REPORT No. 24

AIR FLOW

THROUGH

POPPET VALVES

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

PREPRINT FROM FOURTH ANNUAL REPORT

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AIR FLOW THROUGH POPPET VALVES

A paper in which are discussed the comparative continuous flow characteristics of single and double poppet valves.

Submitted to the National Advisory Committee for Aeronautics by the CLARKE THOMSON RESEARCH.

Prepared by E. M. NUTTING and G. W. LEWIS.

INTRODUCTION.

In the problem of airplane engine design, the question of the number of poppet valves, location of valves, and the valve lift play an important part in the power charteristics and life of the engine. The Clarke Thomson Research in conducting experiments on an air scavaging engine, under the direction of the National Advisory Committee for Aeronautics, attempted to locate data on valve flow characteristics. Very little data was obtainable, and in connection with the scavaging engine problem the experimental data in this paper was obtained. The number and character of the experiments is not such as to render them final and conclusive, but the results afford a direct comparison of valves singly, in pairs, and of different sizes. Further and more extensive data bearing upon the subject should be experimentally obtained and published.

CLARKE THOMSON RESEARCH.[1]

[1] The Clarke Thomson Research was founded by Mr. Clarke Thomson, of Philadelphia, Pennsylvania, September 23, 1916. Mr. Thomson's object in founding the Research was the advancement of aviation by the investigation and development of devices useful to the art. Mr. Thomson placed the resources of the Research at the disposal of the National Advisory Committee for Aeronautics, and all the activities of the Research are under the direction of the National Advisory Committee for Aeronautics.

REPORT No. 24.

By Clarke Thomson Research.

AIR FLOW THROUGH POPPET VALVES.

This discussion deals particularly with the merits of inlet valves in pairs, as compared with the single inlet perhaps more commonly used. The experimental data presented affords a direct comparison of valves singly and in pairs of different sizes, tested in a cylinder designed in accordance with current practice in aviation engines. Unfortunately, necessity limited the investigation to measurements taken under conditions of continuous flow.

This investigation was undertaken after a wholly unprofitable search for accurate information upon the comparative flow characteristics of single and double inlet valves, based upon actual measurement rather than upon some hypothesis, itself largely a matter of opinion.

By way of preliminary analysis, the application of the law of geometrical similarity presents a strong case for valves in pairs. For example, at a given pressure drop and the same lift, one valve would require a diameter of 4 inches to provide an area of opening equal to that of a pair of valves each of 2 inches diameter. The superficial area of the one 4-inch valve is twice the combined area of the two 2-inch valves, and if opened against a pressure in the cylinder, this is a measure of the comparative forces involved. The 4-inch valve would weigh four times the combined weight of the 2-inch pair, and the necessary spring tension would differ in that proportion, for the same lift and the same engine speed. It may be noted here that, while the above is correct upon the assumption of geometric similarity, the effective valve areas differ from the actual, as the coefficient of efflux varies at different lifts; also, that the weight of a well-designed valve increases somewhat less than the third power of the diameter would indicate.

Mr. H. L. Pomeroy [1] in a discussion which he states is wholly analytical, reaches a conclusion decidedly at variance with the above.

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[1] "Engine Design," H. L. Pomeroy, in Automobile Engineering, June, 1912: Consider one $2\frac{1}{4}$-inch valve and two $1\frac{5}{8}$-inch valves of equal area.

Let radius of port $=r$
valve lift $=h$
port area $=\pi r^2$
Valve opening $=2\pi rh$

If V = velocity of gas through port
and V_1 = velocity of gas through valve opening
then

$$V_1=\frac{Vr^2}{2rh}=\frac{Vr}{2h}$$

$$\therefore \frac{V}{V_1}=\frac{2h}{r} \quad \text{. . . . . (1)}$$

]Footnote continued on p. 6.]

Preceding page blank

Briefly stated, he assumes that two valves of 2.83 inch diameter should be substituted for one of 4-inch diameter (equal cross-sectional port area which requires that the smaller diameter be 0.707 of the larger diameter) and that the valves in each case are lifted 31.65 per cent of their respective diameters. He then computes the hydraulic mean radii for the two cases, applies the laws of friction, and reaches the conclusion that the two valves would have a frictional resistance 39 per cent greater than the single valve.

The contrast is sharp. The tentative conclusion geometrically derived is that two valves of *one-half* the cross-sectional port area and equal opening area, as compared to the single valve would afford the same flow. Mr. Pomeroy's tentative conclusion is that two valves having the *same* cross-sectional port area as the single valve, and the same opening area with a lift 0.707 that of the single valve, would have a frictional resistance 39 per cent greater, and therefore less capacity. This discrepancy seemed to afford ample ground for experimentally determining the relative flow in similar combinations of valves.

This work was carried on by the Clarke Thomson Research in connection with problems involving exhaust gas scavenging at the Bureau of Standards and under the general direction of the National Advisory Committee for Aeronautics. Appreciation of the many courtesies extended by the Bureau of Standards is gratefully acknowledged.

APPARATUS.

The apparatus consisted principally of a centrifugal blower, a model cylinder, and U-tubes for measurements of pressure.

The blower was one of special design with a balanced rotor 11.25 inches in diameter, composed of 10 forward curved blades. An electric motor furnished the power, rheostat control permitting speeds from 3,000 to 6,500 revolutions per minute, corresponding approximately to pressures of 9 to 32 inches of water. The number of impulses varied from 30,000 to 65,000 per minute, affording practically continuous flow. The blower was connected to the cylinder with rubber hose, care being taken to see that the alignment of the hose remained perpendicular to the face of the cylinder at point of entrance throughout the tests.

Frictional resistance is proportional to the square of the velocity.

$$\text{Friction} = F = \frac{KLSV^2}{A}$$

Where S=Perimeter
A=Area
L=Length
V=Velocity

$$\therefore \text{ for the port } f = \frac{KLV^2 2\pi r}{\pi r^2} = K\left(\frac{2LV^2}{r}\right)$$

$$\text{for the valve opening } f_1 = \frac{KLV_1^2 4\pi r}{2\pi r h} = K\left(\frac{2LV_1^2}{h}\right)$$

if $f = f_1$

$$\therefore \frac{2KV^2}{r} = \frac{2KV_1^2}{h} \text{ or } \frac{V^2}{V_1^2} = \frac{r}{h}$$

The cylinder is shown in longitudinal cross section in Plate 1. The cylinder head was carved out of white pine by an excellent pattern maker, and carefully finished as to its interior in accordance with dimension drawings. At the entrance end, the passages leading to the valves were cylindrical in form with axis perpendicular to the cylinder axis and 2.5 inches in diameter, the passages then curved as shown to the ports. The approach to the large valve, which had a

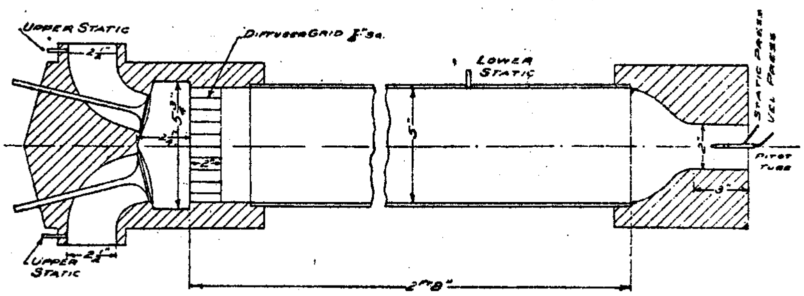

PLATE 1.—Cross through cylinder model.

diameter of 2.5 inches, was circular in cross section at all points. The approach to the pair of valves on the opposite side of the cylinder became narrower in the plane of the cross section shown, and widened laterally to smoothly divide, about 1.5 inches from the ports, into two passages of 1.75 inches diameter. The angle between the valve axis and the cylinder axis was 15 degrees. No valve guides or bushings extended into the passages.

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from equation (1)

$$\frac{V}{V_1}=\frac{2h}{r} \therefore \frac{V^2}{V_1^2}=\frac{4h^2}{r^2}=\frac{r}{h}$$

or $4\,h^3=r^3$

or $h=\frac{r}{\sqrt[3]{4}}= .633\,r$

Assume both valves have a lift of $.633\,r$

Let L=length of pipe of port
V=velocity of gas through port
A=area of port
S=perimeter of port $-2\,\pi\,r$
r=radius of port -1.125

$$f=K\left(\frac{L\,S\,V^2}{A}\right)$$
$$=K\,\frac{2\,L\,V^2}{r}$$
$$=\frac{K\,L\,V^2}{.5625} \text{ for a } 2\tfrac{1}{4} \text{ in. valve.}$$

for $1\tfrac{3}{4}$ in. valve, L and V are the same as before but the hydraulic mean depth A/S

$$=\frac{2\,\pi\,r^2}{4\,\pi\,r}= .405$$

The friction in lbs. per sq. in. is, therefore,

$$\frac{K\,L\,V^2}{.405}$$

$$\text{Hence } \frac{\text{Friction of Double Valve}}{\text{Friction of Single Valve.}}=\frac{.562}{.405}=\frac{1.39}{1}$$

or about 40 per cent more.

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The diameter of the counterbore was 5.75 inches and of the cylinder proper, 5 inches. The valves were seated with a bevel of 30 degrees in the two planes forming the cylinder head. The diffuser shown was constructed of thin brass soldered together and inserted so as to divide the whole area of the cylinder at that point into rectangular passages about seven-eighths inch square and 2 inches long.

The jet at the opposite end of the cylinder was likewise carved out of white pine as shown, and was connected to the cylinder head by a length of 5-inch wrought-iron pipe, smoothly galvanized inside, used to obtain sufficient length for rectification of the air current. Gaskets and shellac were used at the joints and the assembly drawn together with four long bolts extending from end to end, outside the cylinder.

