air bubbles swamped the response due to the valve movement. That this was likely to happen may be shown as follows:-

If the total distance between the two condenser plates is 6 m.m., made up of 1 m.m. glass and 5 m.m. water, then the capacitance of the unit is 5.12 units. If the distance between the plates is now increased to 10 m.m., i.e. 9 m.m. of water, the capacitance is now 4.07 units. There is therefore a capacitance change of 20.5% due to a valve movement of 4 m.m. If one air bubble of diameter 0.5 m.m. is introduced into the original arrangement the capacitance is now 1.45 units, i.e. the change of capacitance due to the bubble is 71.7% of the original vlaue. In practice the unit was so sensitive to the presence of minute quantities of air that this technique might prove useful in researches on air entrainment in water.

Since both air and water may be regarded as equally non-magnetic these difficulties did not occur with a valve lift measurement unit working on the inductive principle.

(ii) Inductive Pick-up Unit. Fig. 34 shows a section through the suction valve with the inductive pick-up unit in place. The construction of these coils had to be carefully carried out to meet the arduous conditions under which they had to operate.

The coils were wound with 120 turns of 38 gauge varnish insulated wire on "Tufnol" formers and were the same for both valves. These coils were then sealed by casting in a thermosetting resin, "Araldite". This material has great mechanical strength and also considerable adhesive strength which was utilised in casting the suction valve coil directly on the valve cover spigot. On the same account release from the moulds was difficult. "D.C. 7 Mould Release Wax" was used. "Silicote" was less successful.



ECTION THROUGH PUMP WITH INDUCTIVE PICK-UP UNITS.

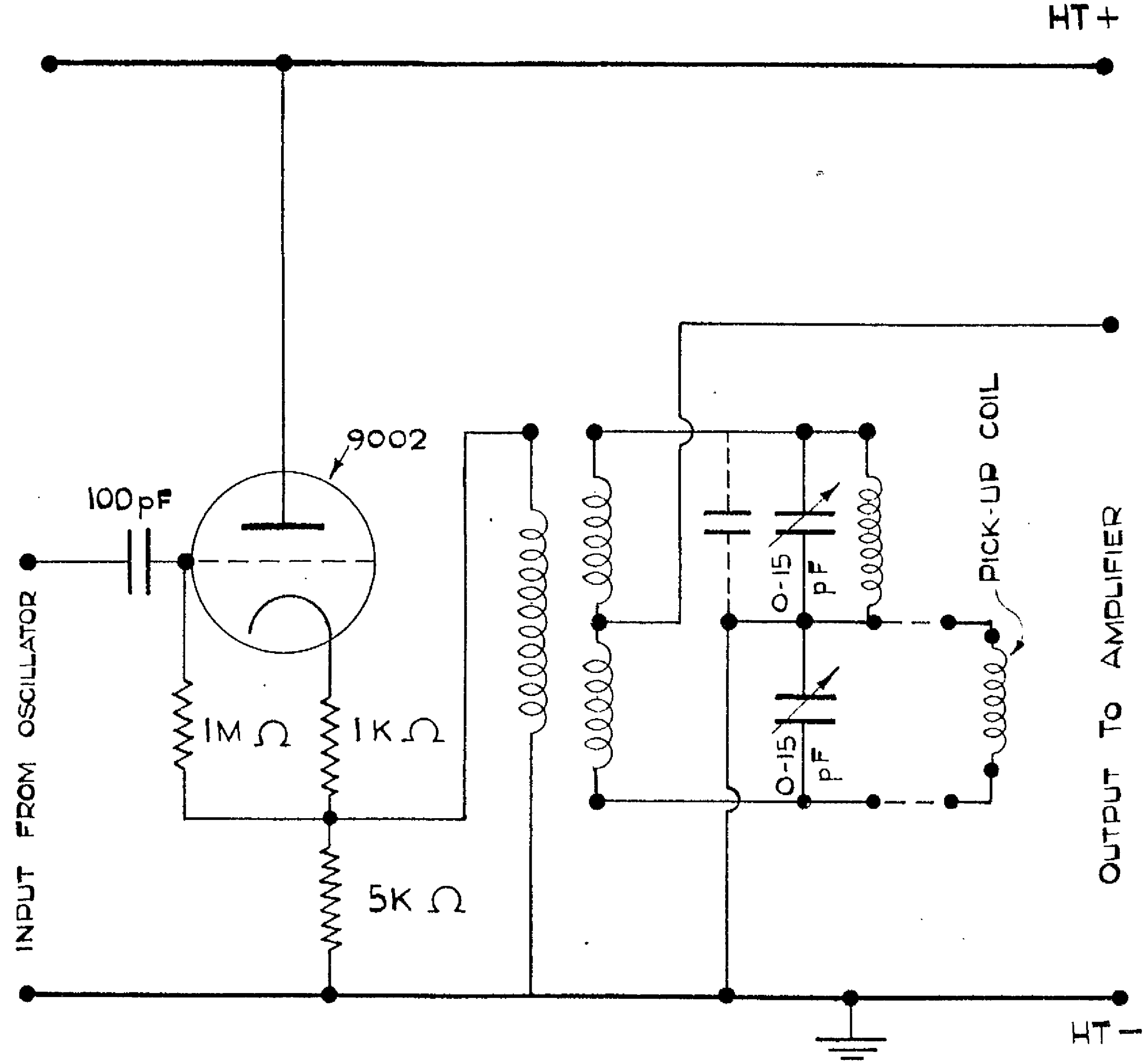
ALE:- HALF SIZE. FIG. No. 34.

The delivery valve pick-up coil could not be cast directly on to the cover and a base was provided with such dimensions that the same moulds could be used in the casting process. In this case the leads from the coils were taken out through a 5/32 in. diameter tube and through a gland seal at the top of the air vessel.

Identical balancing coils were wound for each inductive bridge, but were not cast in Araldite. At full valve lift (about 1/4 in.) the pick-up coils were clear of the valve disc by about 1/32 in. To reduce hydraulic damping effects on the valve, water vents were drilled above the coils.

These inductive pick-up coils were used in the pump when operating with non-magnetic (Tufnol) valve discs. It was found that the coils were impervious to valve motion, cyclical pressure, pressure wave effects, water motion and the presence of air. There was a slight disturbance of the oscilloscope trace due to spring motion. The springs were therefore shielded by thin steel tubes from the elements and this disturbance eliminated. Test results with and without the coils in the pump showed that the presence of the pick-up coils did not affect pump performance. Fig. 55, Appendix IV shows the calibration curves for these pick-up units.

(d) <u>Inductive Bridge and Demodulator</u>. The inductive bridges in each channel were identical (Fig. 35) and were readily interchangeable with the capacitance bridges in the recorder already described (Chapter 4). The bridge was driven through a transformer, the secondary coil of which formed part of the bridge. The phase unbalance of the bridge and the capacitance



INDUCTIVE BRIDGE AND CATHODE FOLLOWER STAGE

FIG. No. 35.

of the lead to the pick-up were compensated for by adjustment of the air trimmer condensers in parallel with the pick-up coil and the balancing coil. All four coils were self-resonant at 500 Kc/s. Since one corner of the bridge could be earthed a single ended output was obtained, unlike the capacitance bridge which required a differential amplifier.

Demodulator stages were inserted in each channel, one half of the 6H6 double-diode valve being used for each. The current flowing through this valve was half wave D.C. with changing amplitude. The smoothing condenser eliminated the R.F. carrier, and since the voltage across the series resistance was proportional to the current through the valve, a varying D.C. voltage output was obtained. The line traces for each valve thus obtained could overlap on the screen without causing ambiguity. The demodulator stage could be readily by-passed. This feature was useful since initial balancing of the bridges was more easily carried out by studying the R.F. carrier.

(e) Valves and Valve Springs. During the first test series, primarily concerned with valve movement, brass valve discs were used with two stiffnesses of spring. The weight of the valves in water W_W were:-

Suction Valve .0845 lb. Discharge Valve 0.0866 lb.

The stiffness of the springs, k, were:-

Light	Suction Valve	Spring	0.587	lb/in.
"	Discharge "	"	0.465	lb/in.
Stiff	Suction "	"	3.32	lb/in.
"	Discharge "	"	3.4	lb/in.

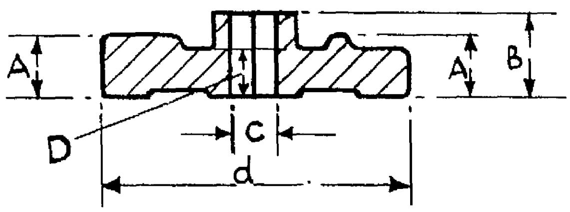
In the second test series to study valve lift, and the effect of valve variables on pump performance, 6 valves of various stiffness were used. Details of these are tabulated in (Fig.36).

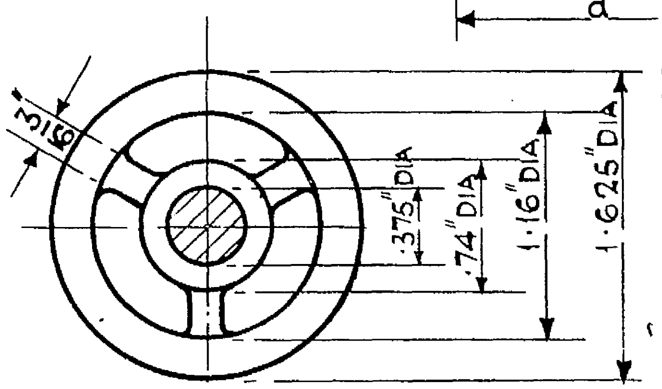
Spring Data

	Identification Tag No.	Symbol	Free Length Inches	Wire Dia. Inches	Mean Coil Dia Inches	W' LB	W'_W LB	K LB/INCH
Suction	1	L	1·200	·036	·860	·0065	·0057	··38
	4	H	1·050	·055	·865	·0111	·0098	3·32
	5	S	1·620	·055	·865	·0154	·0137	2·30
Discharge	2	L	1·330	·036	·785	·0058	·0051	·57
	3	H	1·075	·055	·865	·0113	·0100	2·71
	6	S	1·625	·055	·870	·0154	·0137	2·23

Valve Data

	Ident. No.	Symbol	Material	Dimension as shown below, inches					$\ell = \pi d$ INCH	$a = \frac{\pi}{4}(d^2 - c^2)$ IN^2	W LB.	Ww LB.
				d	A	B	C	D				
Suction	3	'D'	DURAL	1·675	·335	·502	·380	·210	5·26	2·09	·0612	·0395
	6	'd'	"	1·435	·230	·425	·378	·100	4·51	1·51	·0296	·0187
	5	'B'	BRASS	1·436	·209	·390	·378	·200	4·51	1·51	·0975	·0855
Discharge	2	'D'	DURAL	1·659	·325	·508	·378	·230	5·20	2·04	·0624	·0386
	4	'd'	"	1·434	·285	·510	·376	·150	4·50	1·50	·0373	·0213
	6	'B'	BRASS	1·435	·202	·395	·380	·205	4·51	1·50	·0966	·0850





$d_1 = 1 \cdot 16''$	
$\ell_1 = \pi d_1 = 3 \cdot 64''$	
$a_1 = \cdot 63$ INCH^2	
$a_1' = \cdot 49$ INCH^2	

Top of spring to valve seat	
Suction	Discharge
1·090"	1·125"

Valve Seat Dimensions

Dimensions of Water Pump, Springs and Valves — Fig. No. 36.

CHAPTER 8.

WATER PUMP VALVE MOVEMENT.

Rectified oscillograms of the movement of both suction and discharge valves were obtained simultaneously at pump speeds 270, 350 and 430 r.p.m. with no valve springs, light springs and stiff springs at delivery pressures of 0, 75 and 150 $lb/in.^2$ at suction heads of 0, 6 and 12 ft. From these the general behaviour of the valves was studied and the lag at opening and closing of both valves was measured.

8. 1 General Description of Valve Movement.

Fig. 37 explains a typical oscillogram and shows the small effect of the non-linearity of the calibration curves. Superimposed is a displaced sine wave which is a good approximation of the movement under these particular conditions.

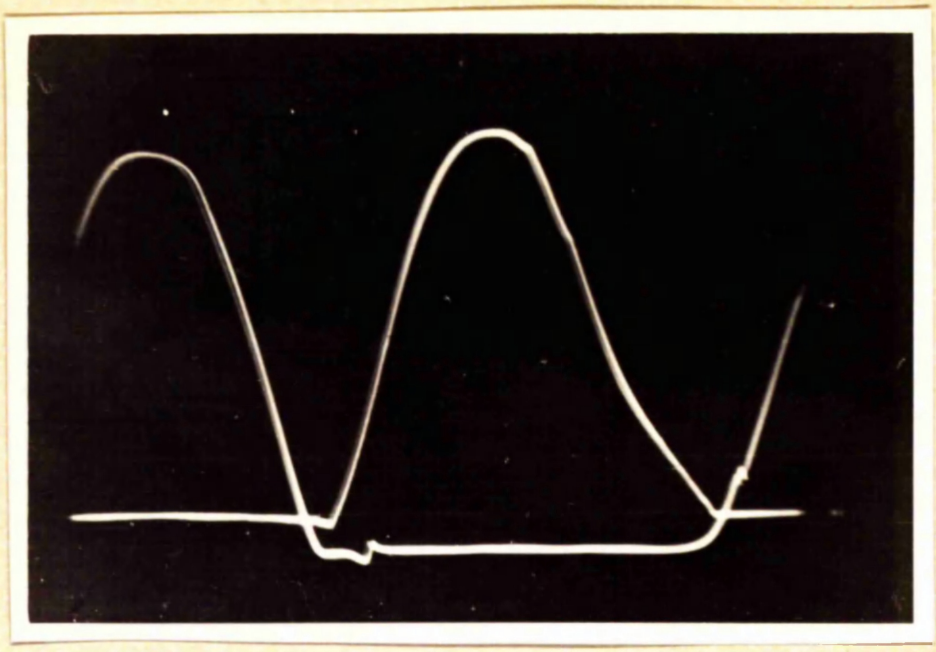
The suction valve always opened after piston reversal. The initial opening of the valve was relatively slow, but the velocity increased rapidly and, sometimes with a few oscillations, the maximum lift was reached close to mid-stroke. The velocity when closing was steady and on approaching the seat was nearly always lower than the velocity when leaving the seat. The valve always closed after piston reversal and came to rest without unduly rapid deceleration.

The relatively slow opening was probably due to the air dissolved in the water and to leakage. After the discharge valve has closed the piston has to move a distance until the pressure drop is sufficient to accelerate both the valve and the suction column. Hence both inertia and elasticity effects will slow the initial suction valve opening. Since the subsequent acceleration was relatively great, equilibrium conditions with

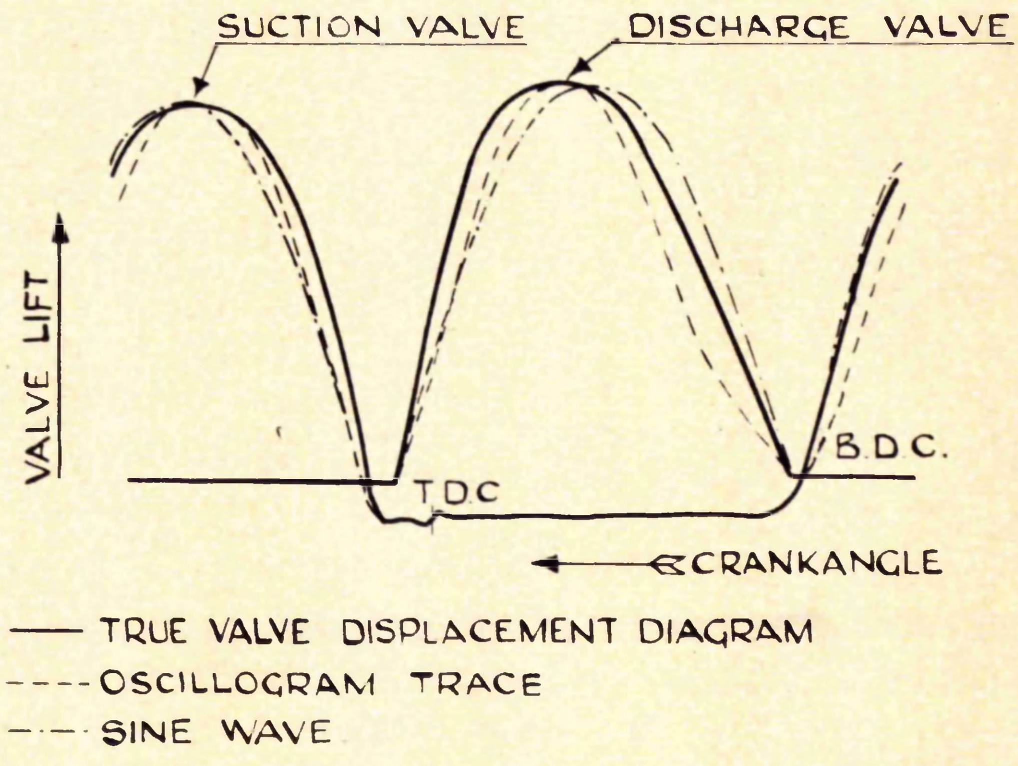
OSCILLOGRAM OF VALVE - DISPLACEMENT

LIGHT SPRING; 430 R.P.M.

