THE EFFECT OF CLEARANCE SPACES IN AIR COMPRESSOR CYLINDERS.

With Notes on Air Compressors and Air Measurement.

Supported by a published paper entitled "Method of Estimating the Slip in Air Compressors by altering the Clearance Volume."

by

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CONTENTS.

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LIST OF PLATES.

INTRODUCTION

AND

SCOPE OF INVESTIGATION.

In comparison with the experimental research which has been devoted to the internal combustion engine there is a lack of thorough research into the problems inherent in air compressors, with the result that the problems met with in the study of air compression cover a wide field. The main purport of this thesis is to investi:gate the beneficial and adverse effects of large and small clearance spaces in air compressor cylinders and to show that altered clearance offers a means of arriv:ing at an approximation to a compressor's air slip.

Since the sizes and types of compressor valves fitted more or less determine the volume of the clear:ance space and since small valves have high, and large valves low air speeds through them, effort usually seeks to achieve a suitable compromise, securing at once a fairly small clearance and a reasonably slow passage of air. The essence is that it is hoped that the tests carried out with altered clearance will prove that the effect of clearance below a certain pressure may be neglected, but that above this pressure clearance has an increasingly adverse influence.

The fact that the theoretical volumetric efficiency is a linear function of the clearance at any delivery pressure suggests a means, if it can be shown that the actual /

actual volumetric efficiency is also of straight line form, of estimating at zero clearance the volumetric efficiency and hence the air slip. (This is discussed in the enclosed publication - "Engineering," 7th August 1936, p.138-139.)

Information has been gathered and recorded regard:ing types of valves, air speeds through them, typical clearance percentages allowed in practice, mean values of the compression exponent and the effect of cooling.

A section has been devoted to reviewing a few of the most common methods of measuring air quantities, particular comparison being drawn between the British and American commercial practices. Several methods are in existence for measuring air, and, although reasonable accuracy can be attained by most of them when correctly applied in a scientific laboratory, it is only within recent years that a standard apparatus has been adopted for commercial tests.

Another section is given to a description of the Compressor and Air Measurement Plant installed in Edinburgh University, for the design of which the author was partly responsible.

GENERAL NOTES ON AIR COMPRESSORS.

Thermodynamic principles show that least work is required when the compression is isothermal. Water jackets round the cylinder, head, valves and covers, cold water sprays, stage compression and finned air cooled cylinders have all been adopted to extract as much as possible of the compression heat.

Cooling is essential, not only from the point of view of saving work, but also to prevent carbonisation of the lubricating oil which might either cause gumming of the valve faces or, at the high temperatures met with, form a dangerous explosive mixture; also high temperatures tend to distort the valve seats.

* Again efficient cooling reduces the air temperat-:ure at the end of the suction stroke.

No matter the nature of the compression, the work done per pound of air is proportional to P_1V_1., i.e., = Constant $\times T_1$ where P_1, T_1 and V_1 are the pressure, temperature and volume of one pound of air at the end of suction.

Thus if T_1 be increased to T_2 by suction heating, the work of compression will be increased in the ratio $\frac{T_2}{T_1}$ e.g., if atmospheric air is increased in temperature from say 15°C. to 50°C., there is an increase of 12% in the work done.

It /

* See E.W.Geyer's translation of Schüle's "Technische Thermodynamik" pp.330 & 331.

It has been found that the gain in economy due to cooling the air during suction is of the same order as that effected by lowering the index of compression.

The value of the compression exponent with water jacket cooling may, with slow and moderate speed machines lie between 1.25 and 1.35, while values around 1.2 may be obtained by using sprays. Occasionally values near 1.2 are found on very slow running machines but it is usual to suspect a leak past the piston or suction valves, as a moderate leak in those two compon:ents reduces the value of the index of compression, allowing the compressor to pass for a perfect isother:mal compressor. In a modern high speed compressor there is very little extraction of heat through the jackets because there is very little time to extract it and because of the relatively thick cylinder walls. In standard work, therefore, it is found that the com:pression line is very near to the adiabatic. The compression exponent is not constant, but is greater at the beginning and less at the end that the mean value of "n", the variation being due to altering temp:erature differences causing different quantities of heat to flow.

Because of the many kinds of compressors made, types of valves differ. They may be either mechanic:ally /

:ally operated or simply spring-loaded. Spring-loaded valves are automatic, pressure differences operating the movement of a thin annular stainless-steel disc between the valve seat and its guard.

In plate valves the valve plate is approximately $\frac{1}{16}$ in. thick and has the advantage of extreme light:ness, especially when considered in proportion to its surface area. As the lift is only about equal to the plate thickness it is obvious that such a valve will not be liable to hammer itself or its seat, and will operate without undue noise. The plate and valve seat have concentric air ports, thus providing the largest possible area for the passage of air.

The maximum lifts generally used in plate valves range from about $\frac{5}{32}$ in. to $\frac{1}{8}$ in. on small compressors, and from about $\frac{1}{16}$ in. to $\frac{5}{32}$ in. on large compressors, i.e., compressors of over 100 cubic feet per minute capacity. If too large a lift is allowed there is a danger of breaking the valve plates, particularly in quick revolution machines.

For quite small machines valves made in the form of a thimble are sometimes used, and are known as Thimble Valves. From the point of view of air tight:ness they function better than plate valves, but at high /

high speeds they are hardly so serviceable,as the plate valve is so much lighter and preferable, therefore, from the point of view of inertia and consequent wear. In high pressure machines, such as the four-stage com:pressors capable of maintaining a pressure of 3,500 lb. per square inch required by the Admiralty for charging torpedoes, the valves in the first and second stages and sometimes in the third stage are plate valves, but the valves in the final stage, in view of the in:creased importance of air tightness, are of the thimble type.

Modern high speed compressors now run at about half the speed of high speed internal combustion engines, the increased speed entailing an increase in air veloc:ities and a decrease in valve lifts, the limit being fixed by valve construction, since decrease in the valve lifts requires the fitting of larger valves which seriously impare the volumetric efficiency.

Commercial considerations often induce makers to depart from proportions which their experience and better judgment would lead them to adopt. Guided by purely theoretical considerations, a machine at a normal pressure of 100 lb. per square inch would have a speed through the suction and delivery valves of approximately 6000 /

6000 feet per minute. When designing reciprocating machines for very low pressures and large volumes, this velocity would be reduced even more, to some 4000 feet per minute. In highly competitive machines higher spring pressures and higher velocities up to 20,000 feet per minute would be adopted.

The provision of generously proportioned valves with low air speeds always involves low volumetric efficiencies, as the clearance volume is considerably increased. If a designer cramps the air spaces and in so doing increases the air velocities through the valves or makes the passage-ways more tortuous, the air losses, from the point of view of power expended, may more than compensate for the clearance saving. On the other hand a low air velocity, other things being equal and within limits, will be found to give a lower horse-power per cubic foot of free air actually delivered. Usually the designer seeks to achieve a suitable compromise, securing at once a fair clearance volume and a reasonably slow passage of air.

In compressor construction it is impossible to avoid clearance spaces round the valves and between the pistons and cylinder covers. These clearance spaces may be as low as . 5 - .6 % in large and 4 - 5 % in small /

PLATE NO.1.

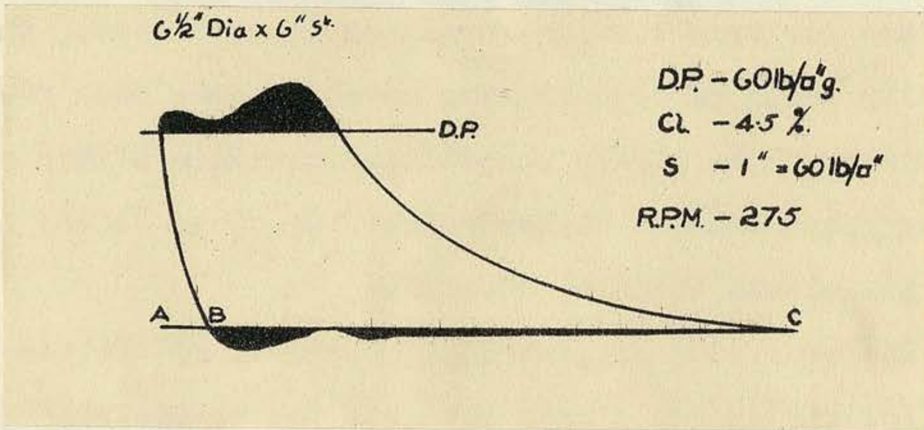
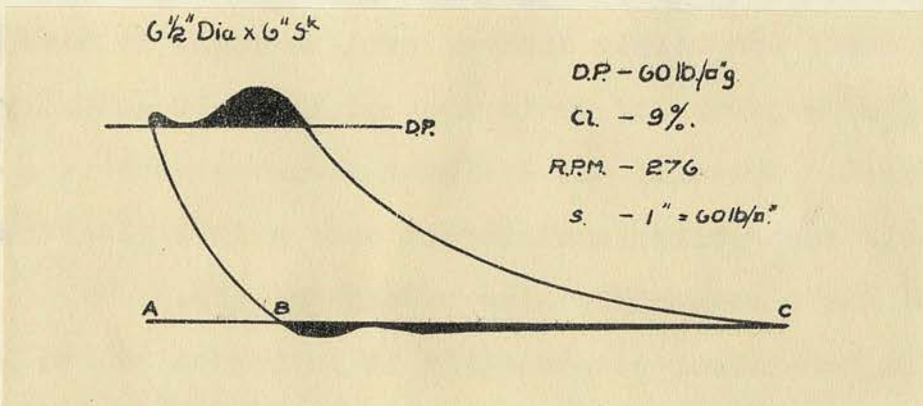


PLATE NO.2.



small compressors.

The modern two-stage vertical compressor has its valves arranged in kidney pockets by the side of the cylinder and in such a design the clearance space is approximately 10%. Where the valves are arranged directly on the heads of the cylinders, the clearance is reduced, with the usual stroke bore ratios adopted in quick revolution two-stage machines, to about 5%. Valves with about 3% clearance are found in long stroke slow revolution compressors where the stroke and cyl-:inder bore are almost equal. Some machines used for injection purposes in Diesel Engines have the clearance in the I.P. and H.P. as high as 15% and 30% respective-:ly, so that when throttled down to a less capacity the excessive work thrown on the H.P. will not be aggravated by a greatly increased discharge temperature.

If we compare two indicator cards taken at the same delivery pressure but with different clearances we find that the card with the larger clearance shows a much later opening of the suction valve. (See Plates 1 and 2). This gives us an approximate method of ascertaining the volumetric efficiency. An equivalent volume of air represented by the length BC is drawn into the cylinder and the ratio $\frac{BC}{AC}$, where AC is a measure of the stroke volume, is known as the apparent volumetric /

volumetric efficiency. The apparent volumetric efficiency, however, is not a true criterion, as it takes into account neither temperature and pressure differences nor leaks past the valves and piston.

It has been shown that the sum of these losses at 60 lb. per square inch is 8.63% for the compressor from which the indicator cards were obtained. (See enclosed publication). Subtracting this loss from the apparent volumetric efficiency, the table below shows that it is possible to obtain an approximation to the actual volumetric efficiency.

Delivery Pressure (lb./sq.in. g.)	Clearance Volume (% of S.V.)	Apparent Volumetric Efficiency (%)	Slip (%)	Approximate Volumetric Efficiency (%)	Actual Volumetric Efficiency (%)
60	4.52	91.5	8.63	82.87	80.41
60	9.04	80.5	8.63	71.87	69.84

The approximate volumetric efficiency is within 2 - 3 % of the actual volumetric efficiency obtained by air measurement and errors as high as 10% may arise if the apparent efficiency is accepted as a measure of the actual volumetric efficiency. Commercial compressors have no provision for taking indicator cards, so that it would appear that the apparent volumetric efficiency is given too much importance as a test result and has no practical value.

The /

The shaded areas on the cards (see Plates 1 and 2) represent the work expended in opening the suction and discharge valves, the discharge valve loss including the friction loss in the aftercooler and pipe lines. Some of these losses, however, are due to the inaccuracy of the indicator itself and its operating mechanism and this inaccuracy is greater the higher the speed of the machine.

Some makers have reduced the cylinder clearance to a minimum, e.g., on Messrs. Reavell's Quadruplex single-stage compressor, the air enters through ports in the gudgeon pin and corresponding slots in the piston, thus eliminating suction valve air spaces. The machine is rated at 672 cubic feet of free air per minute at 120 lb. per square inch delivery pressure, and consists of four cylinders $12\frac{1}{2}$ in. bore × 8 in. stroke discharging into a common air belt. Test results show high volumetric efficiencies and correspondingly low horse-powers per cubic foot of air discharged.

Delivery Pressure lb./sq.inch g.	Volumetric Efficiency %	B.H.P. input per cubic foot of F.A.D.
50	92.1	.142
75	89.1	.172
100	86.3	.205
120	84.6	.223

Designers usually allow a brake horse-power of .25+

on /

on ordinary small machines discharging air at 100 lb. per square inch.

It is generally accepted in practice that the work done in compressing the clearance air is almost completely returned in expanding. On the other hand, either the compressor must make a greater number of strokes, or a compressor of slightly greater dimensions must be installed, (both resulting in a greater loss by friction), so that a desired quantity of air will be delivered in a certain time.

That the work then lost is recovered or that the difference, if any, is of no consequence, is true when the comparison is made between two compressors discharg-:ing at the same pressure, one having, at the expense of an increased air speed through the valves, a small clearance volume, and the other having a large clearance but a low air speed; the loss due to the increased air speed in the one balancing the loss due to the increased clearance in the other. But in the case of two similar compressors differing only in their clearance, the above idea cannot be considered as correct, for it must be remembered that, owing to the metal in the end of the cylinder being very hot at the end of compression, the expanding air will, if the clearance is small, take up more /

more heat from the surrounding metal than if the clear-:ance is large. In other words the expansion index will be less with small clearances, and more work will be recovered per pound of clearance air. Also air leakage and slip will, with large clearances, tend to be greater owing to the pressure taking longer to drop down to the intake pressure during the suction stroke.

Theoretical Effect of Clearance on Work Done:-

Let one pound of air be delivered per stroke, and W_C = weight of clearance air, (lb.)

.·. weight present during compression = $(1 + W_C)$lb.,

and work done during compression = $(1 + W_C)\dfrac{P_2V_2 - P_1V_1}{n - 1}$ (ft.lb.)

(P_2 and V_2 = pressure and volume of one pound after compression (lb./sq.ft., cub.ft.))

(P_1 and V_1 = pressure and volume of one pound before compression (lb./sq.ft., cub.ft.))

Work done during expansion = $W_C\dfrac{P_2V_2 - P_1V_1}{n - 1}$ (ft.lb.)

Work done during suction = P_1V_1, since one pound is taken in during suction. (ft.lb.)

Work done during delivery = P_2V_2, since one pound is discharged during delivery. (ft.lb.)

.·. work done per cycle

$$= P_2V_2 + (1 + W_C)\frac{P_2V_2 - P_1V_1}{n - 1} - W_C\frac{P_2V_2 - P_1V_1}{n - 1} - P_1V_1,$$

and, assuming the indices of compression and expansion to be the same

$$= P_2V_2 + \frac{P_2V_2 - P_1V_1}{n - 1} - P_1V_1 \text{ (ft.lb.)}$$

which /

which is the same as the work done when there is no clearance. We may state, therefore, that theoretically the work done per pound of air delivered is the same whether clearance is considered or not, the assumption being that the pressures and temperatures are constant during suction and discharge and that the law of expansion of the clearance air is equal to the law of compression.

Also:-

The isothermal horse-power (based on delivered air)

$$= \frac{P_1 V_1 \log_e \frac{P_2}{P_1}}{33,000}$$

$(V_1 = \text{F.A.D./min.})$

.'. the isothermal horse-power per cubic foot of F.A.D. per minute

$$= \frac{P_1 \log_e \frac{P_2}{P_1}}{33,000},$$

and the F.A.D. per minute per isothermal horse-power

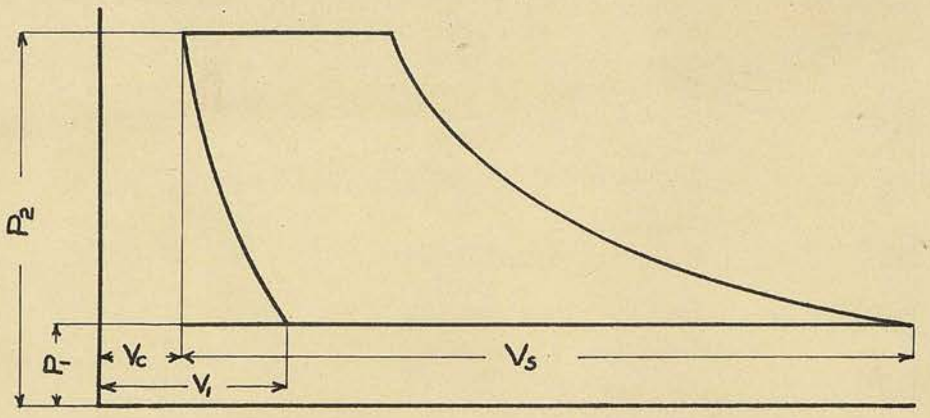
$$= \frac{33,000}{P_1 \log_e \frac{P_2}{P_1}},$$

both being quantities totally unaffected by the cylinder clearance.

Receiver Pressure (lb./sq. in.g.)	Discharge Pressure (lb./sq. in.g.)	Isothermal Horse-power per cu.ft. of F.A.D. per minute	F.A.D. per minute per Isothermal Horse-power (cu.ft.)
0	5	.01879	53.22
10	15	.04512	22.16
20	25	.06374	15.69
30	35	.07816	12.79
40	45	.08991	11.13
50	55	.09984	10.02
60	65	.1084	9.221
70	75	.1161	8.618
80	85	.1228	8.143
90	95	.1289	7.757
100	105	.1345	7.433

The above table lists the isothermal horse-power per cubic foot of F.A.D. per minute and the F.A.D. per minute per isothermal horse-power from zero to 100 lb. per square inch g. receiver pressure, atmospheric pressure being taken as 14.7 lb. per square inch, and are shown plotted to a base of receiver pressure on Plates 33 and 34.

PLATE NO.3.



The Effect of Clearance Volume on Volumetric or Discharge Efficiency:-

Referring to Plate 3, let

V_s = stroke volume;

V_c = clearance volume;

Cl. = clearance volume as a fraction of the stroke volume $= \dfrac{V_c}{V_s}$

P_1 = initial pressure;

P_2 = final pressure.

The theoretical volumetric efficiency

$$= \frac{\text{volume of air taken in per stroke}}{\text{stroke volume}}$$

$$= \frac{V_s + V_c - V_1}{V_s}$$

$$= \frac{V_s + V_c - V_c\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}}{V_s}$$

$$= 1 - \frac{V_c}{V_s}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} + \frac{V_c}{V_s}$$

$$= 1 - \text{Cl.}\left\{\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right\}$$

which is an expression obeying the straight line law $y = -ax + b$, so that if volumetric efficiency be plotted against clearance volume, the curves will be straight lines of gradient $-\left\{\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right\}$ and will cut the volum-

:etric /

PLATE Nº 4

— THEORETICAL VOLUMETRIC EFFICIENCY —

— PLOTTED AGAINST —

— CLEARANCE VOLUME. —

Red where 'n' = 1·4.
Green where 'n' = 1·3.
Blue where 'n' = 1·2.

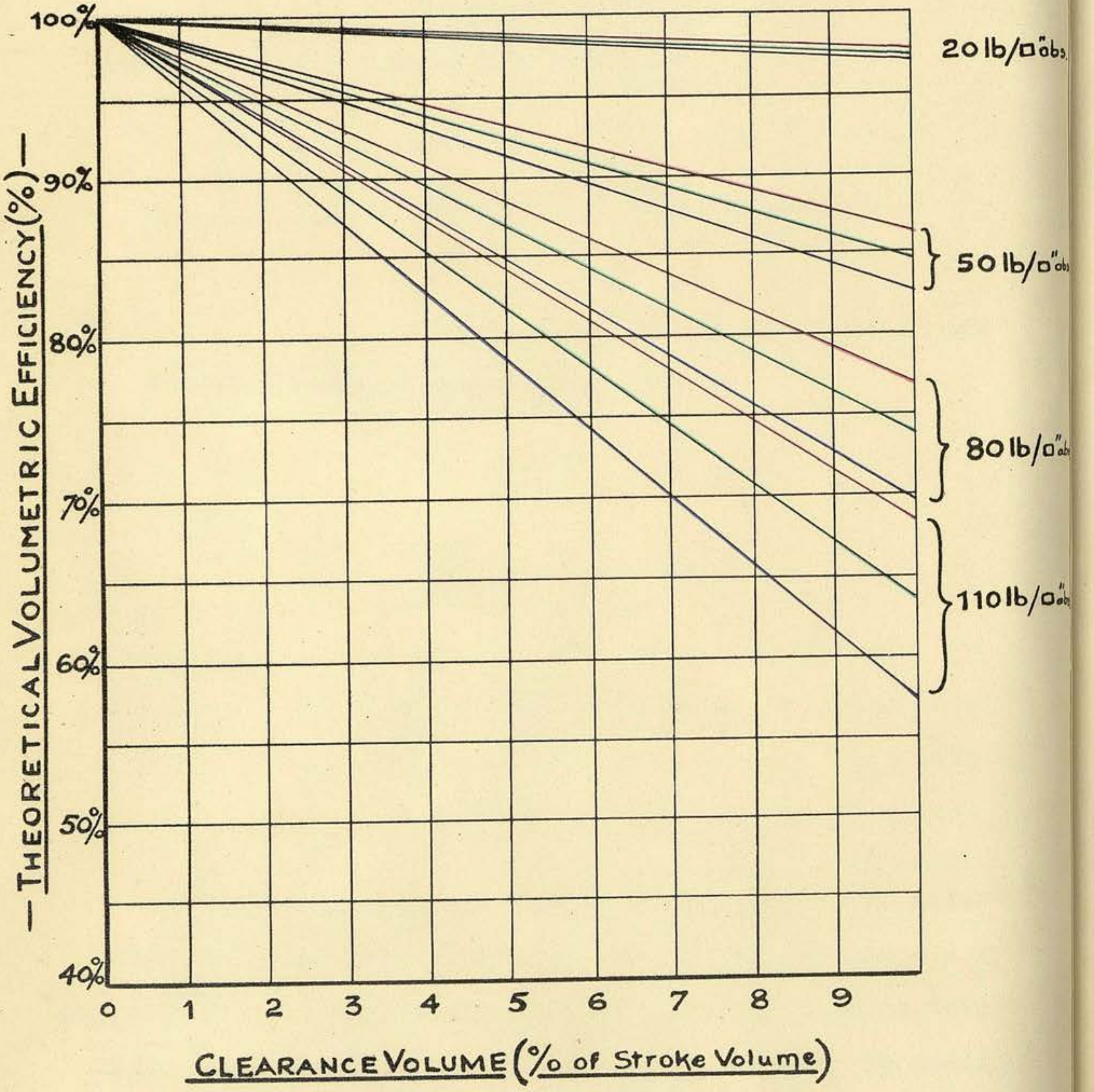
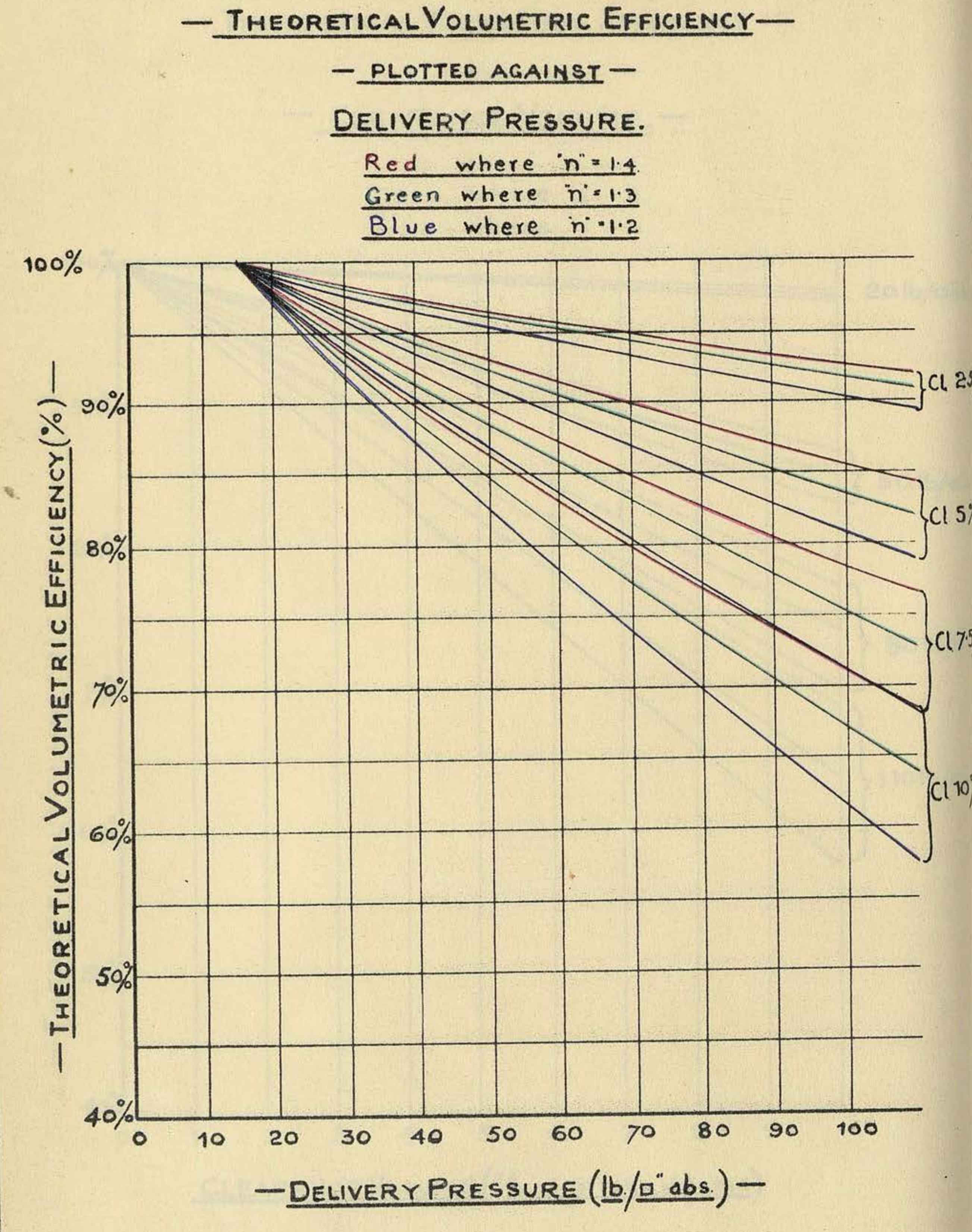


PLATE Nº 5


— THEORETICAL VOLUMETRIC EFFICIENCY —
— PLOTTED AGAINST —
DELIVERY PRESSURE.
Red where "n" = 1·4
Green where "n" = 1·3
Blue where "n" = 1·2
100%
90%
80%
70%
60%
50%
40%
0
10
20
30
40
50
60
70
80
90
100
CL 2.5%
CL 5%
CL 7.5%
CL 10%
— THEORETICAL VOLUMETRIC EFFICIENCY (%) —
— DELIVERY PRESSURE (lb./□" abs.) —

:etric efficiency ordinate at unity. This has been done in Plate 4 for discharge pressures 20 lb., 50 lb., 80 lb., and 110 lb. per square inch absolute and for values of "n" of 1.4, 1.3 and 1.2. (See Appendix III, page 124). Plate 5 shows the theoretical volumetric efficiency plotted to a base of delivery pressure. As the expansion curve approaches the adiabatic, the higher becomes the theoretical volumetric efficiency.

DESCRIPTION OF COMPRESSOR AND AIR MEASUREMENT PLANT INSTALLED IN EDINBURGH UNIVERSITY.

PARTICULARS OF COMPRESSOR.

Makers:- Messrs. Alexander Wilson (Aberdeen) Ltd..

Type:- Three-stage, vertical, totally enclosed, single-acting compressor, having cylinders water-jacketed and intercoolers fitted between the L.P., I.P., and H.P. cylinders.

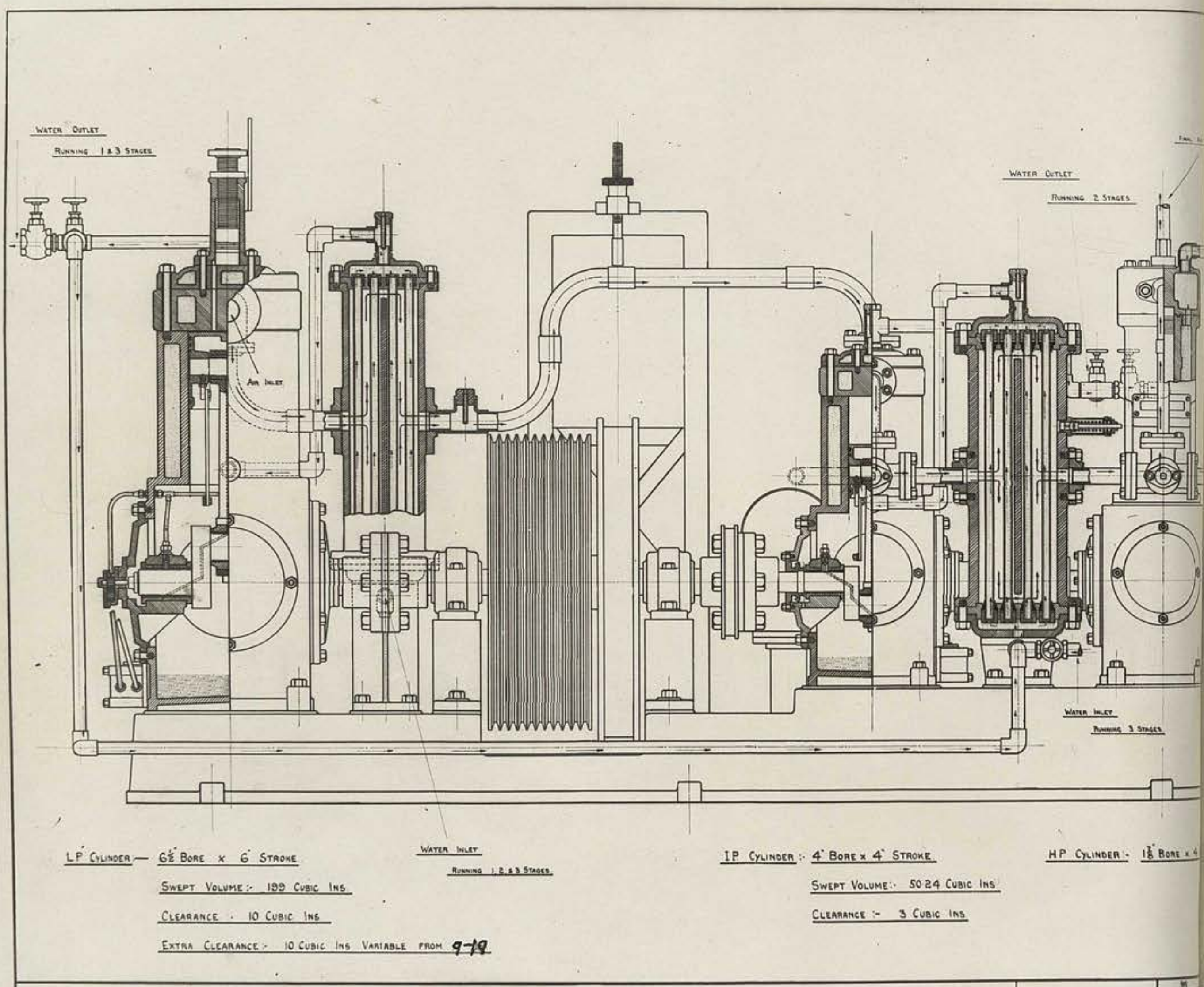
Description:- The first stage cylinder is fitted with a clearance volume adjuster and provision is made in the L.P. and I.P. cylinders for taking indicator diagr-:ams. Provision is also made for ascertaining all air and water temperatures as they enter and leave each cylinder and intercooler.

The driving shaft is fitted, between the L.P., I.P., and H.P. units, with flanged couplings and slip plates, so that each unit may be run separately if required. These couplings and slip plates also allow the compressor to be run both two-stage and three-stage and as there are two couplings between the L.P. and I.P. stages, one on either side of the driving pulley and flywheel, the brake horse-power input to the compressor can be read-:ily determined by means of a rope brake.

The compressed air pipe lines are arranged in such a manner that

(a) when running as a single-stage compressor the air can be led either warm or cold into the anti-pulsator, the intercooler between the L.P. and I.P. here acting as /

PLATE NO.6.



SECTIONAL ARRANGEMENT OF THREE STAGE AIR COMPRESSOR.

as an aftercooler;

(b) when running as a two-stage compressor, having an intercooler between the two units, the air can be led either warm or cold into the anti-pulsator, the inter:cooler between the I.P. and H.P. units acting as an aftercooler.

(c) when running three-stage with two intercoolers, the air is led into a suitable high pressure air bottle, no provision being made in this case for measuring the air.

The cooling water passes through the first inter:cooler, the L.P. jacket, the second intercooler, the I.P. jacket and the H.P. jacket in series, a process simplifying the measurement of the quantity of cooling water passing. Should the cooling water temperature become too high an independent water supply can be taken to the second intercooler and the I.P. and H.P. jackets.

Running single-stage the compressor discharges up to 100 pounds per square inch.

Running two-stage the compressor discharges up to 225 pounds per square inch.