In addition to the single valve with a diameter of 2.5 inches and the pair of valves with diameters of 1.75 inches already mentioned, another pair with diameters of 1.25 inches was tested. False seats were used with this smaller pair, consisting of turned hardwood rings, carefully fitted to the 1.75-inch seats and beveled to receive the smaller valves as shown at Plate 4, Fig. 2. These false seats obviously left a circular shelf or projection 0.25 inch wide immediately above the ports. As a matter of interest, two readings were taken with these shelves projecting above the port, but before running off the main test on these 1.25-inch valves, the lines of the passages were smoothed off by filling in above these projections with putty, giving the approximate stream lines shown.

The valves were all designed on similar lines with the exception that the smallest pair had stems five-sixteenths inch in diameter, to fit the guides used for the larger pair, this dimension being 40 per cent larger than true proportion dictated, equivalent to a reduction of 0.022 square inch or 1.8 per cent of the port area of the smaller pair.

The Pitot tube shown in the jet in Plate 1 was clamped in position at the axis of the jet throughout the tests, velocity readings being taken as later described. The dimensions were three-sixteenths inch outside diameter and about 2.5 inches in length. The impact end was gradually rounded and the static holes were four in number, about 0.02 inch diameter, smoothly perforating the outer wall.

A static tube of one-eighth inch diameter penetrated the central portion of the cylinder, reading static pressure of the air column after passing the valves and the diffuser. This is for convenience termed the "lower static."

Static tubes of one-eighth inch diameter also tapped the flow where the air column entered the passage leading to the valves. These are for convenience termed "upper static," only one being used at a time, as indicated by its position with respect to the valves. All statics were slightly rounded on the inner periphery, and the end kept flush with the inner surface of the cylinder or passage, and so located as to be perpendicular to the direction of air flow.

The upper and lower statics were connected to the two legs of a U-tube to read directly the pressure drop through the valve, and also connected to other U-tubes to read the upper static and lower static head separately.

All U tubes had an inside diameter of about 0.25 inch and were vertical with the exception of one, which was inclined at a slope of 10 to 1 to read with greater accuracy velocity pressures of 3 inches or less.

A centigrade thermometer was clamped with its bare bulb in the air jet at a point about 1.5 inches outside the apparatus. A similar thermometer was hung on the wall for readings of room temperature.

The moisture content recorded is the average for the period indicated, as taken from a recording hygrometer, the variations being but slight, as were those of the barometer. All readings were completed within a period of seven and one-half hours, on May 23, 1918.

MEASUREMENT OF AIR FLOW.

The method used for measuring the velocity and quantity of air is based upon the principles of the impact tube and the jet.[1]

Briefly, the impact tube, when held in and parallel to the air stream, registers a pressure corresponding to the total energy in the air at that point. In case of continuous flow through a pipe of varying cross section, if the impact tube is moved up the axis of the air stream, the pressure registered is constant at all points, except for friction losses. The velocity pressure and static pressure vary with every change of cross section, but the sum of the two, which the impact tube reads, is constant at all points, as the law of conservation of energy indicates. This is similar to Bernouilli's theorem in hydraulics.

Where the section of the pipe is smaller, the velocity of the air must be higher, as the quantity passing all sections of the channel in a given time is constant under conditions of continuous flow. Higher velocity means greater kinetic energy in the moving air particle, and this increment can only arise out of a corresponding diminution of the static pressure.[2]

The jet here used for flow measurement carried this case further, contracting the air column to about one-sixth of its area and discharging into atmosphere at a static pressure equal to atmospheric pressure, or zero U-tube reading, *all* energy in the air being kinetic, read as velocity pressure by the impact tube. This requires that the theo-

[1] An excellent discussion of the use of the impact tube and jet may be found in a paper entitled "The Impact Tube," by Mr. S. A. Moss, vol. 38, Trans. A. S. M. E.

[2] It should not be understood that this conversion of energy (from static to velocity pressure and vice versa) takes place with 100 per cent efficiency, as there is always a conversion loss due to generation of heat by surface and internal friction. In the case of *convergence* or reduction of area, the conversion loss is relatively much less, and the angle of convergence and "stream lining" of the conductor much less important than in case of divergence or increase in area. Convergence reduces static pressure and surface friction and produces a jet effect which rectifies the lines of flow and reduces, or at least does not increase, eddy effects and internal friction. *Divergence*, on the other hand, increases static pressure and surface friction, and unless the angle of divergence be very small, results in conversion losses so large as to indicate great internal friction or eddying, probably in the nature of a rolling motion caused by large velocity differences at different radii. When the angle of divergence reaches 30 degrees on each side of the axis, the theoretical static gain is entirely eliminated by the conversion loss. An excellent discussion of conversion losses with experimental data and efficiency curves may be found in Fan Engineering, pages 120–126, by Willis H. Carrier, member A. S. M. E. Much of the foregoing is quite elementary in character, but it appears that the characteristics of air flow are perhaps less generally understood than most branches of engineering data, and their treatment often seems more complex than illuminating, considered with respect to the average needs of the engineer.

retical orifice be wholly convergent, i. e., that the ratio of absolute pressure of the region into which the jet discharges to the absolute pressure of the region from which the jet discharges be greater than the critical value, 0.5272, for air.

After verifying the fact that throughout the range of velocities used, the impact side of the Pitot tube at any given velocity showed constant readings for various positions in the jet, the Pitot was clamped in position, and readings from the impact side only recorded as velocity pressures. At frequent intervals during the runs the static side of the Pitot was tested, but invariably showed zero reading.

The velocity in the jet roughly equaled the velocity through the average valve opening, being about six times the mean velocity in the cylinder proper. In actual magnitude, the velocities ranged from 1,500 to 19,000 feet per minute, or 25 to 320 feet per second, covering about the extreme range of mean inlet velocities encountered in practice.

Table 1 shows actual and comparative dimensions and areas of the three valve combinations tested.

Table I.

Valve combinations.	Circumference of ports in inches.			Cross sectional area of ports in square inches.		
	One valve.	Total.	Total in per cent of 2.5-inch valve.	One valve.	Total.	Total in per cent of 2.5-inch valve.
2 valves, 1.75-inch diameter	5.498	10.996	140%	2.405	4.810	97.8%
1 valve, 2.5-inch diameter	7.854	7.854	100%	4.909	4.909	100%
2 valves, 1.25-inch diameter	3.927	7.854	100%	1.227	2.454	50%

Diameters and port areas are computed upon the least diameter of the valve or port. In the case of the larger pair, it should be noted that the diameter of 1.75 inches, used for convenience, gives an area about 2 per cent less than that required by the geometrical relation for equal area, namely, $D\sqrt{0.5}=0.7071\ D=1.768$ inches diameter, for the pair to equal the area of the single valve.

The lifts used with each combination of valves were as follows: 0.05, 0.10, 0.20, 0.30, 0.40, 0.50, 0.75, 1.00, and 1.50 inches. These valve lifts were carefully laid off and marked on the stems, and the settings made against fixed indicator points attached to the head of the cylinder. No screw thread or micrometer arrangement was used, and the probable error was relatively much greater at lower lifts. However, independent settings at low lifts checked within the limit of error of about 2 per cent contemplated for the investigation as a whole. Adjustable clamps were used to hold the valves in position when set, and readings taken covering the pressure range available.

After increasing the lift up to 1.5 inches with each valve combination, the valves were reversed; that is, the stems were clamped in the guides so as to project slightly through the ports, the valve heads remaining entirely outside the cylinder, and readings taken to determine the flow through the ports, eliminating the effect of the valve heads as baffle plates in the cylinder. It is often stated in works on design that lifting a valve about one-quarter of its diameter develops a valve area equal to that of the port. This is correct if limited to geometric relations, but seriously misleading if interpreted as providing a substantially equal effective orifice, as will later be developed in the experimental results.

Dr. C. E. Lucke, in his paper on "The problem of aeroplane engine design," presented at the May meeting of the American Society of Mechanical Engineers, 1917, makes the following statement concerning valve lift:

> Coming now to the question of valves, everyone knows that it is of no consequence to lift a poppet valve more than one-quarter of its diameter. It is also true that the valve will work better, and the volumetric efficiency and mean effective pressure be better, the larger the diameter of the valve and the smaller the lift; that is, the valve should not approach the quarter diameter lift. That condition conforms to good principles of gaseous flow.

EXPERIMENTAL DATA.

Tables 2, 3, and 4 contain the data recorded in the tests of the three valve combinations. They are similar in form and refer, respectively, to the 1.75-inch valves, the 2.5-inch valve, and the 1.25-inch valves. The pressure readings are printed as read, in inches of water.

The readings of velocity pressure in the first column were partly taken on a U-tube inclined at a slope of 10 to 1, to facilitate more accurate readings of small quantities, but the decimal point is recorded so as to show pressures in inches of water, vertical head. Readings taken on the inclined tube are given to three places after the decimal point. After reaching the limit of this inclined tube at about 30 inches, or 3 inches actual head, the remaining readings were taken on the usual vertical tubes. This column represents velocity pressure in the jet.

The second column shows the square root of the corresponding reading of velocity pressure in the first column, computed by slide rule. These amounts represent the relative velocities in the jet. The third column or lower static reading refers to the static pressure in the cylinder. Where these readings are small the probable error on account of capillarity or inequality in the tubes is rather large, but they were merely used for a rough check on the pressure drop through the valve tested, shown in the fourth column, which was read from a tube connected to both upper and lower statics.

The square root of pressure drop through the valve, computed by slide rule, appears in the fifth column and is proportional to the theoretical mean velocity through the valve. A separate reading on the upper static appears in the sixth column, and the seventh and eighth columns show in degrees Centrigrade the considerable variations of temperature with the velocity. The ninth column gives the valve lift, and the tenth column the coefficient of efflux, computed on valve areas equal to Π Dh and assuming that the density of the air was atmospheric.