DISCHARGE PRESSURE = 0 SUCTION HEAD = 0



TRUE VALVE DISPLACEMENT DIAGRAM OBTAINED FROM CALIBRATION CURVES



SAMPLE OSCILLOGRAM AND THE EFFECT OF NON-LINEARITY OF PRESSURE PICK-UP ELEMENT

FIG. No 37.

valve load and water flow could be reached before maximum piston speed. The velocity of approach to the seat was relatively low since the descending speed was mainly regulated by the decreasing water flow.

The discharge valve often opened before the suction valve had come to rest and always after piston reversal. This valve opened suddenly and reached its maximum lift about 25^{o} after the crankangle position for maximum piston velocity. Again the velocity on closing was more rapid than on opening. When spring loaded the valve closed with a relatively small lag. Retardation of this valve before reaching its seat was small and it closed with a high velocity.

The sudden opening of the discharge valve was due to the impact of the water forced by the piston into the space beneath the valve which is normally filled with air and water vapour. This sudden initial opening resulted on occasion in the valve vibrating. The inertia of the increasing water flow would cause the continued rise of the valve after maximum piston velocity had been reached. Since the water flow rate decreased after maximum piston velocity the descent was more rapid than the rise.

8. 2 Valve Lag.

The valve lags were found to be virtually independent of discharge pressure. The lags at 0, 75 and 150 lb/in.^{2} discharge pressure were therefore averaged and plotted against pump speed for suction and discharge valves, with light and stiff springs. Generally, the lag increased with pump speed and decreased with increased valve loading, but there was considerable scatter due to both experimental error and irregular valve behaviour. With the light spring the effect of speed on lag was more pronounced. Suction head had relatively small effect on lag at valve closure. The time between discharge valve closure and suction valve opening increased with suction head and with spring stiffness. The time between discharge valve opening and suction valve closure increased with pump speed. The opening lag of one valve depended on the closing lag of the other.

The reason for the greater effect of speed on lag with the light spring was probably that inertia effects represented a larger fraction of the force opposing the valve and the valve therefore did not travel in accordance with the water flow through the valve to the same degree as with the valve with stiff spring. If valve loading or suction head was large, a greater vacuum would be required in the cylinder for the opening of the suction valve. This required a greater piston displacement after discharge valve closure, hence the increased opening lag of the suction valve.

The discharge valve often opened before the suction valve had come to rest. This may have been due to the inertia effect of the water in the suction column and in the cylinder lifting the discharge valve by impact. As would be expected this effect increased with pump speed.

The suction valve lag at B.D.C. with stiff spring and 12 ft. suction head, appeared to be greater at 270 r.p.m. than at 350 r.p.m. perhaps due to irregularity in suction valve closure. According to the oscillogram the valve approached the seat in the normal way at 270 r.p.m. but was delayed a short distance from the seat for some reason. Consistently the opening lag of the discharge valve was somewhat increased.

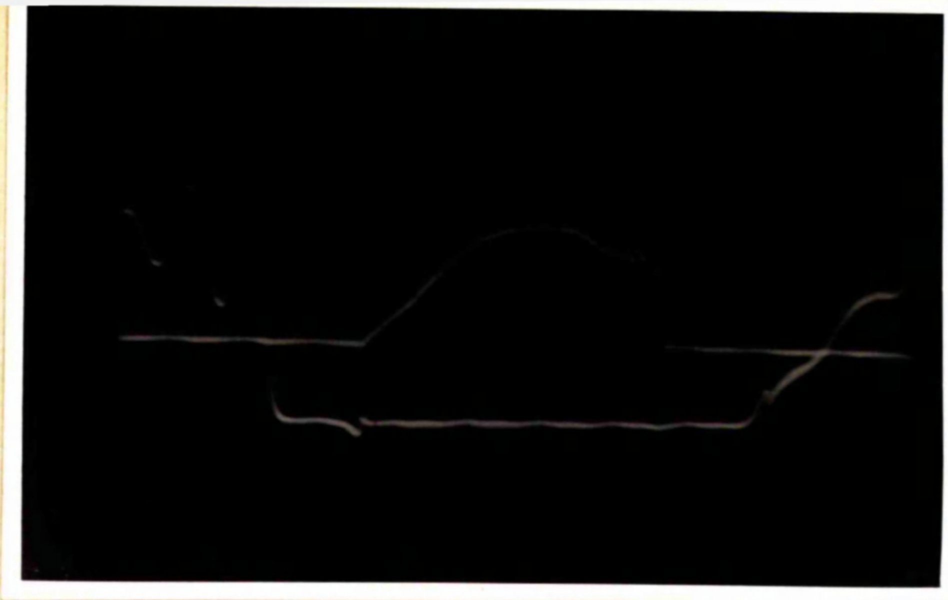
8. 3 Irregular Valve Movement.

Under certain pump conditions quite abnormal valve travel diagrams were obtained.

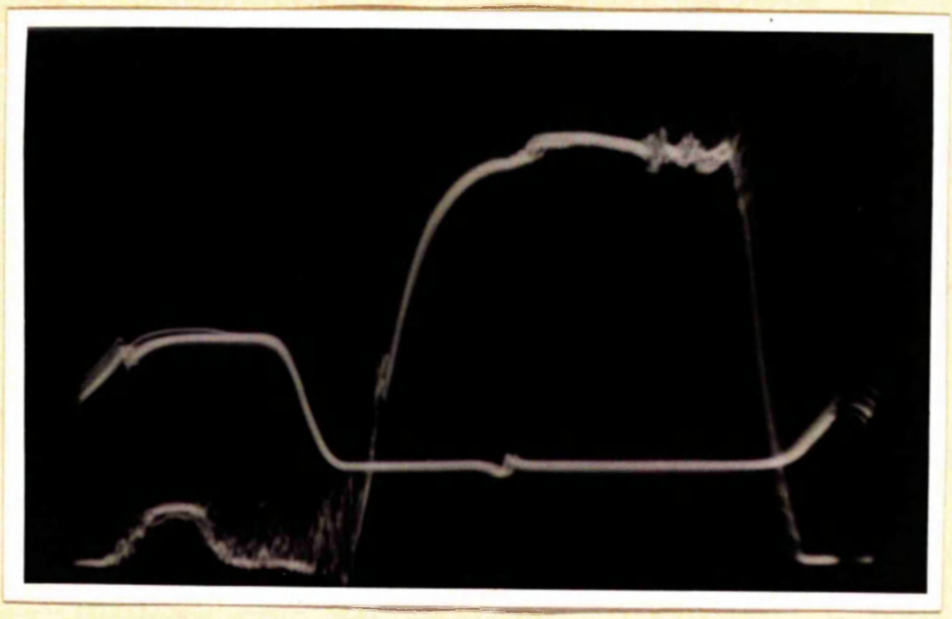
Fig. 38a shows the effect of air in the suction pipe. The valves closed in the normal way but opening was delayed due to the elasticity of the air in the pump cylinder. This caused considerable oscillation of the suction valve. The resulting severe impact on the discharge valve was clearly shown.

Fig. 38b shows the valve movement under severe knocking at 430 r.p.m., 80 lb/in.^{2} delivery pressure and 12 ft. suction head with no valve springs. The discharge valve struck the stop and rebounded. The closure was late and at high velocity, causing rebound from the valve seat. A remarkable feature of this diagram was that the discharge valve left its seat during the suction stroke - against a delivery head of 80 lb/in.^{2}. This may have been due to an instantaneous vacuum over the discharge valve due to pressure waves in the delivery pipe.

Fig. 38c shows the development of an abnormal valve movement under unstable conditions, aggravated by the absence of valve springs. The discharge valve descended in the normal way. In subsequent cycles the valve left the stop and started to descend but probably due to resonance effects, showed a tendency to rise again. This tendency increased and after four pump revolutions the valve, after leaving the stop, rose again and finally closed late on the suction stroke. Due to late closure the suction valve opening was very small and approximately



EFFECT OF AIR
IN SUCTION LINE
STIFF VALVE SPRINGS
270 r.p.m.
DELIVERY HEAD 75 lb/in^2
SUCTION HEAD 12 ft



LIFTING OF
DISCHARGE VALVE
DURING SUCTION
NO VALVE SPRINGS
430 r.p.m.
DELIVERY HEAD 80 lb/in^2
SUCTION HEAD 6 ft.

RAPID CHANGE OF
VALVE TRAVEL DIAGRAM
AT CRITICAL CONDITIONS
NO VALVE SPRINGS
350 r.p.m.
DELIVERY HEAD 15 lb/in^2
SUCTION HEAD 12 ft

IRREGULAR VALVE TRAVEL DIAGRAMS FIG. N^{o} 38 symetrical about B.D.C. No water was delivered until pump conditions were changed: a variation of discharge pressure, within limits, was not sufficient to restore normal operation. During delivery, water was flowing rapidly into the air vessel. As piston displacement decreased, the inertia effect may have created a low pressure in the valve chest which caused flow reversal in the delivery pipe. As the delivery passage was in the plane of the discharge valve disc the inertia effect of the water jet impinging partly under the disc may have kept it open. When the piston decelerated on the suction stroke, the pressure in the valve chest would build up again and the valve consequently descend.

CHAPTER 9.

WATER PUMP AND VALVE PERFORMANCE.

9. 1 Range of Tests.

During the study of the valve displacement diagrams flow quantities were noted and the volumetric efficiency of the pump computed. Maximum valve lifts were noted and compared with the theoretical values. Observations were made of valve noise. These results are not recorded here but from the operating experience gained the following systematic test programme was carried out. Experiment 1 consisted of 108 tests to investigate pump performance using three types of paired valves and two stiffness of springs at suction and discharge, at three speeds (325, 400 and 495 r.p.m.), three discharge pressures (0, 25 and 50 lb/in.^{2}) with the suction air vessel initially (a) full and (b) empty. Experiment II consisted of 12 tests in which dissimilar valves and springs were used on the suction and discharge side. Suction conditions were not varied and water was flowing into the pump under a positive 3 in. head in all tests.

9. 2 Experimental and Calculated Results.

From the observed results of flow quantity, maximum valve lift and valve noise intensity, the quantities; x, the ratio of the radial flow area at perimeter of valve disc to the valve throat area excluding the central spigot and ribs; c'_B and c'_K, the valve discharge coefficient based on valve disc area and throat area respectively, were calculated for the suction valve in all cases. The noise limit, λ, was evaluated for both valves, and the volumetric efficiency computed, for all cases.

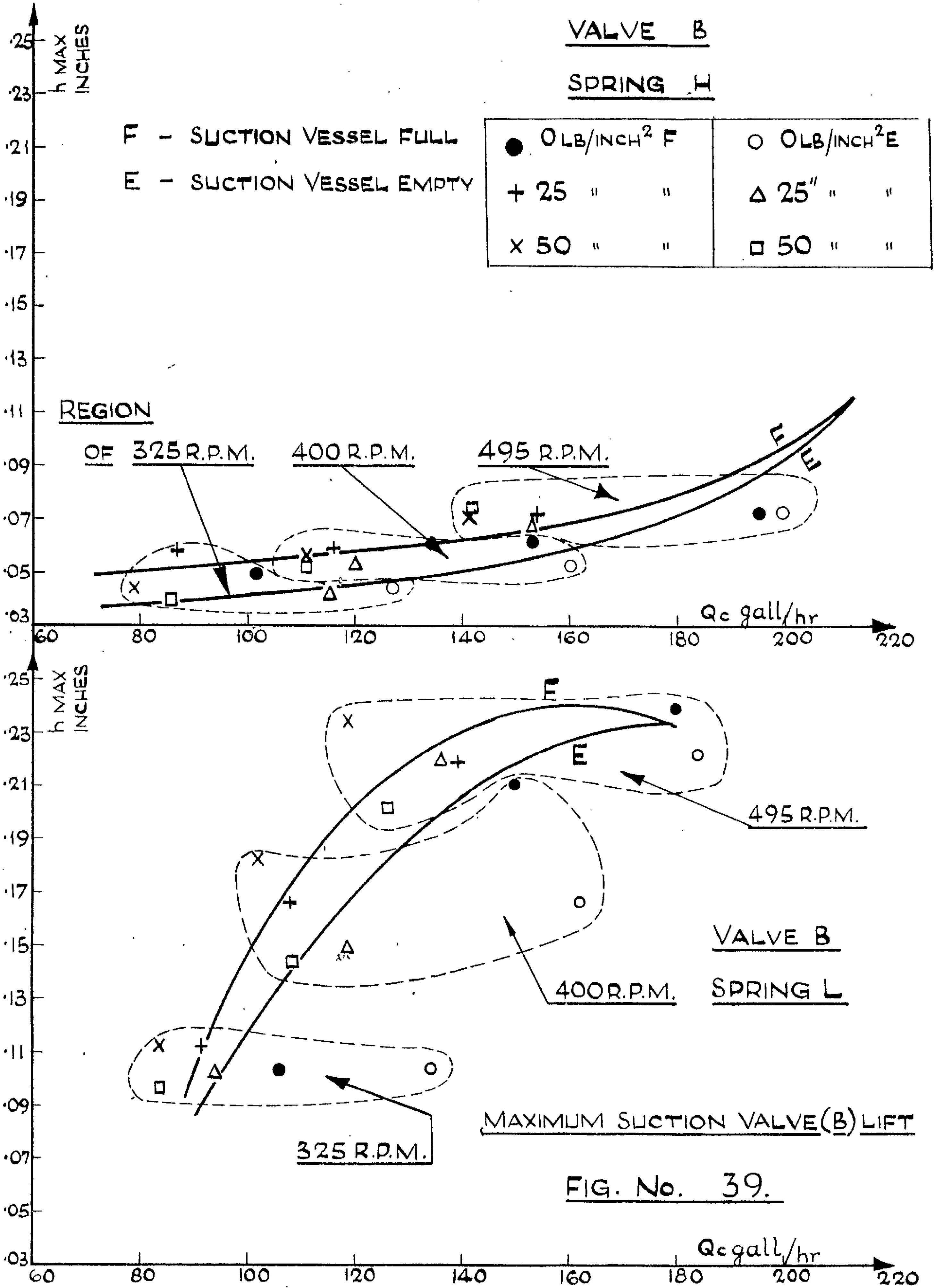
9. 3 Volumetric Efficiency.

In almost every test with zero discharge pressure the volumetric efficiency exceeded unity, sometimes by a considerable amount. The abscissae of the points in equal speed bands in Fig. 39, 40, 41, are directly proportional to the volumetric efficiency.

At a nominal zero delivery pressure, the pressure gradient across the valves was probably insufficient to ensure steady motion. While the valve travel diagrams did not show the valves leaving their seats due to wave action, except in the case with no springs already discussed, it must be deduced that there was a leakage past the valves to obtain values of volumetric efficiency greater than inity. The pressure wave effect causing this must have been small, since the valve lags were almost independent of delivery pressure. However, as the suction side had a small positive head of 3 in. of water, little wave action would be required to cause leakage. Such high values of volumetric efficiency were not obtained in the early tests when using large suction heads.

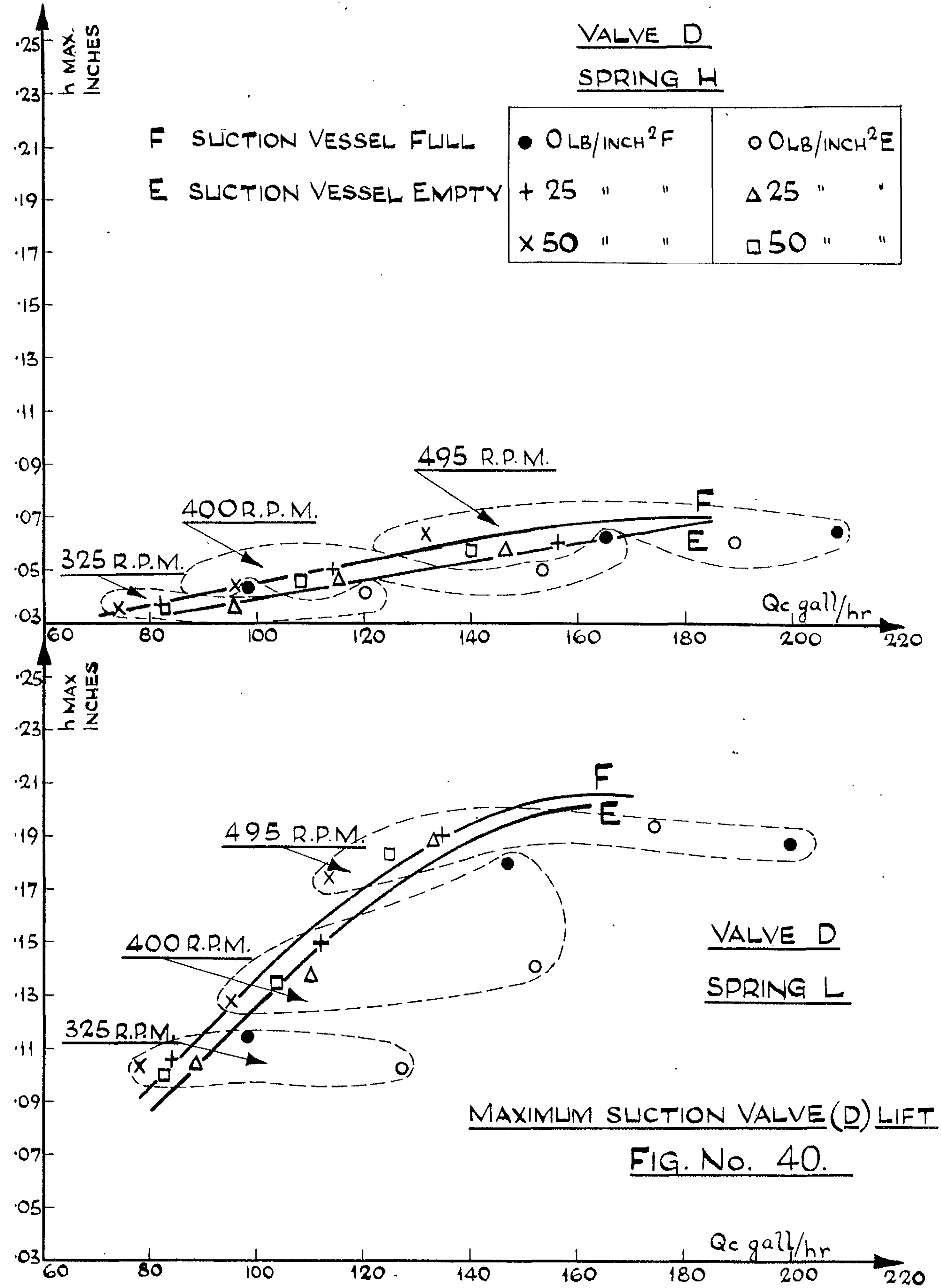
At higher speeds, when shock effects were increased, and there was greater chance of separation in the suction column, greater irregularity in volumetric efficiency was observed. Changes of spring loading and valve mass appeared to have little effect on the irregularity of volumetric efficiency probably because these factors are negligible compared to the effect of shock action shown on the travel diagrams at discharge valve opening.