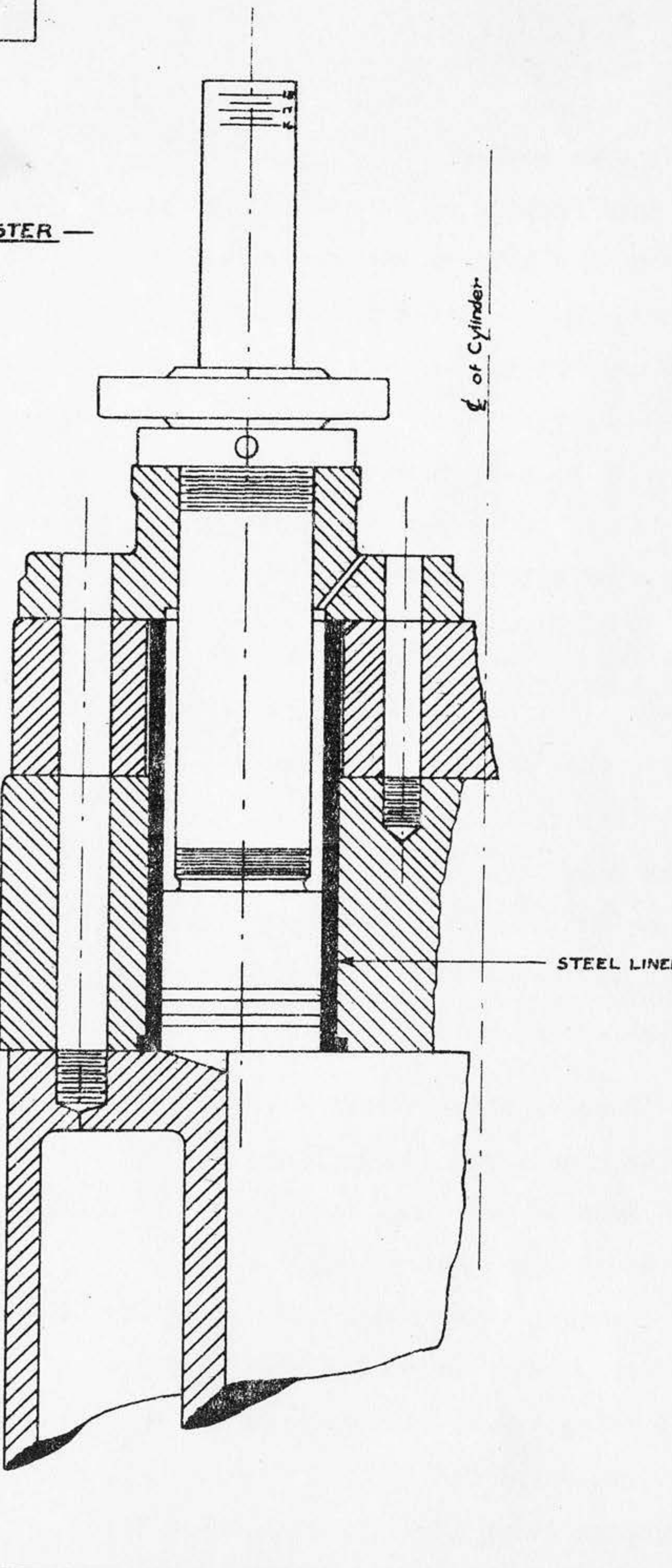
Running three-stage the compressor discharges up to 1000 pounds per square inch, the pressure distribut:ion being - L.P. discharge 56 lb. per square inch, I.P. discharge 215 lb. per square inch, and the H.P. discharge being 1000 lb. per square inch.

The /

PLATE No 7

— CLEARANCE ADJUSTER —

Scale – Half Full Size



The compressor is driven by a Texrope Drive from a 14 H.P. Crompton-Parkinson slip ring motor.

<u>Rating</u>:- 30 cubic feet of free air per minute at 260 r.p.m.

<u>Detail</u>:- L.P. cylinder - Bore $6\frac{1}{2}$", Stroke 6".
(stroke volume = .1152 cu.ft.)

The clearance volume can be varied from 9 to 18 cubic inches, i.e. from 4.52 - 9.04% of the stroke volume.

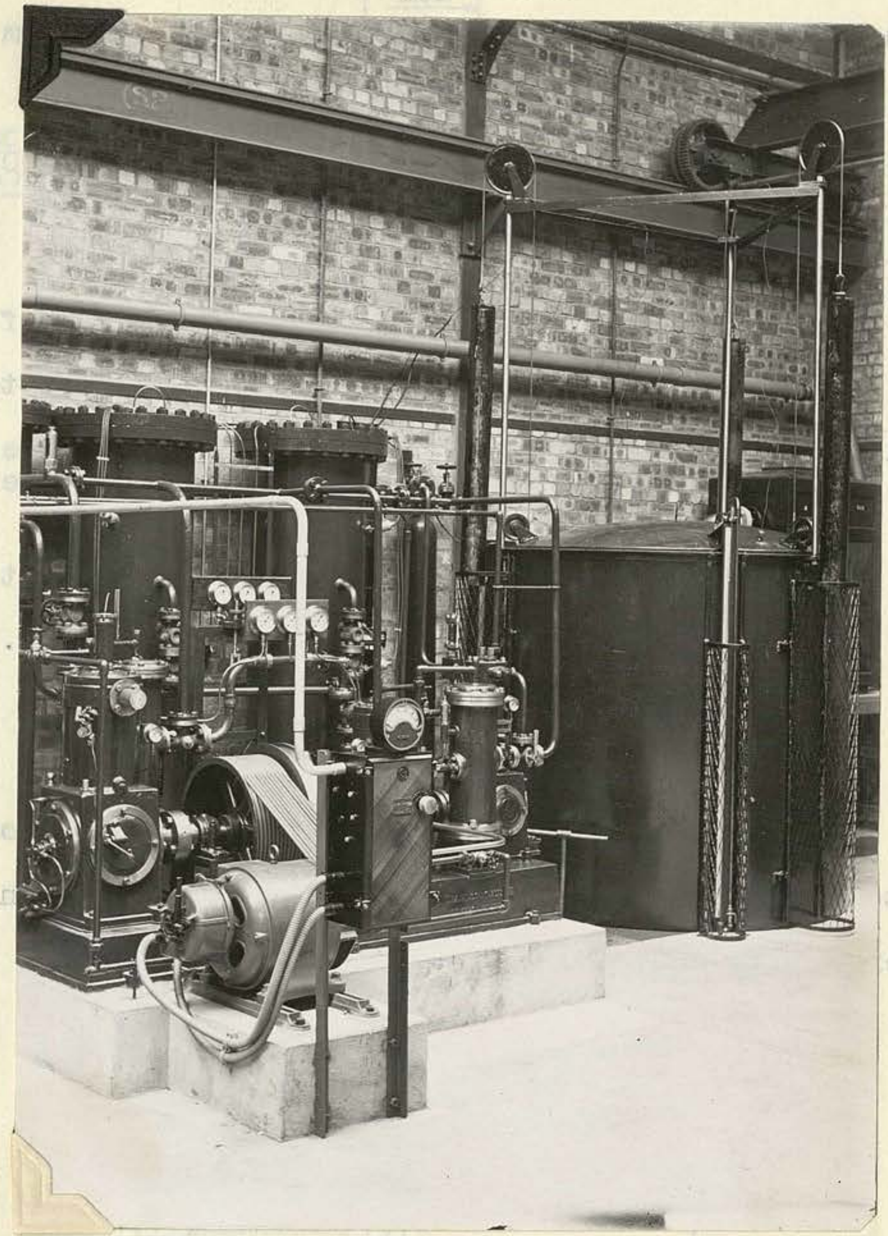
I.P. cylinder - Bore 4", Stroke 4".
(stroke volume = .02909 cu.ft.)

The clearance volume fixed at .001737 cu.ft.

H.P. cylinder - Bore $1\frac{7}{8}$", Stroke 4".

A sectional arrangement of the compressor is shown on Plate 6, but the clearance volume adjuster thereon has been replaced by another (see Plate 7).

PLATE NO.8.*



* See "Civil Engineering," May 1935, p.141.

AIR MEASUREMENT PLANT.

The Air Measurement Plant was designed to fulfil the following conditions, viz., that it should be able to

(a) measure, by means of two receivers, the air dis-:charged from the compressor, so that, by switching from one receiver to the other, a prolonged test could be carried out;

(b) measure the air discharged from the compressor, the two receivers being used as one large combined receiver;

(c) charge the receivers for departmental work;

(d) measure, by a gasometer, the air discharged from the compressor;

(e) discharge the gasometer with the compressor dis-:charging to atmosphere;

(f) use the gasometer to measure the air consumption of the Gas and Oil Engines installed in the Laboratory;

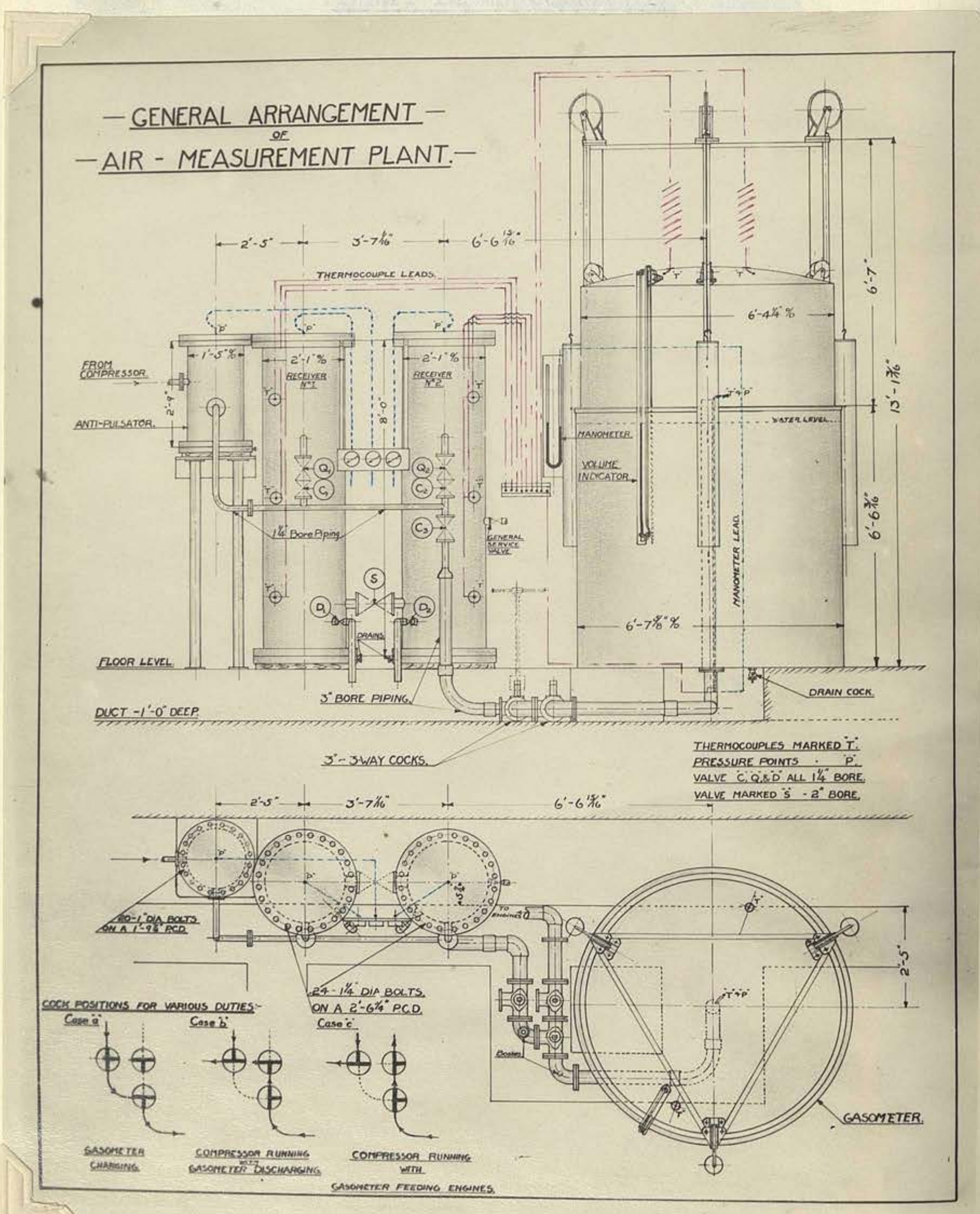
(g) carry out a prolonged test on the compressor by switching from the gasometer to the combined receivers, discharging one while the other is charging, and so on in alternate sequence;

(h) use the gasometer to check other air measurement apparatus, e.g., nozzles, anemometers, etc..

Plate 8 is a photograph of the complete air measurement plant with the air compressor in the fore-:ground, and Plate 9 is a detailed drawing of the plant.

The plant comprises two receivers of equal size, a gasometer, and a small receiver which acts as an anti-pulsator, with valves and pipe lines arranged to fulfil /

PLATE NO.9.



GENERAL ARRANGEMENT
OF
AIR - MEASUREMENT PLANT.
THERMOCOUPLE LEADS.
FROM COMPRESSOR.
ANTI-PULSATOR.
RECEIVER Nº1
RECEIVER Nº2
1¼" Bore Piping
GENERAL SERVICE VALVE
MANOMETER
VOLUME INDICATOR
MANOMETER LEAD
WATER LEVEL
DRAINS
FLOOR LEVEL
3" BORE PIPING
DUCT -1'-0" DEEP
DRAIN COCK
3" - 3-WAY COCKS.
THERMOCOUPLES MARKED "T".
PRESSURE POINTS " "P".
VALVE "C", Q, & D" ALL 1¼" BORE.
VALVE MARKED "S" - 2" BORE.
20-1" DIA. BOLTS ON A 1'-9⅝" P.C.D.
24-1¼" DIA BOLTS ON A 2'-6¾" P.C.D.
COCK POSITIONS FOR VARIOUS DUTIES
Case "a"
Case "b"
Case "c"
GASOMETER CHARGING
COMPRESSOR RUNNING WITH GASOMETER DISCHARGING
COMPRESSOR RUNNING WITH GASOMETER FEEDING ENGINES.
GASOMETER.

fulfil the various conditions detailed above.

THE RECEIVERS.

The position of the various thermocouples for finding the mean temperature within the receivers are shown on Plate No.9. The top thermocouples are in length equal to one-third of the bore, the centre, one-half of the bore, and the bottom, two-thirds of the bore. All the thermocouples are inclined down-:wards towards their tip to prevent water eating into the copper-steel joint.

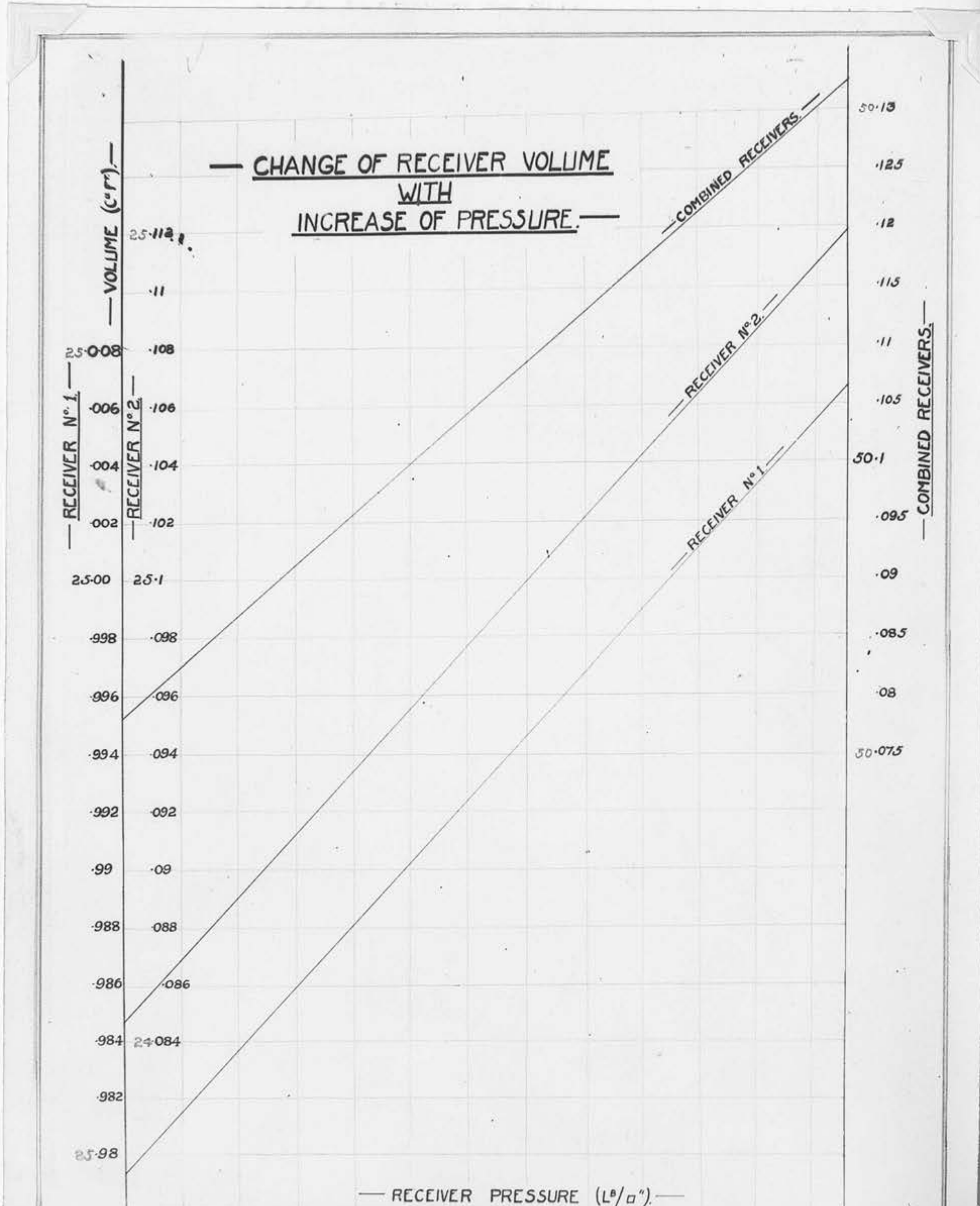
The method adopted to find the correct volume of the receivers at any pressure was as follows(See Plate 10):-

Each receiver was completely filled with water at atmospheric pressure and weighed. The pressure was then raised to 250 lb. per square inch and the receivers again weighed. Each showed a difference in weight of 1.7 lb.. On releasing the pressure, 772 c.c.(1.7 lb.) of water were collected; an amount agreeing with the difference in weight.

As 1.7 lb. of water occupies .0272 cubic feet, therefore, at 250 lb. per square inch, the volume of each receiver increases .0272 cubic feet. The volume measurements obtained are given in tabular form below.

Receiver /

PLATE NO.10.



— CHANGE OF RECEIVER VOLUME
WITH
INCREASE OF PRESSURE. —
— VOLUME (C" F") —
— RECEIVER N° 1. —
— RECEIVER N° 2. —
— COMBINED RECEIVERS. —
— COMBINED RECEIVERS —
— RECEIVER N° 2. —
— RECEIVER N° 1. —
25·0·08
·006
·004
·002
25·00
·998
·996
·994
·992
·99
·988
·986
·984
·982
25·98
25·112
·11
·108
·106
·104
·102
25·1
·098
·096
·094
·092
·09
·088
·086
24·084
50·13
·125
·12
·115
·11
·105
50·1
·095
·09
·085
·08
50·075
— RECEIVER PRESSURE (L^B/□"). —
0
20
40
60
80
100
120
140
160
180
200
220
240
260

	Volume at Atmospheric Pressure. (cub.ft.)	Volume at 250 lb. per square inch. (cub.ft.)
Receiver No.1	24.979	25.007
Receiver No.2	25.085	25.112
Combined Receivers	50.078	50.132

Valves C are ordinary screw down valves, while valves Q are quick acting link valves. This combin-:ation is designed to achieve the following results:-
1) the air entering the receivers is controlled by the relatively slow acting valves C, which can be more fully opened as the pressure in the receivers increases;
2) the air supply is quickly cut off by the valves Q, when the desired pressure is reached, or after a defin-:ite interval of time.

AIR MEASUREMENT BY RECEIVERS.* (See Plate 9.)

CASE A: Using two receivers as one large Combined Receiver.

The valves Q_1, C_1, C_3 and D_2 are closed and the valves S, D_1, Q_2 and C_2 opened. The air from the com-:pressor, having passed into a small receiver, which acts as an anti-pulsator, flows through valve C_2 into the receivers, and out through the discharge valve D_1. The control valve C_2 is then adjusted till the pressure within /

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* See Proceedings of the Institution of Mechanical Engineers, pp.1085 - Nov.1922.
Also Testing of Motive-Power Engines, Royds, ch.14.

within the anti-pulsator reaches the desired discharge pressure at which the compressor is to be tested. The mean temperature in the receivers is noted before the discharge valve D_1 is closed. The receivers are now being charged, and, as the pressure in the receivers increases, the valve C_2 must be gradually opened to keep constant the pressure in the anti-pulsator. A pressure of some 5 lb. per square inch below the anti-pulsator pressure having been reached, the valve Q_2 is quickly shut and the control valve C_3 opened, thus allowing the compressor to discharge to the atmosphere. The final volume (See Plate 10), the final pressure, and the final temperature are noted, and these, together with the atmospheric pressure and the time taken to charge the receivers, furnish sufficient data to arrive at the quantity of air discharged.

CASE B: Using the two receivers separately, charging one while the other is discharging, and so on, over a prolonged period.

The valves D_1, S, C_1 and C_3 are shut and the valves Q_1, D_2, Q_2 and C_2 opened. Having read the mean temp:erature in receiver No.2, shut valve D_2 and charge as described in Case A. When this is charged, quickly close valve Q_2, and, having taken the mean temperature in receiver No.1, open valve Q_1 which is then used to control /

control the anti-pulsator pressure. All necessary readings are taken before opening valve D_2 to discharge receiver No.2. Valve C_2 is now closed and valve Q_2 opened. When receiver No.1 is charged, receiver No.2 is recharged, and the whole cycle of operations is repeated.

In this case it is preferable that each receiver be charged for the same duration of time, rather than up to some pressure slightly below the anti-pulsator pressure as in Case A.

Let w_1 = initial weight of air in receiver No.1 (lb.)

" w_2 = " " " " " No.2 (lb.)

" W_1 = Total weight of air registered over, say, x chargings of receiver No.1 (lb.).

" W_2 = Total weight of air registered over, say, x chargings of receiver No.2 (lb.).

" t = time, in minutes, allowed for charging each receiver.

Then weight of air discharged by the compressor per minute

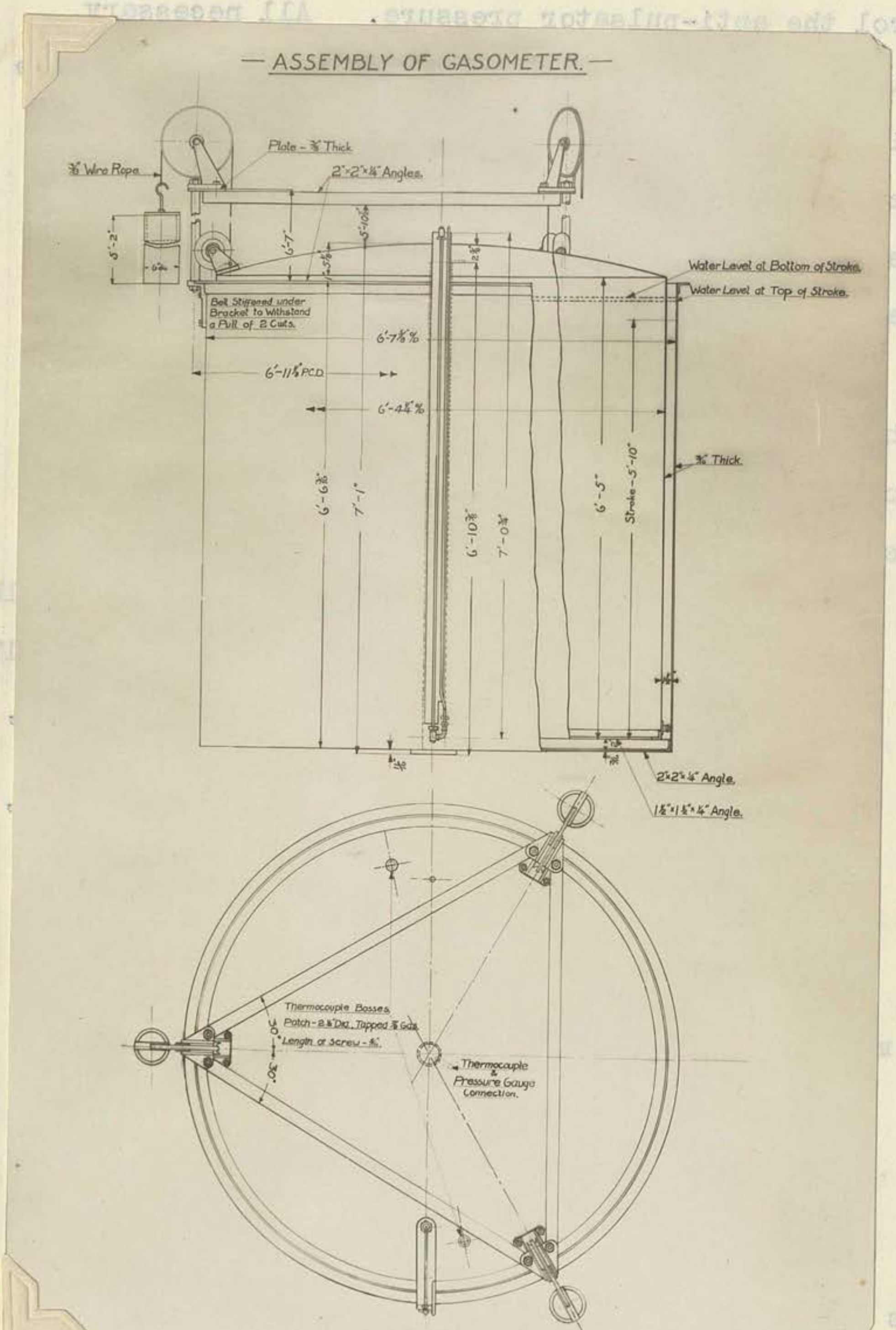
$$= \frac{(W_1 - xw_1) + (W_2 - xw_2)}{2\,xt} \text{ lb.}$$

$$= \frac{(W_1 + W_2) - x(w_1 + w_2)}{2\,xt} \text{ lb.}$$

This formula assumes w_1 and w_2 to be constant. This is not always the case as the initial temperature sometimes varies with each test.

THE /

PLATE NO.11.



— ASSEMBLY OF GASOMETER. —
⅜" Wire Rope
Plate - ⅜" Thick
2"×2"×¼" Angles.
Bell Stiffened under Bracket to Withstand a Pull of 2 Cwts.
Water Level at Bottom of Stroke.
Water Level at Top of Stroke.
6'-11¾" P.C.D.
⅜" Thick.
Stroke - 5'-10"
2"×2"×¼" Angle.
1½"×1½"×¼" Angle.
Thermocouple Bosses. Patch - 2⅛" Dia. Tapped ⅜" Gas. Length of Screw - ¾".
Thermocouple & Pressure Gauge Connection.

THE GASOMETER.

A small scale gasometer, to meet the requirements of the laboratory, was designed, and details of its design are shown in Plate 11. In principle, it is merely an inverted bell, balanced in the manner illustr:ated, and sealed with water. Air enters through a vertical three-inch bore pipe, which protrudes some six inches above the water level, and the bell is so balan:ced as to begin floating when the pressure inside exceeds atmospheric pressure by a half-inch column of water, and by one-and-a-half inches of water when full. Air discharging into the gasometer can, therefore, be reasonably assumed to be at atmospheric conditions. To accomplish this delicate balancing there is, in the top of each weight, a cavity into which lead shot can be introduced to ensure equal loading.

The installation is fitted with three thermo:couples, two on the head of the bell, and one at the top of the intake pipe. There is a pressure point at the top of the intake pipe.

The volume indicator is as shown diagrammatically on Plate 12. It is firmly attached to the bell and travels with it. One end is led into the top of the bell, and the other dips into the water between the bell and the casing. Water is syphoned over this limb, and finds /

PLATE NO.12.

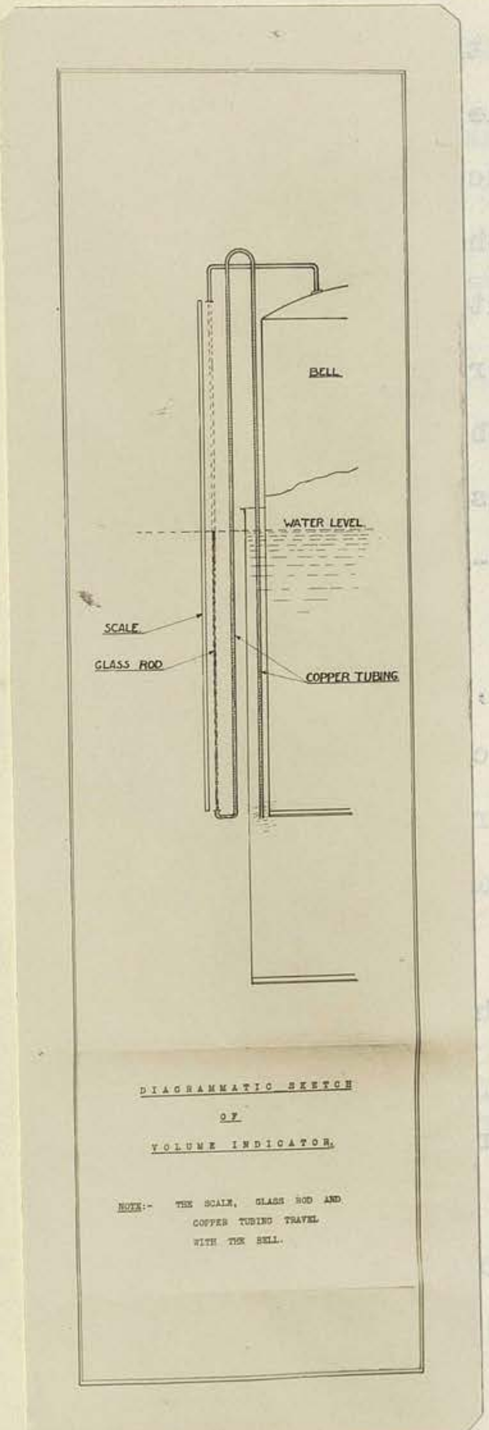
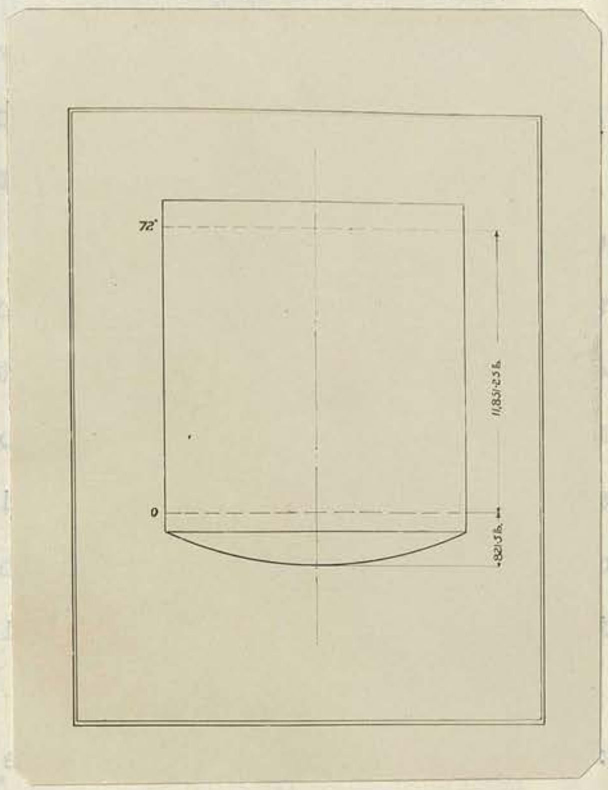


PLATE NO.13.



finds its own level in the glass portion of the indic-:ator.

In calibrating the bell, it was inverted, and a known weight of water poured into the rounded head, until the water attained a three-inch level in the cylinder proper (See Plate 13), which point was marked before the complete filling of the bell with water. Following this, operations were suspended during several days to allow the water to reach the laboratory temperature. The water level was then lowered to the initial mark, being run off in five-inch drops, each of which was carefully weighed. Each five-inch drop was found to vary only $\pm$ 7 lb. in a weight of approx-:imately $7\frac{1}{4}$ cwt., results proving the sides of the bell to be truly cylindrical. Subsequent calculations were:-

Weight registered on 72-inch drop = 11,851 lb.,

i.e. volume between 0 and 72 inches = 189.739 cu.ft.,

$\therefore$ 1 cu.ft. was registered by .3795 inch.

i.e., the scale was divided between 0 and 72 inches into 189.739 parts, each part being .3795 inch long; this .3795 inch was then divided into 10 equal parts.

The weight up to the initial mark was 821.5 lb., i.e., the initial volume was 13.152 cubic feet. To this initial volume the volume of the intake pipe was added, thus making an initial volume of 13.872 cubic feet. /

feet. From this was subtracted the volume occupied by the ring of intake piping projecting above water level, giving the true initial volume, viz., 13.838 cubic feet.

Now 1 cubic foot is registered by .3795 inch,
$\therefore$ .838 cubic foot is registered by .318 inch.