GRAPHICAL COMPARISON OF FLOW.

The data in Table 2, covering the test of the pair of 1.75-inch valves at various openings and at various pressure drops, are shown graphically in Plate 2, the data in Table 3 on the single 2.5-inch valve in Plate 3, and the data in Table 4 on the pair of 1.25-inch valves in

Plate 4. The purpose of this investigation was primarily to secure comparatively accurate comparisons as between the capacities of the different valve combinations, rather than to secure absolute quantitative determination of the flow in any case. It will readily be seen that the velocity and quantity of air flowing through the jet at the outlet of the system will be proportional to the square root of the velocity pressures read by means of the impact tube in the jet. The vertical scale of many of the following graphs is taken from the second columns of Tables 2, 3, and 4, and is termed for convenience, "Proportional flow." It is equally obvious that for accurate quantity determinations, corrections should be made for temperature, pressure, and humidity by the application of well-known thermodynamic formulæ, but this would appear an unnecessary and perhaps misleading refinement, considering the general degree of accuracy here obtainable.

Table 2.

[Kind of measurement, air flow through poppet valves; instrument tested, two valves 1¾ inches diameter, continuous flow; date, May 23, 1918; humidity, 55 per cent; barometer, 755 mm.]

Velocity pressure.	Square root velocity pressure.	Lower static.	Pressure drop.	Square root pressure drop.	Upper static.	Temperature.		Valve lift.	Coefficient efflux.
						Jet.	Room.		
						°	°	*Inches.*	
0.250	0.500	0.20	12.40	3.52	12.75	26.2	23.1	0.05	0.887
.360	.600	.30	16.65	4.09	17.10	26.7	23.1		
.475	.689	.45	22.00	4.69	22.65	29.0	23.3		
.630	.794	.60	28.30	5.33	29.10	31.6	23.5		
.950	.975	1.10	11.05	3.33	12.30	27.9	23.7	.10	.889
1.520	1.233	1.50	14.90	3.86	16.50	29.1	23.8		
1.710	1.31	1.95	19.25	4.39	21.45	30.8	23.9		
2.850	1.69	2.75	26.95	5.19	29.85	34.3	24.0		
2.68	1.64	2.65	8.45	2.91	11.30	29.1	24.0	.20	.818
3.65	1.91	3.50	11.30	3.36	15.00	29.5			
4.50	2.12	4.35	13.95	3.74	18.50	30.3			
6.35	2.52	6.20	19.15	4.38	25.30	32.5			
7.30	2.70	7.10	22.40	4.73	29.65	34.1	24.0		
4.00	2.00	3.80	6.80	2.61	10.85	29.3	24.0	.30	.736
5.20	2.28	5.00	8.90	2.98	14.05	29.5			
6.35	2.52	6.10	10.75	3.28	16.95	30.0			
8.95	2.99	8.60	15.00	3.87	23.65	32.0			
11.20	3.45	10.80	18.80	4.33	29.90	34.3	24.0		
4.80	2.19	4.60	5.70	2.38	10.50	29.0	24.0	.40	.654
6.15	2.48	5.95	7.40	2.72	14.45	29.5			
7.50	2.74	7.30	9.00	3.00	16.40	30.1			
10.50	3.24	10.20	12.35	3.51	22.65	31.9			
12.75	3.56	12.45	14.90	3.86	27.30	33.4	24.0		
5.50	2.34	5.30	4.95	2.22	10.40	28.9	24.0	.50	.608
7.05	2.66	6.90	6.40	2.53	13.45	29.3			
8.70	2.95	8.40	7.55	2.75	15.75	30.0			
11.75	3.42	11.55	10.35	3.22	22.20	31.6			
14.30	3.78	14.15	12.60	3.55	26.90	33.5	24.0		
5.70	2.39	5.50	3.55	1.89	9.30	29 0	24.0	.75	
7.45	2.73	7.30	4.65	2.16	12.05	29.0			.489
8.90	2.98	8.65	5.40	2.32	14.15	29.5			
13.15	3.12	12.95	8.00	2.83	20.95	31.4			
16.20	4.02	16.00	9.65	3.11	25.70	33.5	24.0		
6.25	2.50	6.00	3.05	1.75	9.20	28.7	24.0	1.00	.418
7.80	2.79	7.75	4.00	2.00	11.90	29.0			
9.60	3.10	9.25	4.75	2.18	14.10	29.4			
14.15	3.76	13.80	6.80	2.61	20.65	31.3			
17.20	4.15	16.90	8.30	2.88	25.20	33.0	24.0		
6.40	2.53	6.20	2.80	1.67	9.05	28.1	24.0	1.50	.296
8.30	2.88	8.05	3.55	1.88	11.70	28.9			
9.90	3.14	9.55	4.20	2.05	13.85	29.3			
14.50	3.81	14.10	5.85	2.42	20.05	31.0			
18.05	4.25	17.50	7.20	2.68	24.90	32.5	24.0		
6.20	2.49	6.05	2.15	1.47	8.40	28.6	24.0		
8.20	2.86	8.00	2.80	1.67	10.85	28.8			
10.50	3.24	9.65	3.40	1.84	13.15	29.4		Valve reversed.	
14.35	3.78	14.10	4.85	2.20	19.05	31.0			
17.55	4.18	17.10	5.90	2.43	23.00	32.7	24.2		

Table 3.

[Kind of measurement, air flow through poppet valves; instrument tested, one valve 2¼ inches diameter, continuous flow; date, May 23, 1918; humidity, 56 per cent; barometer, 755 mm.]

Velocity pressure.	Square root velocity pressure.	Lower static.	Pressure drop.	Square root pressure drop.	Upper static.	Temperature. Jet.	Temperature. Room.	Valve lift.	Coefficient efflux.
						°	°	*Inches.*	
0.140	0.374	0.15	12.60	3.55	12.90	29.0	24.6	0.05	0.96
.225	.475	.25	17.10	4.13	17.40	30.8			
.370	.608	.35	23.20	4.82	23.70	32.6			
.435	.660	.45	25.75	5.08	26.30	34.3	25.0		
.505	.710	.65	11.35	3.36	12.25	31.2	25.0	.10	.95
.840	.915	.90	15.50	3.94	16.60	32.0			
1.200	1.097	1.25	20.60	4.55	22.00	33.3			
1.655	1.290	1.70	27.75	5.27	29.50	35.6	25.0		
1.690	1.30	1.70	9.55	3.09	11.50	31.4	25.1	.20	.856
2.330	1.53	2.35	12.90	3.59	15.40	32.0			
3.020	1.74	3.00	16.60	4.07	19.60	33.0			
4.000	2.00	3.85	20.85	4.56	25.40	35.2			
4.750	2.18	4.50	25.00	5.00	29.65	36.3	25.1		
2.70	1.64	2.55	8.35	2.89	11.10	31.2	25.3	.30	.756
3.55	1.88	3.40	11.15	3.34	14.80	31.6			
4.55	2.14	4.40	14.30	3.78	18.85	32.4			
5.80	2.41	5.70	18.40	4.29	24.20	33.8			
7.25	2.69	7.00	22.50	4.75	29.60	35.6	25.3		
3.70	1.92	3.50	6.85	2.62	10.60	30.8	25.6	.40	.728
4.85	2.20	4.70	9.15	3.02	13.95	31.2			
6.05	2.46	5.90	11.45	3.38	17.40	32.1			
7.85	2.80	7.65	14.75	3.84	22.40	33.5			
10.20	3.19	9.85	18.65	4.32	28.20	35.8	25.6		
4.40	2.10	4.20	5.60	2.37	10.00	31.3	25.6	.50	.710
5.80	2.41	5.60	7.40	2.72	13.20	31.5			
7.20	2.68	7.00	9.25	3.04	16.40	32.2			
9.55	3.09	9.30	12.10	3.48	21.50	33.6			
11.85	3.49	11.45	14.75	3.84	26.00	35.2	25.6		
5.40	2.32	5.20	4.20	2.05	9.60	28.9	25.8	.75	.605
7.15	2.67	6.95	5.50	2.34	12.55	30.1			
8.90	2.98	8.60	6.80	2.61	15.45	31.4			
11.90	3.44	11.55	9.05	3.01	20.70	33.5			
14.70	3.88	14.10	10.80	3.28	24.20	35.2	25.8		
5.80	2.41	5.60	3.75	1.94	9.50	31.3	26.3	1.00	.509
7.70	2.78	7.40	4.90	2.22	12.40	31.8			
9.60	3.10	9.25	6.05	2.46	15.40	32.6			
12.90	3.59	12.45	7.95	2.82	20.45	34.3			
15.20	3.90	14.30	9.05	3.01	22.50	35.3	26.3		
6.10	2.47	5.90	3.30	1.82	9.40	31.8	26.6	1.50	.373
8.10	2.85	7.90	4.30	2.08	12.25	32.2			
10.05	3.18	9.80	5.35	2.32	15.30	33.0			
13.50	3.67	13.20	7.05	2.66	20.45	34.5			
15.80	3.98	15.00	7.85	2.80	22.20	35.5	26.6		
6.35	2.52	6.15	2.70	1.64	9.05	31.5	26.6		
8.50	2.92	8.15	3.50	1.87	11.75	32.0			
10.65	3.26	10.30	4.35	2.08	14.75	32.7		Valve reversed.	
14.40	3.80	14.00	5.85	2.42	19.85	34.1			
16.70	4.08	15.80	6.40	2.53	21.20	35.3	26.6		

Table 4.

[Kind of measurement, air flow through poppet valves; instrument tested, two valves 1¼ inches diameter, continuous flow; date, May 23, 1918; Humidity, 53 per cent; barometer, 755 mm.]