At higher delivery pressures the volumetric efficiency reverted to the expected values. Due to improved suction conditions the volumetric efficiency was higher with the suction



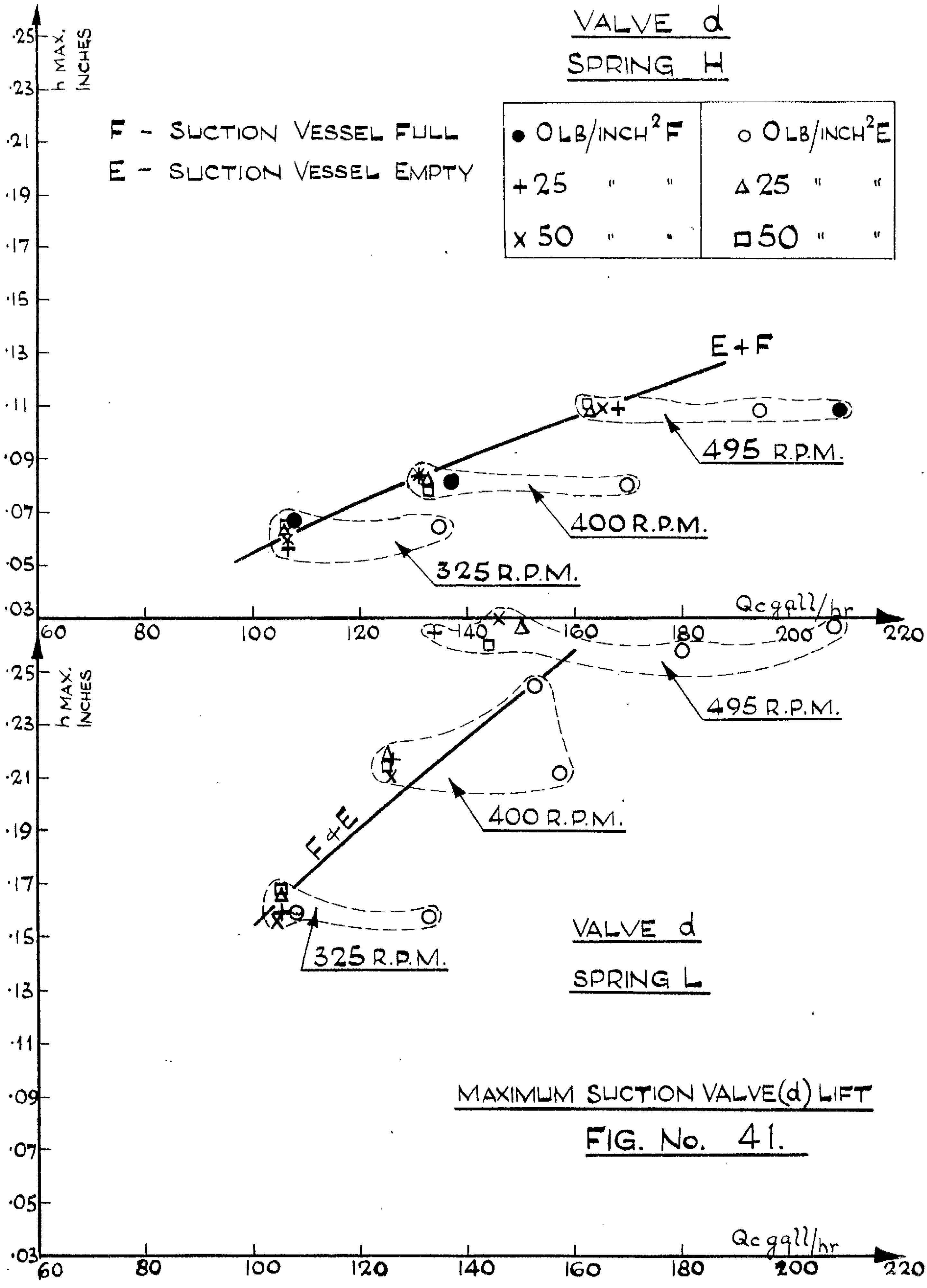
MAXIMUM SUCTION VALVE (B) LIFT

FIG. No. 39.



MAXIMUM SUCTION VALVE (D) LIFT

FIG. No. 40.



MAXIMUM SUCTION VALVE (d) LIFT

FIG. No. 41.

air vessel empty than with it full of water.

The volumetric efficiency was virtually independent of pump speed, probably because even the highest speed used did not greatly affect suction conditions.

There was little difference between the volumetric efficiency when using the brass valve (B) and the large duralium valve (D). The values obtained with the small duralium valve (d) consistently exceeded those of the other two. The volumetric efficiency using the last was hardly affected by suction vessel conditions or delivery pressure. It was considered that this was due to its steady lift characteristic which is discussed later.

The valve movement at zero delivery pressure, which resulted in very high volumetric efficiencies was not associated with excessive valve noise. The lower volumetric efficiencies at high delivery pressures, particularly at high speeds, were always associated with noise. As noise also occurred in the case of the efficient valve (d) at high speeds and high delivery pressures, noise could not be taken as a certain sign of low volumetric efficiency.

9. 4 <u>Maximum Suction Valve Lift.</u>

, The maximum lift of the suction valve is shown against delivery rate in Fig. 39, 40, 41. The vertical height of the equal speed bands showed the effect of delivery pressure and suction valve conditions on the valve lift. As the volumetric efficiency was so little affected by speed changes, the graphs may be regarded also as plots of valve lift against speed, for uniform delivery pressure and suction vessel conditions.

The maximum lift was almost directly proportional to the discharged volume and to the speed, other factors being equal.

Particularly for the valve (B), somewhat lower maximum lifts were obtained when the suction air vessel was empty, than when full. This phenoma, which was hardly noticeable with valves (D) amd (d) was more pronounced with the light spring. It has been seen that the suction valve may oscillate during opening due to shock from combined inertia and lag effects. These oscillations may be considered as superimposed on the normal wave path and they die out as lift increases. Since shock action was less severe with the suction air vessel empty, the resulting oscillations would be less and so also the observed maximum lift attained. Hence the difference in maximum valve lift for the two suction air vessel conditions was numerically larger for the lighter spring.

Valve (D) had a larger volume than valve (B) and had about half its mass, but about the same total valve loading when closed. The increased damping action of the water on the larger volume probably eliminated the oscillation prior to maximum lift and hence no difference in maximum valve lift due to suction air vessel conditions was observed.

Since the valve loading of valve (d) when closed was less than half that for the other two valves, this valve could open with less lag. The better timing, which probably accounted for the good volumetric efficiency obtained, would tend to reduce shock action at opening. The mass of valve (d) which was only 1/4 of the valve (B) assisted in the rapid damping of any small oscillations. Hence the argument advanced for valve (D) holds to an even greater extent for valve (d).

At 400 r.p.m., when using a light spring and zero delivery pressure, there was an exceptional case where the valve lift with the suction air vessel full of water exceeded the lift when empty, by about 0.027 in. for all three valves. This difference was thought to be due to resonance effects in the suction line at this speed, which was a critical speed also as regards valve noise.

The maximum valve lift was affected in an irregular manner by the delivery pressure. It was most obvious in the case of the brass valve (B), much less for valve (D) and least for valve (d). The variation may be neglected in the latter two cases, except at 400 r.p.m. as mentioned. The variation did not exceed 15% for valve (D) and 8% for valve (d) between delivery pressures of 0 to 50 lb/in.^{2} and speed range 325 to 495 r.p.m.

The values of maximum valve lift were virtually independent of valve mass. Without appreciable deterioration in performance, the maximum lifts for the valve (D), with the large sealing surface, were slightly lower. The neglect of the valve mass effect is therefore, compatible with the accuracy expected from design formulae.

9. 5 <u>The Formulae for Maximum Lift.</u>

From the experimental results obtained the validity of the formula for maximum valve lift due to Krauss' modified form of Berg's expression was assessed.

The formula is,

$$h_{max} = \frac{\pi \quad Qv}{c'kl_1 \sqrt{\frac{Ww + Fmax}{a_1 \rho} \cdot 2}}$$

The formulae for valve discharge coefficients are due to Lindner,

$$c'_k = \frac{1}{\sqrt{1 + 5x}}$$

and due to Linton

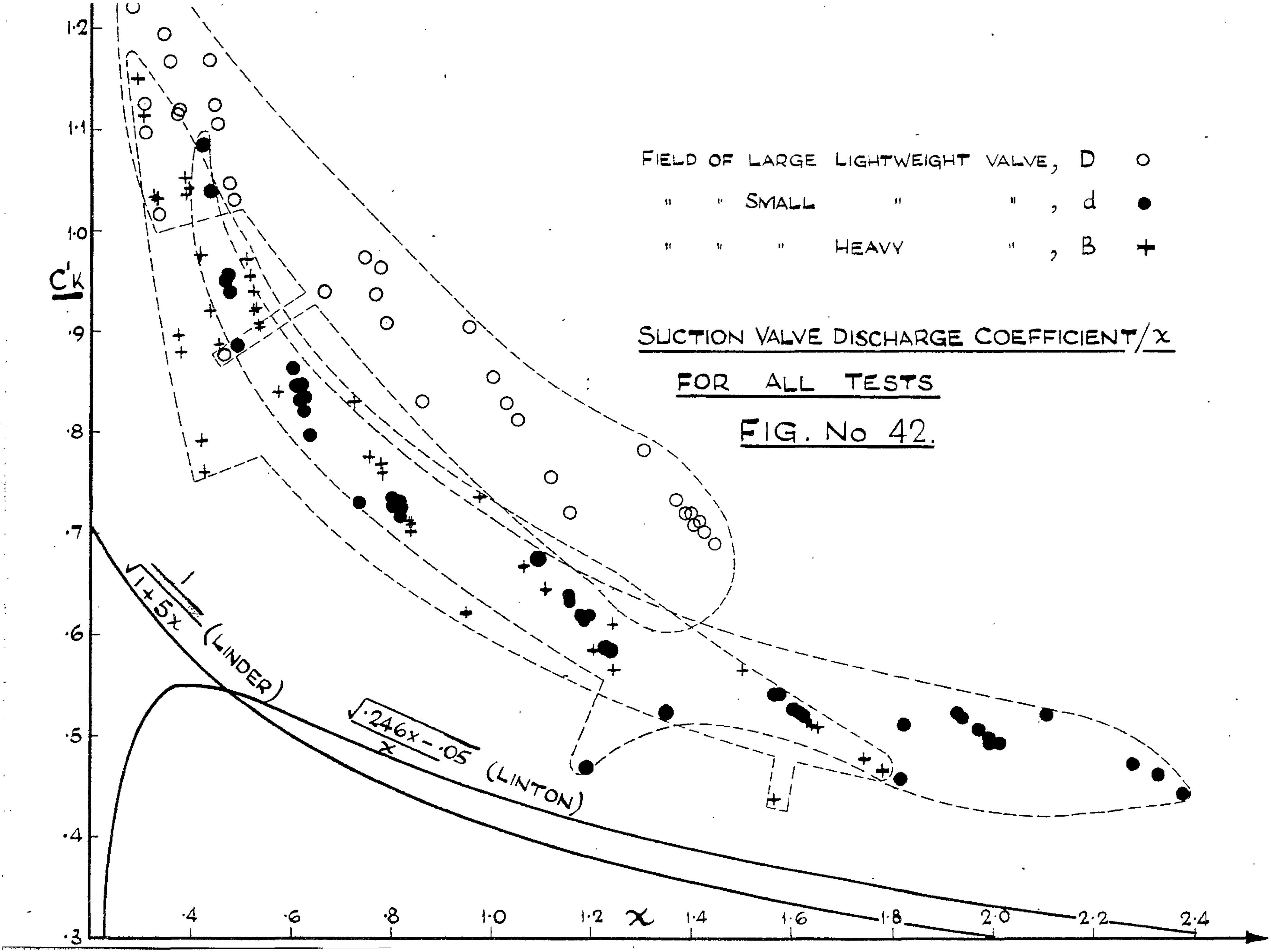
$$c'_k = \frac{\sqrt{.246x - .05}}{x} \text{ for } 0.4 < x < 1.1$$

where $$x = \frac{1.h_{max}}{a}$$

By transposing the formula of Krauss and using the expression x, values of c'_k was plotted against x (Fig. 42) together with Lindner's and Linton's functions, to find if the latter were reasonable approximations. The test results were then replotted for each valve, indicating the speeds, the spring used and the suction vessel conditions (Fig. 43,44,45). The values were only about 60% of the calculated values for c'_k so that the formulae gave lifts almost twice those actually obtained.

Since x is directly proportional to h_{max}, and c'_k is also a function of h_{max} there was a relationship between the valve lifts and the distribution of the points in Fig. 43, 44, 45. The lower lift values for valve (D) and the relatively small difference between those pertaining to valves (B) and (d) were apparent. The distinction between values with light and heavy springs, as well as the exceptional values, at 400 r.p.m. zero delivery pressure, suction vessel full, were clearly shown. The plotted points were reasonably colinear; there was a similarity in the distribution of points for the identically shaped valves (B) and (d). The high values of c'_k with valve (D), which had an excessively large outside diameter, may have been due to the same effects observed by Kahrs with his second diffuser valve.