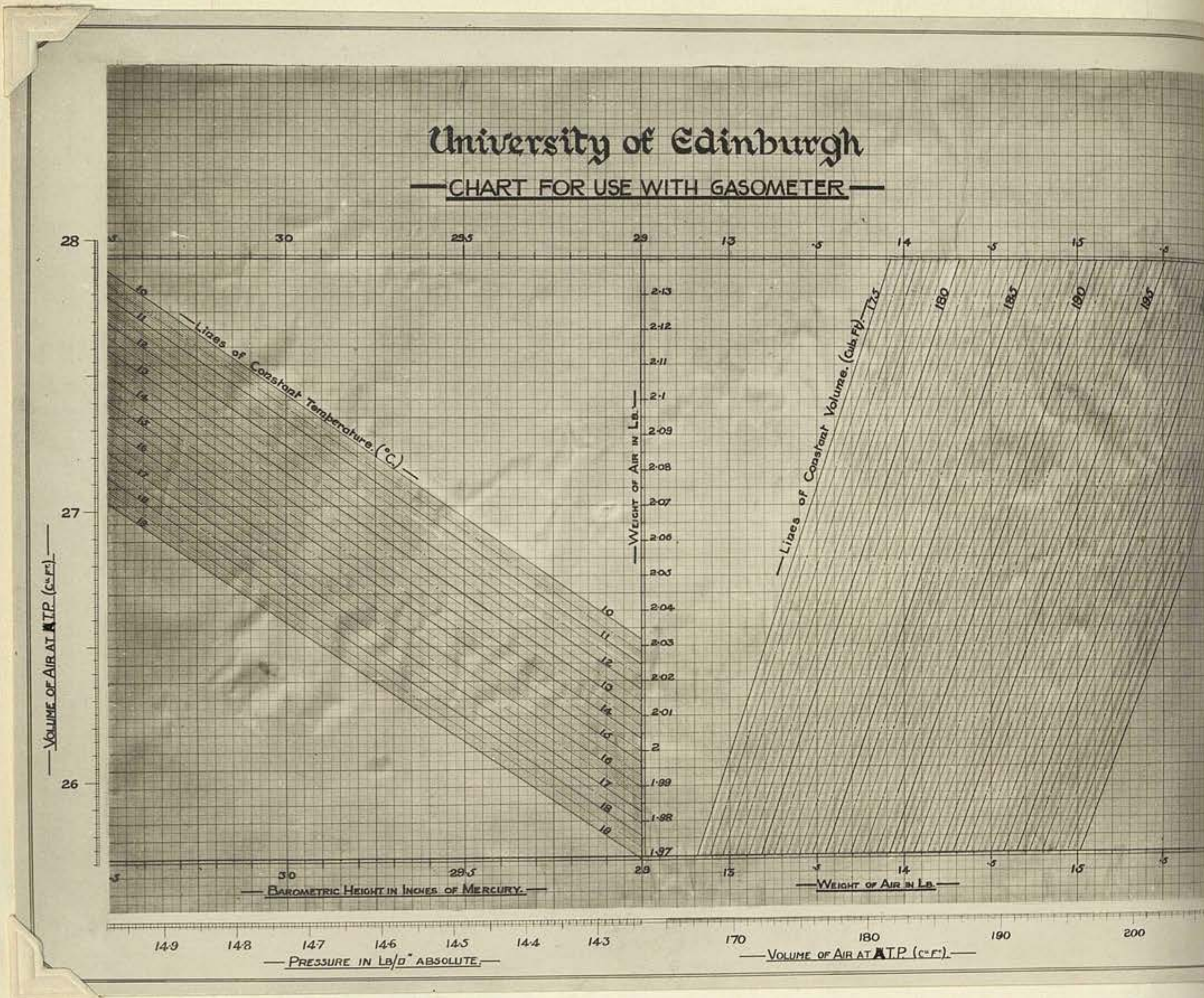
So that, by measuring down .318 inch, the 13 cubic feet mark was obtained and the calibrated scale placed in position.

In considering the question as to whether varying temperatures would affect seriously the volume of the bell itself, and therefore produce errors in the above method of calibration, it must be remembered that the bell, when down, is completely submerged in water, and that, as it takes only from five to seven minutes to fill, it is reasonable to assume that its temperature, when full, is still that of the water. Were the gas-:ometer situated in the open air or in a workshop, places where the temperature variation might be con-:siderable, the variation produced by temperature changes would be a serious factor, but, in a laboratory, working under ideal conditions, temperature changes on volume are negligible.

NOTE:- The temperature of the water over a whole year varied, approximately, only 4°C..

TO /

PLATE NO.14.


University of Edinburgh
CHART FOR USE WITH GASOMETER
Lines of Constant Temperature (°C)
Lines of Constant Volume (Cub.Ft.)
Weight of Air in Lb.
Volume of Air at N.T.P. (cu.ft.)
Barometric Height in Inches of Mercury.
Weight of Air in Lb.
Pressure in Lb/□" Absolute.
Volume of Air at N.T.P. (cu.ft.)

TO USE THE GASOMETER.

Read the initial volume, barometric pressure, and mean temperature, then turn the cocks as in Case "a" on Plate 9 and charge the gasometer for an even number of minutes to secure a final volume between 175 and 203 cubic feet, and then turn cock to allow the compressor to discharge to atmosphere. The gasometer is now full. Read the final volume, mean temperature, and pressure in inches of water above atmosphere.

NOTE:- The cocks, though big, take only a fraction of a second to turn.

CHART FOR USE WITH GASOMETER.

To simplify calculations involved in using the gasometer, a chart to find the weight of air and the volume of air at average temperature and pressure in the gasometer, before and after charging, and also the volume of free air actually delivered, was drawn (See Plate 14.).

After the gasometer had been in use for several months, it was noted that the temperature of the air which it contained rarely exceeded 16°C., and very seldom dropped to 11°C.; also, that the pressure when full was approximately one-and-a-half inches of water above atmospheric pressure.

Most gasometer users prefer that the bell be float-

:ing /

:ing before charging, but in order to construct a suit-:able chart it was decided that the bell should rest on the bottom, thereby indicating a constant volume within. This volume was kept constant at 26.9 cubic feet, and, when due to evaporation of the water within the gasometer, the volume exceeded this, a little water was poured in until the volume again read 26.9 cubic feet. The results obtained were identical whichever method was adopted as to the bell floating or resting on the bottom. It was most important that the volume should remain constant, as the vertical scale, which gives on the chart the weight of air in the gasometer when down, was drawn on that assumption. If, however, several lines of constant volume had been inserted in the neighbourhood of the 26.9 cubic feet line, the diffic-:ulty of change in initial volume could have been over-:come.

The volume of air in the gasometer when full varied from 180 cubic feet to 200 cubic feet.

In view of the above, the limits chosen for the chart were:-

Pressure - 29 to 30.5 inches of mercury;

Temperature - 10 to 19°C.;

Initial Volume fixed at 26.9 cubic feet;

Final Volume - 175 to 205 cubic feet.

CONSTRUCTION /

CONSTRUCTION OF CHART.

CASE A:

When the gasometer is down, its volume is constant at 26.9 cubic feet. Temperature and pressure are, therefore, the only two variables.

$$w = \frac{PV}{RT} = \frac{144 \times 26.9}{96.3} \cdot \frac{P}{T} = 40.24 \frac{P}{T}$$

For upper limit of pressure 30.5 inches mercury,

$$w = 40.24 \times \left(\frac{30.5 \times .491}{T}\right)$$

So that the **weight** of air for all temperatures between $10^{o}C.$ and $19^{o}C.$ can be calculated and indicated on the chart.

Similarly for the lower limit of pressure, 29 inches mercury,

$$w = 40.24 \times \left(\frac{29 \times .491}{T}\right)$$

from which the weights of air for the same range of temperature are also found.

By joining the weights of air on the lower limit of pressure to the corresponding weights of air on the upper limit of pressure for the same temperature, the lines of constant temperature are located.

CASE B: /

CASE B:

When the gasometer is filled, the pressure, the volume, and the temperature are all variables.

The vertical scale for weights of air was drawn on the assumption that the volume remained constant. From it, lines of constant volume for the range 175 to 205 cubic feet can easily be obtained.

Let w = weight of air for range 175 to 205 cubic feet (lb.);

$\overline{w}$ = weight of air on vertical scale, (lb.);

v = volume of air corresponding to w (cubic feet);

and 26.9 = volume of air corresponding to $\overline{w}$ (cubic feet);

Then:-

$$\frac{w}{\overline{w}} = \frac{v}{26.9},$$

$$\text{i.e. } w = \frac{v}{26.9} \times \overline{w}.$$

So that, for the upper limit on the vertical scale, which is 2.14 lb.,

$$w = \frac{2.14}{26.9} \times v,$$

and for the lower limit on the vertical scale, which is 1.97 lb.,

$$w = \frac{1.97}{26.9} \times v.$$

The Lines of Constant Volume are now obtained by joining the corresponding points on the upper and lower limits.

TO /

TO USE THE CHART.

CASE A: when the gasometer is down.

Locate the required pressure on the Horizontal Scale. Trace the perpendicular from this point to meet the required temperature line, then trace a horizontal line from this point to meet the vertical scale of Weights of Air. If the volume of air at A.T.P. is desired, a horizontal line to the left will locate the correct volume at A.T.P.

CASE B: when the gasometer is filled.

Locate the weight of air for a constant volume of 26.9 cubic feet as in Case A, then continue the line to meet the required line of constant volume. A perpendicular from this point will give directly the weight of air on the horizontal Weight of Air Scale. The volume of air at A.T.P. is obtained as in Case A, except that a perpendicular is dropped to the Volume at A.T.P. scale.

CASE C: to find the volume of "free air", i.e., at atmospheric conditions.

As before, obtain the weight of air for a definite volume of 26.9 cubic feet. A horizontal line through this point, to meet the perpendicular from the given weight of air on the horizontal Weight of Air scale, will locate the required volume on the lines of constant volume. /

volume.

Conversion Constants:-

(a) Inches of mercury × .491 = lb. per square inch;

(b) Volume of air at A.T.P. = 13.11 w;

where w = weight of air in lb.,

Average Temperature = 15°C.,

Average Pressure = 14.7 lb. per square inch absolute.

ACCURACY OF CHART.

CASE A:

When the gasometer is down, its volume is constant at 26.9 cubic feet. Temperature and pressure are, therefore, the only two variables.

$$PV = wRT$$

$$\therefore w = \frac{PV}{RT} = \frac{144 \times 26.9}{96.3} \times \frac{p}{t} = K.\frac{h \times .491}{t + 273.1}$$

where w = weight of air in lb.;

P = pressure in lb. per square foot;

V = volume in cubic feet;

T = temperature in °C. absolute;

p = pressure in lb. per square inch;

t = temperature in °C.;

h = height of a mercury column corresponding to pressure p.

Example	h	t	w by Calculation	w by Chart
1	30.25	10	2.111	2.111
2	30.25	19	2.046	2.0458
3	29.25	10	2.041	2.0411
4	29.25	19	1.978	1.978

CASE B:

When the gasometer is filled, the pressure, the temperature, and the volume are all variables.

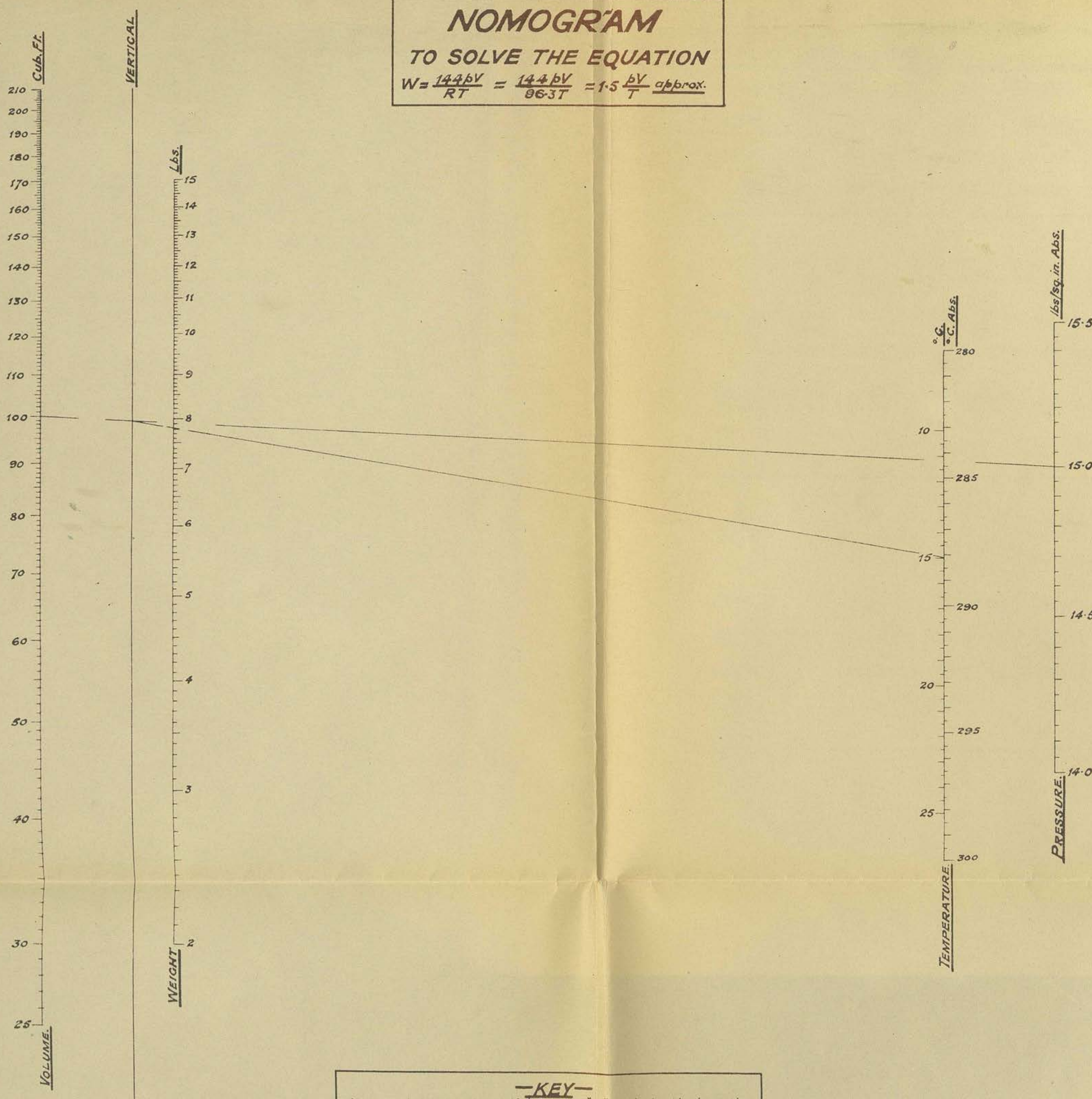
$$PV = wRT, \therefore w = \frac{PV}{RT} = \frac{144}{96.3} \times \frac{pV}{T} = \frac{144}{96.3} \cdot \frac{(h \times .491)V}{(t + 273.1)}$$

Example	h	t	V	w by Calculation	w by Chart
1	30.25	10	175	13.73	13.73
2	30.25	10	205	16.09	16.09
3	29.25	19	175	12.86	12.86
4	29.25	19	205	15.08	15.075

NOMOGRAM

TO SOLVE THE EQUATION

$$W = \frac{144pV}{RT} = \frac{144pV}{96 \cdot 3T} = 1 \cdot 5\frac{pV}{T} \text{ approx.}$$



—KEY—

Lay Straightedge from Value on the "V" Scale to that on the "P" Scale and mark the Intercept on the Mid-Vertical: then from this Intercept across to Value on the "T" Scale and read "W" where this Line cuts the "W" Scale.

R. Thomson

—EDINBURGH UNIVERSITY—
—NOV. 1934.—

LABORATORY SHEET FOR USE WITH GASOMETER.

Results 'A', 'B', and 'C' obtained from Gasometer Chart.

Bar.Pressure............" Hg. Date:

	Time to charge Gasometer (min.)	Pressure above atmosphere (in. of H_2O)	Pressure above atmosphere (in. of Hg)	Absolute Pressure (in. of Hg)	Temperature - (°C.)	Volume - (cu.ft.)	A Weight of air - (lb.)	B Volume of "Free Air" (cu.ft.)	C Volume of Air at A.T.P. (cu.ft.)
END									
BEGINNING						26.9			
						Per....... min.			
						Per minute			

All calculations in the subsequent text regarding weights of air and free air were taken from this orthographic chart, but as it proved rather cumbersome to one not skilled in its use, a suitable Nomogram was constructed. The nomogram complete with construction proof and method of reading is given on Plate 15.

NOTES ON AIR MEASUREMENT.

In making a survey of the various methods of air measurement, the general utility of such a survey, drawn from many sources, brief yet comprehensive, must be kept in view. A catalogue is presented, therefore, of the various methods of air measurement, the approach being made from an engineering angle, but a catalogue with a particular as well as a general application.

(a) Measurement of Air - using Receivers.

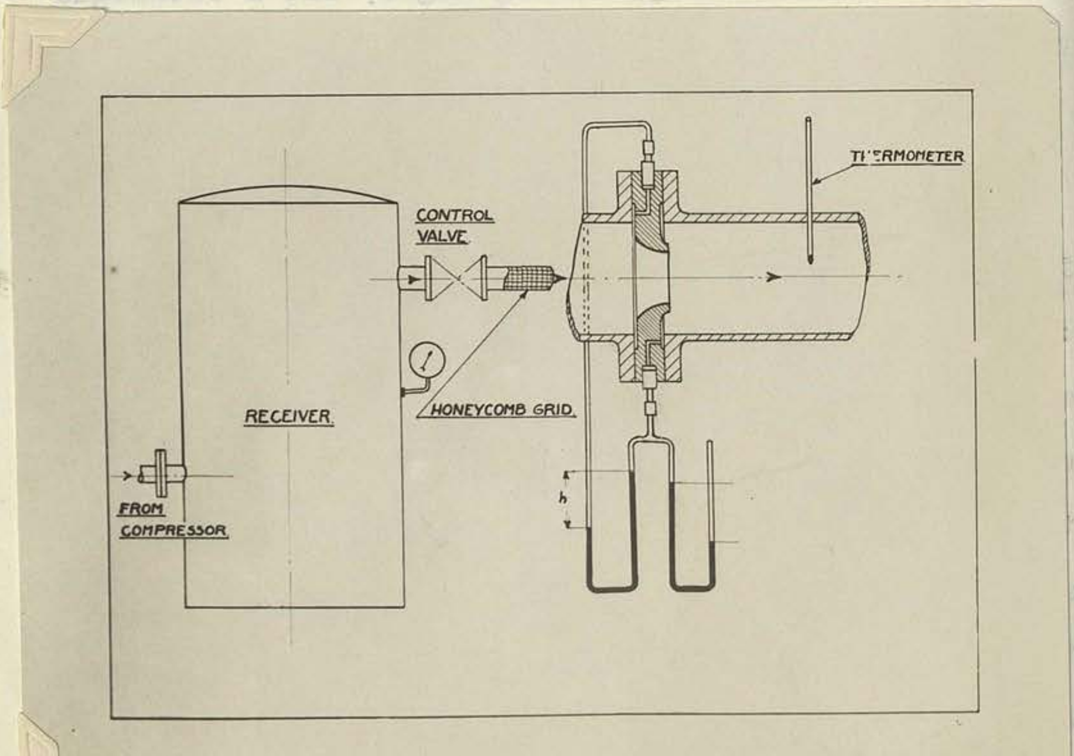
With small compressors this method of measurement is commonly adopted, but its application to large com:pressors is full of practical difficulties. We encounter such difficulties as the ascertainment of the correct volume of the receivers under pressure, of their change in volume produced by varying temperatures and of the actual mean temperature of the air within them. The use of receivers for air measurement is fully treated in the previous section, which explains how the volume at any pressure and the mean temperature are fairly accurately obtained.

(b) Measurement of Air - using a Gasometer.

This method has already been discussed in the section entitled "Air Measurement Plant as Installed in Edinburgh University."

(c) /

PLATE NO.16.


THERMOMETER
CONTROL
VALVE
HONEYCOMB GRID
RECEIVER
FROM
COMPRESSOR
h

(c) Measurement of Air - using Low Pressure Nozzles.

The last few years have seen the universal appli-:cation of this method to the commercial measurement of air. The Report on Air Flow Measurement of the Heat Engine Trials Standing Committee resulted in the adop-:tion, by the British Compressed Air Society, of nine sizes of nozzles of the Interessen-Gesellschaft type, for the range six to thirty thousand cubic feet of free air* per minute.

Plate 16 shows diagrammatically how this method is carried out. Air from the compressor enters a receiver in which all pulsations are damped, it then passes through a flow control valve into a honeycomb grid, issuing thence in stream line flow before passing through the nozzle.

The discharge of free air per minute expressed in cubic feet then $= K \times \dfrac{T_1}{P_1} \times \sqrt{h} \times \sqrt{\dfrac{m_2}{T}}$

where K is a constant

T_1 = temperature of air entering compressor ($^{\circ}$F.abs.);

P_1 = barometric height in inches of mercury;

h = head lost at nozzle (inches of water);

m_2 = absolute pressure on downstream side of nozzle in inches of mercury;

and T = air temperature within pipe after passing through nozzle ($^{\circ}$F.abs.).

A /

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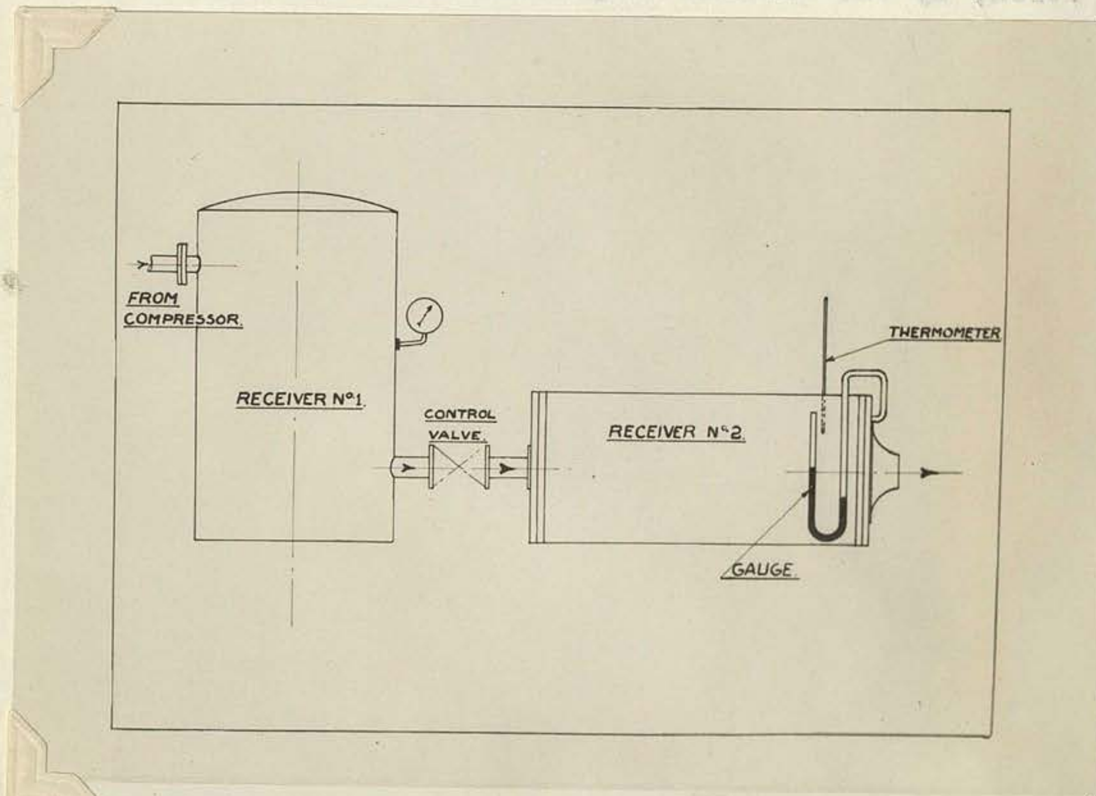
* NOTE: i.e. air at normal atmospheric conditions at the point where the compressor is situated.

PLATE NO.17.


FROM
COMPRESSOR.
RECEIVER Nº1.
CONTROL
VALVE.
RECEIVER Nº2.
THERMOMETER
GAUGE.

A complete list of nozzles, constants, and methods of procedure, is given in the "Standards of the B.C.A.S.", 1932 Edition, pp.20 - 27.

See also:-

"Proc. Inst.Mech.E.," 1914, p.927, (W.E.Fisher);

" " " 1922, p.1090 (W. Reavell);

" " Civil E., Vol.39, p.375(G. Zeuner);

"Proc.Royal Society," Ser.A., Vol.111, p.306 (T.E.Stanton);

"Messung von Gasmengen," published by M. Krayn, Berlin (1913);

"Zeitschrift des Vereines Deutscher Ingenieure," (25th Feb. 1922);

and

"Report of the Heat Engine Trials Standing Committee on Air Flow Measurement," 1931.

Etc., etc..

The method adopted by the American Society of Mechanical Engineers, and by the American Compressed Air Society, differs slightly from the British practice. The nozzle, instead of being placed in a pipe line, is situated at the end of a second receiver, the whirls and eddies caused by the control valve being damped in this second receiver (See Plate 17.). Also, the nozzle used is somewhat longer in the throat than the Interessen-Gesellschaft model.

The /

The volume of free air discharged per minute expressed in cubic feet then =

$$\frac{19.16D^2\ cT_2}{p_2}\sqrt{\frac{(p_1 - p_2)p_2}{T_1}}$$

$$= \frac{2.552D^2\ cT_2}{p_2}\sqrt{\frac{B \times i}{T_1}}$$

where D = diameter of nozzle throat in inches;

c = coefficient of discharge;

p_1 = pressure on upstream side of nozzle (lb./sq.in.abs.);

p_2 = atmospheric pressure (lb./sq.in.abs.);

T_1 = temperature of air on upstream side of nozzle ($^{\circ}$F.abs.);

T_2 = atmospheric temperature ($^{\circ}$F.abs.);

B = value p_2 expressed in inches of mercury; and

$i = (p_1 - p_2)$ expressed in inches of water.

For further details of this method, see the "Standards of the American Compressed Air Society" Fourth Edition, Section 3. See also "Measurement of Flow of Air and Gas with Nozzles," by S.A.Moss, in the "Transactions of the American Society of Mechanical Engineers," vol.50.

The following is a copy of a memorandum prepared by the chief engineers of a leading American air com-:pressor /

:pressor company, dated 7th June 1933, which discusses and criticises the methods recommended in the British Compressed Air Society's publication:-

"Mr. ------- is particularly interested in knowing whether the results produced by the British Compressed Air Society's method will be practically the same as those obtained according to our own standards. As to this, the formula given for computation of the capacity by the British Compressed Air Society agrees exactly with the so-called 'approximate Moss formula' which the A.S.M.E. has adopted, and which is also given in the 4th edition of the Compressed Air Society's 'Trade Standards'.

"With regard to the method proposed for obtaining the readings used in computing the capacity, we are not in thorough agreement with the British method. In the first place, they use a nozzle shape which they adopted from Germany and which disagrees considerably with the standard nozzles now adopted by the A.S.M.E. In particular, the radius of the nozzle is much smaller. Moreover, the British use this nozzle between two flanges and have up- and down-stream connections in the nozzle flange itself for measuring the drop in pressure through the nozzle. When they discharge to atmosphere, they shield the nozzle by a pipe of the same diameter as the upstream pipe, having at least ten nozzle diameters, thus creating a nozzle discharge pressure different from atmosphere. We consider this an unnecessary complication, although it has the advantage that the same nozzle can be used for measurements taken in a pipe not discharging to atmosphere.

"There are a number of other exceptions which we want to take with this British Compressed Air Society'bulletin, and which we will give you in detail in answer to Mr. ----'s request for these comments. For instance, the British propose to make no corrections for humidity and to treat air always as a dry gas. For high atmospheric temperatures and high humidity, this may lead to errors as large as 2%.

"On /

"On the whole, however, and as far as Mr. ---'s question is concerned, I believe we can advise him that the agreement between our methods of testing and those proposed by the British Compressed Air Society will be in close enough agreement for all practical purposes. If any penalty job is involved, I believe it should be agreed beforehand which code is to form the basis of any tests, and if we make any guarantees from here, this will have to be taken into account."

Yet another method, which can be employed to measure air entering a compressor, is the "Throttle-Plate Method", a method successfully used to determine the air consumption of internal combustion engines. Here the nozzles are merely circular holes, drilled through thin sheets of metal, and are thus exception-:ally simple to make.

When a small portion of a fluid falls freely, its theoretical velocity, after falling height "H", is:-

$$V = \sqrt{2gH}.$$

This can be successfully applied to air flowing to or from a receiver under a steady head, "h" inches, of water.

Now, as 1 cubic foot of water at 15°C. weighs 62.4 lb., and 1 cubic foot of air at N.T.P. weighs .08071 lb., therefore the height of a column of air to give /

give the same pressure as a one-inch column of water is $\frac{62.4}{12 \times d}$ feet, where d = density of air at N.T.P.,

$$= \frac{62.4}{12 \times .08071} \text{ feet},$$

$$= 64.4 \text{ feet.}$$

Therefore, at normal temperature and pressure, and under a head of a one-inch column of water, air issues from an orifice with velocity $= \sqrt{2gH}$

$$= \sqrt{2 \times 32.2 \times 64.4}$$

$$= 64.4 \text{ feet per second.}$$

CORRECTION FOR TEMPERATURE.

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} \text{ i.e., } \frac{H_1}{T_1} = \frac{H_2}{T_2} \therefore H_2 = \frac{H_1 \times T_2}{T_1}$$

where $H_1 = 64.4$ feet;

$T_1 = 273^oC.$absolute;

and T_2 is known.

From the above expression we may demonstrate simply that warm air, being lighter, will raise the value of H_2, and, on substituting H_2 for H in the formula $V = \sqrt{2gH}$, the velocity of the air issuing from an orifice will be greater the higher the air temperature, and vice versa.

We know that $V = \sqrt{2gH}$.

$\therefore$ volume delivered per second = Area of Orifice in square feet × Coefficient of Con-:traction × V

$$= A \times c \times \sqrt{2gH} \text{ cubic feet.}$$

The /

The determining of the value of the coefficients of contraction has been the subject of much work by Messrs. Watson, Schofield & Durley, and a complete list of nozzles and quantities of air passing for given pressure drops is given in "Engineering," 1923, p.457 and p.481 (R.O.King).

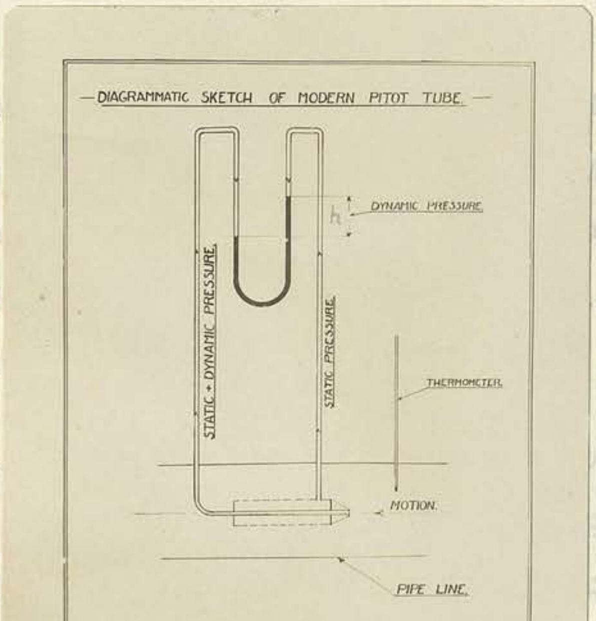
See also:-

"Engineering," September 1910, pp.380-381, (W.E.Dalby);
"Proc. Inst.Mech.E.," 1912, p.517, (Watson & Schofield);
"Proc. Inst.Mech.E.," 1925, p.885, (J.L.Hodgson);
"Proc. Inst.Civil E.," Vol.197, p.243 (H. Gaskell);
"Trans. of A.S.M.E.," 1906, p.193, (R.J.Durley);
"Zeitschrift des Vereines Deutscher Ingenieure," Vol.52, p.285;
"American Machinist," Aug.10th 1905, Vol.28, p.193;
"Der Civilingenieur," Vol.5, p.546; Vol.20, p.14, and
"A Treatise on Chemical Engineering," Ch.I (J. Martin);
See "Engineering," 16th October 1936, p.410.

Plates 18 and 19 show diagrammatically the Throttle-Plate method. The nozzle is held in place by plasticine. To ensure success, the boxes must be free from leaks, and, to procure a steady flow through the nozzle, all pulsat:ions must be thoroughly damped, either by a breather with one box and diaphragm (Plate 18), or by a breather with two boxes connected in tandem (Plate 19).

(d) /

PLATE NO.20.


— DIAGRAMMATIC SKETCH OF MODERN PITOT TUBE. —
DYNAMIC PRESSURE
h
STATIC + DYNAMIC PRESSURE
STATIC PRESSURE
THERMOMETER.
MOTION.
PIPE LINE.

(d) Measurement of Air - using Pitot Tubes.

This method is one favoured chiefly by physicists, and is indeed the standard method of the National Physical Laboratory.

Plate 20 shows diagrammatically the principle of a modern pitot tube, by which means we find the dynamic pressure of the air by balancing the static pressure against the static and dynamic pressures.

The air, by virtue of its motion, supports a column of water "h" inches high, that is, its dynamic pressure corresponds to "h" inches of water. The height of a column of air to balance "h" inches of water must next be determined.

Let H = height of column of air in feet;

d = density of 1 cubic foot of air;

h = height of water column in inches;

and 62.37 = weight of 1 cubic foot of water in lb. at $60^{o}F$..

Then $H \times d = \frac{h \times 62.37}{12}$ $\therefore H = \frac{h \times 62.37}{12d}$

Substituting this value of H in the formula,

$$V = \sqrt{2gH}$$

$$V = \sqrt{\frac{2g \times h \times 62.37}{12d}}$$

i.e., velocity in feet per second $= 18.29 \sqrt{\frac{h}{d}}$ ....(1)

Let /

Let P_1 be normal pressure, (14.7 lb.per square inch);

T_1 be normal temperature, (491.6°F.absolute);

d_1 be density at N.T.P., (.08071 lb.per cubic foot);

and P_2, T_2, and d_2 be respectively the pressure, temperature and density of the air within the pipe.

Now density $\propto \frac{1}{\text{temp.}} \propto$ pressure

$$\therefore\ d_1 = \frac{C.P_1}{T_1}$$

$$\text{i.e. } C = \frac{d_1 T_1}{P_1} = \frac{d_2 T_2}{P_2}$$

$$\therefore\ d_2 = \frac{d_1 \cdot T_1 \cdot P_2}{P_1 \cdot T_2}$$

Substitute this value of d_2 in (1) above, and:-

$$\text{velocity in feet per second} = 18.29\sqrt{\frac{P_1 T_2 h}{d_1 T_1 P_2}}$$

$$= 18.29\sqrt{\frac{P_1 h\ T_2}{P_2(491.6 \times .08071)}}$$

$$= 2.904\sqrt{\frac{P_1 h}{P_1} \cdot T_2}$$

This gives the **velocity** of the air at the point where the pitot tube is inserted, but the velocity is not uniform across the bore of the pipe. If, however, the pitot tube were inserted at various distances from the /

the pipe wall, a curve of velocities could be plotted and the point giving the mean velocity within the pipe determined.

From a knowledge of the mean velocity and of the area of the pipe, the quantity of air passing can readily be determined.

The use of pitot tubes is not common in engineer:ing practice. It should be noted, however, that air:craft employ such tubes in their air speed meters. Here the pitot tube leads to a small bellows, which causes a pointer to rotate on a semi-circular scale, calibrated in velocities of miles per hour.