Velocity pressure.	Square root velocity pressure.	Lower static.	Pressure drop.	Square root pressure drop.	Upper static.	Temperature. Jet.	Temperature. Room.	Valve lift.	Coefficient efflux.
						°	°	*Inches.*	
0.140	0.374	0.20	12.65	3.56	12.90	30.8	27.0	0.05	0.96
.230	.484	.27	17.45	4.18	17.80	32.5			
.350	.591	.40	24.10	4.91	24.60	34.5			
.420	.648	.45	26.35	5.13	26.85	36.0	27.0		
.570	.755	.65	11.40	3.38	12.05	34.0	27.1	.10	.92
.840	.917	.85	16.35	4.04	16.00	34.5			
1.120	1.060	1.10	20.10	4.49	21.35	35.5			
1.440	1.200	1.40	25.60	5.06	26.10	37.5	27.1		
1.230	1.11	1.20	10.50	3.24	11.85	33.5	27.1	.20	.691

Table 4—Continued.

Velocity pressure.	Square root velocity pressure.	Lower static.	Pressure drop.	Square root pressure drop.	Upper static.	Temperature.		Valve lift.	Coefficient efflux.
						Jet.	Room.		
						°	°	*Inches.*	
1.750	1.32	1.70	14.40	3.79	16.25	34.5	....	....	....
2.330	1.53	2.30	19.25	4.39	21.70	36.0	....	....	....
2.930	1.71	2.95	23.60	4.86	26.60	37.0	27.1	....	....
1.940	1.39	1.90	9.35	3.06	11.30	34.0	27.2	.30	.613
2.650	1.63	2.55	12.60	3.55	15.35	34.3	....	....	....
3.550	1.88	3.35	16.50	4.06	20.00	35.0	....	....	....
4.400	2.10	4.20	20.55	4.54	24.75	37.0	....	....	....
5.200	2.28	4.70	24.20	4.92	29.20	39.0	27.2	....	....
2.50	1.58	2.30	8.90	2.98	11.40	34.5	27.2	.40	.528
3.35	1.83	3.15	12.00	3.46	15.35	34.7	....	....	....
4.35	2.09	4.10	15.70	3.96	19.90	35.5	....	....	....
5.30	2.30	5.30	20.10	4.49	25.65	38.5	....	....	....
6.25	2.50	6.05	22.20	4.71	28.25	39.0	27.2	....	....
2.70	1.64	2.50	8.50	2.92	11.30	32.0	27.5	.50	.459
3.75	1.94	3.55	11.55	3.39	15.10	33.5	....	....	....
4.80	2.19	4.70	15.15	3.89	20.00	35.0	....	....	....
5.85	2.42	5.75	18.40	4.29	24.20	36.5	27.5	....	....
3.10	1.76	2.95	8.35	2.89	11.45	33.5	27.9	.75	.327
4.05	2.01	3.90	10.80	3.29	14.80	34.3	....	....	....
5.20	2.28	5.05	14.00	3.74	19.20	35.3	....	....	....
6.60	2.57	6.40	17.70	4.20	24.05	37.0	27.9	....	....
3.10	1.76	2.95	7.80	2.79	10.90	34.0	27.9	1.00	.254
4.20	2.05	4.00	10.45	3.24	14.60	34.6	....	....	....
5.75	2.40	5.60	14.25	3.78	20.00	35.5	....	....	....
7.10	2.66	6.90	17.65	4.20	24.60	37.3	....	....	....
3.35	1.83	3.20	7.90	2.81	11.25	34.0	27.9	1.50	.174
4.35	2.08	4.10	10.25	3.20	14.45	34.3	....	....	....
6.00	2.45	5.85	13.60	3.68	19.30	35.6	....	....	....
7.10	2.66	6.90	16.50	4.06	23.50	37.0	....	....	....
3.35	1.83	3.15	7.50	2.74	10.90	33.5	27.9	Valve reversed.	....
4.60	2.15	4.30	10.05	3.17	14.60	34.2	....		
5.90	2.43	5.75	13.25	3.64	19.10	35.0	....		
7.25	2.69	7.10	16.20	4.02	23.35	36.5	....		

The horizontal scale is laid off to the square root of the pressure drop through the valve combination tested, which is deemed much preferable for the present purpose to the use of the pressure drop itself. With the limited range of pressures available, and the considerable variation in valve areas tested, the curves obtained by plotting to the pressure drop would be so distributed that no single ordinate would intercept all the curves.

The system used, however, produces straight line graphs passing through the center of coordinates, as the velocity through the valve is proportional to the square root of the pressure drop, for any given lift, and these graphs may, therefore, be extended to intercept any particular ordinate without appreciable error, which greatly facilitates the study of the results obtained.

If desired, the slope of each graph may be arithmetically determined by computing the average ordinate and the average abscissa for the points on any one line, applying suitable weights to any readings deemed of unequal value, the slope of the graph being fixed by the ratio of such averages.

Wherever "proportional flow" is used as a basis for plotting in the various plates shown, it should be remembered that this refers to flow through the same jet in all cases, without respect to the valve opening or pressure drop causing the flow, and that the various results so obtained may, therefore, be directly compared, owing to the use of this jet as the common medium of measurement in all tests.

As to these plates 2, 3, and 4, it is true that the scale is small, and that plotting to square roots tends to reduce the magnitude of any irregularities in the points obtained, but the close coincidence of the points with the straight graphs passing through the origin seems to warrant the conclusion that the actual velocity through the valve at any given lift varies directly with the square root of the pressure drop, at least within the limits of these tests, as does the theoretical velocity. It further follows as a general rule within these limits that the coefficient of efflux does not vary with the pressure drop. Certain limitations upon this conclusion may be required, however, and will be discussed in connection with the graphs showing the variation of the coefficient of efflux with the lift.

Curves of equal velocities have been superimposed upon the graphs of plates 2, 3, and 4 to show the approximate velocities through the valves in feet per second. At about the average conditions of the tests, namely, 80° F., 55 per cent humidity, 68° wet bulb, and 29.72 inches barometer, air weighs 0.0717 pounds per cubic foot. Inserting this value in the equation $V=18.275\sqrt{p/w}$ gives $V=68.2\sqrt{p}$. In other words, assuming air at this density, multiplying the vertical scale by 68.2 gives actual velocity through the jet in feet per second, and applying the same correction to the horizontal scale gives *theoretical* velocity through the valves. The actual velocity through the valves may then be obtained either by applying the ratio of areas to the vertical scale of jet velocities or by applying the proper coefficient of efflux to the horizontal scale of theoretical valve velocities.

Curves representing actual velocities of 100, 200, and 300 feet per second through the valves have been laid off by the former method and agree fairly with results obtained by the latter method, except as to irregularities in some of the points used for plotting the coefficient curves later presented.

These curves may be used to approximate the actual pressure drop necessary to produce a given velocity. For example, in plate 2 it is seen that the pair of 1.75-inch valves with 0.20 lift indicates a velocity of 200 feet per second at the ordinate corresponding to 3.55 in the horizontal scale or 12.6 inches of water or 0.455 pounds per square inch as the required pressure drop. It should be noted that these velocity curves are merely approximate and that errors up to 5 per cent or so may be found.

In plate 4, figure 2, will be found an illustration of the manner in which a pair of 1.25-inch valves were seated in the cylinder for testing. The closely shaded sections represent the false seat of hard-

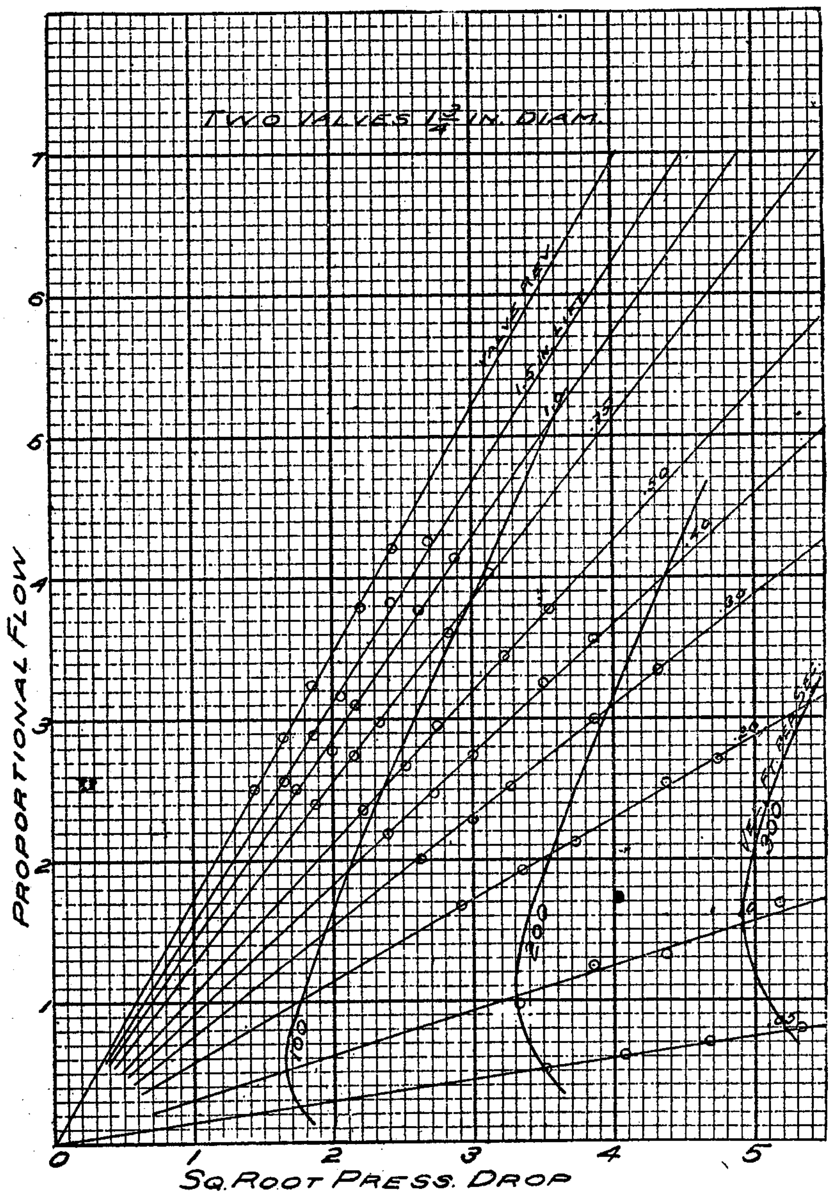

TWO VALVES 1 3/4 IN. DIAM.
PROPORTIONAL FLOW
SQ. ROOT PRESS. DROP
0
1
2
3
4
5
6
7

PLATE 2.