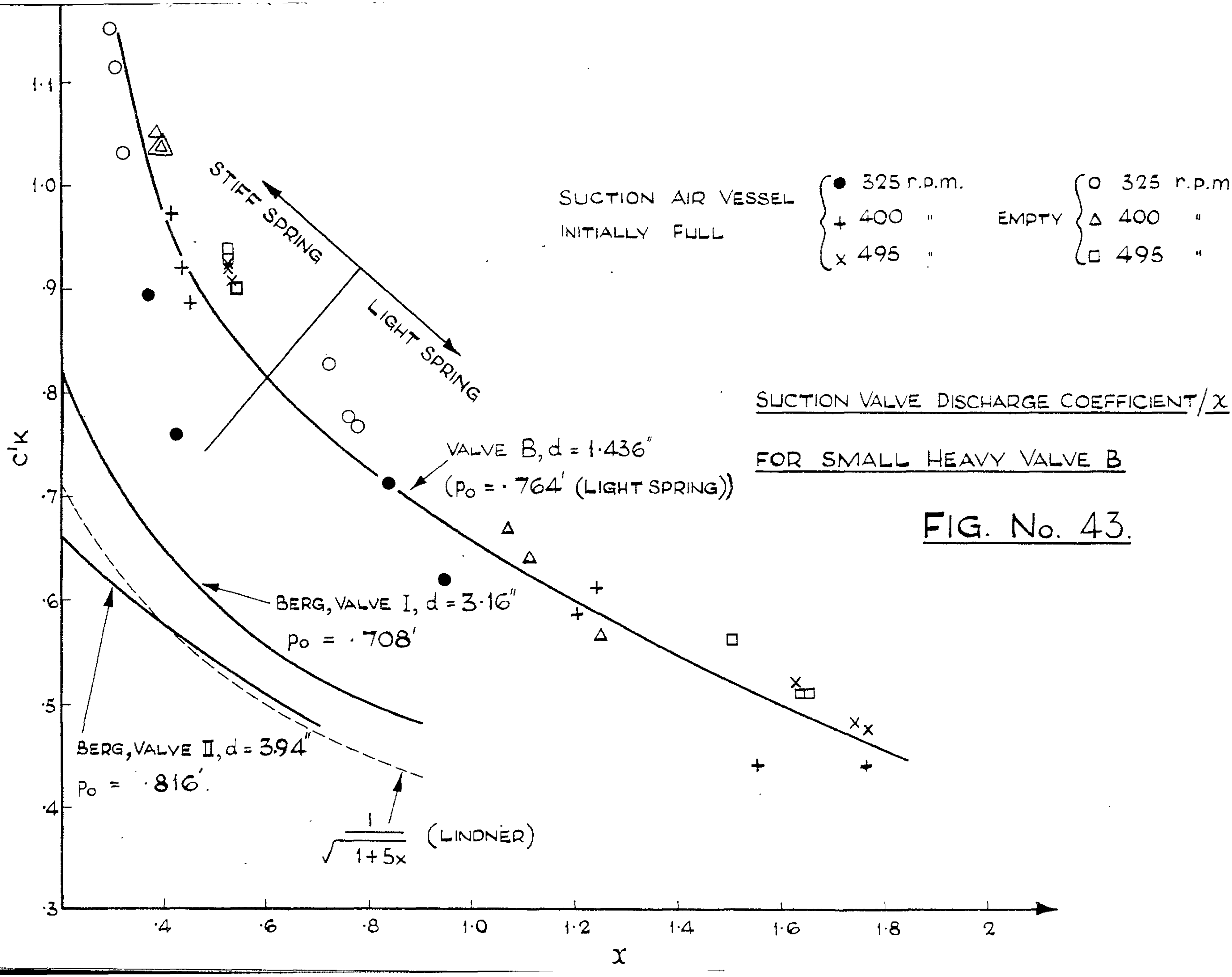
Values of the valve discharge coefficient c'_B, based on Berg's formula, which refers the maximum lift to the disc



SUCTION VALVE DISCHARGE COEFFICIENT/x

FOR ALL TESTS

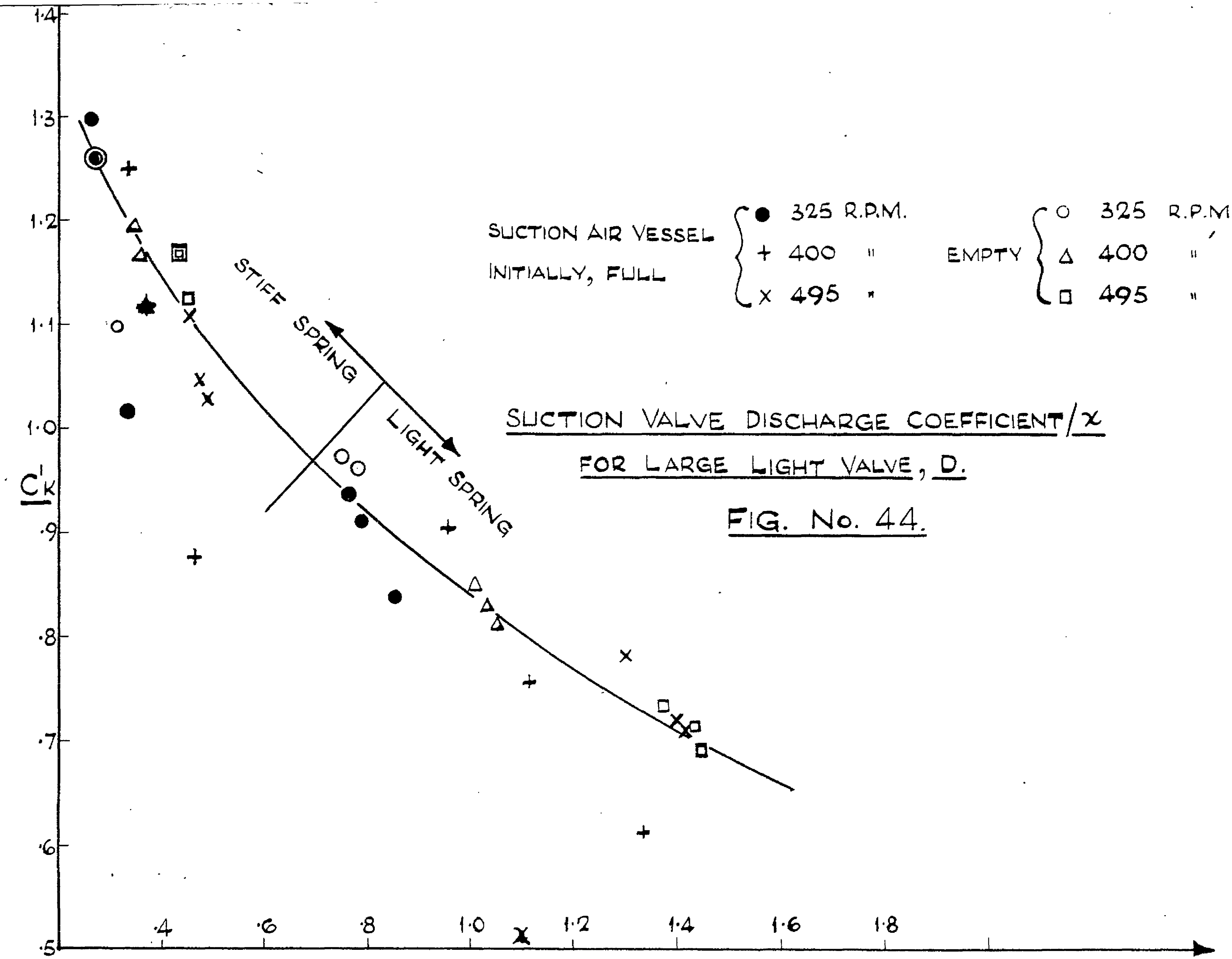
FIG. No 42.



SUCTION VALVE DISCHARGE COEFFICIENT/x

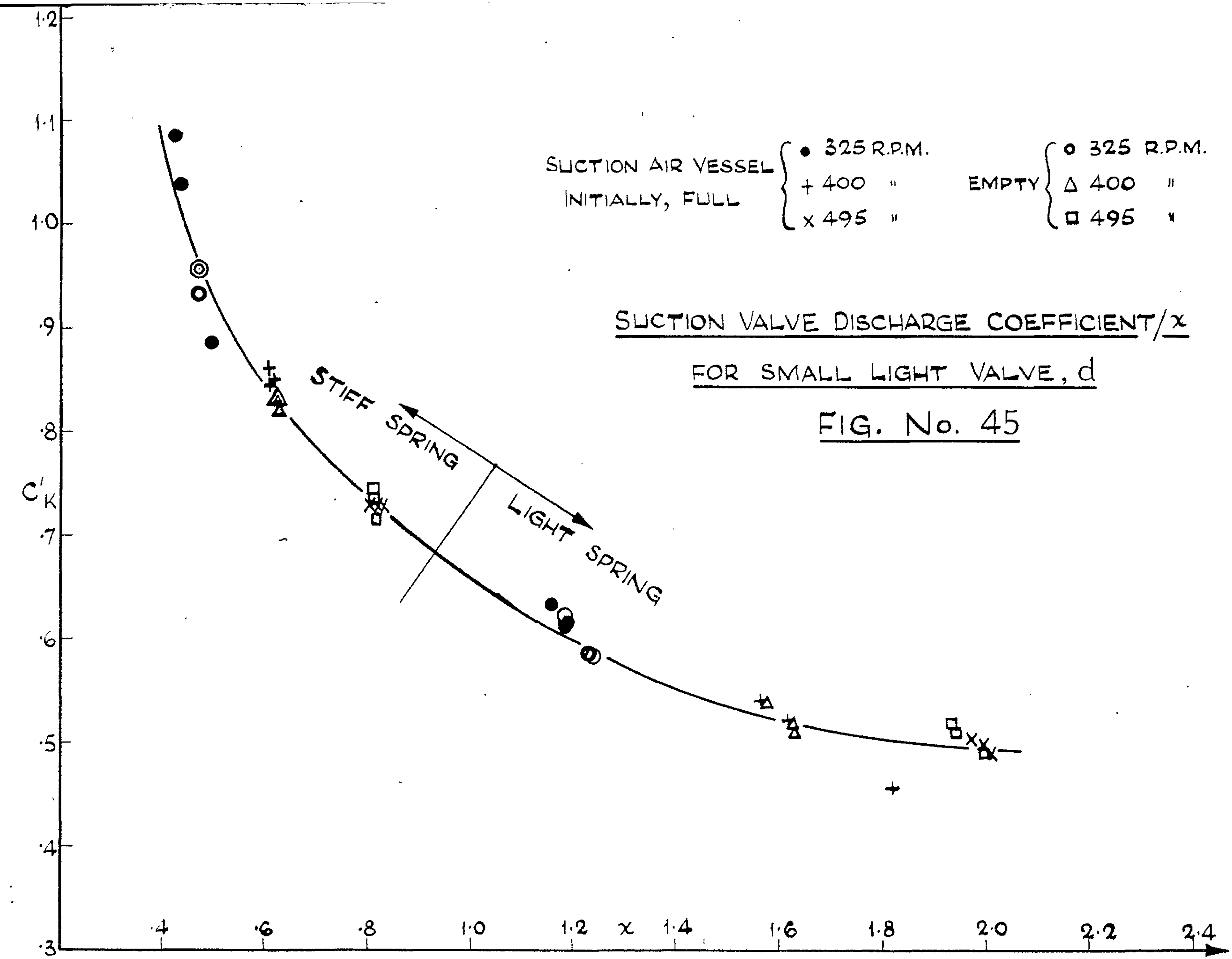
FOR SMALL HEAVY VALVE B

FIG. No. 43.



SUCTION VALVE DISCHARGE COEFFICIENT/x
FOR LARGE LIGHT VALVE, D.

FIG. No. 44.



SUCTION VALVE DISCHARGE COEFFICIENT/x

FOR SMALL LIGHT VALVE, d

FIG. No. 45

dimensions were about 25% higher than the c'_k values, indicating the arbitary numerical magnitude of the coefficient c'. To account for the difference between the results obtained and those calculated by previous formulae the earlier experimental work must be examined. Only Berg tested disc valves which had geometric proportions similar to the present valves. These were the valves I and II, shown in Fig. 125 and 126 of reference 64, which had disc diameters 80 m.m. and 100 m.m. Their c'_k to x curves have been reproduced together with Lindner's approximation in Fig. 44. From this it may be seen that c'_k increased noticeably as the diameter of the disc decreased. The geometrical difference between Berg's valves and valve (B) did not permit extrapolation. However, the trend was towards agreement between the present results and Berg's findings, recalling that Lindner and Linton both based their formulae on valves $3\frac{1}{2}$ in. and 5 in. diameter. Due to the lack of experimental data the effect of disc diameter on the c'_k to x relationship has not been previously observed. x is a complex quantity and it is not obvious why it should vary with disc diameter. Further investigation of this phenomena is required to explain it and to assess its importance on valve design.

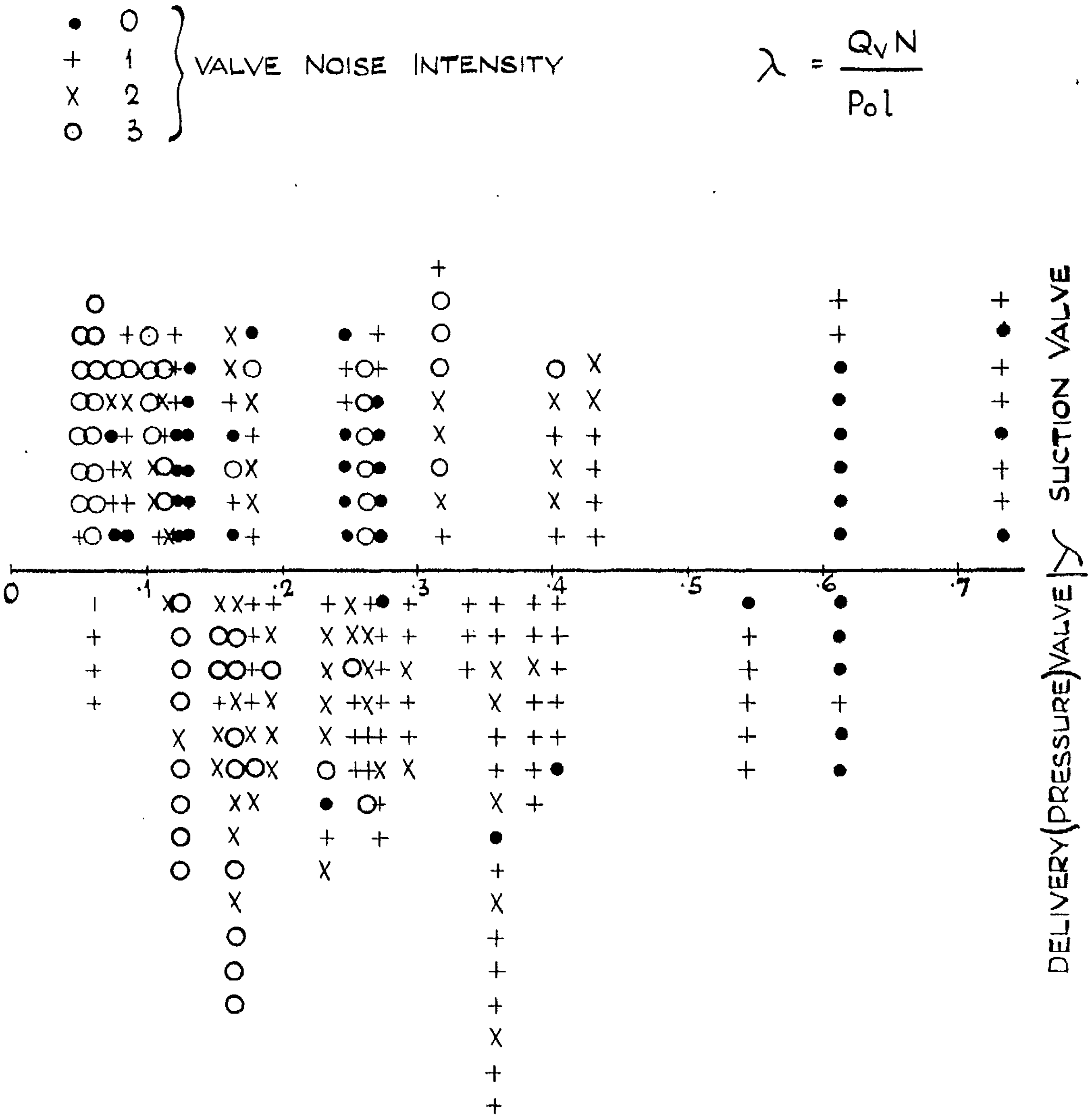
9. 6 Valve Noise.

It was found that in general, higher speeds and delivery pressures resulted in increased valve noise. It was also confirmed that higher spring loadings resulted in a decrease in noise provided that the spring force on the suction valve was not excessive.

Valve (D) was the quietest, probably because its lift was least. The adverse effect on its large seat area on lag appeared to be more than offset by its buffer effect and small mass.

In an attempt to establish the existence of a noise limit Fig. 46 was constructed. This noise limit was denoted by Berg's coefficient $\lambda = \frac{Q_v N}{P_o l}$ and was calculated for all the tests. The noise intensity observed is indicated at this value of λ. The vertical heights of the columns have no significance. It may be seen that high valve noise values were definitely associated with low values of λ, while low noise values tended to be associated with higher values of λ. No high value of valve noise (intensity 2 or 3) occurred at values of λ greater than 0.43 for the suction valve or 0.36 for the discharge valve; the highest noise rating (3) did not occur beyond $\lambda = 0.4$ and $\lambda = 0.27$ respectively. The difference between the two values may be due to the resonant conditions in the suction line and suction tank. Depending on whether the noise "2" or "3" is to be regarded as excessive noise, the mean values of λ are about 0.4 for noise intensity "2" and 0.33 for the greater intensity "3". Berg's value of 0.37 for his disc valve (80 m.m.) is therefore in good agreement. It must be noted, however, that a number of cases of quiet operation of the valves occurred at values of λ considerably below these figures.

There was no distinct noise limit, if this term implies the sudden appearance of valve noise at a certain value of λ. However, a value of λ was obtained, which if exceeded guaranteed a noise level below that regarded as the noise limit. On the other hand, if a lower value of λ was permitted, the valve was not always unduly noisy, provided that the delivery pressure was



VALVE NOISE INTENSITY FROM ALL TESTS/ λ

FIG. No. 46.

not high, the suction conditions reasonable, and the valve shape and mass suitable. The rigorous application of a limiting value of λ may therefore lead to unnecessarily high spring loads with corresponding hydraulic losses.

9. 7 <u>Tests with Dissimilar Valves.</u>

A series of tests was conducted to examine the influence of one valve on the other and to find whether performance could be improved by odd pairing of the valves and springs. Each test was compared with the two corresponding systematic tests in which identical valves and springs had been used at suction and discharge.

The maximum lift of the suction valve and its noise was hardly influenced by the introduction of a dissimilar discharge valve. The direction of any small change in lift depended on whether the discharge valve rose higher or lower than the suction valve when similar valves were used.