See:-

"Proc. Inst.Mech.E," 1904, p.245, (R. Threlfall);

"Flow & Measurement of Air & Gases," Ch.9, (A.B.Eason);

"Treatise on Chemical Engineering," Ch.2 & 3, (J.Martin);

"Transactions of A.S.M.E.," Vol.33, p.1137; Vol.34,p.1019; Vol.34,p.1094; Vol.35,p.633; Vol.38,p.761;

"Proc. Royal Society," Ser.A., Vol.101, 1922, p.435; and

"Advisory Committee for Aeronautics," Reports and Mem:oranda No.71.

Etc., etc..

(e) Measurement of Air - using Mechanical Methods.

Mechanical methods of measuring air flow exist in many forms. We have air displacement meters, venturi meters, orifice meters, electrical meters, anemometers, etc.. /

etc.. Each of such methods has its particular and peculiar advantage, but are not to be recommended where the quantity of air can be determined by other means.

See:-

"Proc. Inst.C.E.," Vol.199, p.391 (Gibson);

"Proc. Inst.C.E.," Vol.204, p.108 (J.L.Hodgson);

"Transactions of A.S.M.E.," Vol.28, p.483; Vol.38, p.799; Vol.42, p.217;

"Engineering," Vol.107, p.261 & p.295, (J.R.Pannell);

"Compressed Air and its Machinery," Ch.5, (Plummer);

"Flow & Measurement of Air & Gases," Ch.8, (A.B.Eason);

"Elements of Thermodynamics," Ch.16, (E.M.Fernald);

"Applied Heat" adapted from Oelschläger's "Der Wärme-:ingenieur" by H. Moss, p.307-310;

Etc., etc..

DESCRIPTION OF ACTUAL TESTS.

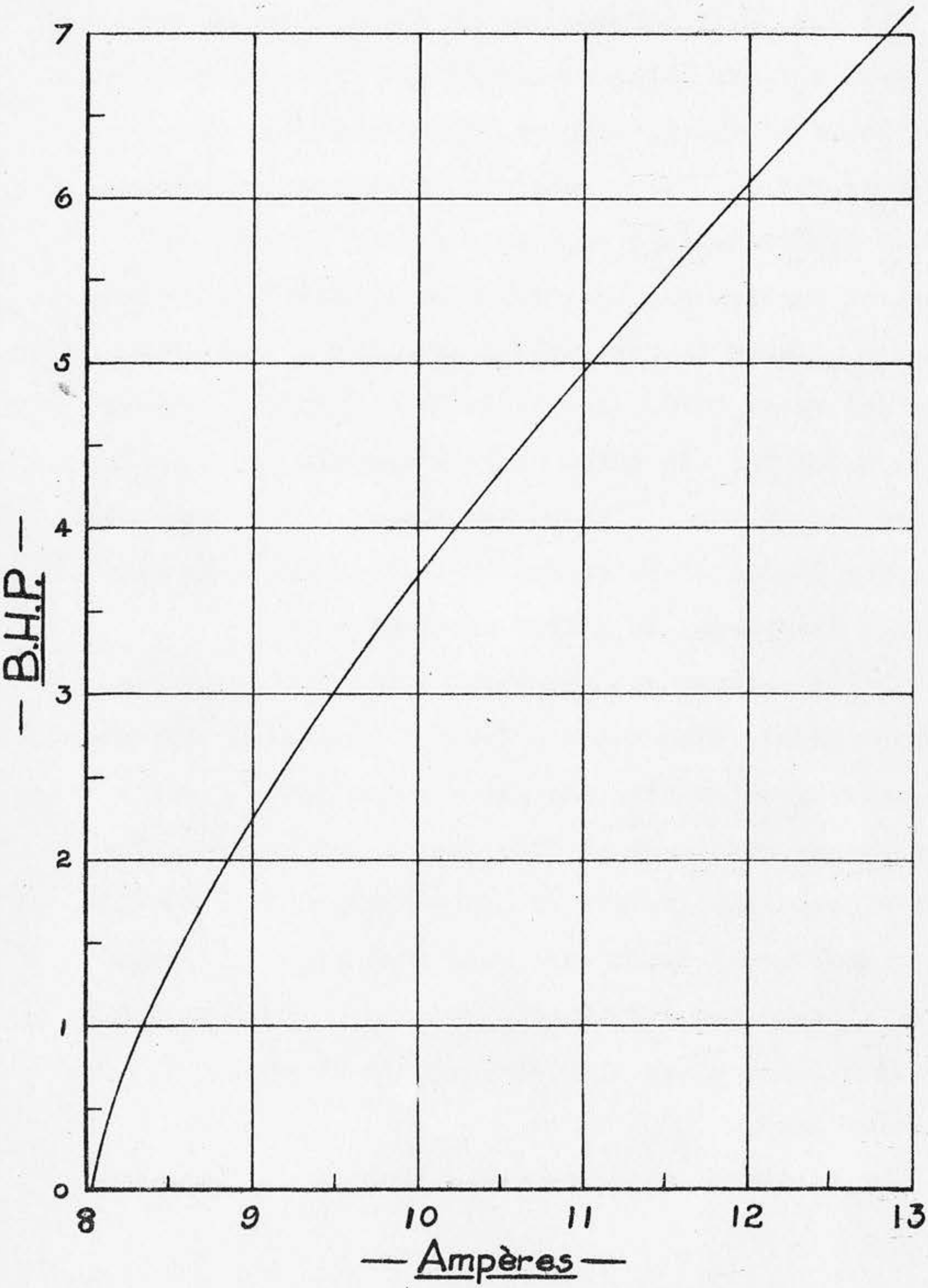
The means adopted to obtain definite results as to the effect of different clearances over a range of delivery pressures was as follows.

The L.P. cylinder was run as a single-stage com:pressor, the air being cooled in the first intercooler, which acted as an aftercooler, before passing into the anti-pulsator and hence into the gasometer for measure:ment. In order that each test should be run under identical conditions, the jacket water outlet temperat:ure was allowed to vary only a fraction of a degree above and below 22°C., and tests were confined to days during which the air inlet temperature was approximately average temperature. These two temperatures, although producing little or no effect on such a result as the free air discharge, do affect heat balances.

The compressor was run under test conditions for one hour before each test. Every test lasted half-an-hour, during which time the air suction and discharge temperatures were read at five minute intervals. The voltage, amperage, weight of jacket water and the r.p.m. of the compressor shaft were also read every five min:utes. There was sufficient test time to charge the gasometer three times with air, and to obtain six indicator cards.

The delivery pressures and clearance volumes chosen were:- /

PLATE Nº 21



were:-

with delivery pressure 0 lb. per square inch gauge,

3 tests at 4.52%, 7.03% and 9.04% clearance;

with delivery pressure 10 lb. per square inch gauge,

5 tests at 4.52%, 6.03%, 7.03%, 8.04% and 9.04% clearance;

with delivery pressures 20, 30, 40, 50, 60, 70, 80 and 90 lb. per square inch gauge,

10 tests at 4.52%, 5.02%, 5.52%, 6.03%, 6.53%, 7.03%, 7.53%, 8.04%, 8.54% and 9.04% clearance;

with delivery pressure 100 lb. per square inch gauge,

3 tests at 4.52%, 7.03% and 9.04% clearance.

To determine the brake horse power input to the compressor a rope brake was fitted round the flywheel and the slip plates removed from either side and the motor current determined in the normal way for many brake horse powers. As the voltage varied very little, it was possible to construct a graph of brake horse power against amperes, from which the brake horse power input to the compressor could be readily obtained, knowing the current required by the motor. (See Plate 21).

(Air speed through valves approximately 3,000 feet per minute.)

RESULTS DEDUCED FROM TESTS.

PLATE Nº 22

—F.A.D/STROKE—
—PLOTTED AGAINST—
—DELIVERY PRESSURE—

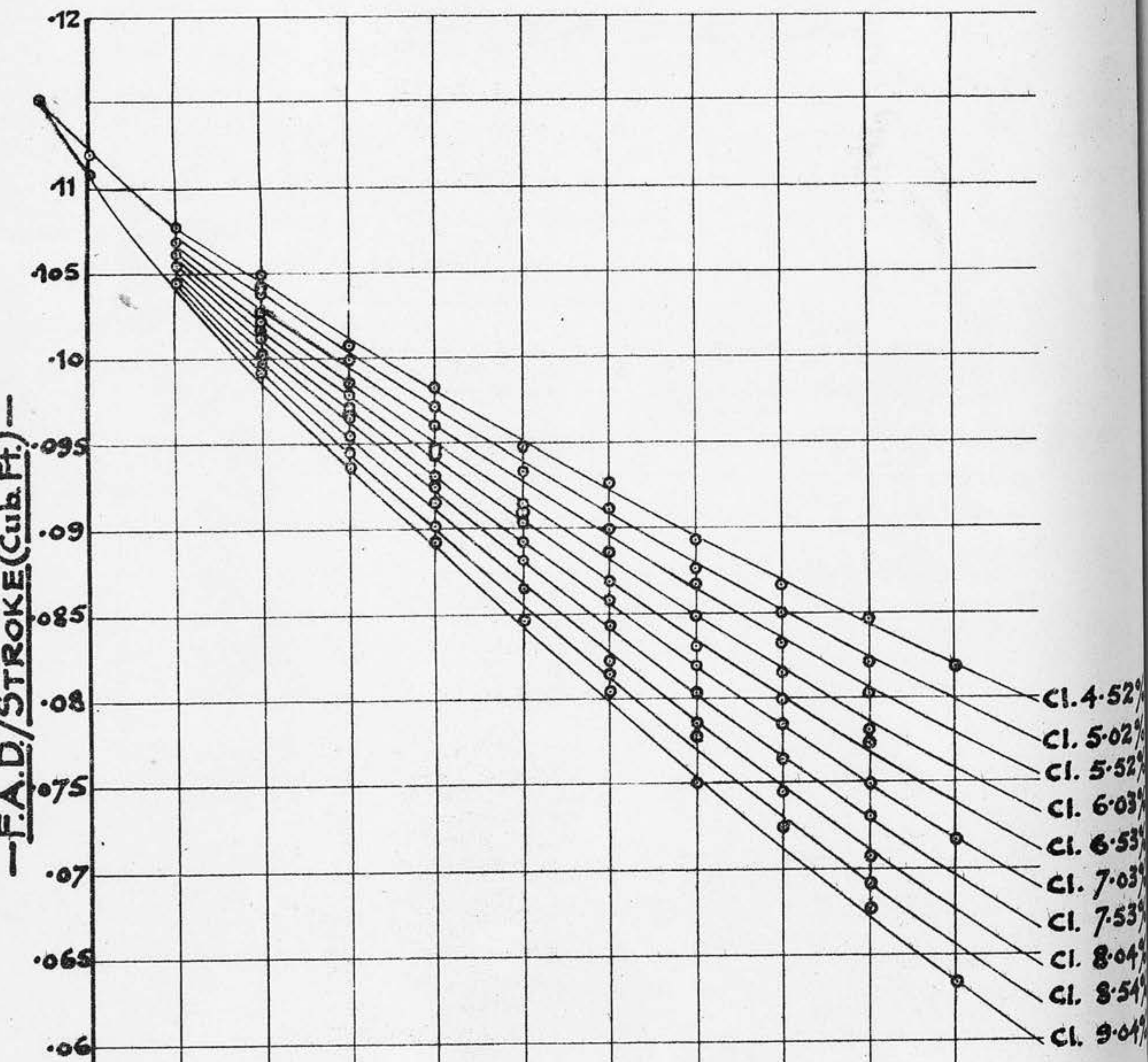
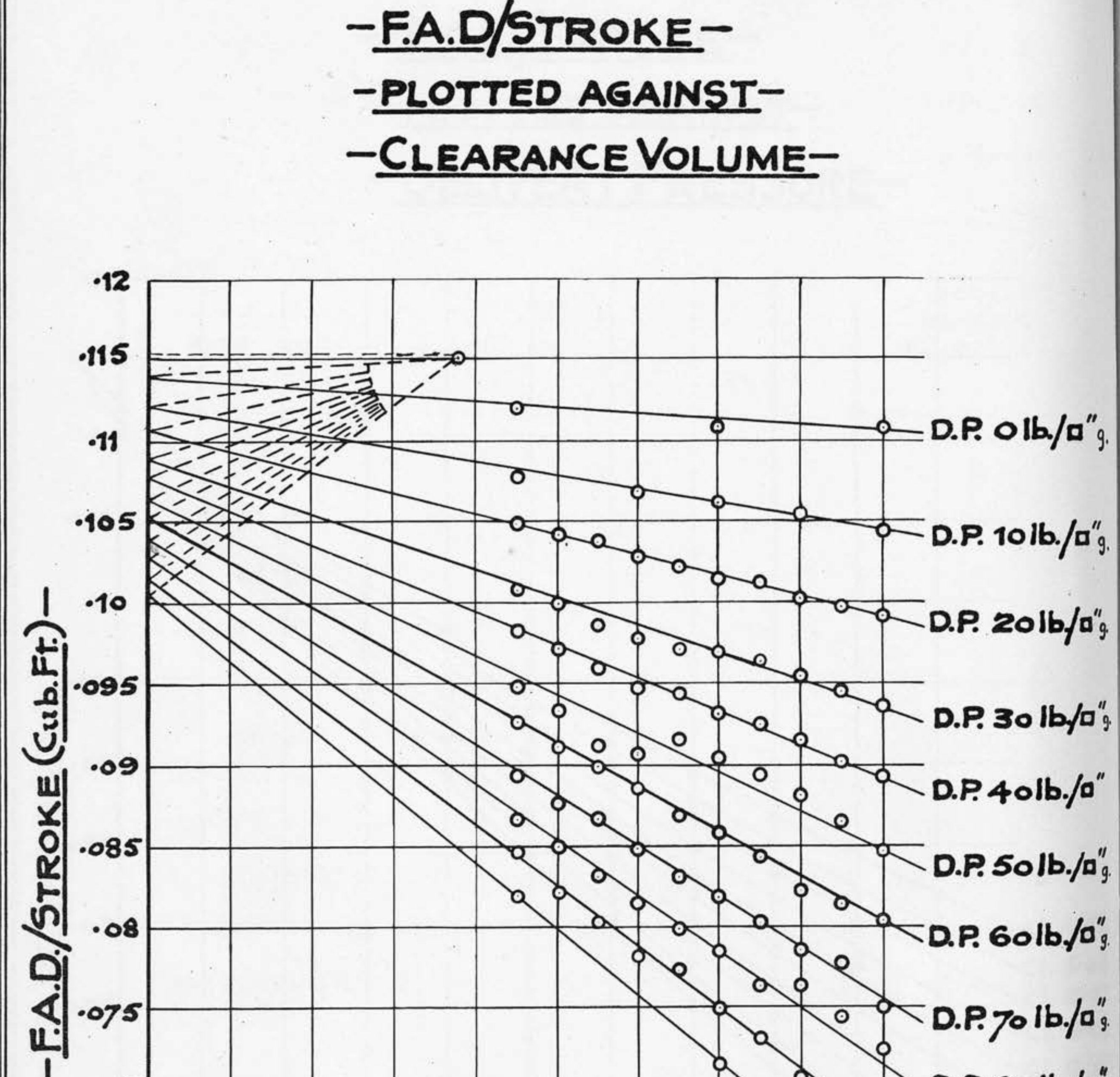


PLATE Nº 23



-F.A.D/STROKE-
-PLOTTED AGAINST-
-CLEARANCE VOLUME-
·12
·115
·11
·105
·10
·095
·09
·085
·08
·075
·07
·065
·06
·055
-F.A.D./STROKE (Cub.Ft.)-
0%
1%
2%
3%
4%
5%
6%
7%
8%
9%
-CLEARANCE VOLUME (% of Stroke Volume)-
D.P. 0 lb./□"g.
D.P. 10 lb./□"g.
D.P. 20 lb./□"g.
D.P. 30 lb./□"g.
D.P. 40 lb./□"
D.P. 50 lb./□"g.
D.P. 60 lb./□"g.
D.P. 70 lb./□"g.
D.P. 80 lb./□"g.
D.P. 90 lb./□"g.
D.P. 100 lb./□"g.

Let us commence the discussion by studying the effect of clearance on the air discharged. The foll-:owing values are taken from the more complete test results in Appendix II, pages110 and 112.

Delivery Pressure lb./sq.in.	F.A.D. per minute (cub.ft.)		Volumetric Efficiency (%)	
	Clearance 4.52%	Clearance 9.04%	Clearance 4.52%	Clearance 9.04%
0	31.33	30.94	97.29	96.12
20	29.13	27.46	91.1	86.06
40	27.12	24.68	85.31	77.56
60	25.47	22.2	80.41	69.84
80	23.84	19.91	75.19	62.87
100	22.43	17.38	71.06	54.95

Large clearance volumes result in less air being discharged, which lowers the volumetric efficiency. At low delivery pressures the difference due to clear-:ance is small, and is greatly increased with increase of pressure.

Plates 22, 23, 24 and 25 show the F.A.D. per stroke and the volumetric efficiency plotted to bases of delivery pressure and clearance volume.

The graphs in Plates 22 and 24 converge to meet at an apparent pressure of - 5 lb. per square inch. All /

PLATE Nº 24

DISCHARGE EFFICIENCY

against

DELIVERY PRESSURE

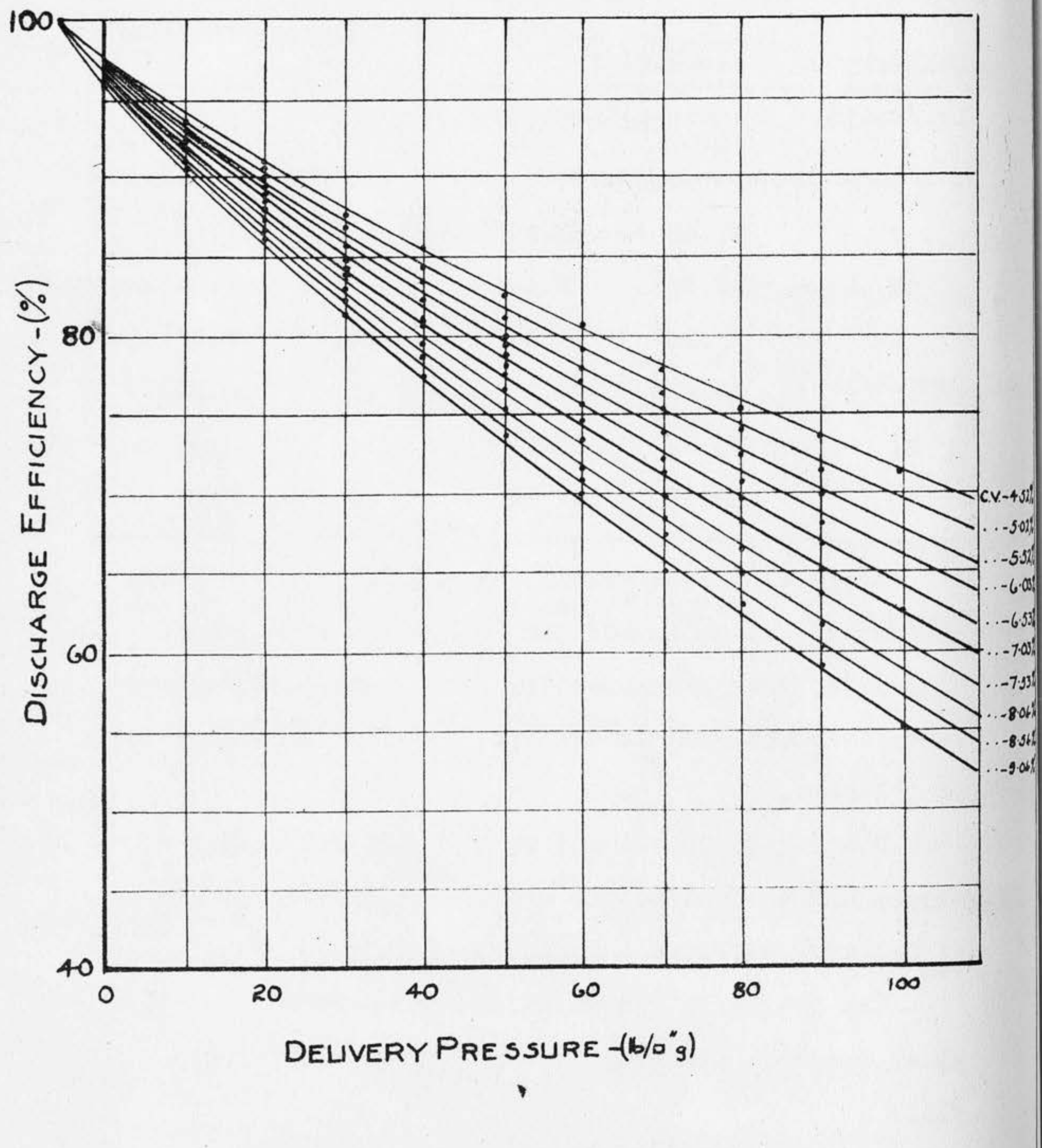
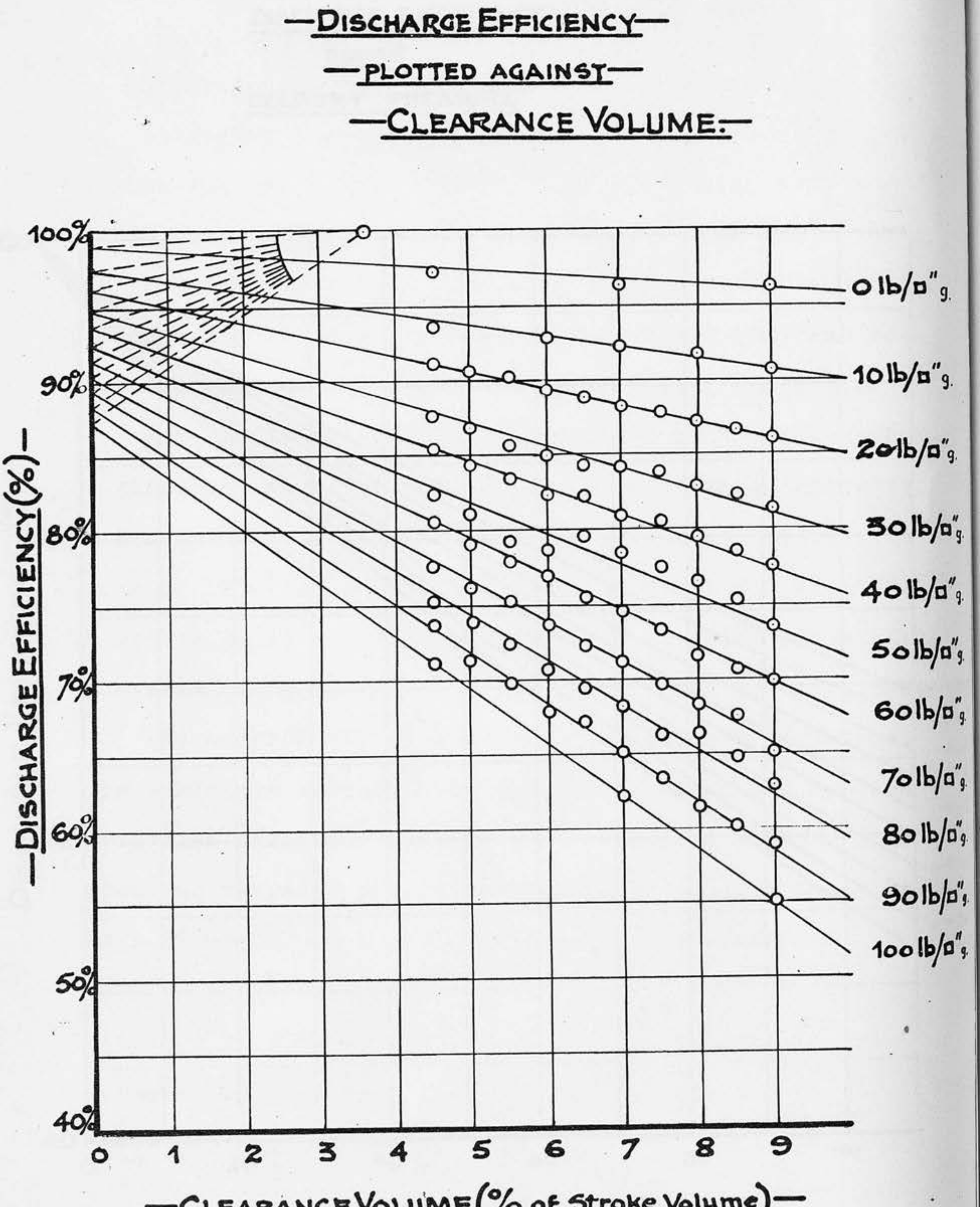


PLATE No. 25

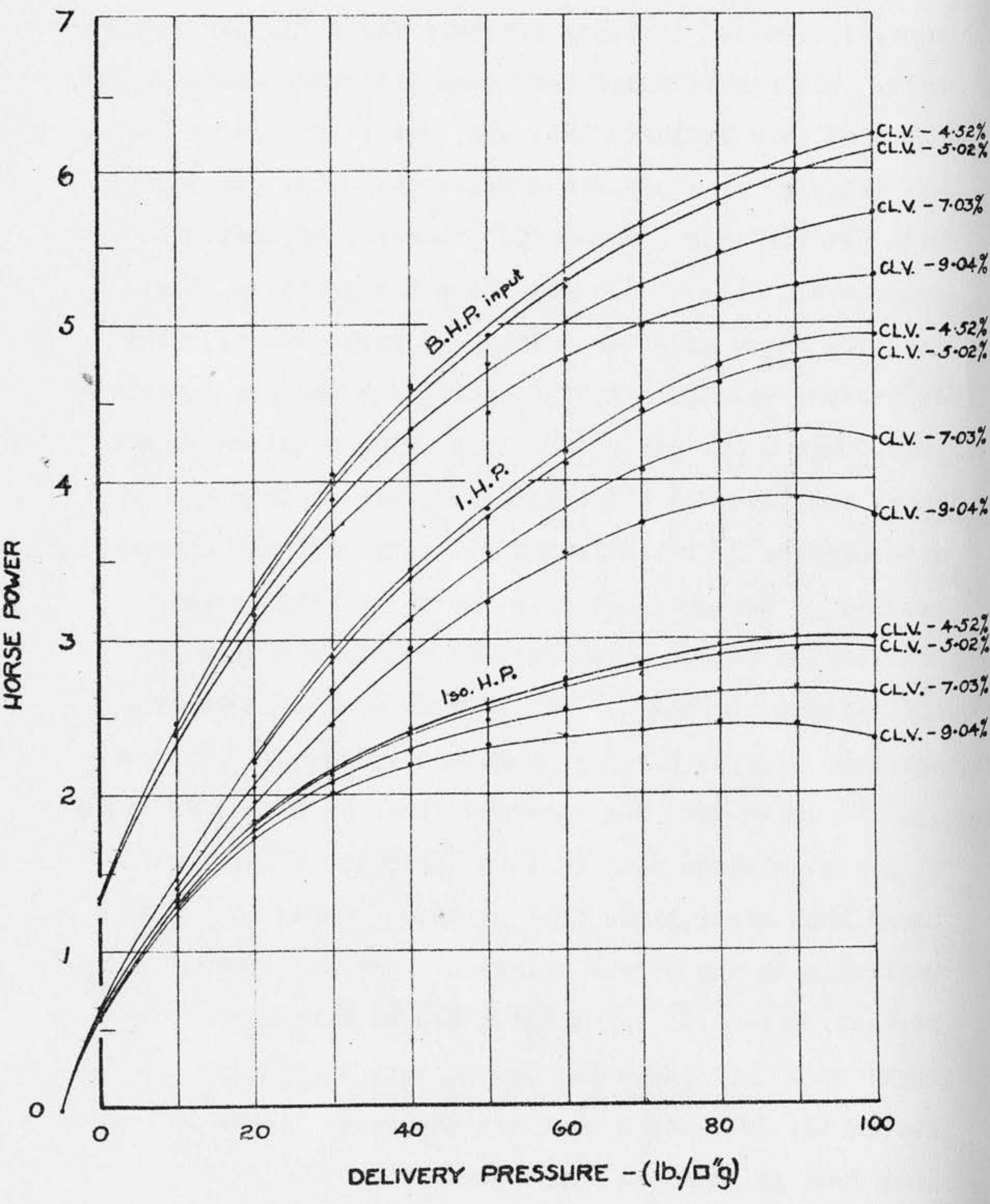

—DISCHARGE EFFICIENCY—
—PLOTTED AGAINST—
—CLEARANCE VOLUME.—
100%
90%
80%
70%
60%
50%
40%
—DISCHARGE EFFICIENCY (%)—
0 lb/□" g.
10 lb/□" g.
20 lb/□" g.
30 lb/□" g.
40 lb/□" g.
50 lb/□" g.
60 lb/□" g.
70 lb/□" g.
80 lb/□" g.
90 lb/□" g.
100 lb/□" g.
0
1
2
3
4
5
6
7
8
9
—CLEARANCE VOLUME (% of Stroke Volume)—

All the graphs in this paper which have a delivery pressure base converge in this manner, demonstrating that although the receiver pressure gauge indicated zero, the actual delivery pressure was 5 lb. per square inch. It may be noted here than indicator cards were taken at zero delivery pressure, and it was found that the delivery pressure was momentarily 5 lb. per square inch greater than atmospheric pressure, so that the compressor, when discharging into the receiver, the pressure gauge of which pointed to zero, was actually delivering air against a pressure of 5 lb. per square inch, this 5 lb. per square inch being required to over-:come the force in the discharge valve springs and to overcome the friction in the delivery pipe and after-:cooler. The pressure drop due to the difference between the actual discharge and receiver temperatures will also be included. To prevent confusion delivery pressure will be taken as meaning the receiver pressure.

Assuming that the straight lines in Plates 23 and 25 are continuous back to zero clearance volume, it is found that every line, when produced, meets at a point in line with the stroke volume. Were the compressor perfect, this pole point which has been thrown to the right of the ordinate for convenience in drawing, would lie on the "y" axis having all discharge curves radiat-:ing from it into the first quadrant.

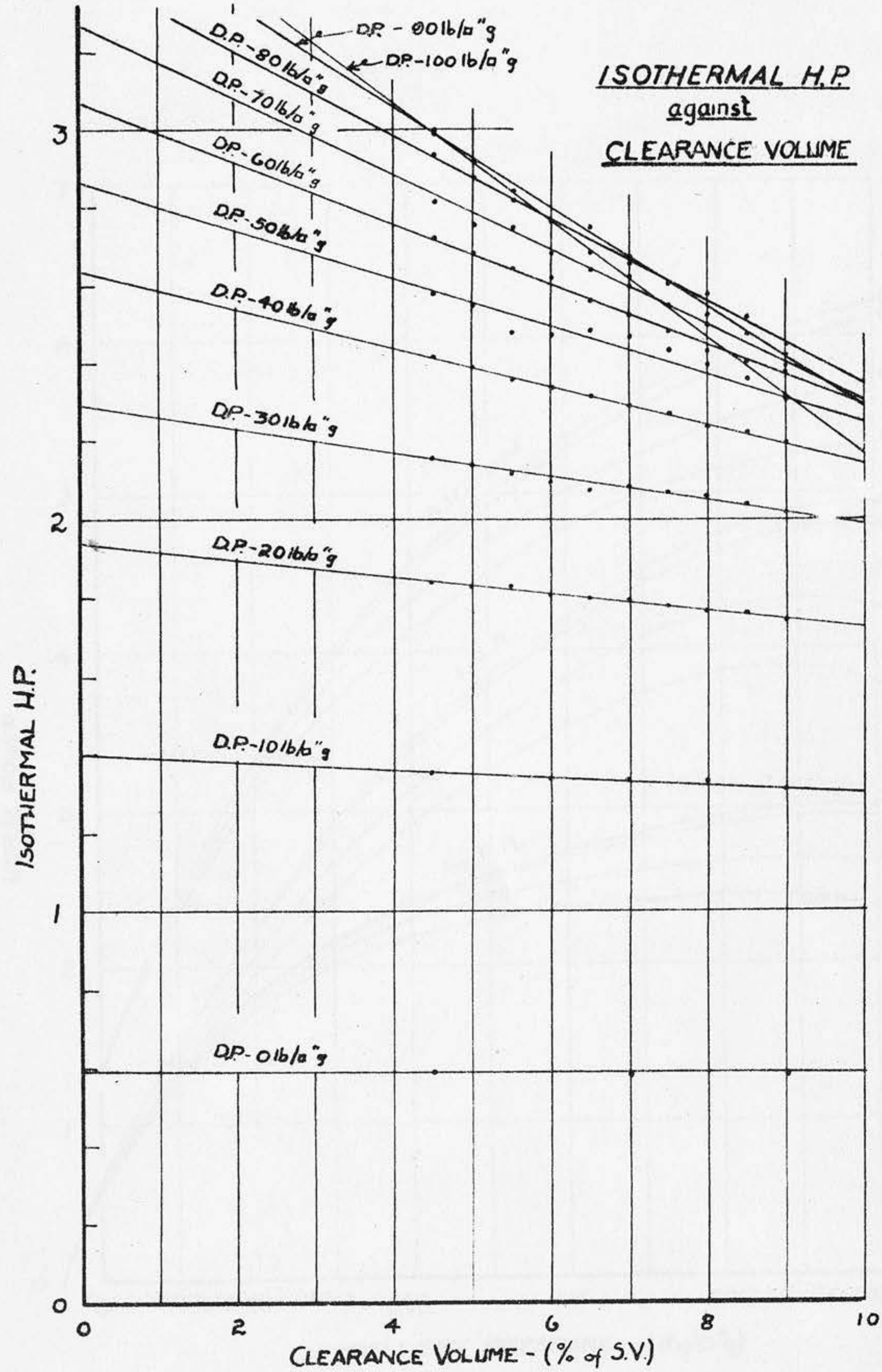
The /

PLATE Nº 26



HORSE POWER
DELIVERY PRESSURE - (lb./□"g)
0
1
2
3
4
5
6
7
0
20
40
60
80
100
B.H.P. input
I.H.P.
Iso. H.P.
CL.V. - 4·52%
CL.V. - 5·02%
CL.V. - 7·03%
CL.V. - 9·04%
CL.V. - 4·52%
CL.V. - 5·02%
CL.V. - 7·03%
CL.V. - 9·04%
CL.V. - 4·52%
CL.V. - 5·02%
CL.V. - 7·03%
CL.V. - 9·04%

PLATE Nº 27


ISOTHERMAL H.P. against CLEARANCE VOLUME
D.P. - 90 lb/□"g
D.P. - 100 lb/□"g
D.P. - 80 lb/□"g
D.P. - 70 lb/□"g
D.P. - 60 lb/□"g
D.P. - 50 lb/□"g
D.P. - 40 lb/□"g
D.P. - 30 lb/□"g
D.P. - 20 lb/□"g
D.P. - 10 lb/□"g
D.P. - 0 lb/□"g
ISOTHERMAL H.P.
0
1
2
3
0
2
4
6
8
10
CLEARANCE VOLUME - (% of S.V.)