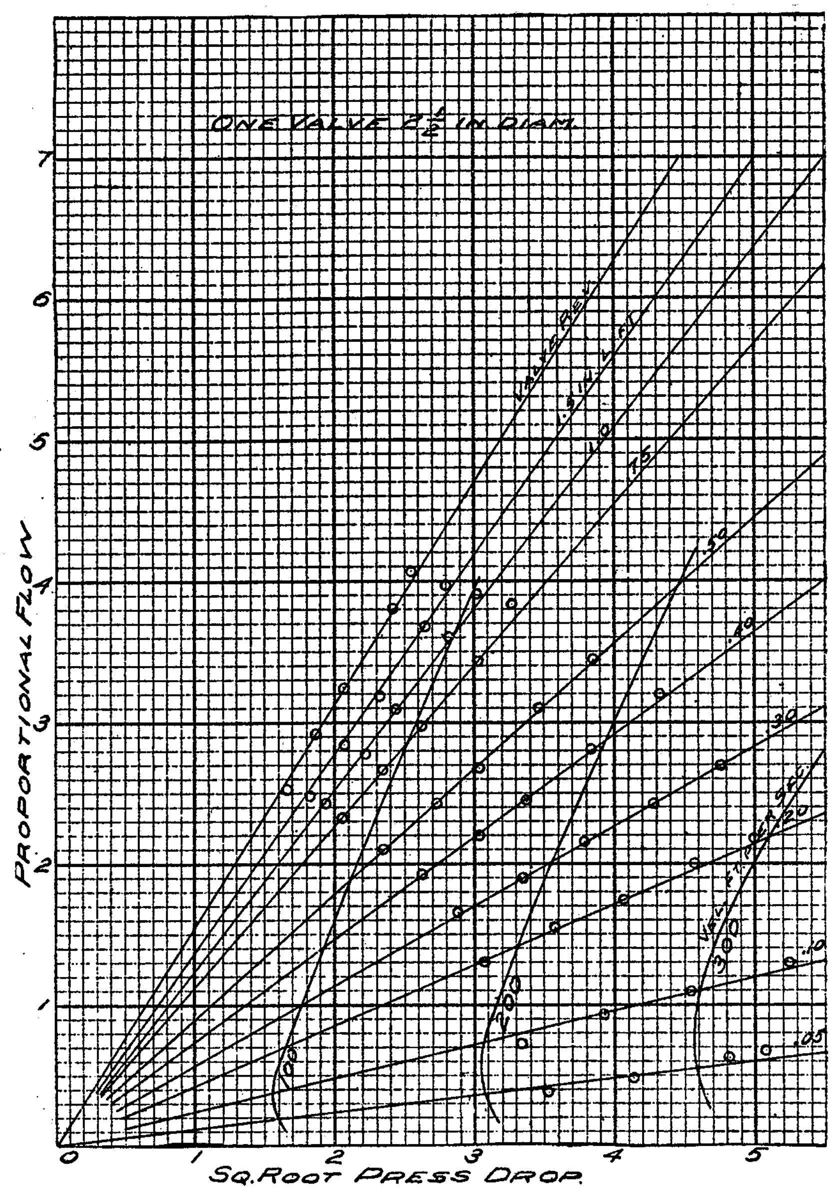
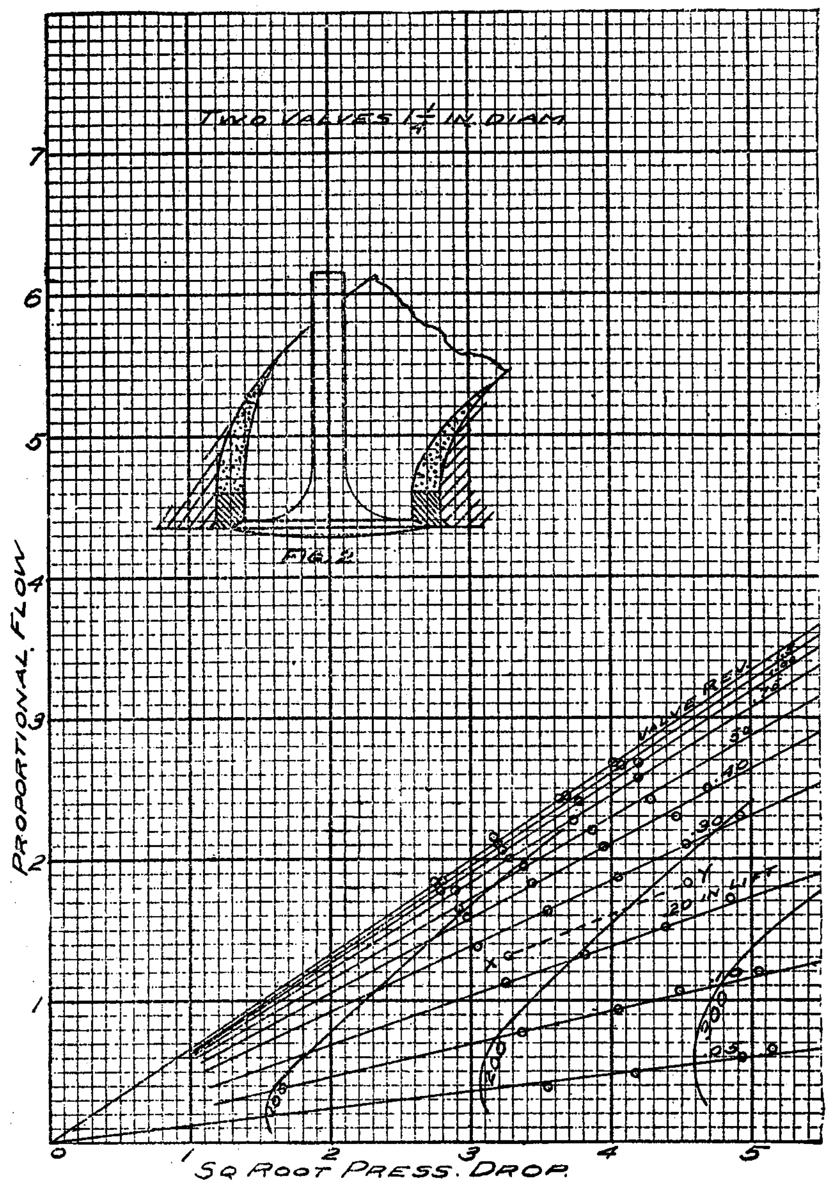

ONE VALVE $2\frac{1}{2}$ IN. DIAM.
PROPORTIONAL FLOW
SQ. ROOT PRESS DROP
0
1
2
3
4
5
6
7
VALVE REV.
.75
.50
.40
.30
.20
.10
.05
VEL. FT. PER SEC.
100
200
300

PLATE 3.



TWO VALVES 1¼ IN. DIAM
FIG. 2
PROPORTIONAL FLOW
SQ ROOT PRESS. DROP
VALVE REV.
IN LIFT
0
1
2
3
4
5
6
7

PLATE 4.

wood, and the dotted sections indicate the putty used to join the ports of the false seats smoothly to the passages.

Before making the putty joint as shown, two tests of purely collateral interest were made at 0.3 inch lift, to show the effect of the sharp ledge in the passage. The results are shown by the dotted line $x-y$, the flow being 13 per cent less for any given pressure drop, than with the passage stream-lined as described. It is, therefore, evident that any projections or sharp angles in the passage tend do greatly reduce the flow, as might well be anticipated.

The very common custom of finishing the inlet passages with two bores meeting at an angle of about 110 degrees certainly puts a heavy restriction upon valve efficiency, but doubtless constructional convenience may be held to justify the practice.

Interesting experimental work could be done on the design of valve guides, possibly joining them to the wall of the passage with a web of stream-line section. Valves with an extremely heavy fillet have been used in the R. A. F. 3a engine, doubtless with the idea of guiding the air current smoothly to the valve opening. Venturi effects in the passage immediately above the valve might be productive of excellent results.

The use of the putty as above described to reduce the size of the passage to the diameter of the small valve, introduced a converging nozzle effect which doubtless tended to direct the air stream inward toward the valve stem and thereby slightly impair the efficiency of this small pair of valves.

It is evident that the intercepts on any ordinate on plate 2, 3, or 4 will represent the variation of the flow with the valve lift, at the pressure drop corresponding to the ordinate selected.

Plate 5, figure 1, presents such curves for the three valve combinations, plotted from intercepts on the ordinates corresponding to a pressure drop of 16 inches of water, the ordinates numbered 4 in the square root scale. The relation would have been the same had any other ordinate been chosen, but the quantities would have been different.

The curve of flow for the single 2.5-inch valve lies between those of the pairs of valves at all lifts. It is found to be very nearly equal to that of the smaller pair for low lifts and approximates that of the larger pair at the higher lifts.