The discharge rate was about the arithmetic mean of the rates obtained during the corresponding tests with paired valves. There were two exceptions to.this general trend. In one case the discharge rate was less than that of the corresponding test with paired valves and springs and one case it was more. These effects were undoubtedly due to the timing of the valves which was governed by their shape and spring loading. The valve noise was greater with the arrangement which gave the lower discharge rate. When the valves were interchanged and the increased discharge obtained, the noise was not greater than in the corresponding tests with paired valves.

It would appear that improvements can on occasion be made by designing valves in dissimilar sets, but further experimental data is required.

CHAPTER 10.

CONCLUSIONS.

10. 1 Automatic Valves for Gas Compressors.

1. The mathematical analysis of the valve movement and the pressure drop across the valve during the cycle were expressed by equations which did not have exact solutions. Due to the simplification of the equations a step-by-step integration could be carried out without resort to the graphical methods of Costagliola.

2. The cantilever reed type valve examined did not vibrate freely under static flow conditions, but in actual operation valve flutter occurred. If the permitted valve lift was reduced, flutter could be avoided, but at the expense of increased valve throttling loss. When valve flutter occurred the labour involved in the solution of the equations was greatly increased.

3. The maximum pressure drop across the suction valve always occurred before the valve reached the stop, if permitted valve lift was adequate to avoid excessive throttling loss. Hence the valve displacement and the maximum initial valve pressure drop during opening were independent of permitted valve lift but were functions of spring stiffness and compressor speed. The pressure difference across the valve was rapidly reducing as the valve approached its stop. The final pressure difference across a valve as it reached the stop was an inverse function of the permitted lift while the spring force was directly proportional to the permitted lift. Hence the amplitude of the initiated flutter increased with increased permitted lift. The decreased

effect of the damping oil on the valve stop, as permitted lift increased, magnified this result when observed experimentally.

4. A portable two-channel valve displacement recorder was developed which did not require attachments to the valve and did not affect compressor clearance volume.

5. A differential pressure capacity type pick-up element was developed to measure the pressure drop across the valve in a high speed compressor. The loss of volumetric efficiency due to suction valve throttling was computed and found to be only about 5% of the difference to be accounted for between the theoretical indicated and actual volumetric efficiency.

6. The volumetric efficiency of the compressor tested was found to be virtually constant over a wide range of permitted suction valve lift. Loss of volumetric efficiency due to excessive valve throttling did not occur until the mean gas velocities through the valve exceeded 230 ft/sec. for air and 130 ft/sec. for Freon 12.

7. The effect on actual volumetric efficiency of a wide range of suction valve spring stiffness was too small to be measured.

8. The difference between the theoretical indicated and actual volumetric efficiency could be as much as 30%. The present work together with recent measurements from the same compressor indicated that less than half this loss was made up of the sum of suction valve throttling, gas leakage past the piston and heat transfer effects (with superheated vapour). Most of the loss was considered to be due to the effect of the discharge valve and to blow-by during suction valve closure. The loss at the discharge valve was considered to be caused by

the high spring loading permitted there on account of the common, but erroneous, assumption that volumetric efficiency is little affected by this valve. The excessive suction valve blow-by at low compressor speeds was caused by undue consideration in design to the dynamics of the valve at opening, which were not important.

9. The estimation of compressor performance with a refrigerant from tests using air was considered to be fundamentally unsound.

10. A ported compressor of the same dimensions was inferior in all material aspects of performance to the valved machine. (Appendix II).

11. Failure of the valve reed due to fatigue was unlikely. High stresses were due to impact, the theoretical values probably being much greater than those actually occurring. Collapse of the valve reed into the port under pressure loading was considered to be the most likely cause of failure in the compressor examined. (Appendix V).

10. 2 <u>Automatic Valves for Water Pumps</u>.

1. Although much valuable work has been carried out, mainly in Germany, to study the water pump valve and its effect on performance, virtually no work has been carried out during the last 30 years - a period in which pump speeds have greatly increased and information on valve behaviour more than ever required.

2. Due to these higher pump speeds older methods of valve indication are obsolete. The attempt to develop a valve movement indicator on the capacity principle was not successful. An indicator working on the inductive principle was successfully

developed and both valves of a small ram pump were indicated simultaneously during a wide range of operating conditions. Tests indicated that the pressence of a carefully designed pick-up coil near the valve did not affect pump performance.

3. Under certain reasonable operating conditions the valve displacement diagram conformed closely to a displaced sine wave. Under other conditions the valve movement was far removed from this wavepath. At certain critical conditions, a great change in the valve displacement curves occurred in a very few pump revolutions. Under certain conditions also, it was possible to have the discharge valve lifting during the suction stroke. The presence of excessive air in the suction line resulted in violent oscillation of the suction valve.

4. The valve lifts calculated by the formula due to Berg and Krauss were almost 100% in excess of those actually obtained.

5. A range of valve weights of ratio 1 to 5 showed that the valve lift was virtually independent of valve mass but not of valve shape.

6. The form of the curves of valve discharge coefficient plotted to x was similar to that of Berg, but revealed a size effect not previously observed since all previous researches have been carried out on much larger pumps and valves.

7. Close agreement was obtained with the later values published by Berg for the noise limit λ. The only reservation was that some cases of quiet valve operation were obtained at values of λ beyond the limit.

8. Only on one occasion in twelve tests did the dissimilar pairing of valves and springs result in pump performance superior to that when using paired valved and springs.

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(c) The Minimum Pressure inside a Reciprocating Pump.
B.H.R.A. Technical Note No. 427.

(d) The Flow through Disc and Metre Valves at Low Lift.
B.H.R.A. Technical Note No. 428.

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APPENDIX II.

COMPARISON OF VALVED AND PORTED TYPE GAS COMPRESSOR.

A comparison of performance was made between the valved compressor when pumping air and one identical, within the bounds of manufacturing limits, except that the suction valve was blanked off and the air admitted through a port 0.0554 in. times 0.69 in. at the end of the "suction" stroke. This provided a flow area 4 times that of the minimum flow area of the valved machine, but was available for only about 1/13 time depending, of course, on the compressor speed and pressure ratio in use.

From the test results shown in Fig. 47 it was apparent that the performance of the ported type (P) compressor did not, in any important respect, match that of the valved (V) machine. The actual volumetric efficiency was consistently about 10% lower in the ported machine over a wide range of pressure ratios. The actual volumetric efficiency tended to a maximum at a particular speed but the excessive losses in the ported machine caused a rapid deterioration in performance with 20% difference between the two at 1750 r.p.m. It may be noted that the indicator diagram was of little service in a study of actual volumetric efficiency. The excessive wire drawing of the ported machine was shown by Fig. 47D. These pressure-volume diagrams were computed from pressure-time oscillograms obtained with a pressure sensitive element with integral diaphragm 0.5 in. diameter, 0.011 in. thick so that great accuracy of the suction loop could not be claimed. The higher power consumption of the ported machine is shown in Fig. 47C.

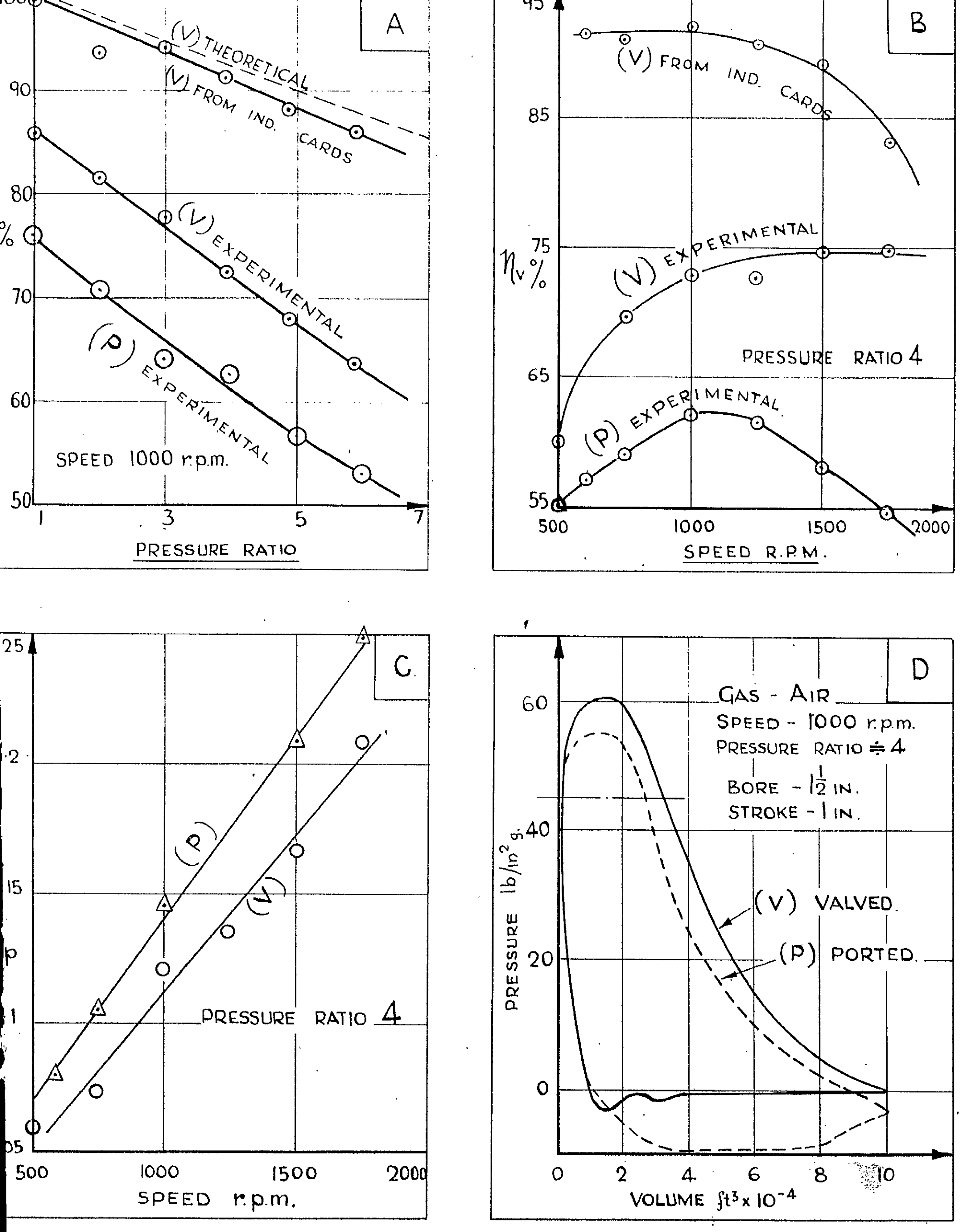


FIG. No. 47 - COMPARISON OF VALVED AND PORTED TYPE COMPRESSORS

APPENDIX III.

INVESTIGATION OF GAS COMPRESSOR VALVE LOSSES WITH STEADY GAS FLOW.

An investigation of the losses due to an automatic valve under steady flow was made, (a) to appreciate the nature of the losses occuring in the valve and the factors affecting them and (b) to provide the coefficients necessary for any mathematical analysis of the valve movement and loss of performance due to the valve in operation in a compressor. From previous findings relating to poppet valves (Stanitz, Lucia and Masselle (29)) and two-stroke engine ports the omission of dynamic and time factors by using steady gas flow does not unduly limit the application of the results to operating conditions.

In conjunction with its function as a seal a valve assembly operates as a flow passage with an inherent throttling effect. Hence it is useful to consider the valve along the lines of convergent-divergent nozzle theory: the radial divergent portion of the passage will serve to recover some of the throat velocity energy as pressure energy.

To compare the test results using a compressible fluid in a reed valve with those of Fuchs, Hoffmann and Schuler (6) using water with ring valves and, to find whether from these present results with air, the valve pressure drop with other fluids (e.g. refrigerants) could be estimated, some basis of equivalence between valve forms and between fluid properties was desirable.

It was presumed that the volume rate of flow, Q, is a function of the variables - parameter of length D, mass density of the gas ρ, viscosity μ, compressibility K, gravitational acceleration g, and pressure difference across the valve Δp_v.

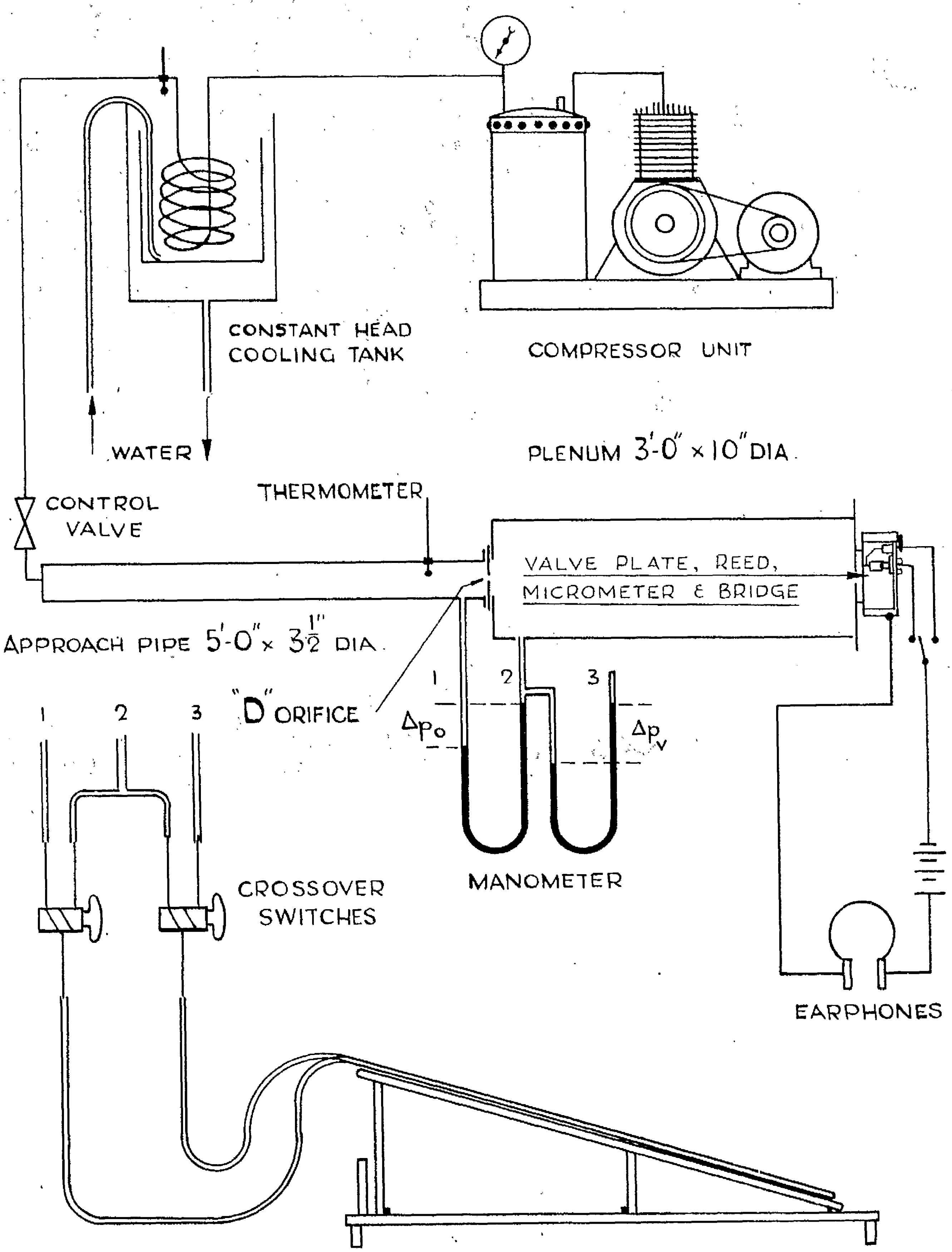
Hence by the method of indices, it was deduced that

$$Q = \sqrt{g\,\Delta p_V}\,D^2 f(Re.M.Fr.)$$

The terms within the functional bracket are representative of a series of which the summation is equivalent to a coefficient of discharge. The dimensionless quantities influence this discharge coefficient and it may show critical values. If these dimensionless quantities all have appreciable effect, then it is not possible to make comparison between different compressible fluids, since it is a physical impossibility to provide all the similarities together.

The Test Circuit. The general layout (Fig. 48) shows the plant used to examine the valve reeds, when mounted on a standard valve plate. The air supply was sufficient to provide a velocity through the valve equivalent to the mean air velocity through the valve in operation, at any valve lift at compressor speeds up to 2000 r.p.m. The mass flow rate of air was measured by a calibrated orifice. The pressure drop across the valve assembly was measured by an inclined water manometer, which was calibrated against an Ascaniameter.

Precision equipment was necessary to measure and control the valve lift accurately. This consisted of a small gantry carrying a micrometer head mounted on a slide. The slide held against the track of the gantry by a spring loaded trolley and permitted a complete traverse of the valve reed. This horizontal travel was controlled by a rod groved at 0.1 in. intervals and accurately located by a spring loaded knife edge. The lift at the valve reed tip was limited by the adjustment of a "stop" arm rigidly clamped to the bridge frame. The "make and break"



CIRCUIT FOR STEADY GAS FLOW TESTS.

FIG. No. 48

electrical contact between the pointed extension to the micrometer head and the valve reed enabled the valve displacement at the tip to be measured to about 0.0002 in.