The isothermal horse power* can be expressed by $\dfrac{P_1V_1 \log_e \frac{P_2}{P_1}}{33,000}$ which is for any one delivery pressure a constant times the F.A.D. per minute in cubic feet, and since the free air discharge decreases with increased clearance, the isothermal horse power will also be de:creased. At low delivery pressures the difference in the free air discharged is not so marked as at high pressures, so that the higher the pressure the greater is the difference in the isothermal horse powers as the clearance is increased. If the pressure compression ratio is increased until no air is discharged, i.e., when the clearance air after expansion completely fills the cylinder, there will be no isothermal horse power. There must exist two pressures at which the isothermal horse power is zero, viz., zero delivery pressure and that pressure at which no air is delivered. The curves will therefore start at zero, reach a maximum and con:tinue downwards to meet the delivery pressure abscissa. (See Plates 26 and 27 which show the isothermal horse power suitably plotted and also show the difference due to clearance increasing with increase of delivery pressure.)

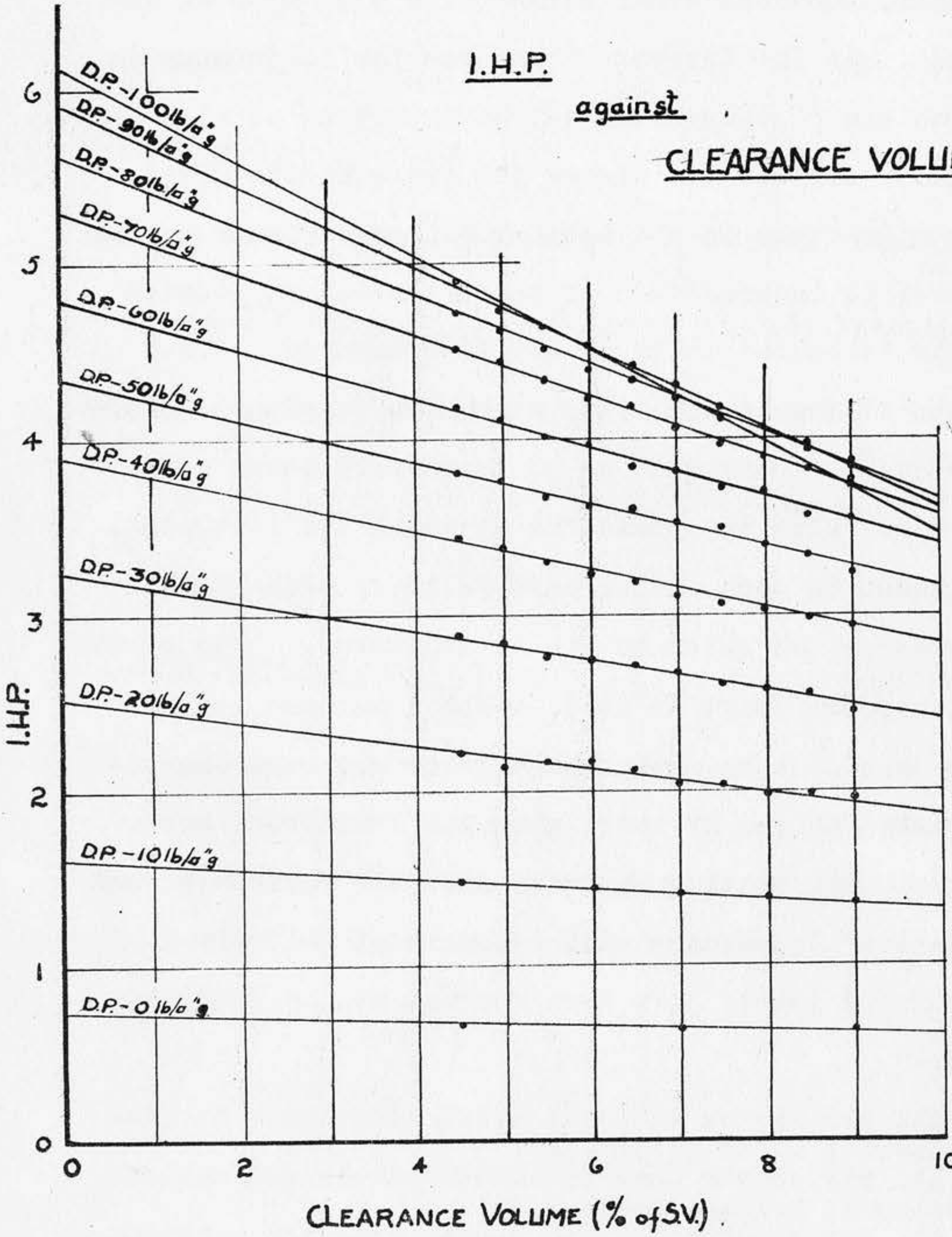
Turning points have been reached at the following values:-

At 96 lb. per square inch and 4.52% clearance, maximum isothermal horse power - 3;
At 94 lb. per square inch and 5.02% clearance, maximum isothermal horse power - 2.96;
At 90 lb. per square inch and 7.03% clearance, maximum isothermal horse power - 2.67;
At 85 lb. per square inch and 9.04% clearance, maximum isothermal horse power - 2.44.

The /

* See Appendix II, page 113.

PLATE Nº 28


I.H.P. against CLEARANCE VOLUME
D.P.-100 lb/□"g
D.P.-90 lb/□"g
D.P.-80 lb/□"g
D.P.-70 lb/□"g
D.P.-60 lb/□"g
D.P.-50 lb/□"g
D.P.-40 lb/□"g
D.P.-30 lb/□"g
D.P.-20 lb/□"g
D.P.-10 lb/□"g
D.P.-0 lb/□"g
I.H.P.
0
1
2
3
4
5
6
0
2
4
6
8
10
CLEARANCE VOLUME (% of S.V.)

The indicated horse power* is somewhat similar to the isothermal horse power in that maximum values are attained, no power being indicated at the compressor's maximum delivery pressure. Similarly altered clearance at the higher pressures increases the difference between the indicated horse powers more than at the lower pressures. The clearance air does work during expansion, thereby reducing the indicated horse power when the clearance is increased. (See Plates 26 and 28.)

The maximum indicated horse powers obtained were:-

4.93 at 98 lb. per square inch and 4.52% clearance;
4.8 at 96 lb. per square inch and 5.02% clearance;
4.32 at 91 lb. per square inch and 7.03% clearance; and
3.91 at 87 lb. per square inch and 9.04% clearance.

The /

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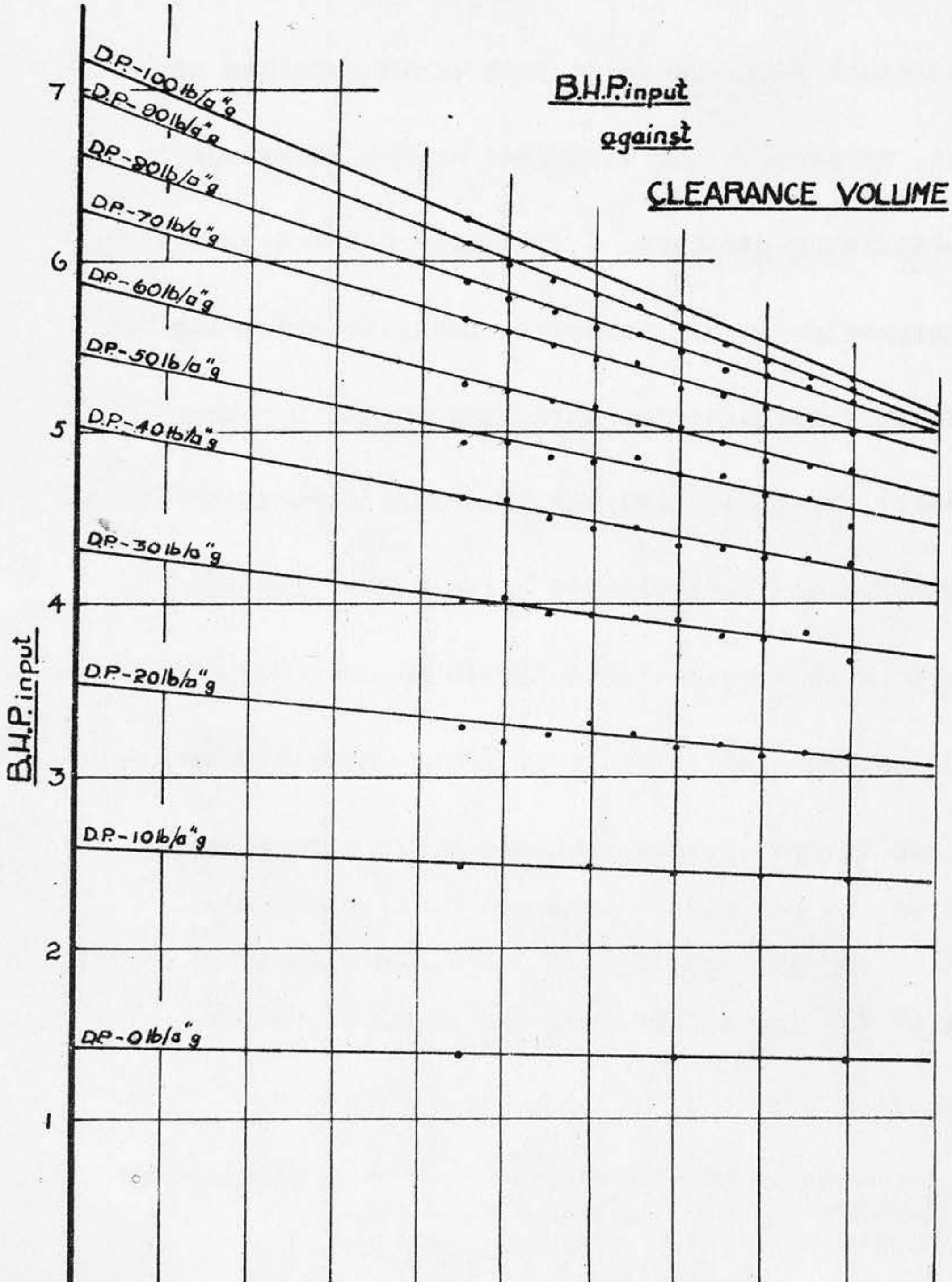
* See Appendix II, page 114.

PLATE Nº 29



B.H.P. input against CLEARANCE VOLUME
D.P.-100 lb/□"g
D.P.-90 lb/□"g
D.P.-80 lb/□"g
D.P.-70 lb/□"g
D.P.-60 lb/□"g
D.P.-50 lb/□"g
D.P.-40 lb/□"g
D.P.-30 lb/□"g
D.P.-20 lb/□"g
D.P.-10 lb/□"g
D.P.-0 lb/□"g
B.H.P. input
0
1
2
3
4
5
6
7
0
2
4
6
8
10
CLEARANCE VOLUME (% of S.V.)

The brake horse power input,* on the other hand, is a positive quantity at zero delivery pressure as power is required to overcome friction. Like the isothermal and indicated horse powers, less power is necessary when the clearance is large, considerable work being recovered from the expanding air. The higher the delivery pressure the more is the work returned, giving a wider divergence of the brake horse power input curves as the delivery pressure increases. Unlike the two previous powers, however, the brake horse power input rises to a maximum and stays there right up to the maximum delivery pressure. (See Plates 26 and 29.)

The following table lists several delivery pressure and clearance volume combinations at similar brake horse powers:-

B.H.P. /

---

* See Appendix II, page 115.

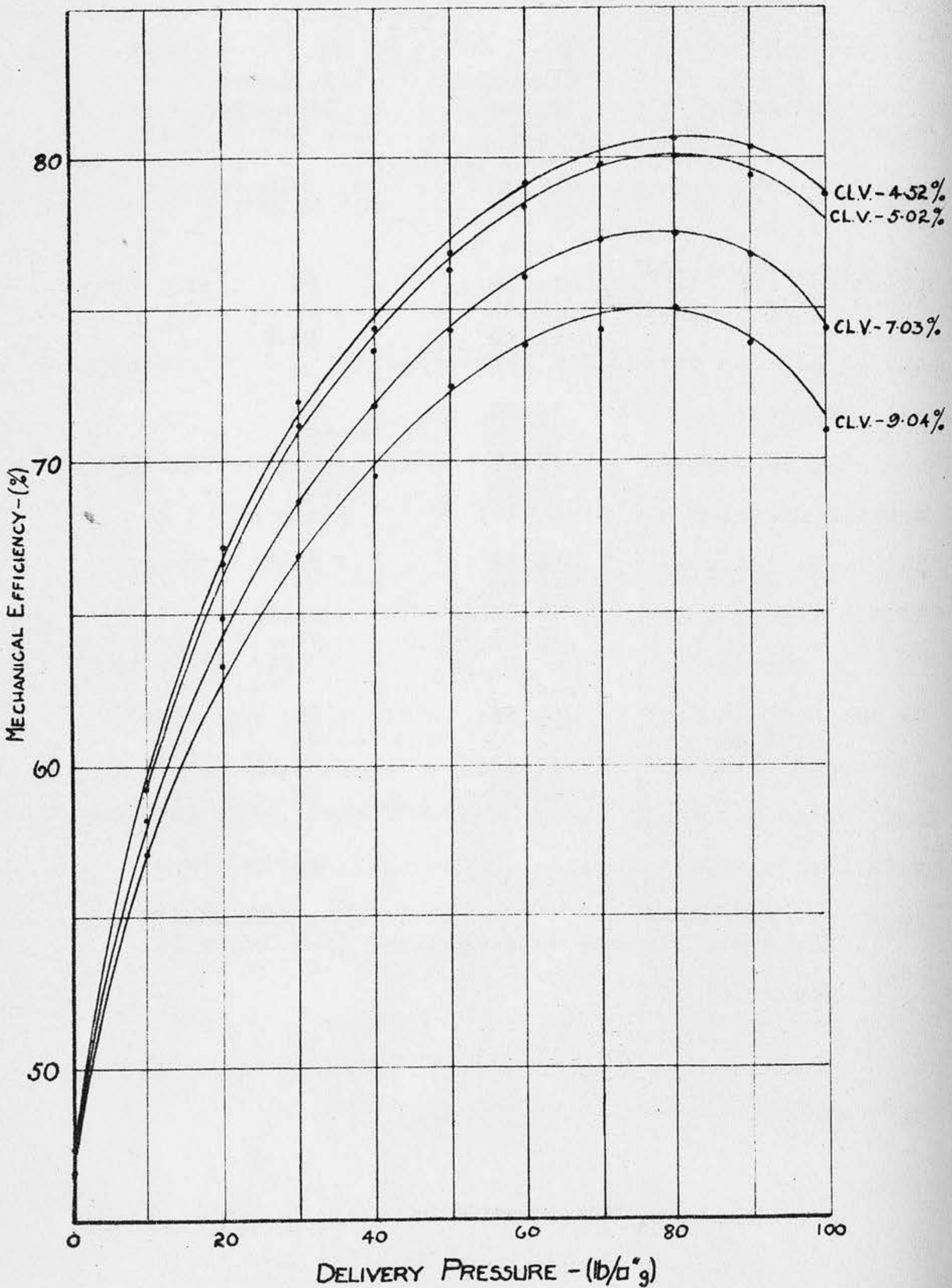
B.H.P. Input	Clearance Volume (%)	Delivery Pressure (lb. per sq.inch)
2	(4.52	5.5
	(9.04	6.5
3	(4.52	16
	(7.03	17
	(9.04	18.5
4	(4.52	30
	(5.02	31
	(7.03	33
	(9.04	37
5	(4.52	50.7
	(5.02	52
	(7.03	59.5
	(9.04	70
5.25	(4.52	57
	(5.02	60
	(7.03	70.5
	(9.04	92

The above figures were obtained from Plate 26.

The /

PLATE Nº 30

MECHANICAL EFFICIENCY against DELIVERY PRESSURE.



The relationships between the various powers are

$\frac{\text{indicated horse power}}{\text{brake horse power input}}$ which is the mechanical effic-:iency;

$\frac{\text{isothermal horse power}}{\text{indicated horse power}}$ which is the compression effic-:iency; and

$\frac{\text{isothermal horse power}}{\text{brake horse power input}}$ which is the overall efficiency

and equals the product of the mechanical and compression efficiencies.

The mechanical efficiency is zero at zero and the maximum delivery pressures for at those pressures no power is indicated and the power to drive the compressor is a definite quantity.

At low delivery pressures the mechanical efficiency* is low owing to the horse power input being not greatly in excess of the power required to overcome friction, and, up to a certain limit, the efficiency will increase with the delivery pressure. The brake horse power input did not reach a maximum within the test limits chosen, but the indicated horse power did, so that there must be some pressure at which the mechanical efficiency is greatest. The curves of mechanical efficiency /

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* See Appendix II, page 116.

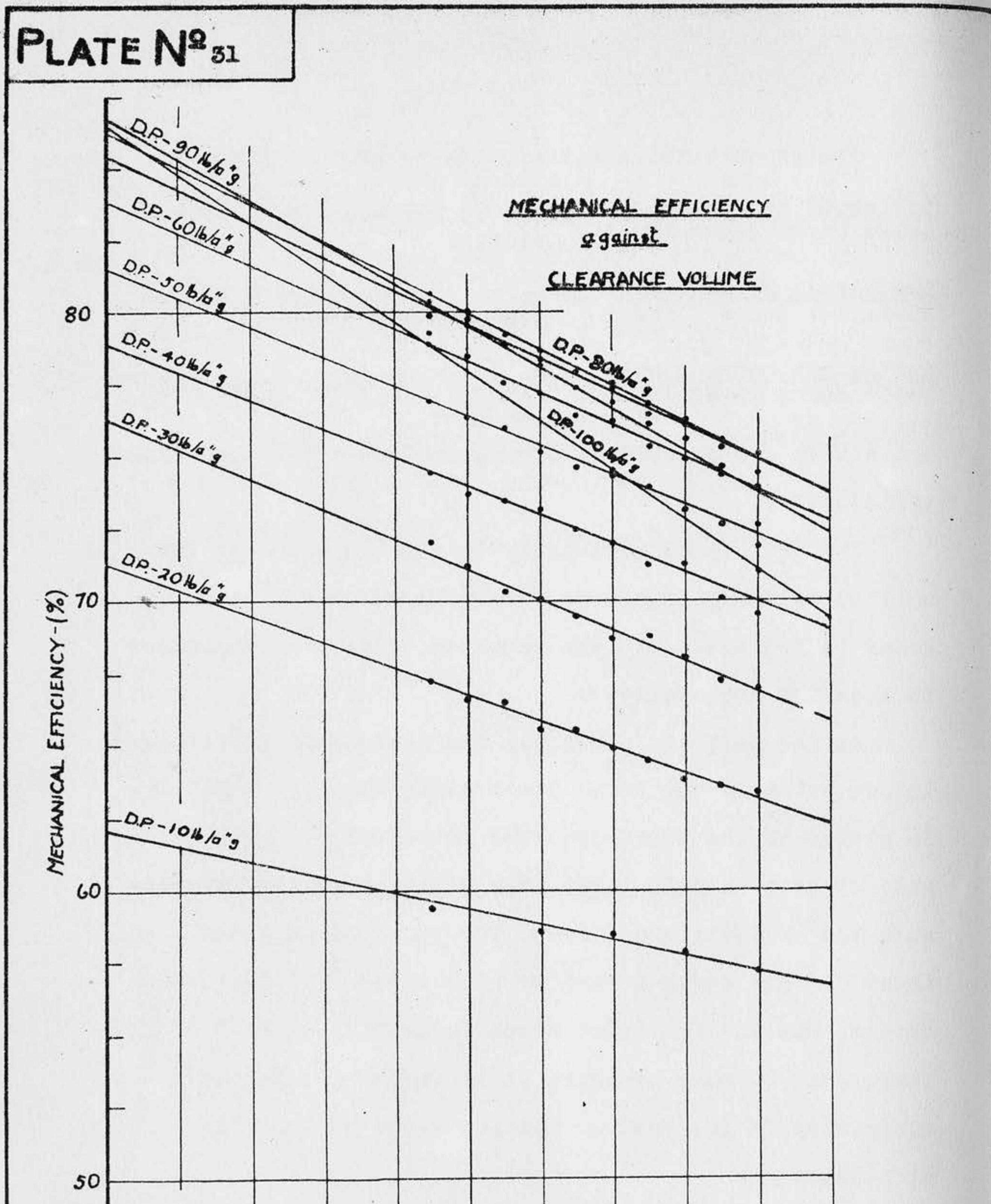


PLATE Nº 31
MECHANICAL EFFICIENCY
against
CLEARANCE VOLUME
D.P. - 90 lb/□" g
D.P. - 60 lb/□" g
D.P. - 50 lb/□" g
D.P. - 40 lb/□" g
D.P. - 30 lb/□" g
D.P. - 20 lb/□" g
D.P. - 10 lb/□" g
D.P. - 0 lb/□" g
D.P. - 80 lb/□" g
D.P. - 100 lb/□" g
80
70
60
50
0
2
4
6
8
10
MECHANICAL EFFICIENCY - (%)
CLEARANCE VOLUME - (% of S.V.)

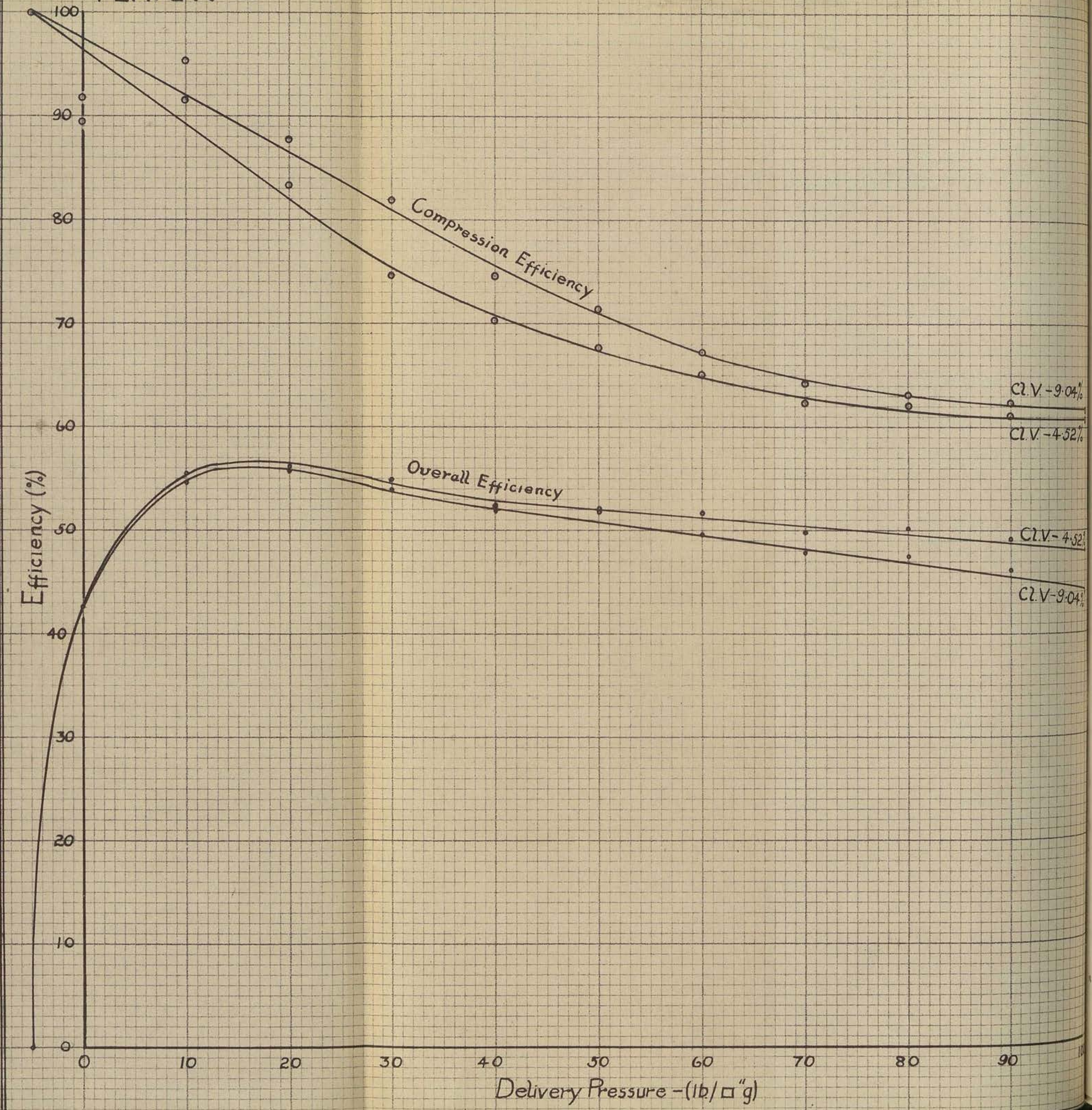
efficiency, therefore, commence at zero delivery pressure, rise to a maximum, and drop to meet the ab-:scissa at the compressor's maximum discharge pressure, which pressure will rise when the clearance is reduced. (See Plates 30 and 31, which show that small clearance volumes give highest mechanical efficiencies.)

The following table proves conclusively that the smaller the clearance the better is the mechanical efficiency:-

Delivery Pressure (lb.per sq.in.)	Clearance Volume (%)	Maximum Mechanical Efficiency (%)
82.5	4.52	80.7
82	5.02	80.1
79	7.03	77.5
77	9.04	75

Were /

PLATE Nº 32



Compression Efficiency
Overall Efficiency
Cl.V. - 9·04%
Cl.V. - 4·52%
Cl.V. - 4·52%
Cl.V. - 9·04%
Efficiency (%)
Delivery Pressure - (lb/□"g)
100
90
80
70
60
50
40
30
20
10
0
0
10
20
30
40
50
60
70
80
90

Were the compression efficiency constant, the above delivery pressures would be the compressor's best working pressures, as the overall efficiency would then be greatest.

The compression efficiency*, however, is by no means constant. (See Plate 32.) It is unity at both zero and the maximum delivery pressure and is highest near these two pressures, a minimum value lying between them. (Reconsult Plate 26, which shows the indicated horse power curves lying very close to the isothermal horse power curves at low delivery pressures and diverging as the pressure increases. Beyond the points of maximum power they converge to meet ultimately at the point of maximum delivery pressure.) There is a gain in compression efficiency with increased clearance, and a minimum value of 61% has been reached at 100 lb. per square inch delivery pressure and 4.52% clearance.

At low delivery pressures the compression efficiency is erratic due to incomplete knowledge of the exact delivery pressure for use in the isothermal horse power formula. As previously mentioned, it was found that the actual delivery pressure was some 5 lb. per square inch in excess of the receiver pressure. Unless this figure /

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* See Appendix II, page 117.

figure is known to about two places of decimals, errors exist in the isothermal horse powers at very low del:ivery pressures, which errors are greatly increased in the compression efficiency, since no light-spring indic:ator card is accurate. With increase of delivery pressure the error disappears.

The best compressor would be one in which the com:pression work is a minimum and the expansion work a maximum, i.e., when the compression and expansion are isothermal. Against this adiabatic expansion of the clearance air has its advantages; it increases the volumetric efficiency and lowers the air temperature at the end of the suction stroke. The average compression exponents for the compressor under test are 1.36 (clearance, 4.52%) and 1.34 (clearance 9.04%) and there is a tendency for the value to fall with increase of delivery pressure, e.g.,

Delivery Pressure (lb./sq.in.)	Compression Exponent	
	Clearance 4.52%	Clearance 9.04%
30	1.365	1.345
60	1.36	1.34
90	1.355	1.34

Increased /

Increased clearance lowers the value of the com-:pression exponent, since there is a larger area for the extraction of heat. Less work will therefore be indic-:ated and the compression efficiency will be increased.

The overall efficiency*, on the other hand, is zero at zero and the compressor's maximum delivery pressure, and rises to a maximum at that particular pressure at which the product of the mechanical and compression efficiencies is greatest. (See Plate 32.) The overall efficiency rises very quickly between zero and 10 lb. per square inch, owing to the rapid rise in mechanical efficiency and the relatively slow fall in compression efficiency. It is greatest between 10 and 30 lb. per square inch, which are the compressor's most economical working pressure limits.

Whatever may be the discharge pressure, the over-:all efficiency is highest when the clearance is small, although up to 45 lb. per square inch the effect of clearance could be neglected,as the difference is but one per cent. for the large increase in clearance 4.52 - 9.04%. Beyond this pressure clearance must needs be considered. (The difference at 100 lb. per square inch is 3.8% for the same clearance range.) Further, an increase in delivery pressure is not marked by a sudden drop in efficiency. The overall efficiency only drops from 54.5% to 48% between 30 and 100 lb. per square inch (clearance 4.52%), and from 53.7% to 44.2% between 30 and 100 lb. per square inch (clearance 9.04%), the diff-:erence with the larger clearance being the greater.

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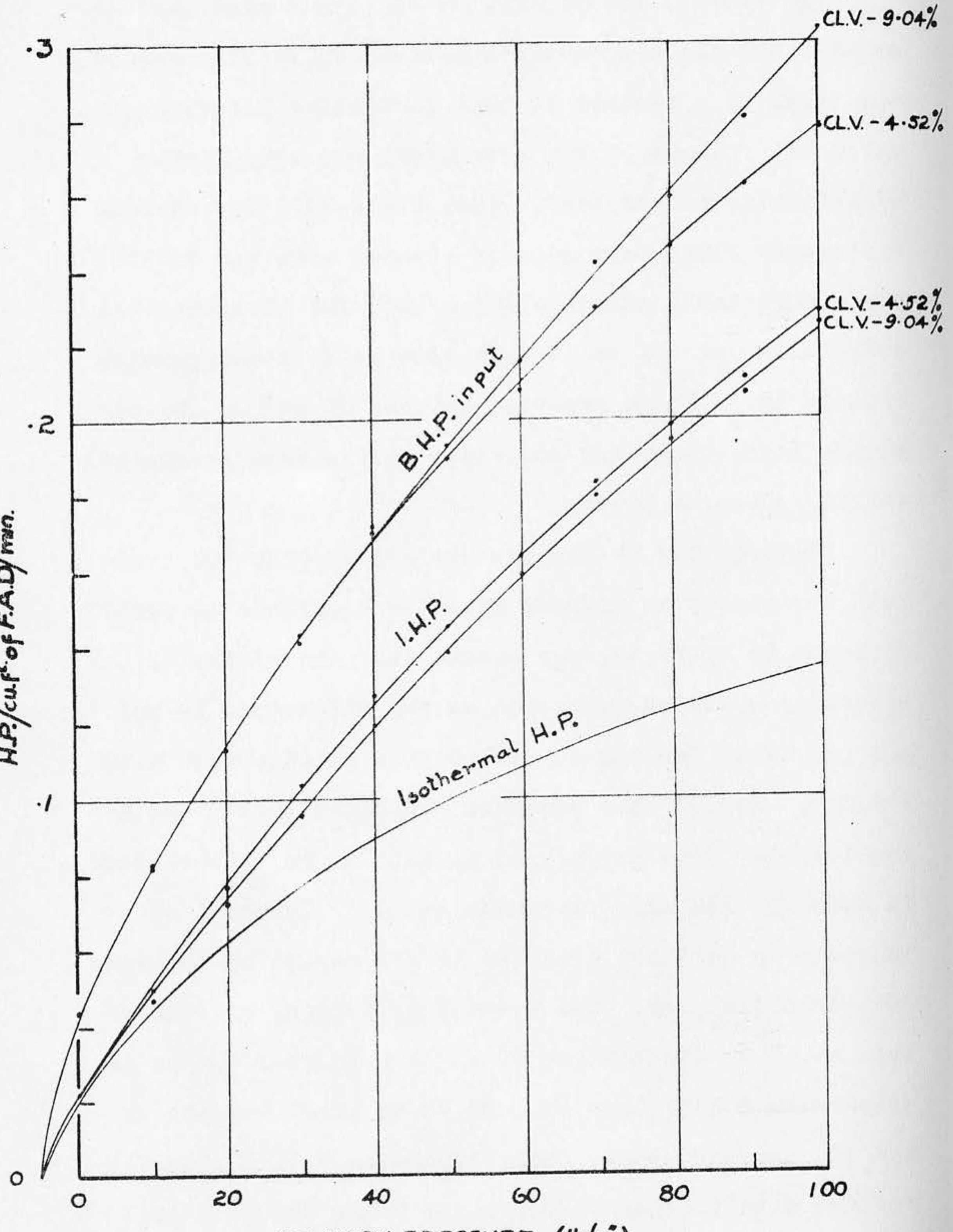
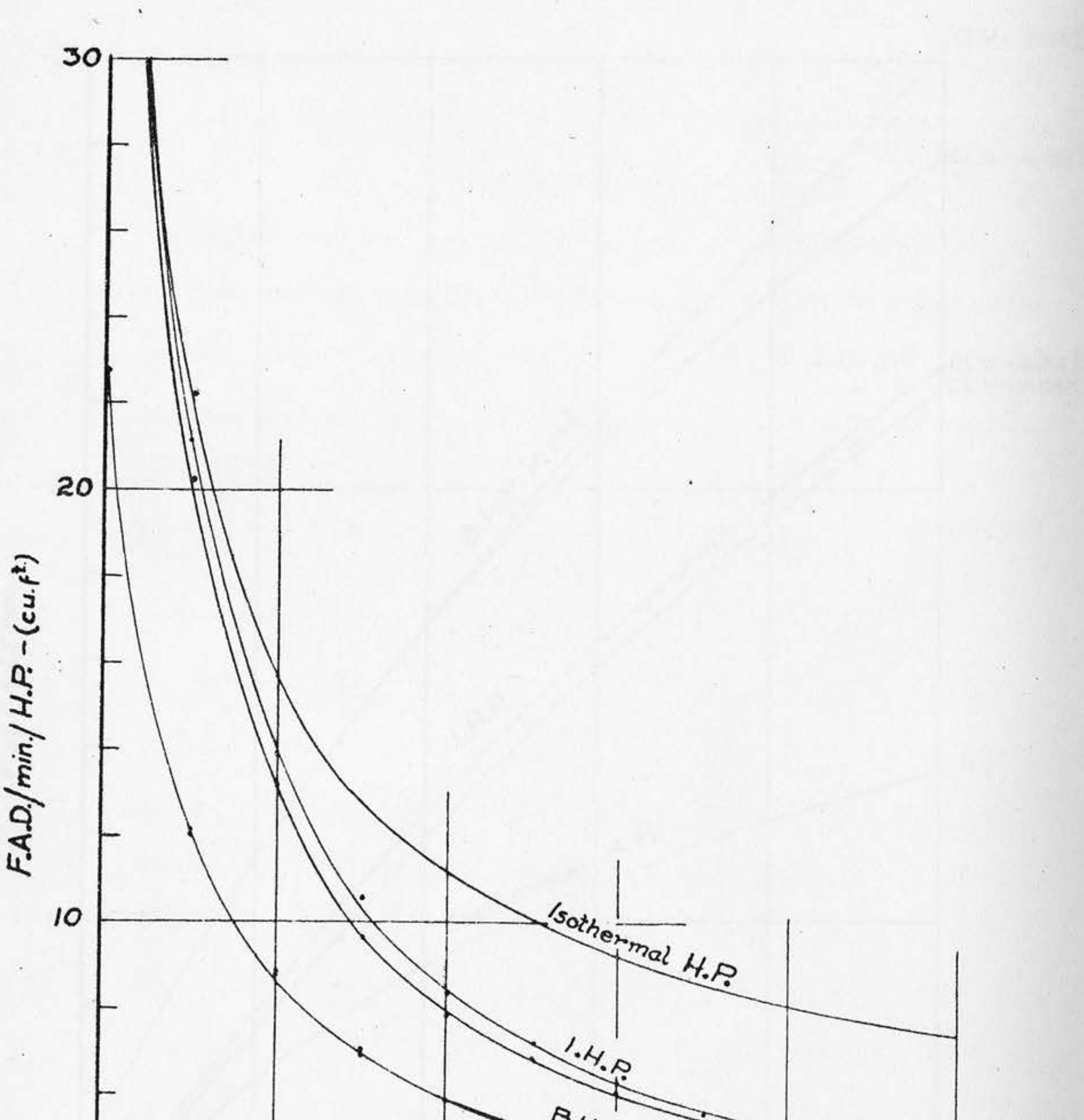
* See Appendix II, page 118.