These curves are plotted against valve lift in inches, but for convenience the points equal to one quarter and one-half diameter have been marked on each curve. By interpolation between these points and others similarly located, the approximate curve of flow for two valves of 1.5 inch diameter is presented. This indicates a flow quite closely equal to that of the single 2.5-inch valve, up to a lift of about 0.6 inch.

The vertical intercepts of these four curves on ordinates corresponding to various valve lifts are compared in Table 5, in percentages of the flow of the single valve.

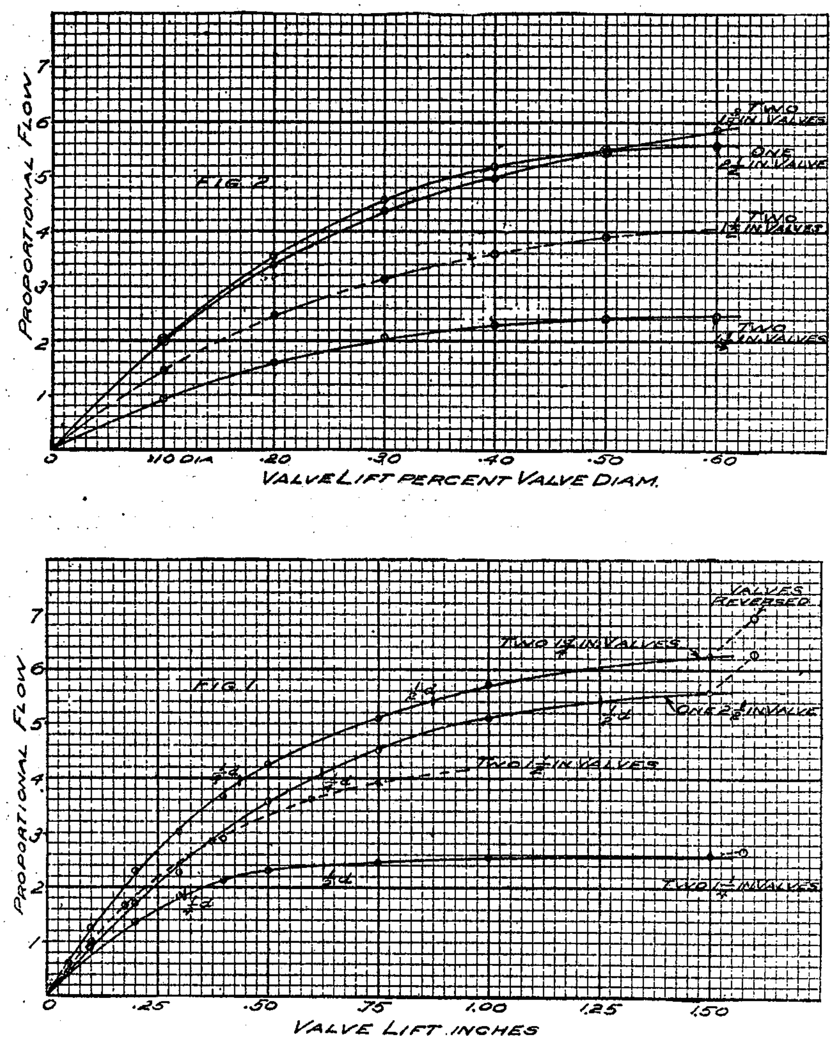


PLATE 5.

Table 5.

Lift in inches	0.125	0.25	0.375	0.500	0.625	0.750
2 valves, 1.75 inches	137%	129%	125%	119%	116%	113%
[1] 2 valves, 1.50 inches	114%	106%	101%	94%	90%	87%
1 valve, 2.5 inches	100%	100%	100%	100%	100%	100%
2 valves, 1.25 inches	85%	77%	72%	65%	59%	54%

[1] Interpolated.

The points connected to the curves by broken lines indicate, to vertical scale only, the flow with the valves reversed. This might be considered equivalent to the flow with the valves at an infinite lift, which agrees with the horizontal trend of the curves, but more practically, these points represent the maximum limit of flow through the respective ports at this pressure drop. These curves, comparing performance upon a basis of equal lift in inches, are particularly applicable where it is conceded that mechanical features generally limit the possible lift regardless of valve diameter.

On the other hand, it is often asserted that the proper limit of valve lift is a function of the diameter, and for purposes of comparison on this basis, figure 2 of plate 5 has been prepared from the curves last discussed, changing the horizontal scale to read in per cent of the diameter of each valve. In the case of pairs of valves, the flow of both is plotted against the lift, expressed in per cent of the diameter of one valve only.

The result of this transposition is at once apparent. The intercepts on any ordinate very closely agree with the proportionate cross-sectional port areas of the several valve combinations, and in the case of the two curves corresponding to valve combinations with equal cross-sectional port area, the curves coincide within the probable error of the work. Up to a lift of 0.5 diameter the coincidence is all the more exact if it be remembered that the two 1.75-inch valves have an area about 2 per cent less than the single 2.5-inch valve.

From this it would appear reasonable to infer that under fairly similar conditions different valves or combinations of valves have capacities in proportion to their respective cross-sectional port areas, when the lift in each case is same per cent of their respective diameters.

It also seems logical to infer that the theory of the hydraulic mean radius has but little application to the losses in poppet valves, it being more properly applicable to what may, for convenience, be termed surface friction, or actual rubbing of the moving fluid upon the surrounding wall, whence its derivation—the relation of cross sectional area to perimeter in contact with the moving fluid.

In the case of continuous flow through a pipe or conduit, pressure losses may be classified as friction losses and dynamic losses, although no sharp distinction can be drawn. Dynamic losses are due to change in direction, either of the whole column or its lesser parts, as at elbows, nozzles, or offsets; and friction of the fluid against the walls undoubtedly causes a rolling motion with change of both direction and velocity in the adjacent particles. The change in direction at an elbow will cause a greater pressure with greater friction on the outer side. An easy-radius elbow is ordinarily estimated to cause a pressure loss equal to the friction loss in 10 diameters of straight pipe, but a right-angle or mitered joint in the pipe will cause a loss equal to the friction loss in nearly 50 diameters.[1] It is thus evident that where marked changes in direction take place in a length of but two or three diameters, the dynamic losses may be many times as great as the losses due to friction, and the case of the inlet passage terminating in a poppet valve falls in this class.

[1] "Loss of Pressure Due to Elbows," Frank L. Busey, Proc. of Am. Soc. of Heating and Ventilating Engineers, 1913.

As a rough comparison of probable friction loss and dynamic loss, it may be assumed that the friction in the passage and at the lip of the valve is equivalent to that of 5 diameters of straight pipe at the same velocity. From Mr. Busey's experiments dynamic losses might be expected equal to the friction loss in about 8 diameters due to the curvature of the passage, and further dynamic losses equal to the friction in at least 30 diameters due to the sharp change of direction at the valve seat, —60 degrees at low lifts with a 30-degree seat. If this comparison is within the limits of fair approximation, Mr. Pomeroy's 39 per cent greater friction loss is only applicable to about 15 per cent of the total loss, or about 6 per cent less capacity would be expected from a pair of valves having 0.7 the diameter and 0.7 the lift of a single valve. From the data here obtained, it appears that the two valves have only about 2 per cent less capacity at 0.7 the lift.

For ready comparison, Table 6 has been prepared from the vertical intercepts on the curves of figure 2, plate 5, showing the relative cross-sectional port areas and capacities in per cent of the area and capacity of the single 2.5-inch valve, each valve being lifted the same per cent of its diameter.

Table 6.

	Relative area.	Relative flow at lift equal to—			
		0.1 diameter.	0.15 diameter.	0.20 diameter.	0.25 diameter.
2 valves, 1.75 inches diameter	98%	96%	96%	96%	96%
2 valves, 1.50 inches diameter[1]	72%	70%	69%	69%	69%
1 valve, 2.5 inches diameter	100%	100%	100%	100%	100%
2 valves, 1.25 inches diameter	50%	46%	45%	45%	45%

[1] By interpolation.

It is evident from an inspection of the curves on plate 5 that a lift equal to one-quarter diameter develops less than 67 per cent of the full capacity of the port, and that a lift of one-half diameter develops 80 to 90 per cent of the full capacity.

The coefficient of efflux is taken as the ratio of the observed mean velocity through the valve to the mean velocity which would theoretically result from an equal pressure drop. Assuming that the temperature, density, and humidity of the air are the same at the valve as at the jet, this coefficient may be obtained directly from the relation of the areas and the proportional velocities set forth in Tables 2, 3, and 4. The proportional velocity at the jet multiplied by the ratio of the jet area to valve area gives the proportional velocity through the valve. If the ratio of this velocity to the square root of the pressure drop be taken, the result is the coefficient of efflux. To be more exact, this should be multiplied by 0.99, the coefficient of the jet.

The above short method may be justified by developing the usual equation $V=\sqrt{2\,g\,h}$ in the units here most convenient:

V=velocity in feet per second.

g=acceleration constant of gravity in feet per second.

h=head of air in feet causing the flow.

Substituting the head in inches of water:

$$V=\sqrt{2\,g\,p\,\frac{62.31}{12w}}=\sqrt{2\,g\,p\,x\,\frac{5.193}{w}}=18.275\sqrt{\frac{p}{w}}$$

where w is the weight of water in pounds per cubic foot, 62.31 is the weight of water in pounds per cubic foot, and p is the pressure head in inches of water.

This equation is deemed sufficiently accurate for the low pressures here subjected to examination.

Now, if A = the jet area, and
a = the valve area,

the mean velocity through the valve is $18.275\,\frac{A}{a}\sqrt{\frac{p}{w}}$ where p is the velocity pressure and w the density of the air at the jet.