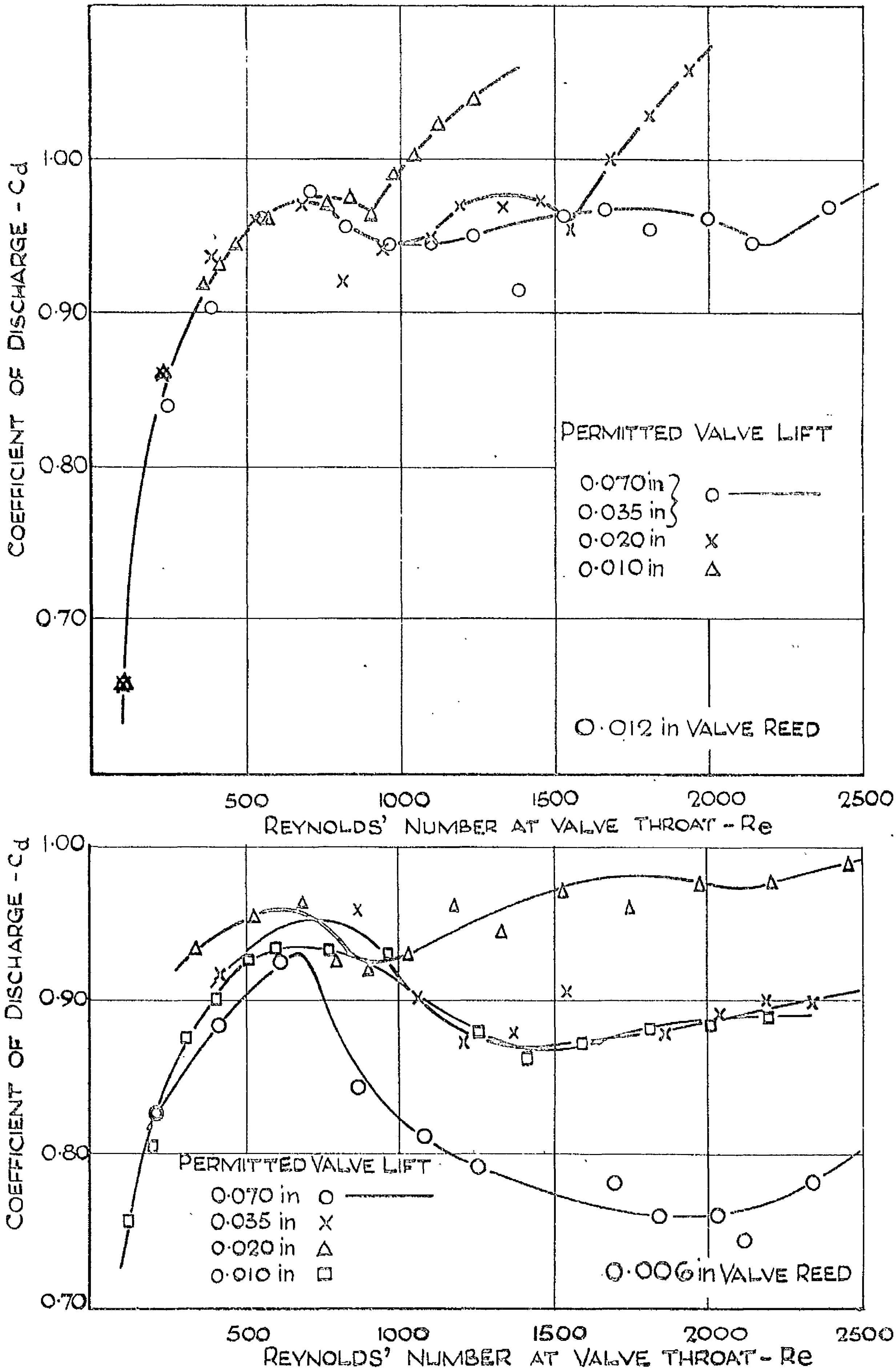
The air mass flow readings were taken at intervals of pressure heads across the valve. Until the reed contacted the limit stop only the lift over the centre of the port was measured; after contact readings were also taken at 0.2 in. on either side of the centre to take account of the slight distortion of the reed now acting as a beam restricted at the ends.

Observations were made as the head over the valve increased and again as it decreased with reduction of the air mass flow. Contrary to the results obtained in tests by Shrenk (6) with water pump valves there was no measurable difference between the readings.

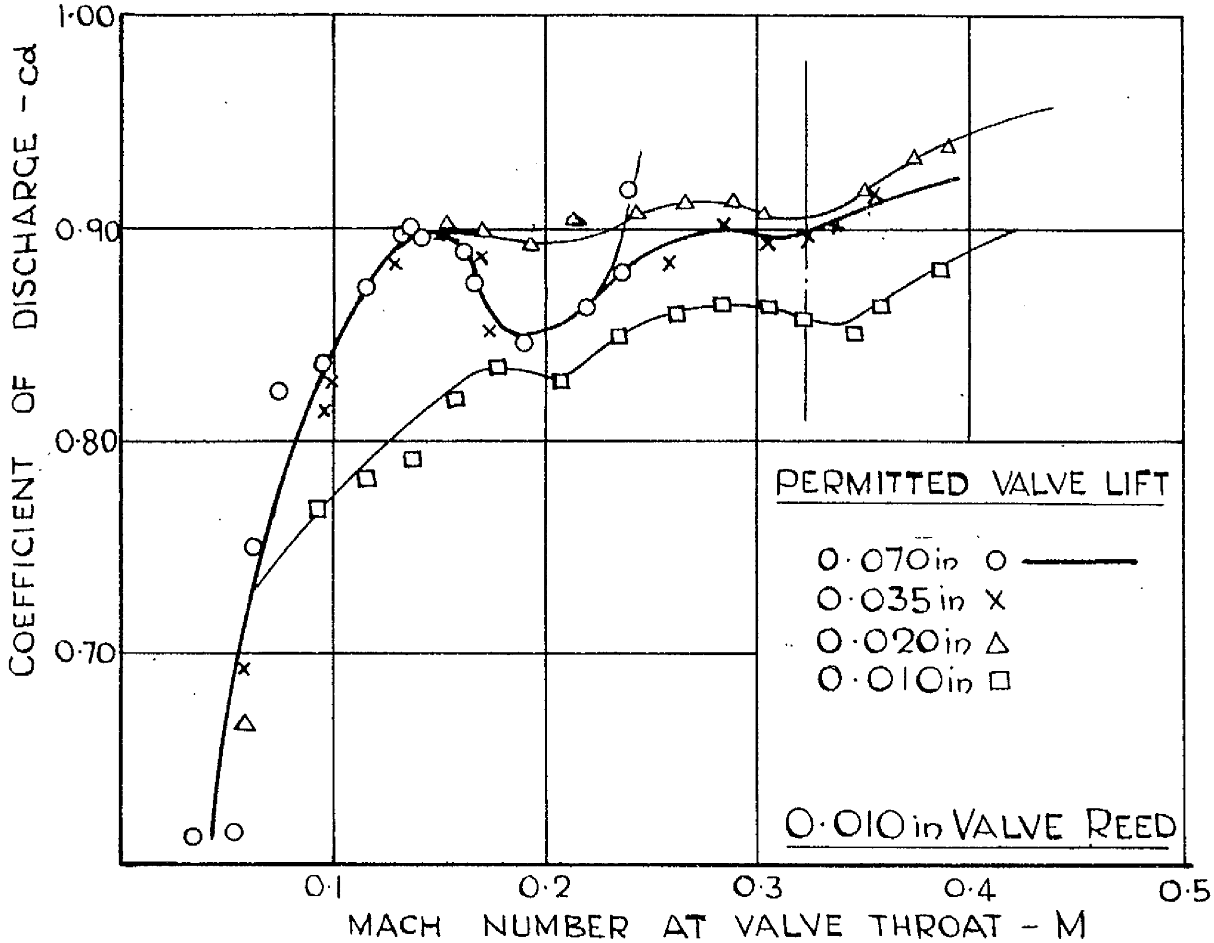
One measure of the efficiency of the valve process, which has been used in previous work, is "Valve Resistance Coefficient" Z defined as $\frac{\text{pressure head}}{\text{kinetic head}}$, and, in general, this should be as low as possible. Since the pressure drop across the valve is small it may be shown that $Z \doteqdot \frac{1}{(c_d)^2}$. The present results are graphed to show c_d. The coefficient Z has only been computed for comparison with the results of Fuchs, Hoffmann and Schuler. Since it is not practicable to obtain a pressure plot through the valve, the calculations have been based on the overall pressure drop across the valve (Δp_v); it will be possible, therefore, to have values of $Z < 1$, just as c_d may be greater than unity in a convergent-divergent nozzle when the overall pressure ratio is greater than the critical. The coefficient of discharge was based on the minimum flow area, which, within the range of valve lifts, was always the annular area at entry to the radial passage. In computing the Reynolds' Number the characteristic length D was taken as $\frac{\text{flow cross section}}{\text{wetted perimeter}}$. The wetted perimeter is practically independent of the valve lift (h) and the flow corss section is (wetted perimeter) times (valve lift/2.) Hence D becomes h/2.

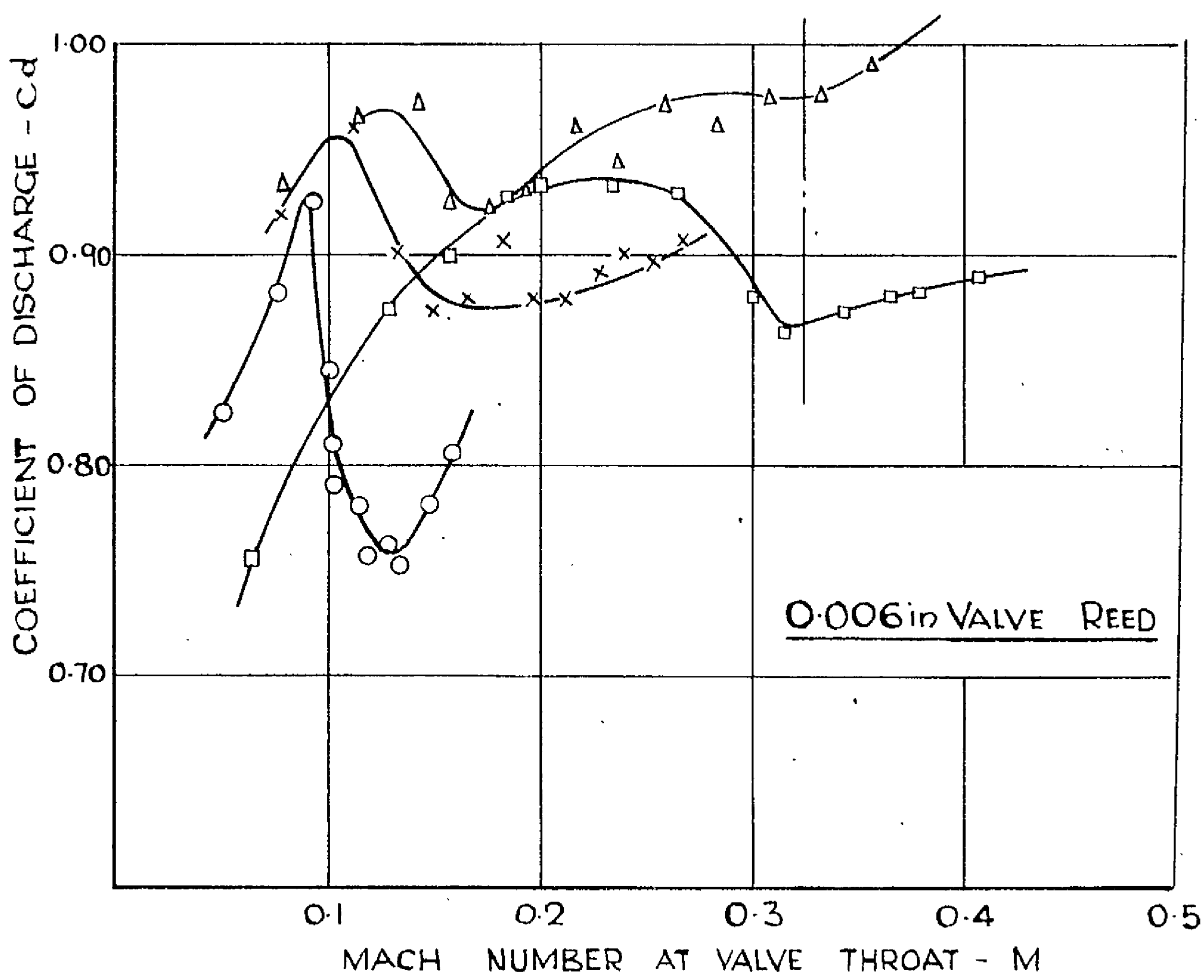
<u>Discussion of Results</u>. Considering the unsuitable entry, the abrupt formation of a throat by the valve reed, the change of direction and the rapid divergence in the diffuser the coefficient of discharge, except at very small gas flows, was surprisingly high. This was partly due to basing the calculations on the throat area; the value of 0.6 for c_d used by Costagliola is close to that obtained here if defined in the same way. At a critical value of Re about 700 the rapid increase in c_d during increasing velocity under laminar flow conditions abruptly ceased. The tortuous flow path would be the cause of this early breakdown of laminar flow. A transition stage between laminar and turbulent flow resulted in a decrease in c_d and at a less clearly defined value of Re, between 900 and 1700, turbulent flow was established, after which c_d remained fairly constant, as would be expected. A further small change in c_d occurred consistently at a Mach number 0.32, (Fig. 50) a lower value than would be anticipated for Mach number effects to be apparent.

The plot of Zand Δp_v to a base of unrestricted valve lift (Fig. 51) showed the same characteristics as were obtained by Fuchs, Hoffmann and Schuler testing annular port valves using water. One notable difference was that in no case did they obtain values of Z less than unity; this was possible here at

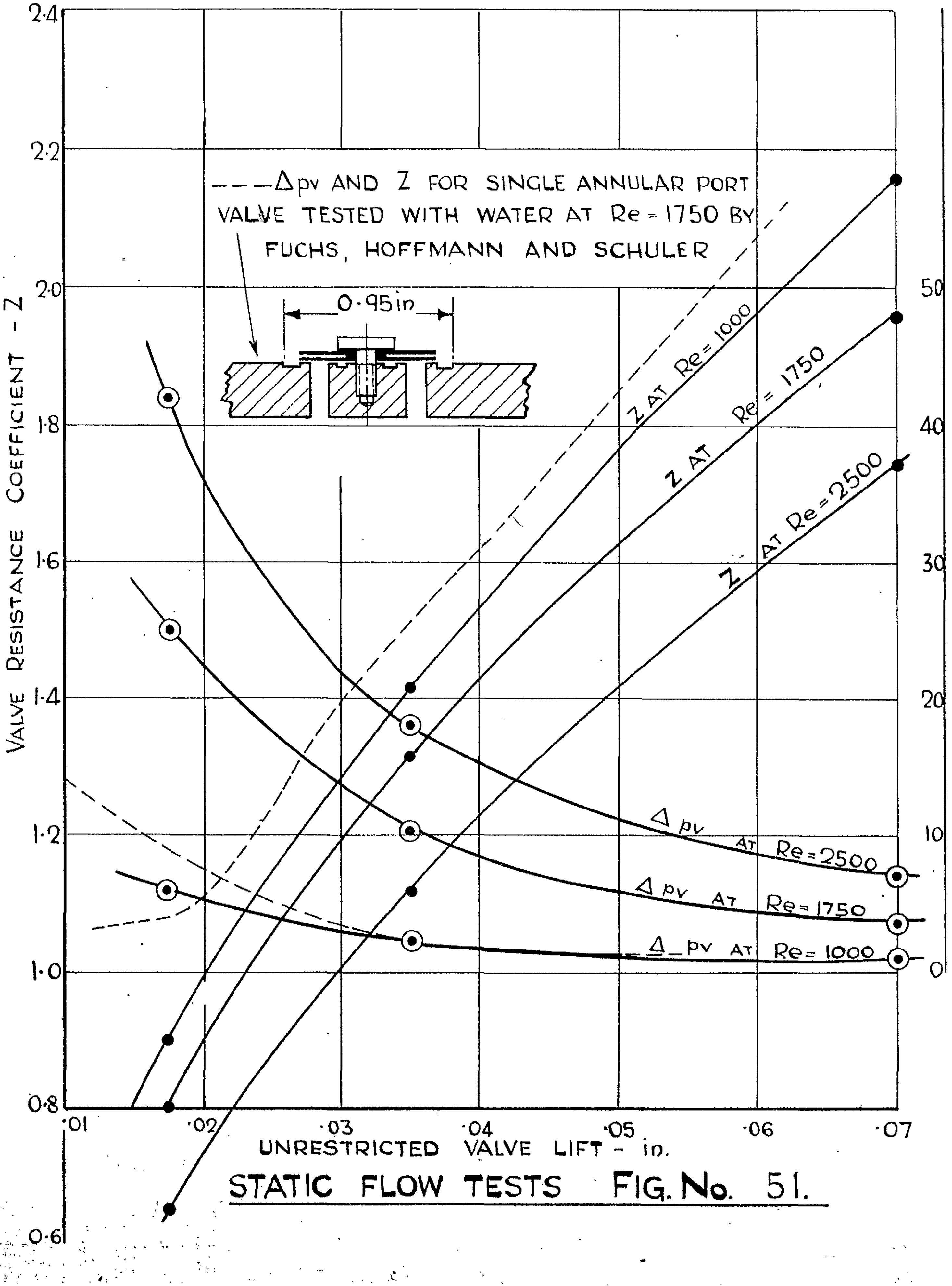


STATIC FLOW TEST RESULTS FIG. No. 49.