PLATE No 33



CL.V. - 9·04%
CL.V. - 4·52%
CL.V. - 4·52%
CL.V. - 9·04%
B.H.P. input
I.H.P.
Isothermal H.P.
·3
·2
·1
0
0
20
40
60
80
100
H.P./cu.ft. of F.A.D./min.
DELIVERY PRESSURE - (lb/□"g)

PLATE No. 34


30
20
10
0
0
20
40
60
80
100
F.A.D./min./H.P. - (cu.ft.)
DELIVERY PRESSURE - (lb/□"g.)
Isothermal H.P.
I.H.P.
B.H.P.
CL.V.-9·04%
CL.V.-4·52%
CL.V.-4·52%
CL.V.-9·04%

Since cylinder clearance does not affect the iso-:thermal horse power per cubic foot of F.A.D. per minute, it is necessary to consider its effect on the brake in-:put and indicated horse powers per cubic foot of F.A.D. per minute and their reciprocals.

The following values are typical of the results obtained and are shown graphically interpreted on Plates 33, 34, 35 and 36.

Delivery Pressure (lb./sq.in.)	B.H.P.input per cub.ft. of F.A.D. per minute *		F.A.D./min. per B.H.P. input (cub.ft.) *	
	Clearance 4.52%	Clearance 9.04%	Clearance 4.52%	Clearance 9.04%
0	.0441	.0439	22.7	22.75
20	.1132	.1138	8.82	8.79
40	.1705	.172	5.865	5.82
60	.208	.216	4.81	4.63
80	.246	.259	4.065	3.86
100	.2775	.305	3.605	3.28

At zero delivery pressure clearance has no effect on the brake horse power input per cubic foot of F.A.D. per minute; there should exist, therefore, some pressure below which the effect of clearance is of no practical consequence. All the graphs up to and including 40 lb. per /

* See Appendix II, pages 119 and 120.

PLATE No 35

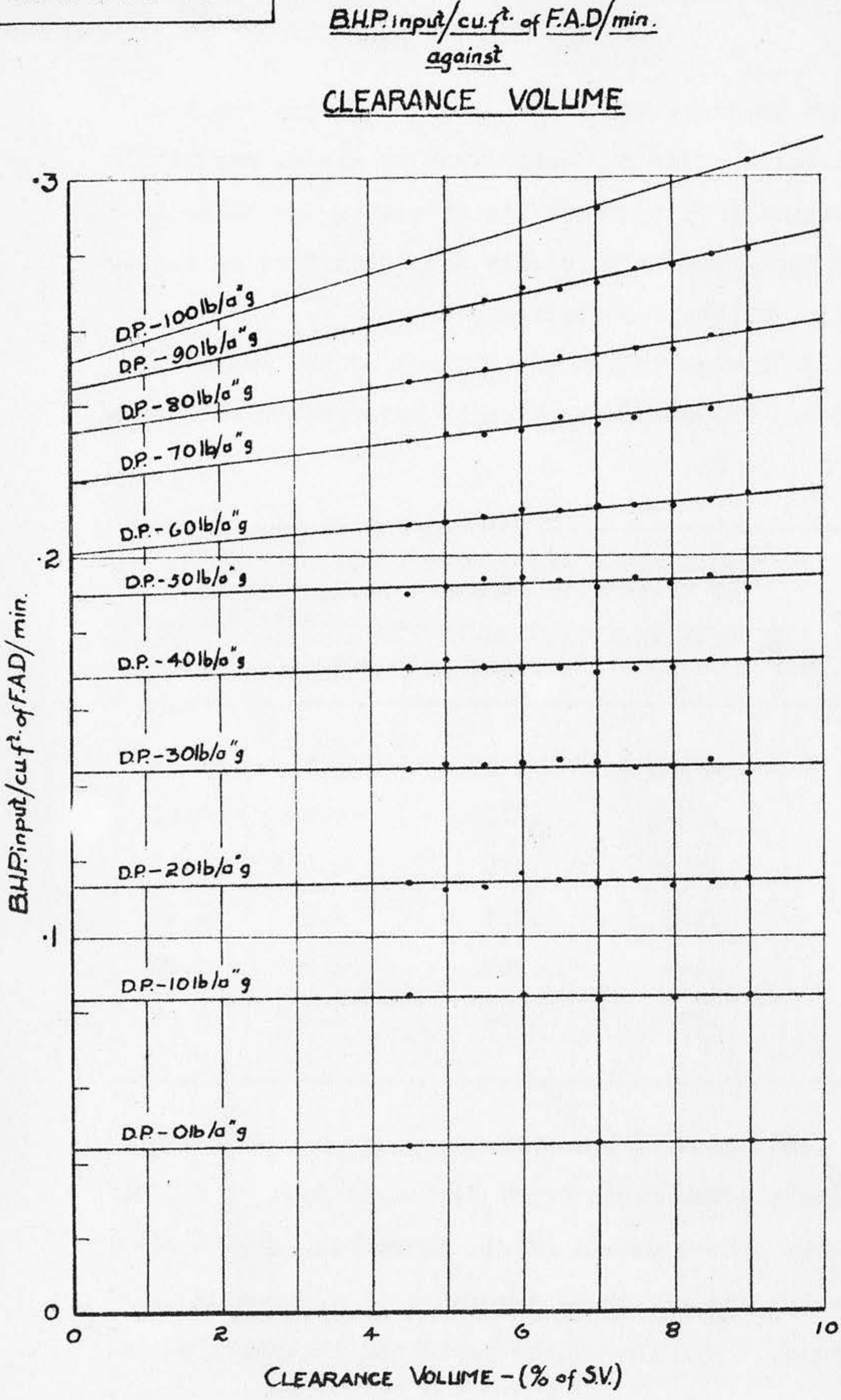
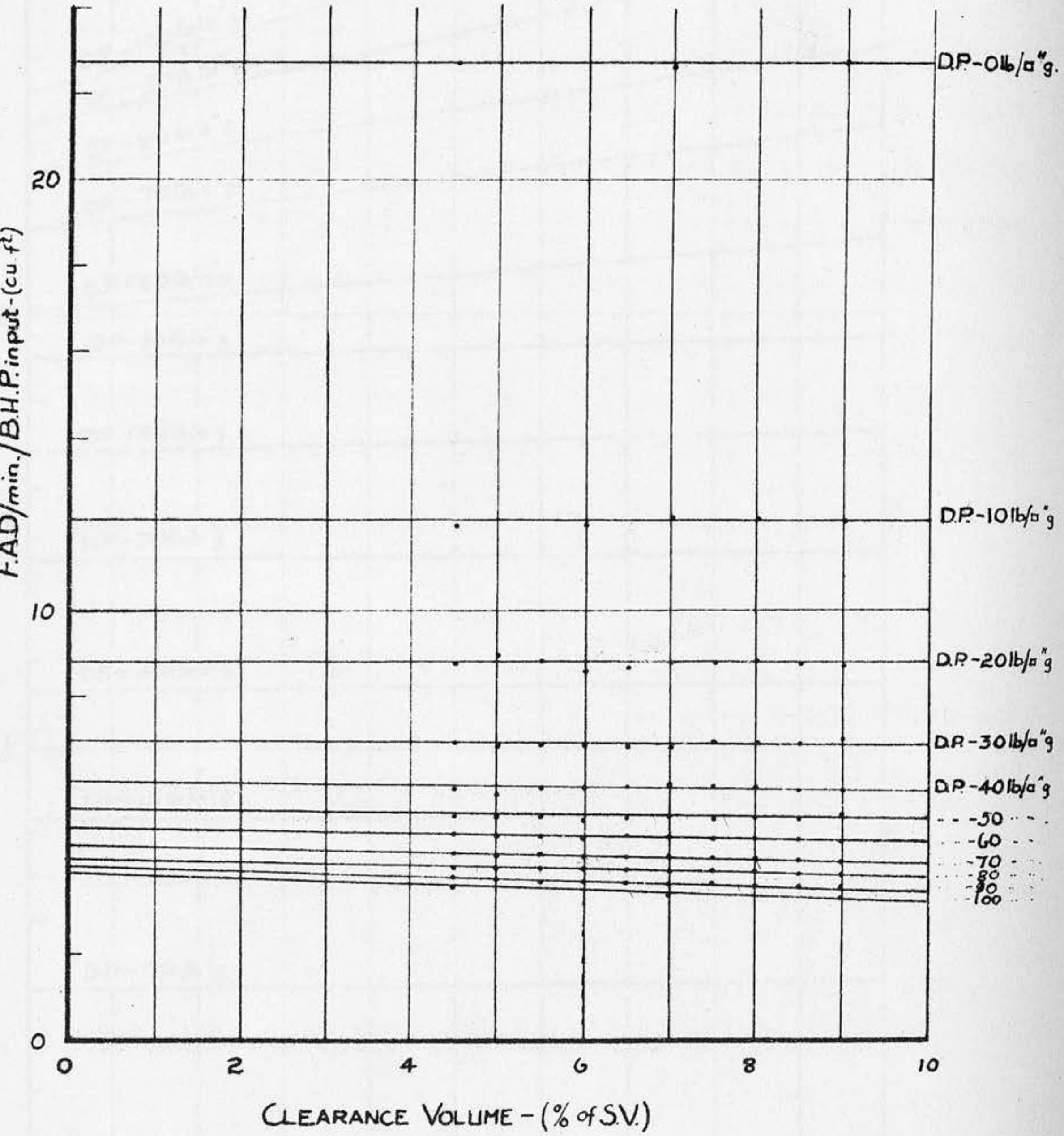

B.H.P. input/cu.ft. of F.A.D/min.
against
CLEARANCE VOLUME
D.P. - 100 lb/□"g
D.P. - 90 lb/□"g
D.P. - 80 lb/□"g
D.P. - 70 lb/□"g
D.P. - 60 lb/□"g
D.P. - 50 lb/□"g
D.P. - 40 lb/□"g
D.P. - 30 lb/□"g
D.P. - 20 lb/□"g
D.P. - 10 lb/□"g
D.P. - 0 lb/□"g
·3
·2
·1
0
0
2
4
6
8
10
B.H.P. input/cu.ft. of F.A.D/min.
CLEARANCE VOLUME - (% of S.V.)

PLATE No 36

F.A.D./min. / B.H.P. INPUT against CLEARANCE VOLUME.



per square inch, which are drawn to a base of clearance volume, are parallel to the base, resembling in this respect equivalent graphs of isothermal horse power. Above 40 lb. per square inch they begin to slope at increasing gradients, retaining their straight line form. It could be stated that up to a delivery pressure of 45 lb. per square inch the effect of clearance between the limits 4.5 and 9% could be neglected when considering the relationship between brake horse power input and free air discharge.

Over 45 lb. per square inch the power required per unit of air discharged <u>is</u> affected by altered clearance, less power being required when the clearance is small, e.g.,

At delivery pressures	60, 80 and 100 lb. per square inch
the differences between the B.H,P. input per cub.ft. of F.A.D. per minute at 9.04% and 4.52% clearance are ...	.008, .013 and .0275; and
the differences between the F.A.D. per minute per B.H.P. input at 4.52% and 9.04% clearance are	.18, .205 and .325.

Increase of delivery pressure increases the difference.

Opposed /

PLATE No 37

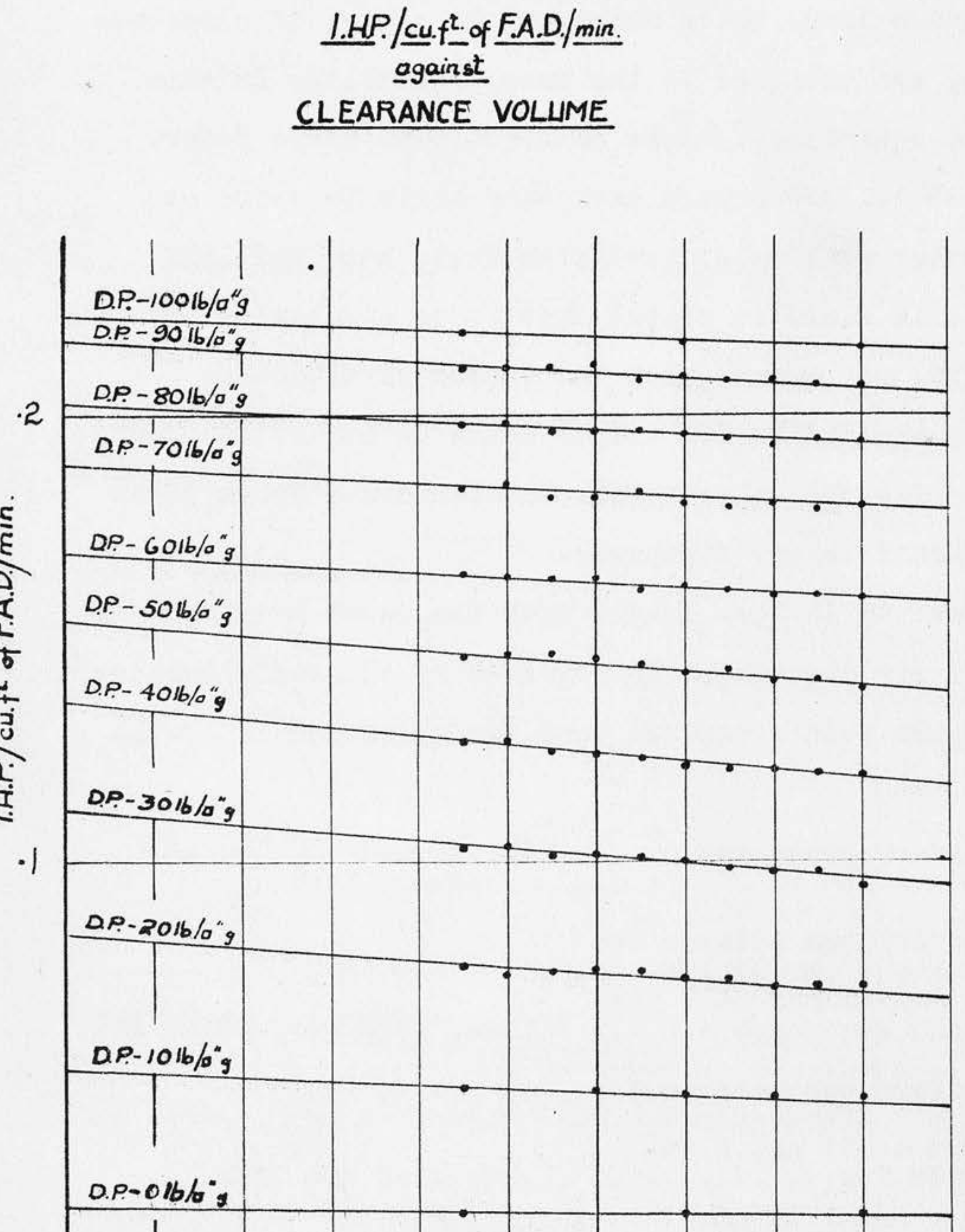
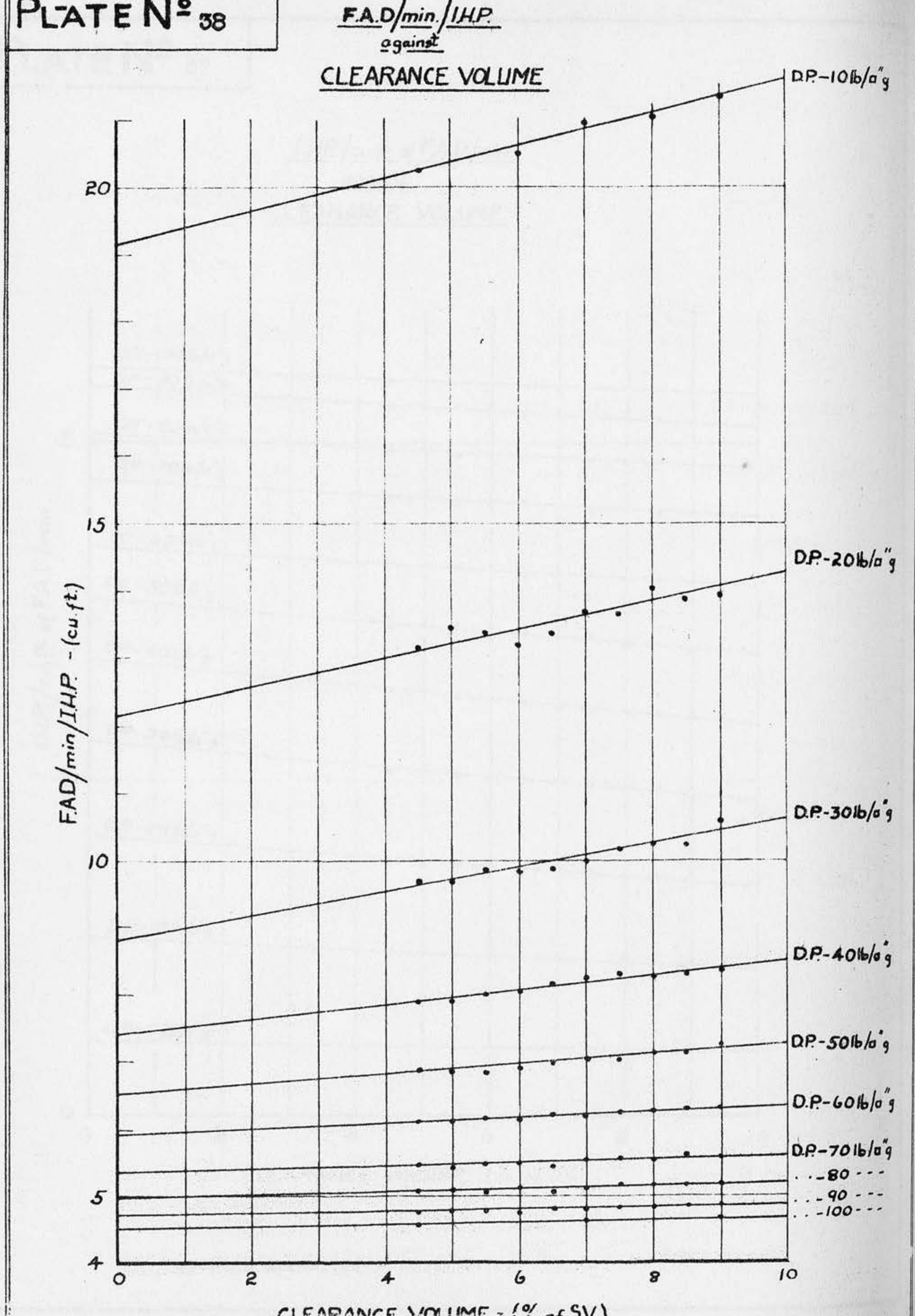

I.H.P./cu.ft. of F.A.D./min.
against
CLEARANCE VOLUME
D.P.-100 lb/□"g
D.P.-90 lb/□"g
D.P.-80 lb/□"g
D.P.-70 lb/□"g
D.P.-60 lb/□"g
D.P.-50 lb/□"g
D.P.-40 lb/□"g
D.P.-30 lb/□"g
D.P.-20 lb/□"g
D.P.-10 lb/□"g
D.P.-0 lb/□"g
I.H.P./cu.ft. of F.A.D./min.
·2
·1
0
0
2
4
6
8
10
CLEARANCE VOLUME (% of S.V.)

PLATE Nº 38


F.A.D/min./I.H.P.
against
CLEARANCE VOLUME
D.P.-10lb/□"g
D.P.-20lb/□"g
D.P.-30lb/□"g
D.P.-40lb/□"g
D.P.-50lb/□"g
D.P.-60lb/□"g
D.P.-70lb/□"g
-80
-90
-100
20
15
10
5
4
0
2
4
6
8
10
F.A.D/min/I.H.P. - (cu.ft.)
CLEARANCE VOLUME - (% of S.V.)

Opposed to the adverse effect of large clearances on the brake horse power input per cubic foot of F.A.D. per minute, large clearance volumes reduce the indicated horse power per cubic foot of F.A.D. per minute,* and this is primarily due to the reduction in the compression exponent which lowers the indicated horse power. (See Plates 33, 34, 37 and 38.) At all delivery pressures the difference in the indicated horse power per cubic foot of F.A.D. per minute is approximately constant at .005 between the limits of clearance 4.52 and 9.04%.

Increased /

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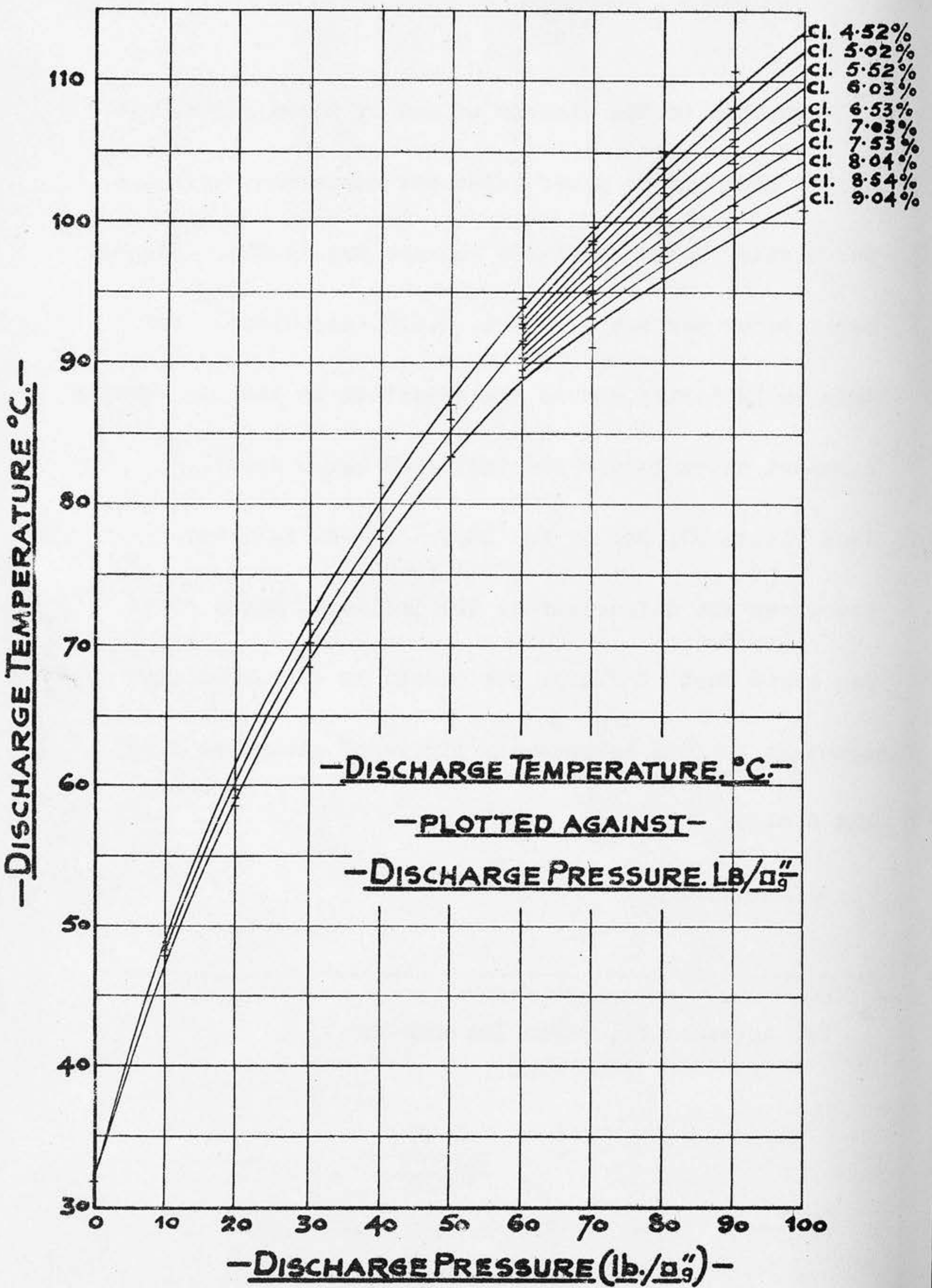
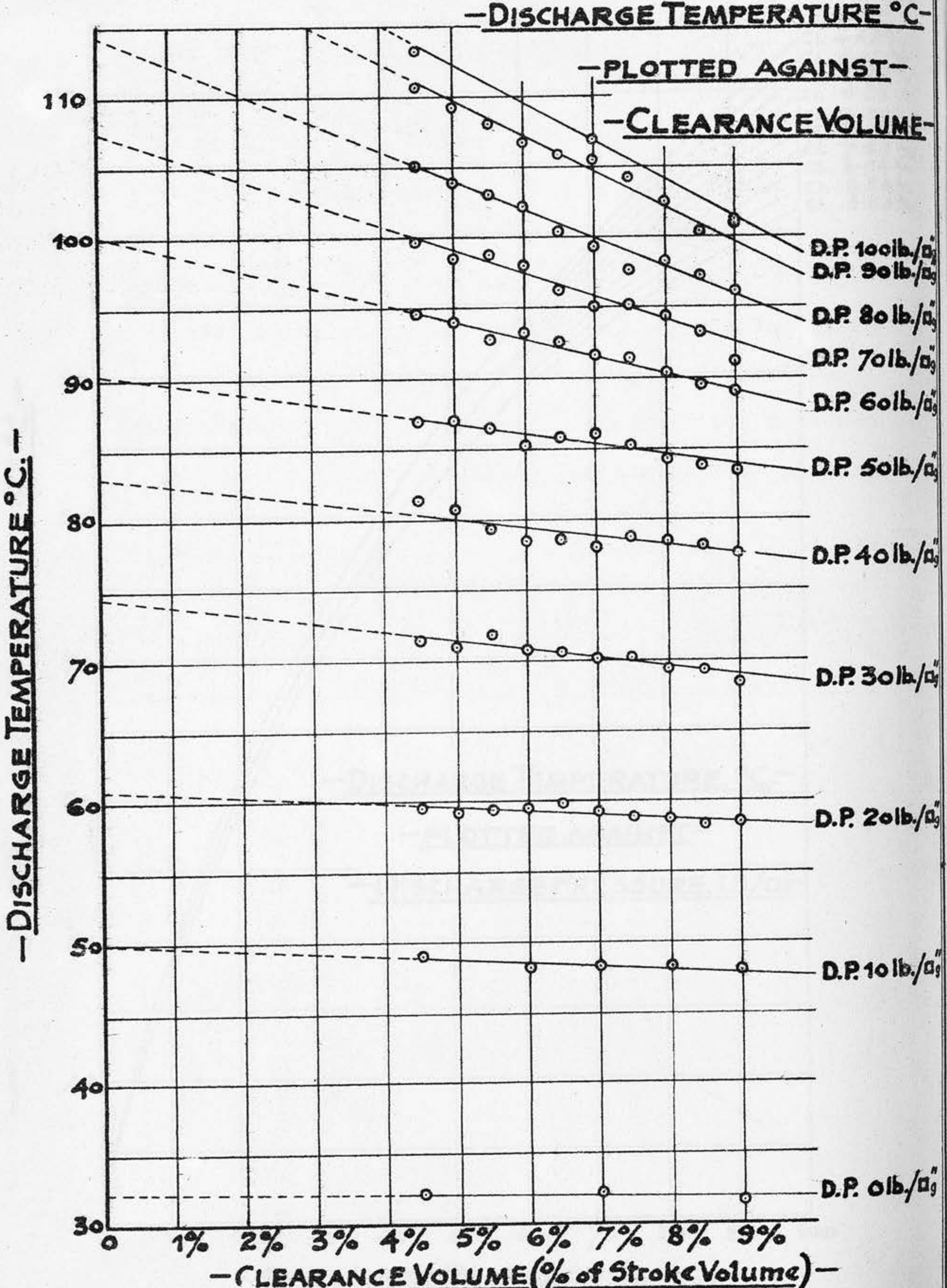
* See Appendix II, pages 121 and 122.

PLATE Nº 39



-DISCHARGE TEMPERATURE °C.-
-PLOTTED AGAINST-
-DISCHARGE PRESSURE. LB/□"g-
-DISCHARGE TEMPERATURE °C.-
-DISCHARGE PRESSURE (lb./□"g)-
110
100
90
80
70
60
50
40
30
0
10
20
30
40
50
60
70
80
90
100
Cl. 4·52%
Cl. 5·02%
Cl. 5·52%
Cl. 6·03%
Cl. 6·53%
Cl. 7·03%
Cl. 7·53%
Cl. 8·04%
Cl. 8·54%
Cl. 9·04%

PLATE No. 40



-DISCHARGE TEMPERATURE °C-
-PLOTTED AGAINST-
-CLEARANCE VOLUME-
110
100
90
80
70
60
50
40
30
0
1%
2%
3%
4%
5%
6%
7%
8%
9%
-DISCHARGE TEMPERATURE °C.-
-CLEARANCE VOLUME (% of Stroke Volume)-
D.P. 100 lb./□"g
D.P. 90 lb./□"g
D.P. 80 lb./□"g
D.P. 70 lb./□"g
D.P. 60 lb./□"g
D.P. 50 lb./□"g
D.P. 40 lb./□"g
D.P. 30 lb./□"g
D.P. 20 lb./□"g
D.P. 10 lb./□"g
D.P. 0 lb./□"g

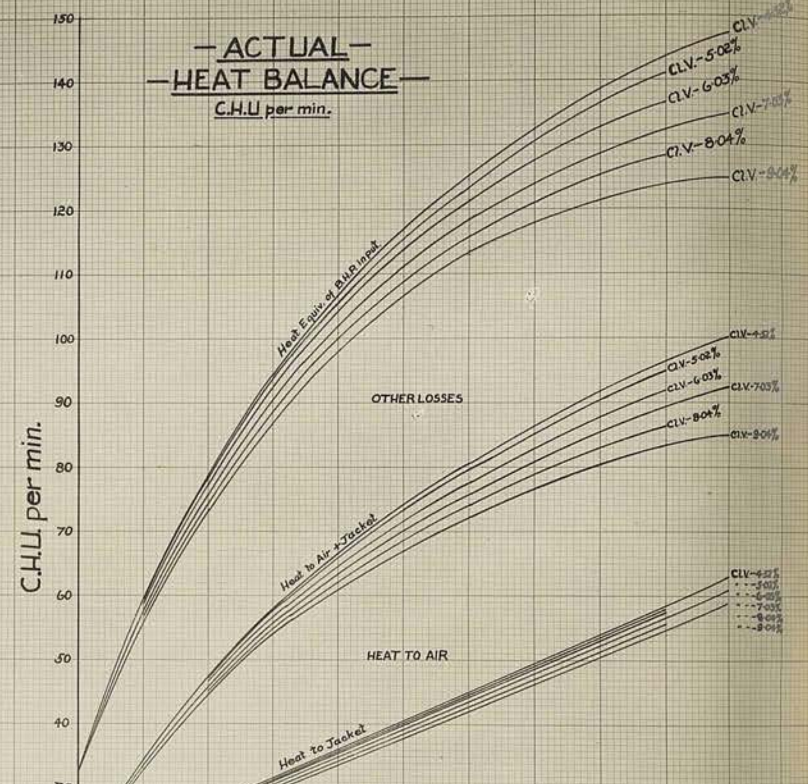
Increased clearance at all delivery pressures sees a decrease in the air discharge temperature.* No matter the clearance, the pressure compression ratio is unalter:ed, and the lowered temperature at large clearance volumes is due to the reduction in the compression ex:ponent and to less air passing through the discharge valve per stroke, which reduces the friction heat. (See Plates 39 and 40.) The following table gives the observed air discharge temperatures at the minimum and maximum test clearance volumes:-

Delivery Pressure (lb./sq.inch)	Air Discharge Temperature (°C.)	
	Clearance 4.52%	Clearance 9.04%
0	31.9	31.4
10	49	48
20	59.7	58.6
30	71.5	68.4
40	81.4	77.6
50	87	83.4
60	94.6	88.9
70	99.7	91.1
80	105.1	96.4
90	110.7	101.1
100	113.3	100.9

* See Appendix II, pages 98 to 108 (Observation Sheets).

PLATE NO.41.