The theoretical mean velocity through the valve is $18.275\sqrt{\frac{P}{W}}$, where P and W are, respectively, the pressure drop and the density of the air at the valve.

$$\text{The Coefficient of Efflux}=\frac{18.275\,\frac{A}{a}\sqrt{\frac{p}{w}}}{18.275\sqrt{\frac{P}{W}}}\text{ or }\frac{A\sqrt{p}}{a\sqrt{P}}$$

where the density of the air is the same in the jet and in the valve

In computing the coefficient the valve area has been taken as $\pi D\,h$ for all lifts. It is realized that for small lifts the aid of trigonometrical formulæ may be invoked to determine accurately the least area of opening, but the same formulæ are not applicable at higher lifts. Moreover, they are only justifiable upon the theory that the lines of flow are parallel to the slope of the valve seat, a condition which certainly does not obtain for any except the smallest lifts.

In plate 6 the coefficients of efflux will be found, plotted against valve lift in inches in figure 1, and against lift in per cent of diameter in figure 2. These coefficients are considerably higher at low lifts, a feature somewhat difficult to explain satisfactorily. Both friction and dynamic losses should be greater at low lifts, as the ratio of perimeter to area is then greater and the angular deflection sharper. It seems probable that there is an approximation to a jet action at low lifts, the discharge taking place into a region of relatively low pressure, somewhat after the manner of the true jet used for measurement at the outlet end of the cylinder. The comparatively high discharge efficiency of any such jet seems to make this the most probable explanation of the high coefficients.

If such jet action takes place, the pressure in the valve area should approximate that of the cylinder itself, and the theoretical velocity through the valve should be computed upon the lower pressure rather than the higher. This would reduce the error involved in computing the theoretical flow and coefficient upon the assumption of atmospheric density in the valve, as has been done.

The maximum static pressure in the cylinder was 17.5 inches of water. As a pressure of 1 inch of water is equal to a pressure of

0.5768 ounce per square inch, this would equal a pressure of 10.09 ounces, or 0.631 pound, per square inch, or an absolute pressure of 15.23 pounds per square inch, 755 millimeters observed atmospheric pressure being equal to 14.60 pounds per square inch. The density of the air varying with the absolute pressure and the ratio of abso-

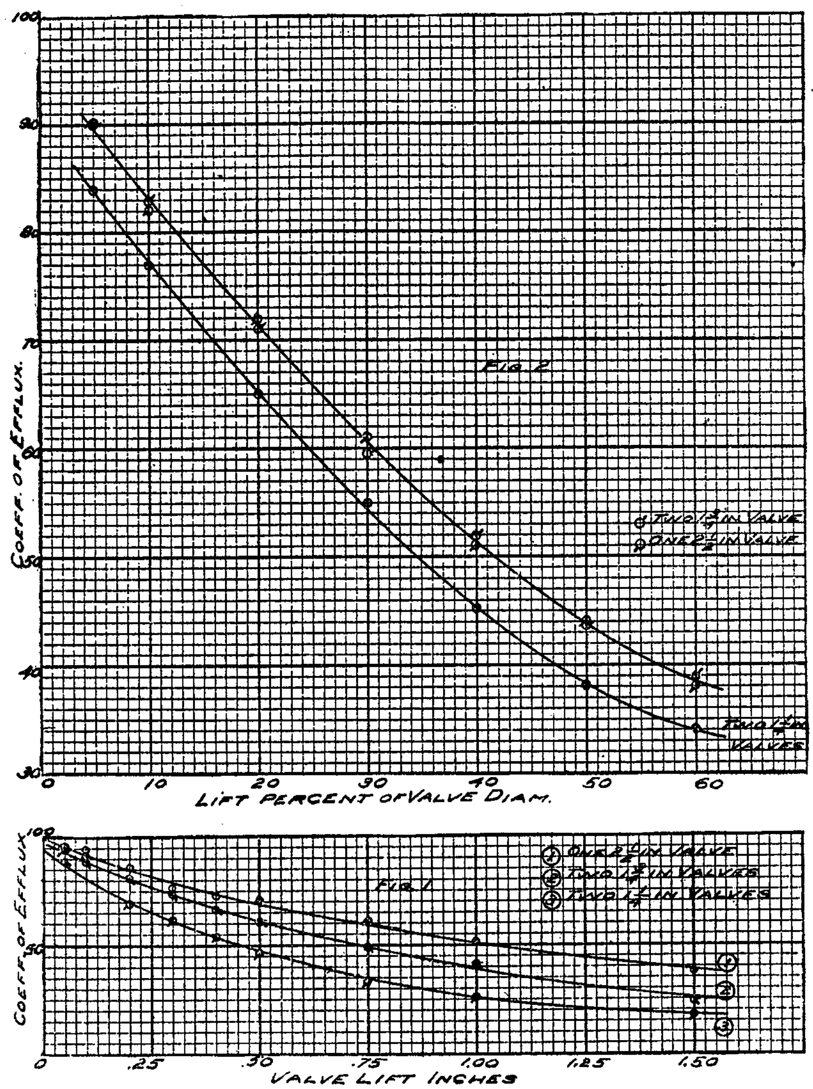


PLATE 6.

lute pressures being 1.046, the error involved under the above assumptions would be about 2.2 per cent as the density of the air enters the equation under the radical sign. This error would be materially less at the lower lifts, the pressures in the cylinder were then being considerably less. No appreciable error would appear to be introduced by assuming equal temperature and equal humidity at valve and jet for any given valve opening and pressure drop.

Referring again to plate 6, it will be noted that in figure 1, where the coefficients are compared at the same absolute lift, the differences between the three valve combinations are quite considerable, and that at the very low lifts the points plotted present some irregularities. The curves have been drawn to conform to the greatest number of points reasonably possible, and the curves in figure 2 have been plotted from those in figure 1. The points for the two larger combinations so nearly coincide in figure 2 that but one line has been drawn.

The relative intercepts of the coefficient curves in figure 1 at various absolute lifts, expressed in per cent of the values for the single 2.5-inch valve, are presented in Table 7.

Table 7.

	Relative coefficient of efflux.				
Valve lift in inches	0.125	0.25	0.375	0.50	0.625
2 valves, 1.75 inches diameter	96%	94%	91%	89%	86%
1 valve, 2.5 inches diameter	100%	100%	100%	100%	100%
2 valves, 1.25 inches diameter	87%	79%	73%	69%	64%

In figure 2 it will be seen that when compared on a basis of equal valve lifts, expressed in per cent of diameter, the coefficients are much more nearly equal, the curves for the two larger combinations coinciding, and that for the small valves being but little lower. It seems entirely probable that even this small difference is largely caused by the converging lines of the passages leading to these small valves, as before explained. The comparative values are here shown.

Table 8.

	Relative coefficient of efflux.				
Valve lift in per cent of diameter	0.10	0.15	0.20	0.25	0.30
2 valves, 1.75 inches diameter	100%	100%	100%	100%	100%
1 valve, 2.50 inches diameter	100%	100%	100%	100%	100%
2 valves, 1.25 inches diameter	93%	92%	91%	90%	90%

GENERAL CONSIDERATIONS.

The only experimental investigation of the flow of air through poppet valves of which record was found in the technical publications was carried out as a thesis by Mr. R. M. Strong and Mr. F. W. Hollman, and later published by Prof. C. E. Lucke under the title, "Pressure Drop Through Poppet Valves," Vol. 27, Transactions of the American Society of Mechanical Engineers (1905). Prof. Lucke seems to have been the first to call attention to two noteworthy characteristics, which are found to be supported by the data here presented; first, that the coefficient of efflux, computed for air at atmospheric density, is nearly constant for all pressure drops; and, second, that this coefficient is much larger for low lift's Tests were made both with continuous flow, and intermittent flow, the latter being as nearly as possible similar to actual operating conditions for the two gas engines tested, and it was found that the coefficient of efflux for continuous flow was not the same as that for intermittent flow, even at the point of zero acceleration.

It is patent that extreme care should be exercised in any attempt to apply the results of continuous-flow experiments to flow under operating or intermittent conditions, since inertia and resonance effects in the inlet manifold will obviously make great differences in the absolute quantities, and these effects will vary with the type of manifold used. Moreover, the pressure drop, velocity, and coefficient will obviously vary with many other factors as between different engines, different speeds for the same engine, and as to instantaneous values at different points of the stroke for a given engine at a given speed.

However, in the question of design as to whether two inlet valves or one should be used, it is believed the comparative results here presented may be made to serve a real purpose. It is difficult to perceive any reason why the comparative relations obtaining between these three valve combinations for continuous flow should not find some parallel in the comparative relations between the same three combinations for intermittent flow, if no other variables are permitted to affect the comparative results in the latter case. Only inherent differences between the three combinations, effective with intermittent flow and noneffective with continuous flow, or vice versa, would appear capable of affecting this parallel, and it is improbable that such differences, if any, are of great magnitude.

It is hoped that these modest experiments will arouse interest in the question of multiple valves, and certainly the discussion of any direct comparisons obtained in practice would be very interesting. Aeronautic engines of today have so nearly approached the theoretical limit of efficiency that even small improvements may be well worth while, but it seems probable that the mechanical advantages of dual or multiple valves may be of even more importance.

The dimensions of the cylinder model used for these experiments offer a ready basis for discussion, and are commonly encountered in aviation engine practice, the bore being 5 inches and the diameter of combustion chamber 5.75 inches. A combustion chamber of this size permits the use of two valves of 2.5 inches diameter, or four valves of 1.875 inches diameter, inclined at 15 or 20 degrees to the cylinder axis in both cases. Four 1.75-inch valves can be placed in a 5.5-inch cylinder head inclined, or a 5.75-inch cylinder head vertical; and four 1.5-inch valves are even more readily accommodated in a 5-inch cylinder head, or a cylinder having the combustion chamber the same diameter as the cylinder proper. These valves may be placed vertically, and the cylinder is much more easily machined. The combustion chamber will have better proportions, and the slight increase in cylinder height will be more than offset as to over-all height by the saving in spring length.