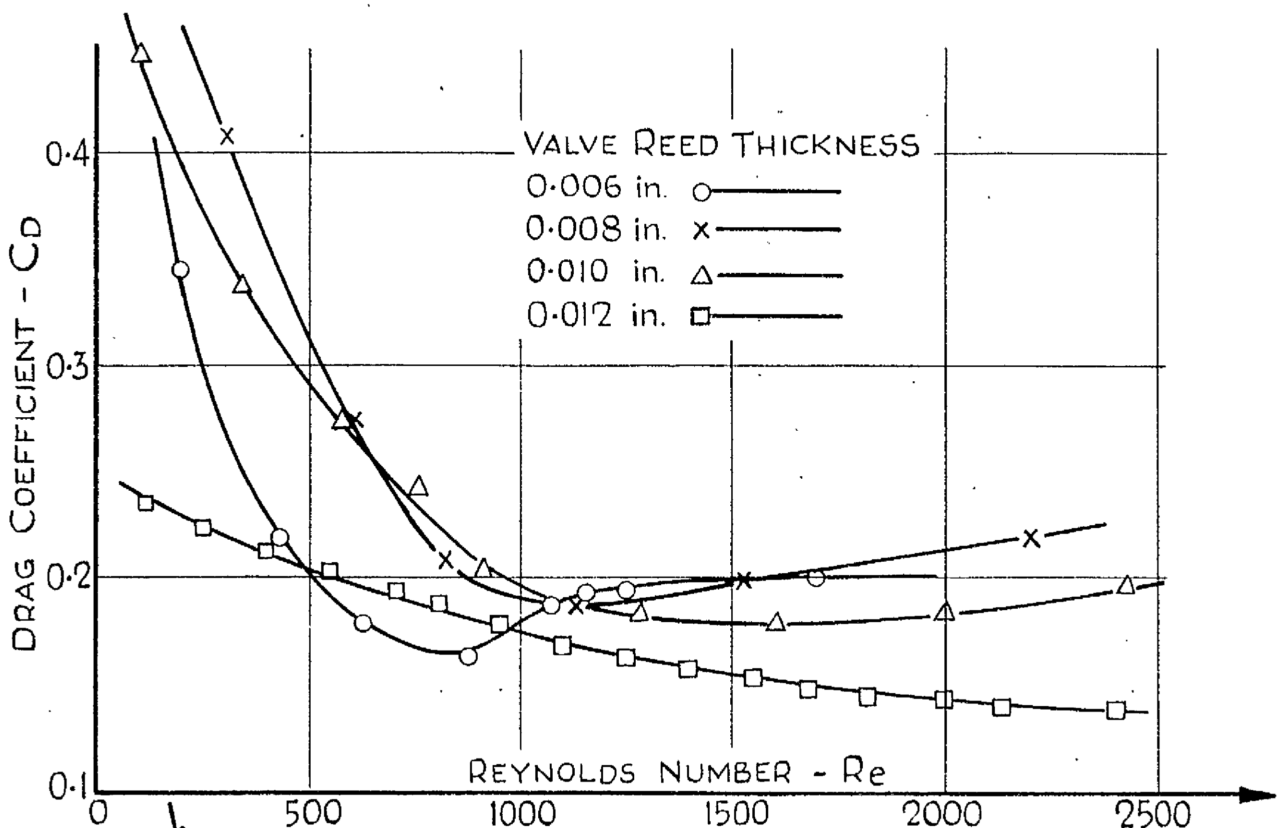


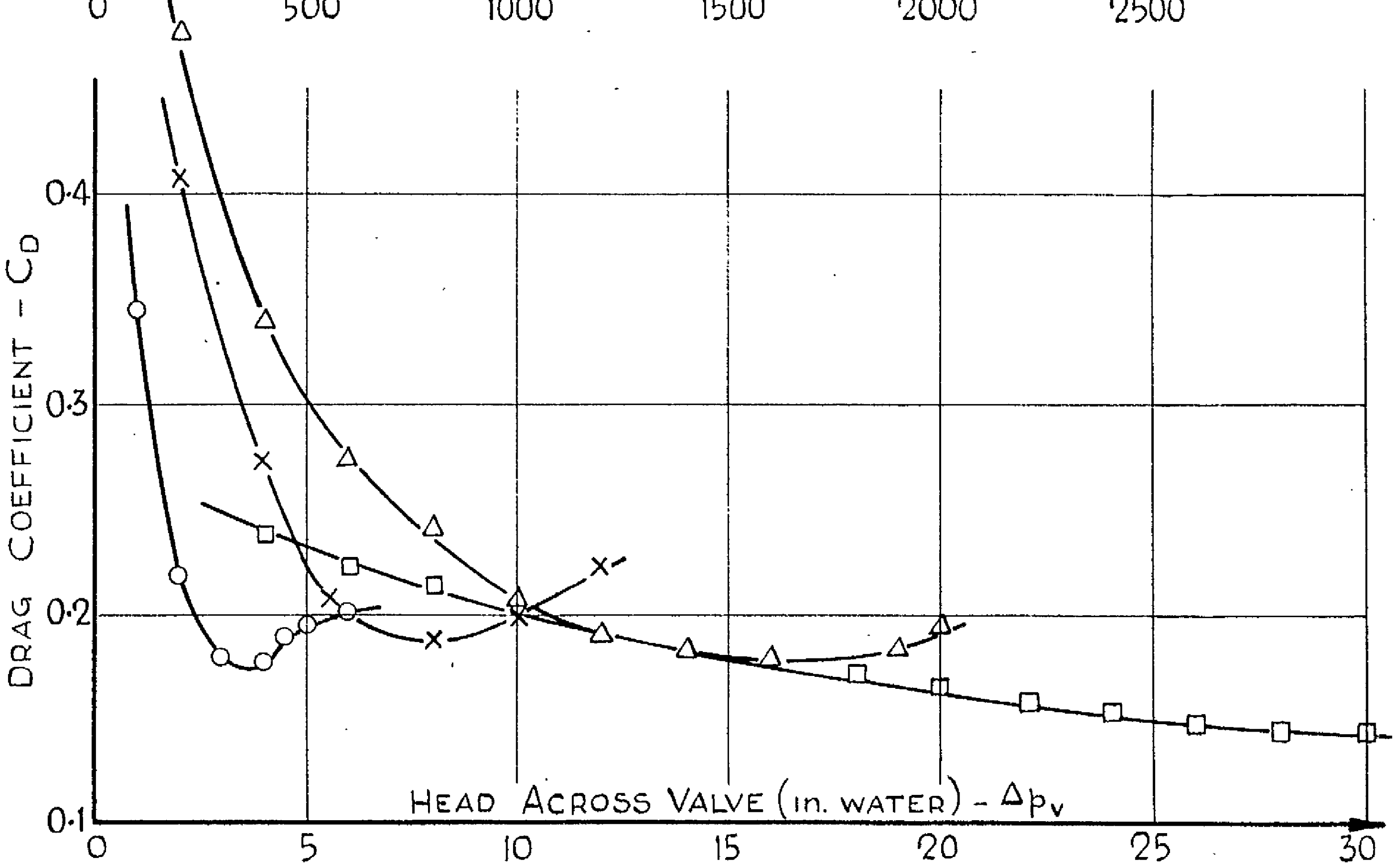


STATIC FLOW TEST RESULTS FIG. Nº 50.



STATIC FLOW TESTS FIG. No. 51.





Experimental Values of Valve Pressure Drag Coefficient

Fig. Nº 52.

low lifts due to the appreciable diffuser effect of the radial flow passage. An attractive valve would have low values of both Z and Δp_v since both affect compressor performance, Z by its relationship with c_d and Δp_v by its effect on both volumetric and pumping efficiencies. Since Z increased as Δp_v decreased there is the suggestion of an optimum valve lift, apart from any consideration of valve dynamics.

The pressure drag coefficient, C_D decreased (Fig. 52) with increasing gas flow until turbulent flow was established. Thereafter this coefficient was reasonably constant at 0.2. The coefficients must be assumed constant in the development of any practicable mathematical statement of the valve behaviour. Hence the coefficient of discharge was taken in Chapter 3 as 0.9 and the pressure drag coefficient as 0.2.

APPENDIX IV.

CALIBRATION OF EXPERIMENTAL APPARATUS.

(a) Valve Reed Displacement Pick-up Unit. The capacitance, C, of the pick-up unit is given by $C = \frac{K_p A}{4\pi h}$ where K_p is the permittivity, A the plate area and h the distance between the plates, or dielectric thickness. $\frac{dC}{dh} = \frac{-K_p A}{4\pi h^2}$ indicates that the sensitivity falls as the distance between the plates increases. Thus a linear capacitance/displacement relationship can only be approached at the expense of sensitivity. The clearance to prevent earthing of the insulated plate and the valve reed in the closed position was set at 0.010 in. providing reasonably linear capacity/displacement except close to valve closing and opening (Fig. 53). Due to this bedding of the insulated plate in the valve plate and since the ratio of electrode periphery to gap was a relatively low value of 350, fringe effects were not necessarily negligible, so the calibration was made experimentally. The capacitance of the actual unit was so small that a geometrically similar model four times larger was constructed and the calibration curve obtained using a Marconi Circuit Magnification Meter. The capacitance of the actual unit was therefore 1/16 of the measured values. The corresponding theoretical curve assumed K_p for a space as 8.855 Farads/metre and the corresponding figures for air and Freon 12 both approximate closely to this.

(b) Differential Pressure Pick-up Unit. Due to the delicate nature of this unit, and to the drift possible with electronic recording apparatus, the arrangement was calibrated in situ, with the gas compressor stopped, before and after each test. The compressor inlet valve was closed, the suction valve lift reduced to zero by the external valve lift device, and the small needle

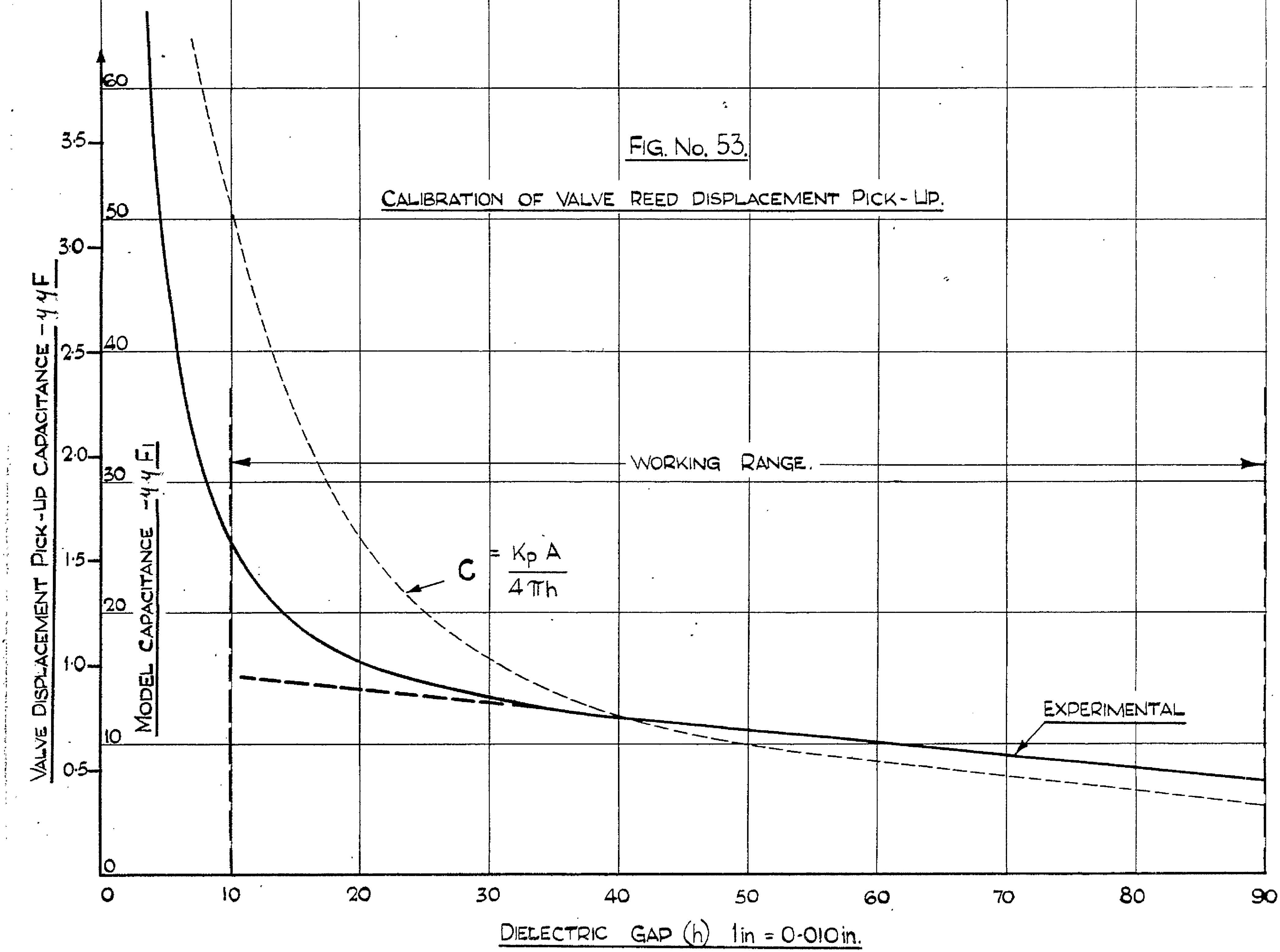


FIG. No. 53.

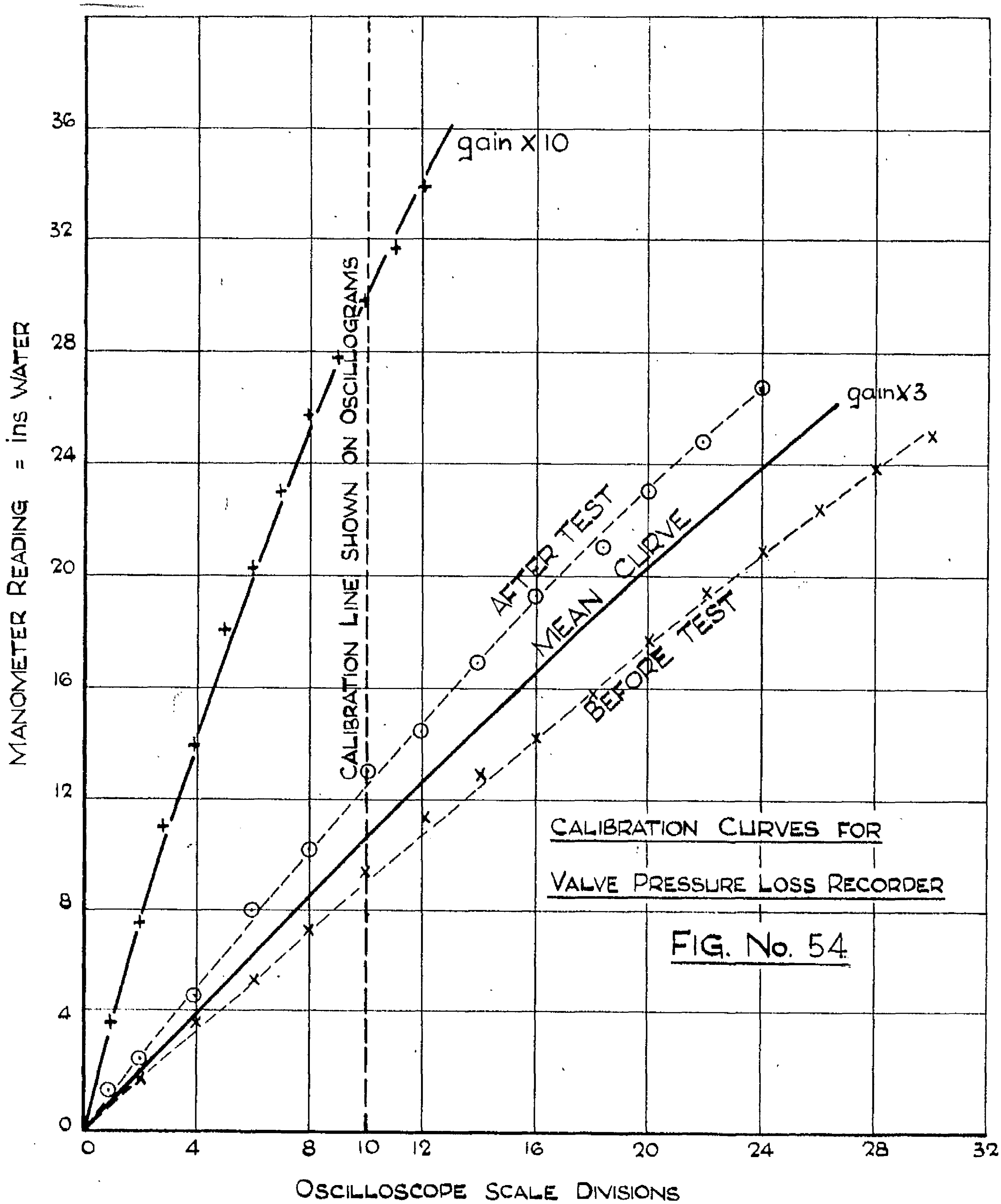
CALIBRATION OF VALVE REED DISPLACEMENT PICK-UP.

stop valve B (Fig. 13) opened. The pressure tappings A and B, connected to the suction head and cylinder respectively, led to a vertical water manometer. Compressed air was throttled and supplied to the suction head through C, thus displacing the diaphragm of the pick-up. The Y oscilloscope deflections were recorded against the corresponding manometer reading for both ascending and descending pressures. Before restarting the compressor calibration lines were imposed on the film by adjusting the Y shift on the graduated oscilloscope screen (Fig. 21). The zero position of the oscilloscope trace was checked again at the conclusion of the test and the calibration repeated. The gain x 10 calibration, if required, was obtained from the mean gain x 3. A specimen calibration curve is shown in Fig. 54.