	Clearance Volume (% of S.V.)									
	4·52	5·02	5·52	6·03	6·53	7·03	7·53	8·04	8·54	9·04
D.P. – 0 lb/□"g										
Heat to B.H.P. input (CHU/min)	32·5					32·3				32·1
" " Jacket (CHU/min)	11·57					11·46				12·48
" " Air (")	7·24					7·67				7·49
Other Losses (")	13·69					13·17				12·13
D.P. – 10 lb/□"g										
Heat to B.H.P. input (CHU/min)	58·9			58·4		57		57		56·5
" " Jacket (")	18·33			18·75		18·62		17·59		20·43
" " Air (")	16·29			15·91		15·79		16·16		15·43
Other Losses (")	24·28			23·74		22·59		23·27		20·62
D.P. – 20 lb/□"g										
Heat to B.H.P. input (CHU/min)	77·7	75·6	76·5	78	76·5	74·9	75·2	73·5	74	73·6
" " Jacket (")	25·72	24	26·9	26·8	26·55	25·6	24·3	26·53	26·92	27·31
" " Air (")	21·4	21·29	20·76	20·5	20·83	20·68	20·48	19·1	19·09	19·37
Other Losses (")	31·08	30·31	28·84	30·7	29·12	28·62	30·42	27·87	27·99	26·72
D.P. – 30 lb/□"g										
Heat to B.H.P. input (CHU/min)	94·5	95	93·1	93·1	92·4	92·1	89·7	89·5	90·2	86·6
" " Jacket (")	31·46	30·8	30·18	31·32	31·08	31·46	31·33	31·08	30·67	30·67
" " Air (")	26·29	26	26·63	26·02	25·77	25·83	25·49	25·09	25·01	24·36
Other Losses (")	36·75	38·2	36·29	35·76	35·55	34·81	32·88	33·34	34·52	31·57
D.P. – 40 lb/□"g										
Heat to B.H.P. input (CHU/min)	108·9	108·6	106	104·6	104·4	102	101·8	100·9	100·9	100
" " Jacket (")	35·92	35·08	34·08	33·72	33·8	34·7	35·48	35·95	35·3	36·58
" " Air (")	31·65	31·09	30·49	29·79	29·07	28·36	28·67	28·08	27·53	26·87
Other Losses (")	41·33	42·43	41·43	41·09	41·55	38·94	37·65	36·87	38·07	36·55
D.P. – 50 lb/□"g										
Heat to B.H.P. input (CHU/min)	116·6	116·6	114·5	113·8	116·5	112·1	111·9	109·6	108·9	104·8
" " Jacket (")	39·22	39·45	40·45	39·1	38·9	39·12	38·5	38·3	37·5	37·5
" " Air (")	34·34	34·07	33·04	31·96	31·45	31·24	30·42	29·61	29·34	28·61
Other Losses (")	43·04	43·08	41·01	42·74	44·15	41·74	42·98	41·69	42·06	38·69
D.P. – 60 lb/□"g										
Heat to B.H.P. input (CHU/min)	124·7	123·2	122·3	121·5	118·7	118·2	116·1	113·6	113·1	113
" " Jacket (")	44·65	44·75	44·1	44·3	43·35	42·4	42·8	42·5	42·9	43·6
" " Air (")	35·9	34·82	33·85	33·75	32·96	32·2	31·52	30·63	29·22	28·71
Other Losses (")	44·15	43·63	44·35	43·45	42·39	43·6	41·78	40·47	40·98	40·69
D.P. – 70 lb/□"g										
Heat to B.H.P. input (CHU/min)	133·1	131·5	129·6	127·8	127·2	123·9	122·8	121·1	120	117·6
" " Jacket (")	47·3	47·3	48·6	49·25	49·5	49·1	47·4	47·4	46·5	45·5
" " Air (")	36·72	35·61	34·92	33·38	32·05	31·38	31·49	30·29	29·68	28·11
Other Losses (")	49·08	48·59	46·08	45·17	45·67	43·42	43·91	43·41	43·82	43·99
D.P. – 80 lb/□"g										
Heat to B.H.P. input (CHU/min)	138·2	136	134·3	131·9	131·3	128·9	126·2	125·8	124·4	121·5
" " Jacket (")	54·1	53·45	53·4	52·4	55	52	51·15	53·7	53·4	53·1
" " Air (")	37·53	36·4	35·45	34·75	33·47	32·54	31·13	28·76	27·85	27·27
Other Losses (")	44·57	44·15	45·45	44·75	44·83	44·36	43·92	43·34	43·15	41·13
D.P. – 90 lb/□"g										
Heat to B.H.P. input (CHU/min)	144	141	139·8	136·7	135	132	129·9	127·2	125	123·7
" " Jacket (")	58·6	58·7	56·65	56·45	57·5	57	56·2	53·9	54·25	56·15
" " Air (")	38·5	36·98	35·95	34·77	33·1	31·93	30·65	29·62	28·34	29·05
Other Losses (")	46·9	45·32	44·2	45·48	44·4	43·07	43·05	43·68	42·41	39·5
D.P. – 100 lb/□"g										
Heat to B.H.P. input (CHU/min)	146·8					134·8				124·9
" " Jacket (")	62·9					61				58·9
" " Air (")	38·99					32·15				26·77
Other Losses (")	44·91					41·65				39·23



HEAT BALANCES.

The heat lost in friction is equal to the difference between the brake and indicated horse-powers while the indicated horse-power sums the heat to the jacket, air and other losses. A balance struck in this manner is not strictly correct, as some of the friction heat enters the air and jacket, a negative balance occasionally resulting, particularly at low discharge pressures. A truer and more accurate balance would be -- heat equivalent of the brake horse-power input is equal to the sum of the heat to the jacket, heat to the air and heat to all other losses which includes that friction heat which neither enters the air nor jacket and the heat lost by radiation and leakage.

The actual heat balances are tabulated on Plate 41. From them we see that for any one delivery pressure the heat given to the air and jacket is greatest when the clearance is least. This we would expect since the brake horse-power input, air discharge temperature and weight of air discharged are then greater. Also at low delivery pressures there is a slow fall in air and jacket heat and at high pressures a quick drop in air heat but a relatively slow fall in jacket heat when the clearance is increased. The following table illustrates this: /

this:

D.P. - 30 lb. per square inch:-

	Cl. 4.52%	Cl. 9.04%	Difference
B.H.P. input (C.H.U. per min.)	94.5	86.6	7.9
Jacket (C.H.U. per min.)	31.46	30.67	.79
Air (C.H.U. per min.)	26.29	24.36	1.93
Other Losses (C.H.U. per min.)	36.75	31.57	5.18

D.P. - 90 lb. per square inch:-

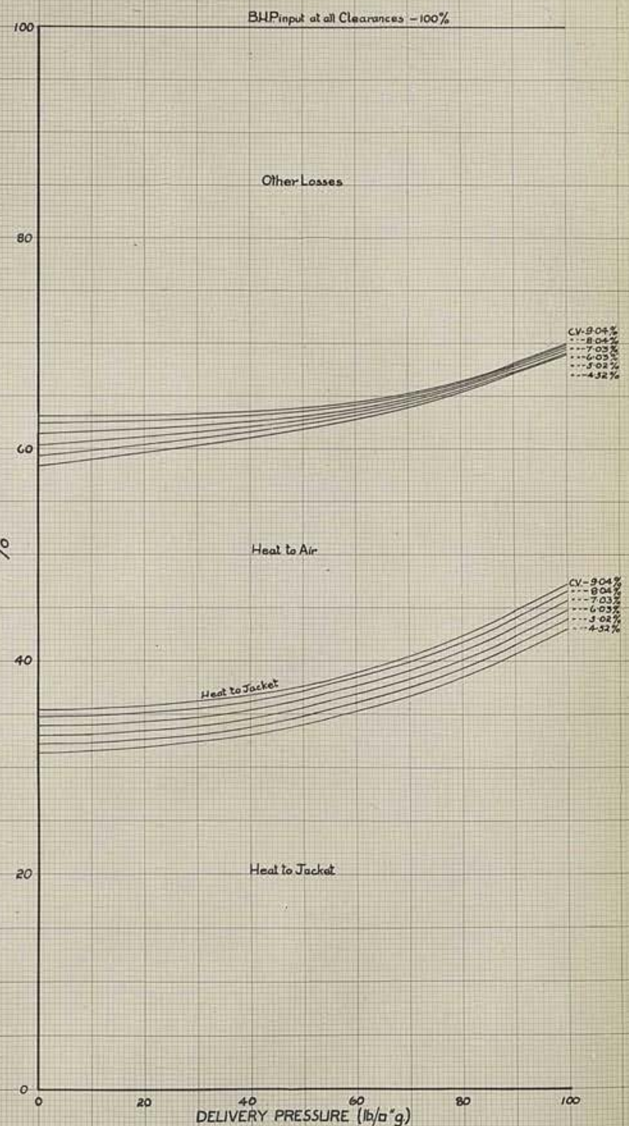
	Cl. 4.52%	Cl. 9.04%	Difference
B.H.P. input (C.H.U. per min.)	144	123.7	20.3
Jacket (C.H.U. per min.)	58.6	56.15	2.45
Air (C.H.U. per min.)	38.5	28.05	10.45
Other Losses (C.H.U. per min.)	46.9	39.5	7.4

We could say therefore that up to about 40 lb. per square inch delivery pressure increase of clearance between the limits 4.5 and 9% does not seriously alter the actual heat to the air. Over 40 lb. per square inch, however, owing to a wider range of volumetric efficiency /

PLATE NO.42.

—PERCENTAGE—
—HEAT BALANCE—

		Clearance Volume (% of S.V.)									
		4·32	5·02	5·52	6·03	6·33	7·03	7·53	8·04	8·54	9·04
DP-0 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	39·6					35·3				38·9
	Air	23·3					23·7				23·3
	Other Losses	37·1					40·8				37·8
DP-10 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	31·1			32·1		32·7		30·9		36·2
	Air	27·7			27·2		27·7		28·3		27·3
	Other Losses	41·2			40·7		39·6		40·8		36·5
DP-20 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	32·5	31·7	35·2	34·4	34·7	34·2	32·3	36·1	36·4	37·4
	Air	27·5	28·2	27·1	26·3	27·2	27·6	27·3	26	25·8	26·3
	Other Losses	40	40·1	37·7	39·3	38·1	38·2	40·4	37·9	37·8	36·3
DP-30 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	33·3	32·4	32·4	33·6	33·6	34·2	34·9	34·7	34	35·4
	Air	27·8	27·4	28·6	28	27·9	28·1	28·4	28	27·7	28·1
	Other Losses	38·9	40·2	39	38·4	38·5	37·7	36·7	37·3	38·3	36·5
DP-40 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	33	32·3	32·1	32·2	32·4	34	34·8	35·6	35	36·58
	Air	29	28·6	28·8	28·5	27·8	27·8	28·2	27·8	27·3	26·87
	Other Losses	38	39·1	39·1	39·3	39·8	38·2	37	36·6	37·7	36·55
DP-50 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	33·6	33·8	35·4	34·4	34	34·9	34·4	35	34·4	35·8
	Air	29·5	29·2	28·8	28	27·5	27·9	27·2	27	27	27·3
	Other Losses	36·9	37	35·8	37·6	38·5	37·2	38·4	38	38·6	36·9
DP-60 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	35·8	36·3	36·1	36·4	36·5	35·9	36·8	37·4	37·9	38·6
	Air	28·8	28·3	27·7	27·8	27·8	27·2	27·2	27	25·8	25·4
	Other Losses	35·4	35·4	36·2	35·8	35·7	36·9	36	35·6	36·3	36
DP-70 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	35·5	36	37·5	38·5	38·9	39·6	38·6	39·2	38·8	38·7
	Air	27·6	27·1	26·9	26·1	25·2	25·3	25·6	25	24·7	23·9
	Other Losses	36·9	36·9	35·6	35·4	35·9	35·1	35·8	35·8	36·5	37·4
DP-80 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	39·2	39·3	39·8	39·7	40·3	40·3	40·5	42·7	42·9	43·7
	Air	27·1	26·8	26·4	26·4	25·5	25·3	24·7	22·8	22·4	22·4
	Other Losses	33·7	33·9	33·8	33·9	34·2	34·4	34·8	34·5	34·7	33·9
DP-90 lb/□"g	B.H.P. input	100	100	100	100	100	100	100	100	100	100
	Jacket	40·7	41·6	40·8	41·3	42·6	43·2	43·3	42·3	43·4	45·4
	Air	26·7	26·2	25·4	25·4	24·5	24·2	23·6	23·3	22·7	22·7
	Other Losses	32·6	32·2	33·8	33·3	32·9	32·6	33·1	34·4	33·9	31·9
DP-100 lb/□"g	B.H.P. input	100					100				100
	Jacket	42·8					45·2				47·2
	Air	26·6					23·8				21·4
	Other Losses	30·6					31				31·4



efficiency and delivery temperature, much less heat is taken away in the air with increased clearance. The actual heat to the jacket on the other hand, shows, with increased clearance at all discharge pressures, but a very small decrease and the importance of this is at once apparent when the jacket heat is reduced to a per:centage of the heat input.

The curves of Heat Equivalent of B.H.P. input are similar to the B.H.P. input curves on Plate 26 and those of Heat to Air + Jacket resemble the curves of I.H.P. on the same plate, since the sum total of the heat to air and jacket is approximately equal to the heat equivalent of the indicated horse-power. The jacket water curves, unlike the others, continue up:wards and will meet the heat to air + jacket curves at the compressor's maximum delivery pressure, for at that pressure the air heat will be zero and most of the heat input will enter the jacket water.

Reducing the heat distribution to a percentage of the heat input demonstrates better the significance of each heat balance (See Plate 42).

The percentage heat to the air rises to a maximum and drops to zero at the maximum delivery pressure, e.g., with clearance 4.52% the air heat is 22.3% at zero delivery /

delivery pressure, rising to 29.5% at 50 lb. per square inch and falling to 26.6% at 100 lb. per square inch, and with clearance 9.04% is 23.3% at zero delivery pressure, rising to 28.1% at 30 lb. per square inch and falling to 21.4% at 100 lb. per square inch. Since large clearances give lower maximum delivery pressures, the air heat will reach zero earlier and the maximum percentage air heat is lower in value and is reached earlier. They are:-

rance lume	4.52%	5.02%	5.52%	6.03%	6.53%	7.03%	7.53%	8.04%	8.54%	9.04%
- 30 per re inch.					27.9	28.1	28.4	28	27.7	28.1
- 40 per re inch.			28.8	28.5						
- 50 per re inch.	29.5	29.2	28.8							

The following extract from the tabulated values on Plate 42 lists the range of air heat from 20 to 100 lb. per square inch delivery pressure and shows that increased clearance lowers the percentage heat to the air.

D.P. /

	Cl. 4.52%	Cl. 9.04%
D.P. - 20 lb. per square inch g.	27.5%	26.3%
D.P. - 40 lb. per square inch g.	29%	26.87%
D.P. - 60 lb. per square inch g.	28.8%	25.4%
D.P. - 80 lb. per square inch g.	27.1%	22.4%
D.P. -100 lb. per square inch g.	26.6%	21.4%

On the other hand more heat enters the jacket when the clearance is increased,

	Cl. 4.52%	Cl. 9.04%
D.P. - 20 lb. per square inch g.	32.5%	37.4%
D.P. - 40 lb. per square inch g.	33%	36.58%
D.P. - 60 lb. per square inch g.	35.8%	38.6%
D.P. - 80 lb. per square inch g.	39.2%	43.7%
D.P. -100 lb. per square inch g.	42.8%	47.2%

The heat to the jacket water becomes greater with increased delivery pressure, the highest value being reached at the compressor's maximum delivery pressure and the trend of the large clearance curves is to approach the brake horse-power input quicker than those with small clearance volumes.

NOTE: The compression exponent was lowered with both increased clearance and increased delivery pressure.

CONCLUSION.

An attempt has been made at the beginning of this thesis to bring together some hitherto unrecorded information on air compressors and to co-relate a few of the most common systems of air measurement. It should be mentioned here that particular attention should be paid to the code which is to form the basis of air measurement tests if a penalty job is involved, and should be agreed to beforehand. The compressor and air measurement plant installed in the University of Edin:burgh has been deemed worthy of permanent recording in complete detail.

It is recommended that apparent volumetric effic:iencies have no standing as test results, unless good compressor slip approximations are known.

Theoretically the work done per pound of air is the same whether clearance is considered or not, nor does clearance in any way affect the isothermal horse power per cubic foot of F.A.D. per minute.

The tests performed with altered clearance prove that the beneficial effects of small clearances are

higher volumetric efficiencies;

higher mechanical efficiencies;

higher overall efficiencies; and

lower brake horse powers per cubic foot of F.A.D. /

F.A.D. per minute, and that the beneficial effects of large clearances are

higher compression efficiencies;

lower discharge temperatures;

lower compression exponents; and

lower indicated horse powers per cubic foot of F.A.D. per minute.

The sole aim of the designer, within economical limits, being to procure a greater air discharge per unit of power, his interest will lie principally in the brake horse power per cubic foot of F.A.D. per minute and the overall efficiency and, keeping in view the above comparative list, will favour small clearance spaces.

The best and most economical working pressures lie between 10 and 30 lb. per square inch, since the over-:all efficiency is then highest, and these two pressure limits could be accepted as typical of a small compressor.

The tests definitely establish that up to a delivery pressure of 50 lb. per square inch the effect of clear-:ance between the limits 4.5 and 9% can be neglected; therefore, when designing compressors to discharge up to 50 lb. per square inch, fit as large valves as poss-:ible, even though their installation increases the clearance /

clearance to 10%. The result would be that the volum- :etric efficiency would be reduced but air velocities and discharge temperatures would be lowered, more heat would enter the jacket water and more air would be delivered per unit horse power. The initial compressor cost would be greater, as a slightly larger machine would require to be installed to deliver a definite quantity of air in a certain time, but a reduction in the running cost would be effected. Commercial con- :siderations induce makers to depart from valve propor- :tions which their experience would lead them to adopt, but an experienced purchaser would be more easily temp- :ted by a compressor's performance than by its initial cost.

Over 50 lb. per square inch delivery pressure retain a minimum practical clearance at the expense of an in- :creased air speed.

The heat balances show that by raising the delivery pressure, the percentage heat to the jacket is increased, and that the percentage air heat rises to a maximum and falls to zero at the compressor's maximum delivery press- :ure. At each delivery pressure increase in clearance volume lowers the percentage air heat but raises the percentage of jacket water heat. Also the maximum per- :centage air heat is lowered in value and occurs at a lower /

lower delivery pressure when the clearance is increased.

A review of all the plates in the preceding section which have a base of clearance volume shows every graph without one exception to be a straight line.

The theoretical volumetric efficiency

$= 1 - Cl. \left\{ \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1 \right\}$ which is a linear function of the clearance, and the tests showed both the actual volumetric efficiency and the F.A.D. per minute to be also of linear form.

The isothermal horse power $= \dfrac{P_1 V_1 \log_e \frac{P_2}{P_1}}{33{,}000}$

$=$ Constant $\times$ F.A.D. per minute,

so that the graphical interpretation of the isothermal horse power is linear and by actual test both the indicated and brake input horse powers are proved to lie in a straight line.

Having established that the air discharged and fundamental horse powers obey straight line laws when the clearance is altered, it follows that the relationships between the air discharged and the powers, and between the powers themselves, viz., the isothermal, indicated and brake horse powers per cubic foot of F.A.D. per minute and their reciprocals, and the mechanical, compression and overall efficiencies are all straight lines /

lines when plotted to a base of clearance volume. At all delivery pressures the isothermal horse power per cubic foot of F.A.D.per minute is parallel to the clearance volume abscissa, (See page 23), while the indicated and brake horse powers per cubic foot of F.A.D. per minute are parallel to the abscissa at low delivery pressures, but slope at ever increasing gradients as the delivery pressure is increased.

It could be stated, therefore, that the following results:-

theoretical volumetric efficiency;

actual volumetric efficiency;

F.A.D. per minute;

isothermal horse power;

indicated horse power;

brake horse power input;

isothermal horse power per cubic foot of F.A.D. per minute and its reciprocal;

indicated horse power per cubic foot of F.A.D. per minute and its reciprocal;

brake horse power input per cubic foot of F.A.D. per minute and its reciprocal;

mechanical efficiency;

compression efficiency;

overall efficiency;

and the discharge temperature obey at each delivery

pressure /

pressure a straight line law when the clearance is altered.

Let us grasp the significant importance of this. From a knowledge of a compressor's performance at two different clearance volumes, the performance at any practical clearance can be accurately determined. It furnishes a means of estimating at zero clearance results which cannot be found by practical experiment. Com:pressor designers with a wealth of previous test data could, when designing a new type of compressor having a new clearance, arrive at a fair approximation to the compressor's test performance. Air velocities might be different, but his experience and judgment would aid him in his choice of information.

The enclosed published paper introduces a hitherto unsuspected yet simple experimental means of ascertain:ing a compressor's air slip by altering the clearance volume. The clearance can be increased temporarily in a commercial machine by placing a distance piece between the cylinder and cylinder head and in an experimental machine by removing a compression plate from the big end. The practicability of this new method is assured by several engineering firms and compares favourably with the normal method of air intake and output measurement.

Its /

Its primary advantage is that only one air measurement nozzle is required. Air measurement by nozzles toler-:ates an error of $\pm$ 2%, so that where two nozzles are in use, the error may be as high as 4%, and at low del-:ivery pressures and small slip values it might appear that more air leaves than enters the compressor. Against this the use of one nozzle and a clearance ad-:justment would at the most be subject to an error of $\pm$ 2%.

It would be better to denote slip measured in the normal way by the term 'leakage,' for by slip proper is meant all the losses except that caused by the clearance volume. (That air which fails to enter the compressor is also a loss.) It could be argued that slip is, as in pumps, the difference between the actual and theoret-:ical discharge, but air is a compressible fluid and slip should sum all the losses except that due to clear-:ance.

A P P E N D I C E S.

APPENDIX I.

POINTS IN THE DESIGN OF AIR COMPRESSOR VALVES WORTHY OF INVESTIGATION.

On a low pressure compressor, of some 5 - 8 lb. per square inch, the resistance to air flow through the valves becomes, probably, the all-important factor, and an investigation into this resistance might yield profitable results. On account of the expense involved,* tests would require to be made under conditions of uniform flow, that is, valves would be in-:serted in a pipe and the loss of head measured for various air speeds. Three points worthy of investigation are:-

a) Effective Net Port Area: i.e., the opening through the ports in the valve seat in relation to the area of the hole in the cylinder. This is a limiting factor in compressor design.

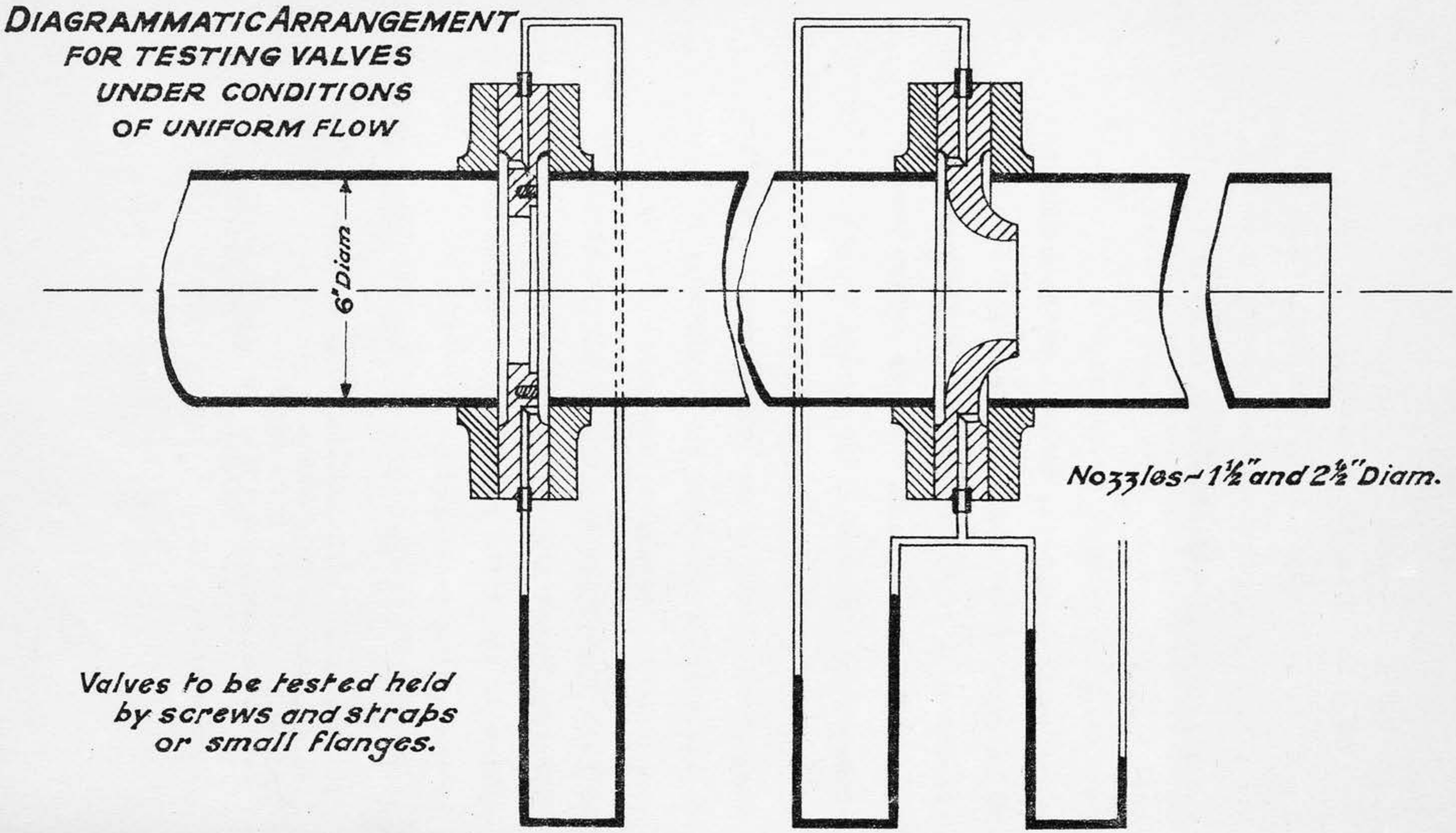
b) Flow Resistance at Various Air Speeds: Presumably this shows variance according to the extent to which the ports are stream-lined, and to the spring pressure.

c) /

---

* In an attempt to develop a new type of valve, a certain American firm spent some $50,000, the net result being a valve no better than those already in use. They had, of course, a lot of information.

PLATE No. 43


DIAGRAMMATIC ARRANGEMENT
FOR TESTING VALVES
UNDER CONDITIONS
OF UNIFORM FLOW
6" Diam
Nozzles – 1½" and 2½" Diam.
Valves to be tested held
by screws and straps
or small flanges.

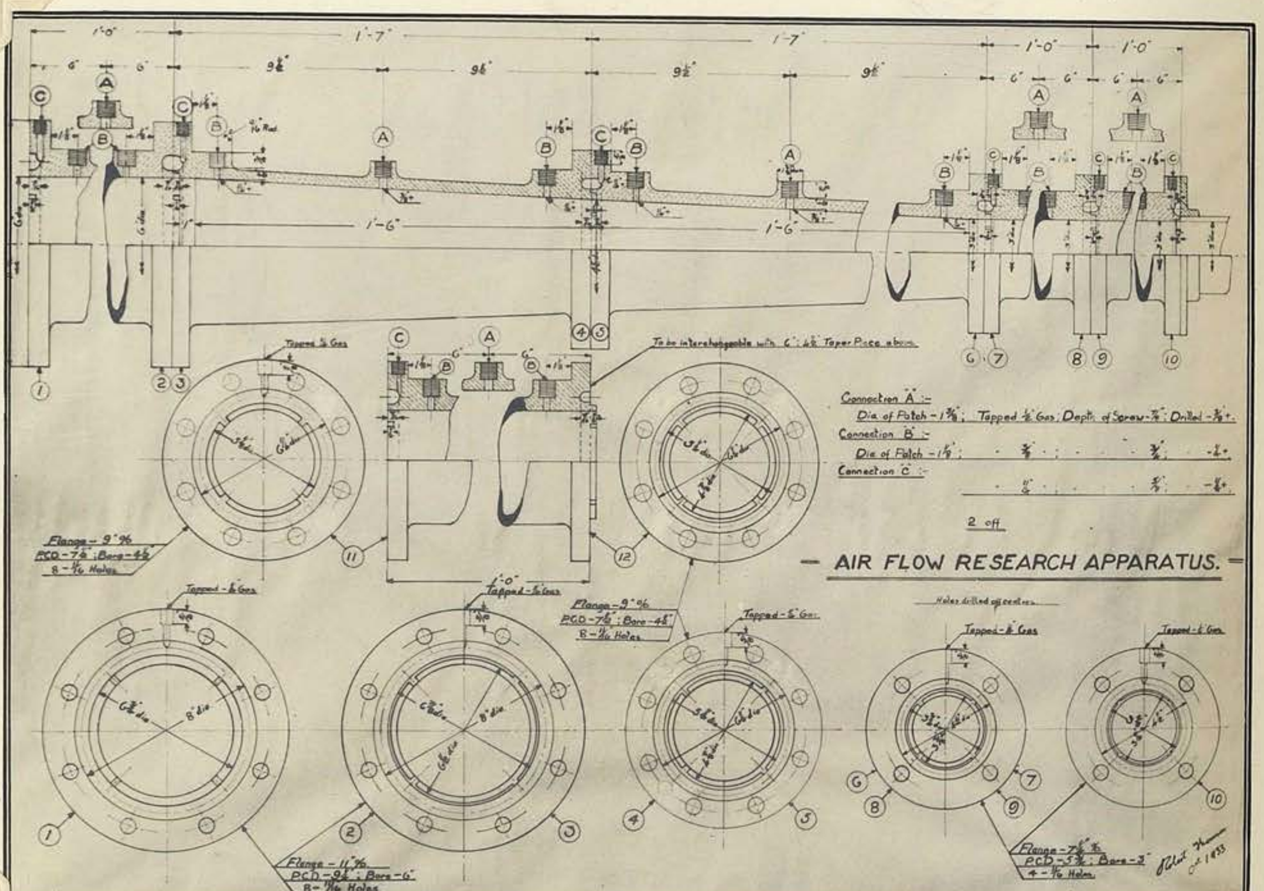
c) Effect of Restricting the Lift: It is generally accepted that a lift area of 75-80% of the port area meets all requirements, but some curves showing the result of restricting the lift would, doubtless, be illuminating.

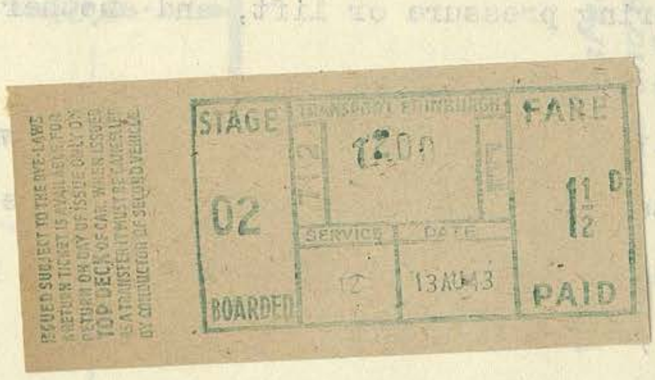
Plate 43 shows roughly a method which might be adopted, and Plate 44 the actual apparatus in more complete detail.

High pressure air might be used as an injector to induce a large volume of low pressure air for test purposes, and a photograph of the three U-tubes would give simultaneous readings. For quickness an air meter in the inlet pipe would serve to give results of such accuracy as can be expected in any reasonable time, as some twelve to fifteen tests per hour could be made on one size of valve. Conditions could then be changed in some respect, in the spring pressure or lift, and another series of tests run.

It may be that little information would be obtained from such tests on air valves as opposed to fitting them in their proper place, viz., /

PLATE NO.44.


AIR FLOW RESEARCH APPARATUS.
Connection "A" :-
Dia of Patch - 1 3/8"; Tapped 1/2" Gas; Depth of Screw - 7/8"; Drilled - 3/8"+.
Connection "B" :-
Connection "C" :-
2 off
Holes drilled off centres.
Tapped - 1/8" Gas
To be interchangeable with 6"; 4 1/2" Taper Piece above.
Flange - 9" %
PCD - 7 1/2"; Bore - 4 1/2"
8 - 9/16 Holes
Flange - 11 %
PCD - 9 1/2"; Bore - 6"
8 - 9/16 Holes


STAGE
02
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TOP DECK

viz., at the end of a cylinder in which a piston is reciprocating in the normal way. Against this, losses or points in design, hitherto unsuspected, might be more easily visualised under conditions of uniform flow.

O B S E R V A T I O N S

	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height (" of Hg)	29.91					29.91				29.91
R.P.M. of Compressor shaft	279.5					279.5				279.5
Average Piston Speed(ft/min)	279.5					279.5				279.5
Volts.	418					417				417
Amps.	8.52					8.51				8.5
Air Discharge per min.(lb.)	2.3793					2.3553				2.3516
Jacket water used per min.(lb)	4.436					3.617				4.222
" " inlet temp. (°C)	19.295					19.445				19.3
" " outlet " (°C)	21.903					22.513				22.256
Air Inlet Temperature (°C)	19.233					18.425				18.16
Air Discharge Temperature (°C)	31.9					32				31.416

Delivery Pressure 10 lb. per sq.in.g.

	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height (" of Hg)	29.91			29.91		29.91		29.91		29.91
R.P.M. of Compressor Shaft	279			279		279		279		279
Average Piston Speed(ft/min)	279			279		279		279		279
Volts.	417			417		417		416		416
Amps.	9.15			9.14		9.1		9.1		9.08
Air Discharge per min.(lb.)	2.2944			2.272		2.2586		2.246		2.2221
Jacket Water used per min(lb)	7.062			6.279		6.279		6.188		6.308
" " inlet temperature (^oC)	19.87			19.495		19.715		19.593		19.253
" " outlet " (^oC)	22.466			22.481		22.68		22.436		22.495
Air Inlet Temperature (^oC)	19.41			18.88		19.19		18.32		19.05
Air Discharge Temperature(^oC)	49			48.05		48.316		48.25		47.966

Delivery Pressure 20 lb. per sq.in.g.