Two 1.5-inch valves will have a flow capacity equal to one 2.5-inch valve at the same pressure drop and the same lift, will present but 72 per cent as much area to any pressure in the cylinder at the time of opening, and will weigh but 56 per cent of the weight of the single valve, assuming that the weights vary as $D^{2.5}$, which is approximately correct for these sizes. Assuming any reasonable pressure in the cylinder at the time of valve opening, and spring tensions in proportion to valve weights, it is evident that the two small valves will require less than half the power to open them, and this will be a direct saving of mechanical loss, as valve action is not the type of reciprocating motion which can return during one portion of the stroke energy stored during another portion, excepting only the energy stored in the spring.

It has been said that valves in pairs are more difficult to cool than single valves, but this does not appear to stand analysis. The proportion of the 5-inch cylinder head occupied by the small valves is only about 95 per cent of the proportion of the 5.75-inch head occupied by the large valve. The circumference of the two valves is 20 per cent greater than that of the single valve, and although the seats would have somewhat less width, the distance of heat flow in this direction would be but 60 per cent as great. As to the portion of the heat which flows to the guide, the conditions are also somewhat in favor of the small valves, the distance to the water-cooled portion of the guide being less and the proportion of water-cooled guide greater.

In one example of foreign engine design dual valves of about this size are lifted to one-half diameter, and give entirely satisfactory operation at speeds up to 2,200 revolutions per minute. The possibilities in this direction are largely untried, but the negative work used in overcoming valve resistance to inlet flow might be reduced with small valves at high lifts and the volumetric efficiency increased, without introducing serious mechanical difficulties. This, of course, is contrary to the principle of using low lifts to secure a higher coefficient, but still the over-all result might be beneficial.

The comparison of a single 2.5-inch valve to a pair of 1.75-inch valves may be analyzed in much the same manner, and as to heat conditions the result would seem slightly in favor of the pair. If lifted 0.375 inch, the capacity will be 25 per cent greater than that of one 2.5-inch valve, according to the experimental results shown in Table 5, or the resistance will be but 64 per cent as great, the resistance varying approximately with the square of velocity or capacity. This should result in higher volumetric efficiency. The superficial area of the two combinations would be practically equal, but the weight of the pair would be but 82 per cent of that of the single valve, with correspondingly reduced total spring tension and slightly reduced mechanical loss.

Interesting comparisons may be drawn from data published by the Automobile Engineer, London, Volume VII, Nos. 105–6–8–9 (1917) covering Benz and Mercedes engines, each make being constructed in both 2-valve and 4-valve models. Except for the valve changes and an increase in compression ratio from 4.50 to about 4.90, the design of the 4-valve models is much the same as that of the respective

2-valve types. The data are represented in Table 9, the ratio of volume to horsepower and brake mean pressure being given for the rated power at 1,400 revolutions per minute for each engine. The "valve factor" is one-half the product of inlet-valve opening area by the number of degrees open divided by the displacement of one piston, affording a ready index of relative valve capacity.

Table 9.

Engine.	Benz 4-valve.	Mercedes 4-valve.	Benz 2-valve.	Mercedes 2-valve.
Bore, inches	5.71	6.30	5.12	5.51
Stroke, inches	7.48	7.09	7.09	6.30
Piston displacement per cylinder, cubic inches	191.38	220.82	146.05	150.20
Piston displacement, total cubic inches	1,148.30	1,324.90	876.30	901.20
Valve port diameter, inches	2.04	2.17	2.42	2.67
Valve lift, inches	.465	.398	.433	.440
Inlet valve opening, square inches	2.99	2.72	3.29	3.70
Rated horsepower	230	260	160	160
Rated revolutions per minute	1,400	1,400	1,400	1,400
Maximum horsepower	250	270	164	162.5
Maximum revolutions per minute	1,650	1,650	1,400	1,400
Inlet-valve opening, degrees	245	228.3	240	213
Area inlet pipe, square inches	3.65	6.85	2.96	3.54
Cubic inches piston displacement per horsepower	4.99	5.10	5.48	5.63
Compression, ratio	4.91	4.94	4.50	4.50
Valve factor	3.80	2.82	2.70	2.62
Brake, mean effective pressure	113	107.5	103	102

The valve factor for the 4-valve Mercedes is but slightly larger than that of the 2-valve, and the mean effective pressure is increased only 5 per cent, which is practically accounted for by the increase in compression ratio from 4.50 to 4.94. In the Benz 4-valve, the factor is increased 35 per cent and the mean effective pressure increased 10 per cent, only about one-half of which can be due to the increase in compression ratio from 4.50 to 4.91.

In plate 7 a comparison is made of the power output of these four engines plotted against gas velocity through the inlet valve. These velocities are computed for this comparative purpose, as the ratio of piston displacement per explosion to one-half the product of valve-opening area by the time of the opening. The broken curves represent the 4-valve Benz and Mercedes, respectively, reduced approximately to compensate for difference in compression ratio.

In conclusion, a summary of the results experimentally derived is presented. It should be borne in mind that the number and character of the experiments is not such as to render them final and conclusive. It is earnestly hoped that further and more extensive data bearing upon this subject will be experimentally obtained and published, and it is believed that the results here presented will be found substantially correct in the light of later research. Caution should be exercised in the application of these results, for apparent similarity with respect to air flow is often most deceptive.

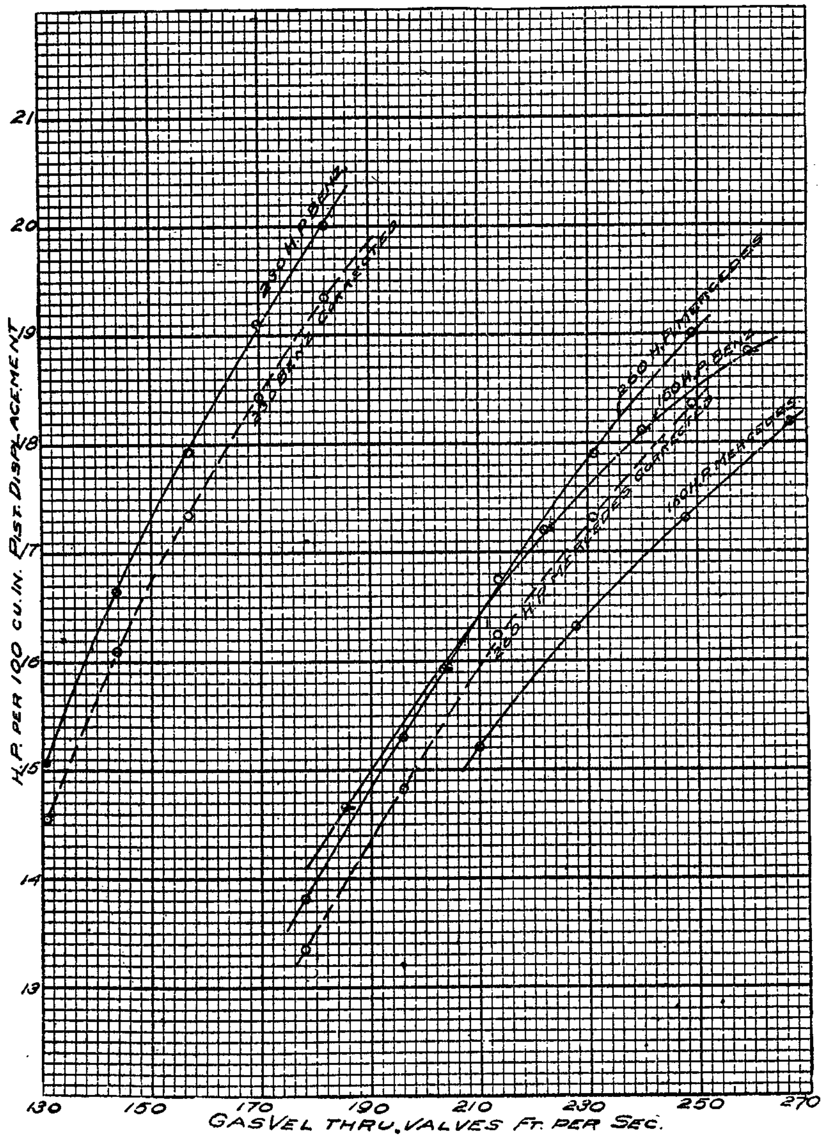

H.P. PER 100 CU. IN. PIST. DISPLACEMENT
21
20
19
18
17
16
15
14
13
230 H.P. BENZ
200 H.P. MERCEDES
100 H.P. BENZ
200 H.P. MERCEDES CORRECTED
100 H.P. MERCEDES
130
150
170
190
210
230
250
270
GAS VEL THRU. VALVES FT. PER SEC.

PLATE 7.

CONCLUSIONS.

1. The coefficient of efflux is practically constant, for all pressure drops (at least below 1 pound per square inch) where the lower pressure is approximately atmospheric, and the theoretical flow is computed upon air at atmospheric density.

2. Under conditions of general similarity, the coefficient of efflux is very nearly the same for valves of different sizes, at equal lifts expressed in per cent of their respective diameters.

3. Lifting a valve one-quarter of its diameter may develop an area of opening geometrically equal to its port area, but affords a capacity less than 67 per cent of that of the unobstructed port, at the same pressure drop; a lift equal to one-half diameter develops 80 to 90 per cent of this maximum capacity.

4. At the same pressure drop, one valve of diameter D and lift h is equal in capacity to:

First. A pair of valves of diameter 0.707 D (equal port area) and lift 0.707 h.

Second. A pair of valves of diameter 0.6 D and lift h, for values of h not exceeding about 0.25 D.

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