The reduction in diaphragm thickness and increase in diameter progressively made during development to increase sensitivity required that a check be made that the natural frequency of the diaphragm was sufficiently high. The fundamental frequency of a circular plate clamped at the boundary is given by Timoshenko.* The calculated fundamental frequency was thus 8,800 cycles/sec. Account may be taken of the mass of fluid in which the plate vibrates, (Lamb*). This reduced the calculated frequency by 1% in air and 8% in Freon 12, but this reduction would be offset by an increase due to the stretching of the middle surface of the plate. The natural frequency of the assembled unit was measured by electromagnetic oscillator and

* Timoshenko, S — Vibration Problems in Engineering, p.314.

* Lamb, H — Proc. Roy. Socy. Vol. 98 (1921), p.205.



CALIBRATION CURVES FOR

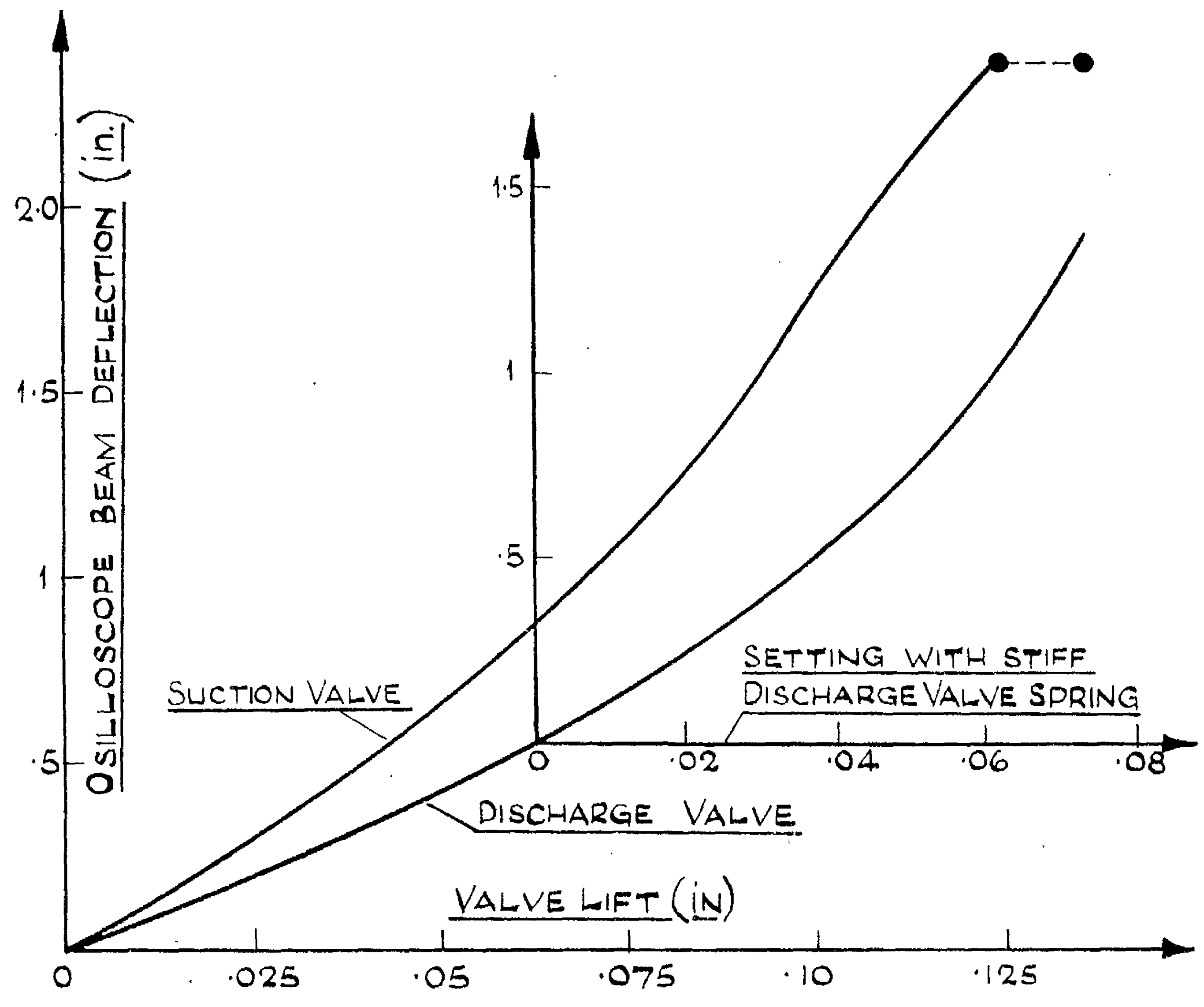
VALVE PRESSURE LOSS RECORDER

FIG. No. 54

found to be considerably less than the calculated value: the discrepancy was probably partly due to the imperfect clamping of the diaphragm and the limited rigidity of the valve plate.

(c) Water Pump Valve Displacement Pick-up Unit. For convenient calibration of the suction valve pick-up unit a 6 BA screw with dial attachment was fitted as shown in Fig. 34. The calibration curve was obtained during pump operation by changing the permitted valve lift and noting the corresponding beam deflection. To calibrate the discharge pick-up unit the valve lift was restricted by inserting five rings of varying thickness between the valve disc and stop. With stiff valve springs the beam deflection for the discharge valve was relatively small. For this case the coil was moved nearer the valve to obtain increased sensitivity and calibration was carried out as before (Fig. 55).

In later tests more precise determination of suction valve lift was obtained by a precision device which included an electrical make and break arrangement similar to that used with the gas compressor valves. The micrometer head was clamped in a bridge and a probe, which passed through a gland into the valve chamber, attached to it. When the probe was screwed down by a large nut until contact was just made with the valve at rest, a zero reading was provided. Screwing back the large nut provided a permitted valve lift for calibration purposes, or the maximum lift attained during operation could be measured.



CALIBRATION CURVES FOR PICK-UP UNITS

FIG. No. 55.

APPENDIX V.

A NOTE ON GAS COMPRESSOR VALVE FAILURE.

The causes of failure are usually difficult to diagnose. The complexity of this important problem, together with the fact that those with most experience of it have little opportunity for investigation, perhaps accounts for the dearth of published knowledge. A paper by Hoerbiger (12) has been the only attempt to explain failures and not merely attribute them to faulty material, finish or heat treatment. Inter-crystalline stressing due to local variation in temperature within the valve during the cycle, was said to appear as "an infectious disease" in the metal, which might even spread to the backing plate. It is surprising that there was no subsequent discussion to this paper to the 7th International Congress of Refrigeration. It is possible to support the theory by evidence not advanced by Hoerbiger; for example, there appears to be an increased tendency to failure when pumping wet vapour, when thermal shock due to local evaporation is probably intensified. On the other hand, the remedy which is most often effective is a reduction in lift which increases these local temperature variations. Our examination of the surfaces of several ring valves which had failed in a large ammonia compressor did not show any pitting, and the D.P. hardness of the material remained uniform close to the fracture.

Because metal becomes brittle at low temperature it is often assumed that there will be an increased tendency to fatigue failure where there is impact loading. It is of interest to note therefore, that recent work by Kudravtsav (72) showed that fatigue strength materially increased as brittleness increased at low temperatures.

In most forms of automatic valves for small gas compressors, where for simplicity the valve spring is provided by the elasticity of the valve reed material, the valve is subject to stressing due to (i) flexing, (ii) punching into the valve port, (iii) impact on (a) the stop, (b) the valve seat.

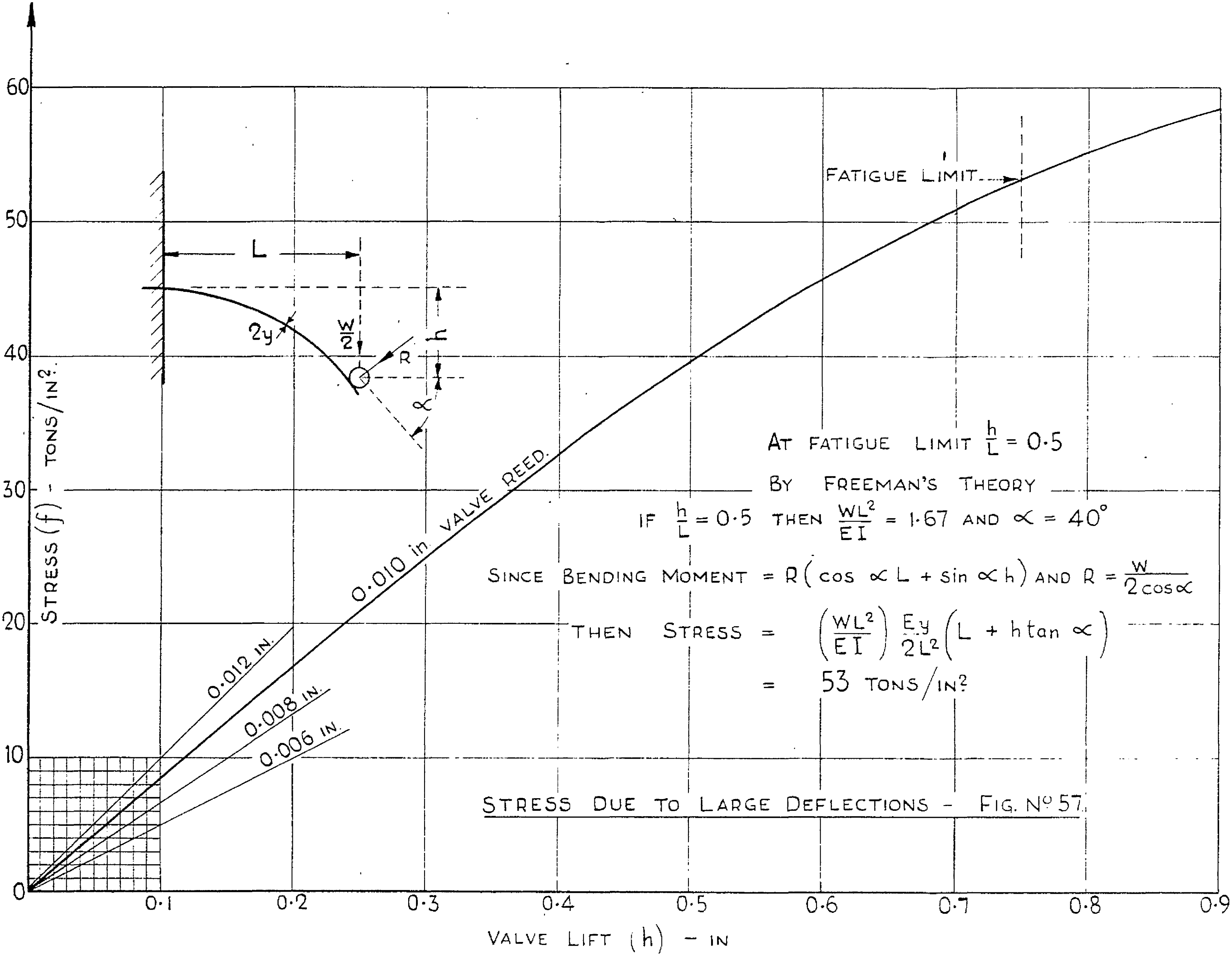
The maximum stress due to lift will depend on the form of the valve reed. The brittle lacquer technique was used by Hétényi and Young (70) to obtain peak stress values, stress concentration factors and the relative strength of different valve reed designs. No quantative information was given and the authors, experts in this technique, stated that this was their most difficult application of it.

To determine the fatigue limit of the valve steel, ten valves were flexed by point loading by a suitably modified Wöhler fatigue testing machine. The results obtained are given in Fig. 56. To obtain the stress in the material to cause fracture the simple bending theory was not adequate, since the deflection was 0.75 in. over a specimen length 1.5 in. A recent mathematical theory of beam deflection by Freeman (69) was modified to fit this case exactly, and the fatigue stress was calculated as 53 Tons/in.^{2} (Fig. 57). Since the valve lifts used (0-0.08 in.) were so much less there was little chance of failure due to primary flexing. When the valve motion was directly observed by stroboscope (Chapter 4. 5) it was noted that only primary flexing of the reed occured. No secondary flexing was evident even at 3000 r.p.m., nor was it expected since at this high compressor speed the valve was still operating below its natural frequency.

SPEED OF REVERSAL 1500/min. VALVE REED THICKNESS 0·010 in.

VALVE Nº	VALVE TIP DISPLACEMENT (in.)	$\frac{\text{RESTRICTED AREA}}{\text{PORT AREA}}$	STRESS TON/in²	$\frac{\text{STRESS}}{\text{FATIGUE STRENGTH}}$	MILLION REVERSALS	REMARKS
1	0·035	0·4	3	1/18	16	NO FRACTURE
2	0·172	2	15	1/3·5	16	NO FRACTURE
3	0·344	4	29	1/1·8	16	NO FRACTURE
4	0·75	8·7	53	1	'a' 2 'b' 6·5	'a' - FRACTURE ONE LEG 'b' - DISCONTINUED
5	0·75	8·7	53	1	16·8	FRACTURE ONE LEG.
6	0·75	8·7	53	1	18·7	FRACTURE ONE LEG.
7	0·75	8·7	53	1	87·3	SIMULTANEOUS FRACTURE BOTH LEGS
8	0·85	9·9	57	1/93	0·9	SIMULTANEOUS FRACTURE BOTH LEGS
9	0·85	9·9	57	1/93	5·2	SIMULTANEOUS FRACTURE BOTH LEGS
10	0·8	9·3	55	1/96	30	UNBROKEN - TEST DISCONTINUED

FATIGUE TEST RESULTS - FIG. Nº 56.



STRESS DUE TO LARGE DEFLECTIONS - FIG. No 57.

The stress due to punching of the valve into the port during compression and delivery may be estimated by assuming the valve reed to be a circular plate simply supported at the edge and uniformly loaded. If the friction at the valve seat was very great so that the valve could be considered clamped the stress would be reduced 40%; inertia effects would tend to increase the stress.

The more severe impact stress may be either at the stop or the seat depending on the spring stiffness, valve lift, pressure ratio and compressor speed. If the valve spring is stiff the valve may not reach the stop. This impact increases with compressor speed, pressure ratio, and reduction in valve weight. The principal cause of a high terminal velocity at impact on the seat will probably be spring stiffness, since the piston velocity is low over B.D.C.

The stresses due to these effects have been estimated for this compressor, for the worst combination of conditions and also for the best.

Type of Stressing.		Conditions for Large Stress.	Max. Stress. Ton/in^2	Min. Stress. Ton/in.^{2}
Flexing		Large Lift, Heavy Valve Reed	8.5	1.5
Pressure Load Bending.		Light Valve Reed Large pressure Ratio	36 60	6
Impact	At Stop	Light Valve Reed Large Pressure Ratio High Compressor Speed	30	0
	At Seat	Large Valve Lift Heavy Valve Reed High Compressor Speed	5	1.5

It was considered that the estimate of stress due to impact on the stop was too high. By calculation this was greatest with the light valve. However, due to its flexibility it readily bent after contact with the stop at its tip. The damping due to the presence of oil would also be most effective with the light valve.

It is apparent that failure was most likely to occur with the light valve due to pressure load bending. Stresses were small due to flexing, at any practical lift, the effect against which the stop is usually provided.

The combination of best conditions consistent with no deterioration of the compressor performance showed that failure was unlikely.

A compressor, pumping air, in a water cooled circuit, closed to conserve the oil, was run at 2000 r.p.m. at a high pressure ratio with a valve lift of 0.105 in. for over 300 hours. Valve failure did not occur. The valve surface was examined at intervals by Talysurf recorder. While the usual coppering of the valve reed seat occurred, no measurable indentation due to impact was observed.

The failures which occur in general are probably due to sufficiently diverse reasons that no panacea can be offered. The quantative analysis of stresses in the valves of a particular machine, consideration of corrosion, thermal shock etc., may point to the effect causing the failure. The complexity and importance of the matter warrant a major investigation, if there is access to compressors where valve failures have occurred.

ACKNOWLEDGEMENTS.

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