	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height (" of Hg)	29.76	29.76	29.69	29.69	29.69	29.69	29.69	29.69	29.69	29.69
R.P.M. of Compression Shaft	277.5	277	278	277.75	277.2	277.25	277.5	277.5	277.25	277
Average Piston Speed(ft/min)	277.5	277	278	277.75	277.2	277.25	277.5	277.5	277.25	277
Volts	418	419	419	420	421	418	418	420	420	417
Amps.	9.69	9.62	9.65	9.7	9.65	9.6	9.61	9.56	9.58	9.57
AirDischarge per min.(lb.)	2.1915	2.17075	2.1658	2.1476	2.1334	2.12247	2.11413	2.1008	2.0882	2.0726
Jacket Water used per min. (lb.)	9.14	9.28	10.37	10.4	10.3	10.28	10.34	10.01	11.19	12.95
" " inlet temp.(°C)	19.641	19.678	18.853	18.988	19.395	19.648	19.768	19.313	19.341	19.561
" " outlet " (°C)	22.4	22.265	21.446	21.568	21.975	22.138	22.12	21.963	21.746	21.685
Air Inlet Temperature (°C)	18.98	18.325	19.61	19.61	19.2	18.52	18.35	20.77	20.12	19.6
Air Discharge Temp. (°C)	59.666	59.183	59.55	59.4	59.866	59.15	58.733	58.65	58.166	58.55

O B S E R V A T I O N S.

Delivery Pressure 30 lb. per sq.in.g.

	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height (" of Hg)	29.14	29.14	29.14	29.14	29.14	29.6	29.6	29.6	29.6	29.6
R.P.M. of Compressor Shaft	277,25	277.5	277.83	277.2	276.33	276.75	276.5	277.5	277.75	278
Average Piston Speed(ft/min)	277.25	277.5	277.83	277.2	276.33	276.75	276.5	277.5	277.75	278
Volts	418	418	416	418	416	417	417	417	417	417
Amps.	10.23	10.26	10.18	10.18	10.16	10.15	10.07	10.06	10.01	9.98
Air Discharge per min.(lb.)	2.0605	2.0422	2.027	2.0093	1.987	2.0248	2.0048	1.9902	1.9791	1.9591
Jacket Water used per min. (lb)	8.6	8.63	8.52	9.25	9.02	7.8	7.71	7.75	7.75	7.91
" " inlet temp.(°C.)	18.406	18.506	18.573	18.421	18.54	17.935	17.926	17.888	17.85	17.828
" " outlet temp.(°C)	22.065	22.07	22.113	21.808	21.986	21.966	21.991	21.9	21.805	21.706
Air Inlet Temperature (°C.)	18.31	17.89	17.18	16.725	16.62	16.95	17.17	16.95	16.65	16.6
Air Discharge Temp. (°C.)	71.466	70.95	71.9	70.7	70.666	70.1	70.133	69.45	69.316	68.433

O B S E R V A T I O N S.

Delivery Pressure 40 lb. per sq.in.g.

	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height (" of Hg)	29.57	29.57	29.57	29.57	29.28	29.28	29.28	29.29	29.29	29.29
R.P.M. of Compressor Shaft	276	275.7	276	276.25	276	275.75	275.5	275.5	275.5	276.25
Average Piston Speed (ft/min)	276	275.7	276	276.25	276	275.75	275.5	275.5	275.5	276.25
Volts	416	416	416	416	417	418	419	421	422	420
Amps.	10.72	10.71	10.63	10.57	10.56	10.49	10.48	10.45	10.45	10.42
Air Discharge per min.(lb.)	2.0394	2.01442	1.9994	1.9754	1.946	1.91466	1.9031	1.8793	1.8583	1.8381
Jacket Water used per min(lb)	9.22	9.2	8.81	8.59	8.265	8.7	8.7	8.95	9.135	10.4
" " inlet temperature (°C)	18.223	18.298	18.391	18.353	17.945	17.711	17.715	17.96	18.026	17.886
" "outlet temp. (°C)	22.118	22.115	22.255	22.278	22.033	21.701	21.788	21.976	21.89	21.403
Air Inlet Temperature (°C)	16.7	16.4	15.8	15.675	16.43	16.3	16	16.23	16.41	16.7
Air Discharge Temperature(°C)	81.383	80.716	79.33	78.483	78.666	78.033	78.766	78.5	78.15	77.633

O B S E R V A T I O N S.

Delivery Pressure 50 lb. per sq.in.g.

	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height(" of Hg)	29.35	29.35	29.35	29.35	29.58	29.58	29.58	29.58	29.56	29.56
R.P.M. of Compressor Shaft	275.5	276.2	275.7	275.25	275.25	275.5	275	275	275.5	276
Average Piston Speed(ft/min)	275.5	276.2	275.7	275.25	275.25	275.5	275	275	275.5	276
Volts	418	417	416	419	419	420	421	420	418	418
Amps.	11.01	11.01	10.93	10.9	10.93	10.85	10.84	10.75	10.72	10.58
Air Discharge per min.(lb)	1.966	1.943	1.8943	1.86066	1.8963	1.8733	1.8465	1.8226	1.7985	1.7614
Jacket Water used per min. (lb)	9.83	10.16	10.325	9.61	10.2	10.16	10.07	10.2	10.19	10.12
" " inlet temperature (°C)	17.98	17.968	18.013	18.151	18.083	18.095	18.095	18.103	18.23	18.226
" " outlet temp.(°C)	21.97	21.848	21.933	22.22	21.9	21.946	21.918	21.858	21.91	21.931
Air Inlet Temperature (°C)	14.2	13.975	13.85	13.7	16.5	16.6	16.6	16.56	15.8	15.7
Air Discharge Temp. (°C.)	86.966	87.033	86.516	85.25	85.733	86.083	85.25	84.25	83.75	83.366

O B S E R V A T I O N S.

Delivery pressure 60 lb. per sq.in.g.

	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height (" of Hg)	29.45	29.45	29.45	29.44	29.44	29.44	29.44	29.44	29.32	29.32
R.P.M. ofCompressor Shaft	275	275	274.7	275	275	275.25	275	275.25	275.75	276
Average Piston Speed(ft/min)	275	275	274.7	275	275	275.25	275	275.25	275.75	276
Volts	417	417	417	419	420	417	416	416	418	419
Amps.	11.3	11.24	11.21	11.18	11.08	11.06	10.99	10.89	10.88	10.87
Air Discharge per min.(lb)	1.9098	1.8756	1.84986	1.8264	1.7921	1.7714	1.7374	1.7015	1.6725	1.6552
Jacket water used per min. (lb)	10.99	10.94	11.06	11.68	11.63	11.57	11.66	11.12	10.94	12.03
" " inlet temp.(°C)	18.19	18.166	18.078	18.101	18.263	18.448	18.623	18.731	18.495	18.355
" " outlet temp.(°C)	22.251	22.255	22.065	21.896	21.99	22.113	22.291	22.551	22.413	21.973
Air Inlet Temperature (°C)	16.3	16.6	16.48	16.25	16.04	15.94	15.8	15.3	16.74	16.64
Air Discharge Temp. (°C)	94.616	93.966	92.7	93.233	92.666	91.666	91.383	90.316	89.55	88.916

OBSERVATIONS.

Delivery Pressure 70 lb. per sq.in.g.

	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height(" of Hg)	29.32	29.32	29.58	29.58	29.58	29.58	29.59	29.59	29.59	29.59
R.P.M. of Compressor Shaft	275.17	274.75	275	275	275.25	275.33	276.4	275.5	275.75	276.1
Average Piston Speed(ft/min)	275.17	274.75	275	275	275.25	275.33	276.4	275.5	275.75	276.1
Volts	416	416	417	418	418	415	415	417	415	415
Amps.	11.62	11.56	11.48	11.42	11.4	11.27	11.23	11.17	11.12	11.04
Air Discharge per min.(lb)	1.8352	1.79896	1.7888	1.7402	1.71066	1.6881	1.669	1.6307	1.616	1.5622
Jacket Water used per min. (lb)	13.98	14.08	13.26	12.94	12.98	13.16	13.01	13.72	14.37	14.84
" " inlet temp.(°C)	18.813	18.93	18.448	18.4	18.328	18.28	18.393	18.393	18.64	18.76
" " outlet temp.(°C)	22.196	22.29	22.11	22.205	22.14	22.01	22.035	21.851	21.873	21.828
Air Inlet Temperature (°C)	16.3	16.04	17.44	18.07	18.13	17.6	16.575	16.975	16.6	16.1
Air Discharge Temperature (°C)	99.683	98.516	98.766	98	96.133	95.066	95.216	94.366	93.133	91.083

OBSERVATIONS.

Delivery Pressure 80 lb. per sq.in.g.

	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height (" of Hg)	30.05	30.05	30.05	30.07	30.07	30.07	30.07	30.12	30.12	30.12
R.P.M. of Compressor Shaft	275.33	275.25	275.5	275.5	275.25	276	276.25	275.17	275	275
Average Piston Speed(ft/min)	275.33	275.25	275.5	275.5	275.25	276	276.25	275.17	275	275
Volts	415	415	415	415	416	416	415	415	416	415
Amps.	11.82	11.72	11.66	11.57	11.55	11.46	11.36	11.34	11.29	11.19
Air Discharge per min.(lb)	1.8104	1.7765	1.7422	1.7101	1.678	1.6514	1.61075	1.5788	1.54194	1.50406
Jacket Water used per min. (lb)	14.24	14.12	14.27	14.18	15.93	15.95	15.96	13.12	12.94	12.2
" " inlet temp.(^oC)	18.458	18.401	18.328	18.398	18.421	19.02	19.513	18.376	18.256	18.135
" " outlet temp.(^oC)	22.255	22.185	22.07	22.091	21.751	22.283	22.718	22.463	22.38	22.483
Air Inlet Temperature (^oC)	18.7	18.5	18.3	17.45	17.24	17.27	17.15	22.36	21.875	20.86
Air Discharge Temperature (^oC)	105.05	103.883	103.066	102.133	100.35	99.366	97.683	98.233	97.133	96.433

O B S E R V A T I O N S.

Delivery Pressure 90 lb. per sq.in.g.

	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height(" of Hg)	30.14	30.14	30.14	30.14	30.2	30.2	30.2	30.2	30.2	30.2
R.P.M. of Compressor Shaft	274.5	274.7	274.1	274.3	275	275	275	276	276	276
Average Piston Speed(ft/min)	274.5	274.7	274.1	274.3	275	275	275	276	276	276
Volts	415	415	416	415	415	415	416	415	416	415
Amps.	12.03	11.92	11.84	11.75	11.69	11.57	11.5	11.4	11.32	11.26
Air Discharge per min.(lb)	1.76311	1.713	1.6764	1.6346	1.613	1.5623	1.5241	1.48544	1.45355	1.4252
Jacket Water used per min. (lb)	15.6	15.53	15.6	15.74	13.94	13.3	12.72	12.68	15.81	15.81
" " inlet temp.(^oC)	18.17	18.385	18.856	19.136	18.255	18.34	18.316	18.216	17.911	18.403
" " outlet temp.(^oC)	21.925	22.165	22.488	22.718	22.375	22.625	22.733	22.47	21.341	21.951
Air Inlet Temperature (^oC)	19.7	19.2	18.608	18.083	20.333	20.375	20.3	19.317	19	19.133
Air Discharge Temp. (^oC)	110.7	109.15	107.95	106.7	105.833	105.511	104.111	102.383	100.233	101.133

OBSERVATIONS.

Delivery Pressure 100 lb. per sq.in.g.

	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
Barometric Height(" of Hg)	29.56					29.56				29.56
R.P.M. of Compressor Shaft	274					274.25				274.5
Average Piston Speed(ft/min)	274					274.25				274.5
Volts	417					417				417
Amps.	12.13					11.68				11.31
Air Discharge per min.(lb)	1.6854					1.4766				1.3103
Jacket Water used per min.(lb)	14.15					13.12				11.76
" " inlet temp.(°C)	17.473					17.743				17.043
" " outlet temp.(°C)	21.92					22.39				22.053
Air Inlet Temperature (°C)	16.87					16.2				15.77
Air Discharge Temp. (°C)	113.266					106.916				100.9

R E S U L T S

FREE AIR DISCHARGE PER MINUTE(CU.FT.)

VERY SSURE	CLEARANCE VOLUME(% of Stroke Volume)									
q.in.g.	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
[illegible]	31.33	-	-	-	-	31	-	-	-	30.94
[illegible]	30.08	-	-	29.79	-	29.62	-	29.46	-	29.14
[illegible]	29.125	28.86	28.84	28.55	28.35	28.15	28.05	27.86	27.667	27.46
[illegible]	27.95	27.675	27.4	27.11	26.82	26.825	26.666	26.467	26.292	26.025
[illegible]	27.12	26.76	26.483	26.175	26.11	25.683	25.5	25.2	24.914	24.68
[illegible]	26.12	25.775	25.133	24.933	25.192	24.893	24.536	24.207	23.843	23.35
[illegible]	25.47	25.03	24.67	24.36	23.88	23.586	23.13	22.62	22.436	22.2
[illegible]	24.586	24.071	23.83	23.3	22.85	22.514	22.164	21.63	21.431	20.706
[illegible]	23.843	23.379	22.914	22.421	21.979	21.644	21.081	21.006	20.481	19.91
[illegible]	23.23	22.536	22.013	21.413	21.25	20.588	20.075	19.511	19.067	18.7
[illegible]	22.43	-	-	-	-	19.613	-	-	-	17.378

FREEE AIR DISCHARGE PER STROKE(CU.FT.)

VERY SSURE .in.g.	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
	.1120	-	-	-	-	.1109	-	-	-	.1108
	.1077	-	-	.1068	-	.1062	-	.1055	-	.1044
	.1049	.1042	.1038	.1028	.1023	.1015	.1011	.1003	.09977	.09913
	.1008	.09997	.09863	.09781	.09705	.09692	.09642	.09537	.09464	.09363
	.09826	.09707	.09596	.09473	.09461	.09313	.09255	.09147	.09044	.09~~8~~033
	.09479	.09333	.09118	.09059	.09152	.09036	.08923	.08804	.08656	.08461
	.09262	.09101	.08978	.08859	.08686	.08570	.08412	.08219	.08136	.08044
	.08935	.08758	.08668	.08474	.08303	.08179	.08021	.07852	.07773	.07499
	.08662	.08496	.08318	.08137	.07986	.07843	.0763	.07635	.07447	.07241
	.08463	.08204	.08028	.07803	.07727	.07489	.073	.07068	.06908	.06775
	.08185	-	-	-	-	.07152	-	-	-	.0633

DISCHARGE EFFICIENCY (%)

IVERY SSURE sq.in.g.	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
0	97.29	-	-	-	-	96.29	-	-	-	96.12
0	93.58	-	-	92.7	-	92.17	-	91.66	-	90.67
0	91.1	90.44	90.07	89.23	88.8	88.14	87.76	87.14	86.62	86.06
0	87.52	86.78	85.63	84.92	84.26	84.14	83.73	82.79	82.17	81.28
0	85.31	84.28	83.32	82.24	82.13	80.85	80.35	79.41	78.52	77.56
0	82.3	81.02	79.16	78.65	79.45	78.45	77.47	76.43	75.15	73.45
0	80.41	79.02	77.94	76.91	75.41	74.4	73.03	71.36	70.63	69.84
0	77.57	76.03	75.25	73.57	72.07	71.01	69.63	68.17	67.48	65.1
0	75.19	73.75	72.21	70.65	69.33	68.1	66.24	66.28	64.66	62.87
0	73.47	71.22	69.69	67.75	67.08	65.01	63.37	61.37	59.98	58.82
0	71.06	-	-	-	-	62.09	-	-	-	54.95

ISOTHERMAL HORSE POWER.

Isothermal HorsePower is the theoretical horse power required to compress the Free Air Discharge assuming isothermal compression

$$= \frac{P_1V_1 \log_e \frac{P_2}{P_{1-}}}{33,000}$$

where P_1 = inlet pressure (lb./sq.foot)

" P_2 = discharge pressure (lb./sq.foot)

and " V_1 = F.A.D./min. (cu.ft.)

IVERY SSURE sq.in.g.	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
	.587	-	-	-	-	.581	-	-	-	.58
	1,36	-	-	1.34	-	1.34	-	1.33	-	1.31
	1.85	1.83	1.83	1.81	1.8	1.79	1.78	1.77	1.76	1.74
	2.16	2.14	2.12	2.1	2.08	2.09	2.07	2.06	2.04	2.02
	2.42	2.39	2.36	2.34	2.32	2.29	2.27	2.24	2.22	2.2
	2.58	2.55	2.48	2.47	2.49	2.47	2.43	2.4	2.36	2.31
	2.73	2.69	2.65	2.62	2.56	2.53	2.48	2.43	2.4	2.38
	2.82	2.76	2.75	2.69	2.64	2.6	2.55	2.5	2.47	2.39
	2.94	2.88	2.82	2.76	2.71	2.66	2.6	2.58	2.52	2.45
	3.0	2.92	2.85	2.77	2.75	2.67	2.6	2.53	2.47	2.42
	2.99	-	-	-	-	2.62	-	-	-	2.32

INDICATED HORSE POWER.

IVERY SSURE sq.in.g.	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
	.6549	-	-	-	-	.6437	-	-	-	.6315
	1.484	-	-	1.452	-	1.413	-	1.402	-	1.37
	2.219	2.145	2.164	2.172	2.128	2.062	2.058	1.99	2.00	1.98
	2.892	2.869	2.778	2.761	2.72	2.688	2.621	2.592	2.578	2.463
	3.442	3.399	3.303	3.25	3.206	3.12	3.075	3.047	2.999	2.951
	3.809	3.778	3.69	3.625	3.625	3.537	3.512	3.401	3.355	3.23
	4.189	4.102	4.02	3.982	3.85	3.819	3.712	3.612	3.571	3.538
	4.512	4.455	4.342	4.241	4.201	4.069	3.979	3.911	3.833	3.72
	4.728	4.621	4.515	4.4	4.331	4.246	4.111	4.07	3.992	3.879
	4.91	4.75	4.645	4.542	4.429	4.311	4.189	4.08	3.958	3.88
	4.91	-	-	-	-	4.26	-	-	-	3.76

BRAKE HORSE POWER

VERY SURE q.in.g.	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
	1.38	-	-	-	-	1.37	-	-	-	1.36
	2.5	-	-	2.48	-	2.42	-	2.42	-	2.4
	3.3	3.21	3.25	3.31	3.25	3.18	3.19	3.12	3.14	3.125
	4.01	4.03	3.95	3.9~~2~~5	3.91	3.91	3.81	3.8	3.83	3.68
	4.62	4.61	4.5	4.44	4.43	4.33	4.32	4.28	4.28	4.24
	4.95	4.95	4.86	4.83	4.86	4.76	4.75	4.65	4.62	4.45
	5.29	5.23	5.19	5.16	5.04	5.02	4.93	4.82	4.8	4.79
	5.65	5.58	5.5	5.42	5.4	5.26	5.21	5.14	5.09	4.99
	5.87	5.77	5.7	5.6	5.57	5.47	5.36	5.34	5.28	5.16
	6.11	5.98	5.89	5.8	5.73	5.6	5.51	5.4	5.31	5.25
	6.23	-	-	-	-	5.72	-	-	-	5.3

MECHANICAL EFFICIENCY. (%).

IVERY SSURE sq.in.g.	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
	47.4	-	-	-	-	47.0	-	-	-	46.4
	59.3	-	-	58.6	-	58.4	-	57.9	-	57.1
	67.2	66.7	66.6	65.6	65.5	64.8	64.5	63.8	63.7	63.3
	72.1	71.1	70.3	69.9	69.4	68.8	68.8	68.1	67.3	66.9
	74.5	73.8	73.4	73.1	72.3	72.0	71.1	71.2	70.1	69.6
	76.9	76.3	75.9	75.0	74.6	74.3	73.9	73.1	72.6	72.5
	79.2	78.4	77.4	77.1	76.4	76.1	75.2	74.9	74.3	73.8
	79.8	79.8	78.9	78.2	77.8	77.3	76.4	76.1	75.2	74.4
	80.6	80	79.2	78.5	77.8	77.5	76.7	76.2	75.5	75.1
	80.3	79.4	78.8	78.3	77.3	76.8	76.1	75.6	74.5	73.9
	78.8	-	-	-	-	74.4	-	-	-	71.0

COMPRESSION EFFICIENCY.

The compression efficiency is the ratio of the isothermal horse power to the indicated air horse power.

IVERY SSURE sq.in.g.	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
0	89.5	-	-	-	-	90.2	-	-	-	91.8
10	91.6	-	-	92.2	-	94.7	-	94.8	-	95.5
20	83.3	85.3	84.5	83.3	84.6	86.8	86.3	88.9	88	87.9
30	74.7	74.5	76.2	76.1	76.4	77.5	79	79.5	79	82
40	70.3	70.3	71.4	72	72.4	73.3	73.8	73.4	74	74.6
50	67.7	67.4	67.2	68.1	68.6	69.9	69.2	70.5	70.2	71.5
60	65.1	65,6	65.9	65.8	66.5	66.2	66.8	67.3	67.2	67.2
70	62.5	61.9	63.3	63.4	62.8	63.8	64.1	63.9	64.4	64.2
80	62.1	62.3	62.4	62.7	62.6	62.7	63.2	63.3	63.1	63.1
90	61.1	61.5	61.3	61	62	61.9	62	62.1	62.4	62.3
100	60.9	-	-	-	-	61.5	-	-	-	61.7

OVERALL EFFICIENCY.

The overall efficiency is the ratio of the isothermal horse power to the brake horse power input,

$$= \frac{\text{Isothermal H.P.}}{\text{B.H.P. input}} = \text{Compression Efficiency} \times \text{Mechanical Efficiency.}$$

IVERY SSURE sq.in.g.	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
0	42.5	-	-	-	-	42.4	-	-	-	42.6
0	55.4	-	-	54	-	55.3	-	54.9	-	54.6
0	56.1	57	56.2	54.7	55.4	56.3	55.8	56.8	56.1	55.7
0	53.9	53.1	53.6	53.2	53	53.4	54.3	54.2	53.2	54.9
0	52.4	51.9	52.4	52.6	52.4	52.8	52.5	52.3	51.9	51.9
0	52.1	51.5	51	51.1	51.2	51.9	51.1	51.6	51	51.9
0	51.6	51.4	51	50.8	50.8	50.4	50.3	50.4	50	49.6
0	49.9	49.5	50	49.6	48.9	49.4	49	48.6	48.5	47.8
0	50.1	49.9	49.5	49.3	48.6	48.6	48.5	48.3	47.7	47.5
0	49.1	48.8	48.3	47.7	48	47.6	47.1	46.9	46.5	46.1
0	48	-	-	-	-	45.8	-	-	-	43.8

B.H.P. Input per cu.ft. of FREE AIR
delivered per minute.

VERY SSURE	CLEARANCE VOLUME(% ofStroke Volume)									
.in.g.	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
	.0441	-	-	-	-	.0442	-	-	-	.0439
	.0831	-	-	.0832	-	.0817	-	.0821	-	.0823
	.1132	.1112	.1126	.116	.1145	.113	.1138	.112	.1135	.1138
	.1434	.1457	.1442	.1457	.146	.1456	.143	.1436	.14$\overset{5}{\cancel{3}}$5	.1414
	.1705	.1725	.17	.1695	.1695	.1685	.1695	.17	.172	.172
	.19	.192	.1935	.194	.193	.191	.1935	.192	.194	.1905
	.208	.209	.2105	.212	.211	.213	.213	.213	.214	.216
	.23	.232	.231	.233	.2365	.234	.2355	.238	.238	.2415
	.246	.247	.249	.25	.253	.253	.254	.254	.258	.259
	.2625	.265	.2675	.271	.27	.272	.275	.277	.279	.281
	.2775	-	-	-	-	.292	-	-	-	.305

FREE AIR DISCHARGE PER MINUTE PER B.H.P. INPUT
(CU. FT.)

ERY SURE .in.g.	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
	22.7	-	-	-	-	22.6	-	-	-	22.75
	12.03	-	-	12.01	-	12.24	-	12.18	-	12.14
	8.82	8.99	8.88	8.62	8.72	8.85	8.79	8.92	8.81	8.79
	6.975	6.86	6.94	6.86	6.85	6.86	6.99	6.97	6.87	7.07
	5.865	5.8	5.89	5.9	5.9	5.93	5.9	5.89	5.82	5.82
	5.27	5.21	5.17	5.165	5.18	5.23	5.17	5.21	5.16	5.25
	4.81	4.785	4.75	4.715	4.74	4.69	4.69	4.69	4.68	4.63
	4.35	4.31	4.33	4.3	4.23	4.28	4.25	4.21	4.21	4.14
	4.065	4.05	4.02	4.00	3.95	3.95	3.94	3.94	3.88	3.86
	3.81	3.77	3.74	3.69	3.71	3.675	3.64	3.61	3.59	3.56
	3.605	-	-	-	-	3.427	-	-	-	3.28

I.H.P. per cu.ft. of FREE AIR
delivered per minute.

ERY URE .in.g.	CLEARANCE VOLUME (% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
	.0209	-	-	-	-	.02076	-	-	-	.0204
	.04935	-	-	.04874	-	.04769	-	.04758	-	.04701
	.07618	.07432	.07502	.07607	.07506	.07324	.07336	.07145	.0723	.07211
	.1035	.1037	.1014	.1018	.1014	.1002	.09828	.0979	.09806	.09462
	.1269	.1271	.1248	.1242	.1227	.1214	.1206	.1209	.1203	.1196
	.1458	.1466	.1468	.1454	.144	.1421	.1431	.1405	.1407	.1384
	.1644	.1639	.1629	.1635	.1612	.1619	.1604	.1597	.1592	.1593
	.1835	.1851	.1822	.182	.1839	.1807	.1795	.1808	.1788	.1797
	.1983	.1976	.1971	.1963	.1971	.1962	.1951	.1937	.1949	.1948
	.2113	.2108	.2111	.2121	.2084	.2093	.2086	.2091	.2075	.2075
	.219	-	-	-	-	.2172	-	-	-	.2164

FREE AIR DISCHARGE PER MINUTE PER I.H.P.(CU.FT.)

ERY SURE .in.g.	CLEARANCE VOLUME(% of Stroke Volume)									
	4.52	5.02	5.52	6.03	6.53	7.03	7.53	8.04	8.54	9.04
	47.83	-	-	-	-	48,16	-	-	-	49.01
	20.26	-	-	20.51	-	20.96	-	21.01	-	21.27
	13.13	13.46	13.33	13.15	13.32	13.66	13.63	14.00	13.83	13.87
	9.665	9.645	9.863	9.819	9.858	9.979	10.17	10.21	10.2	10.57
	7.879	7.872	8.019	8.052	8.145	8.231	8.29	8.269	8.308	8.362
	6.857	6.823	6.813	6.877	6.947	7.037	6.987	7.119	7.108	7.228
	6.08	6.101	6.137	6.117	6.203	6.176	6.232	6.263	6.284	6.275
	5.449	5.401	5.489	5.494	5.441	5.534	5.571	5.531	5.591	5.567
	5.044	5.06	5.075	5.094	5.075	5.098	5.126	5.162	5.13	5.134
	4.732	4.744	4.739	4.714	4.797	4.776	4.792	4.78	4.818	4.819
	4.567	-	-	-	-	4.605	-	-	-	4.622

APPENDIX III.

SUPPLEMENT TO ARTICLE ON AIR SLIP IN COMPRESSORS.

"ENGINEERING," 7th August 1936.

PLATE Nº 45

-F.A.D./STROKE (CORRECTED FOR SLIP)-

-PLOTTED AGAINST-

-DISCHARGE PRESSURE-

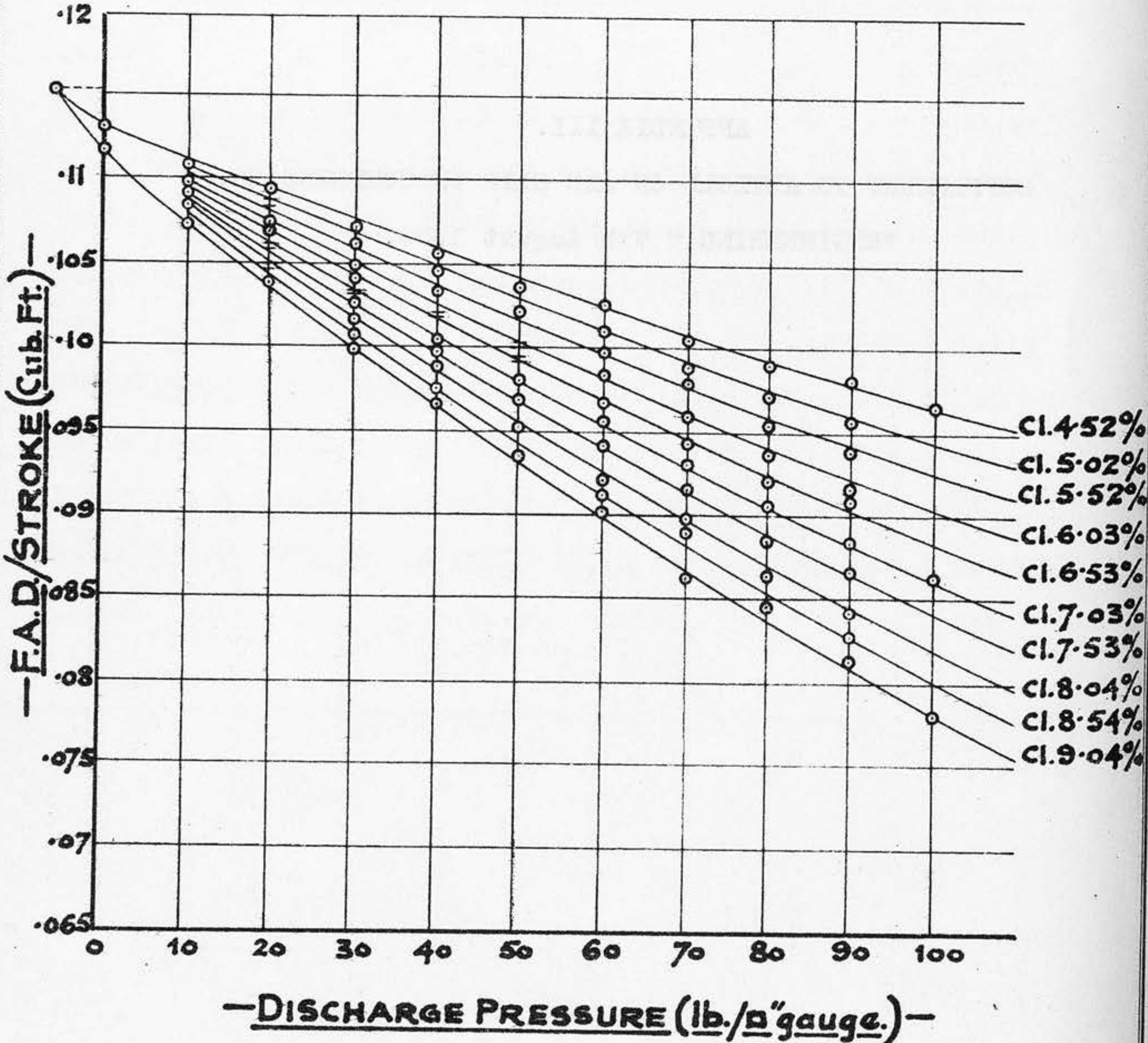

PLATE Nº 46

DISCHARGE EFFICIENCY (CORRECTED FOR SLIP)
against
DELIVERY PRESSURE

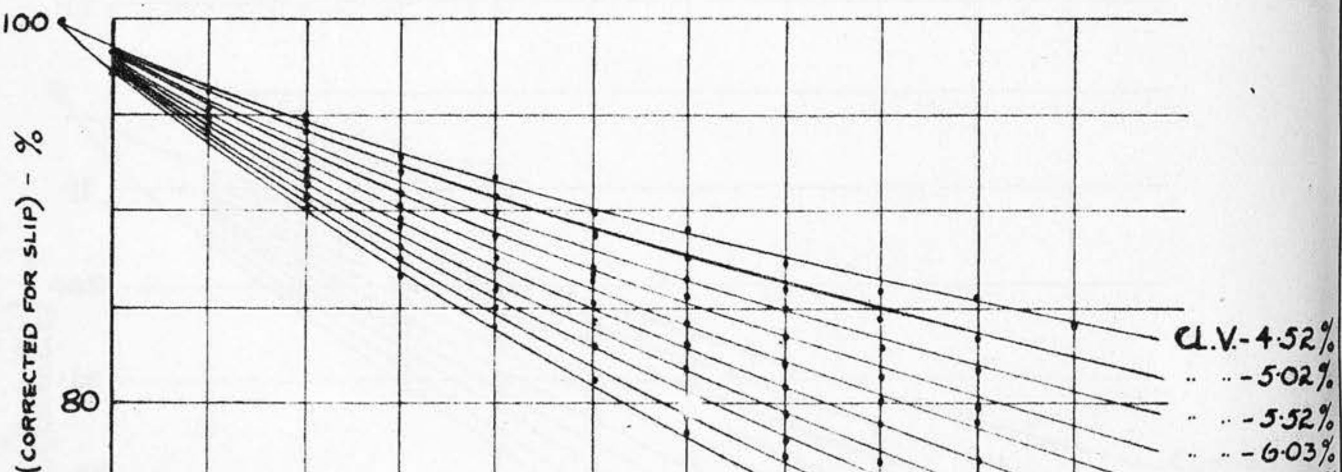

TABLE OF THEORETICAL VOLUMETRIC EFFICIENCIES.

	VOLUMETRIC EFFICIENCY				
	Clearance Volume as a Percentage of Stroke Volume.				
n = 1.2	0%	2.5%	5%	7.5%	10%
harge Pressure lb./sq.in.abs.	100	99.32	98.65	98.07	97.29
" " "	"	95.68	91.37	87.05	82.73
" " "	"	92.42	84.83	77.24	69.66
" " "	"	89.35	78.7	68.04	57.39
n = 1.3					
harge Pressure lb./sq.in.abs.	100	99.38	98.77	98.15	97.53
" " "	"	96.19	92.38	88.57	84.76
" " "	"	93.44	86.88	80.33	73.77
" " "	"	90.92	81.85	72.77	63.7
n = 1.4					
harge Pressure lb./sq.in.abs.	100	99.43	98.86	98.29	97.72
" " "	"	96.59	93.19	89.78	86.38
" " "	"	94.23	88.47	82.7	79.64
" " "	"	92.12	84.25	76.38	68.5

TE: The Volumetric Efficiency is highest when the expansion is adiabatic.