AUTOMOBILE ENGINE CHARACTERISTICS

BY

E. C. COOK

A. N. GAIL

ARMOUR INSTITUTE OF TECHNOLOGY

1920

THE REPRESENTATION OF AUTOMOBILE ENGINE CHARACTERISTICS BY SURFACES

A THESIS

PRESENTED BY

ELLIS C. COOK AND ARTHUR N. GAIL

TO THE

PRESIDENT AND FACULTY

OF

ARMOUR INSTITUTE OF TECHNOLOGY

FOR THE DEGREE OF

BACHELOR OF SCIENCE

IN

MECHANICAL ENGINEERING

MAY 27, 1920

APPROVED:

Geo. F. Gebhardt

Professor of Mechanical Engineering

Dean of Engineering Studies

Dean of Cultural Studies

*** INDEX ***

INDEX - Cont.

*** PART I ***

Object of the Investigation.

OBJECT OF THE INVESTIGATION.

The analysis of the economic operation of automobile engines is dependent upon consistent carburetion in each test. The carburetor is considered as a strictly automatic metering device which properly proportions the amounts of air and fuel to obtain the best conditions for combustion. Up to the present time, variable speed automobile engines have been tested with no attention to the quantity of air entering the cylinders. In previous tests the rate of fuel consumption was determined and the carburetor was called upon to maintain the proper mixture for combustion.

For maximum power output and efficiency the engine should take in the maximum amount of properly proportioned air and fuel mixture.

It is the purpose of this progress thesis to present a mathod of determining the volum-

etric efficiency and the weight of dry air entering the engine. The original purpose called for a continuation of the previous test work with the addition of the air determination but the available apparatus was unsuited to the conditions and the work resolved itself into a solution of the air metering problem, of which this is submitted as a solution.

***   Part II   ***

Description of Apparatus.

Engine

Dynamometer

Orifice Meter

Fuel Weighing Apparatus

Water Cooling

Engine Speed Counting

Manifold Depression

DESCRIPTION OF APPARATUS.

Engine:-

The automobile engine for which the orifice meter was originally planned and designed, and the engine which was to be tested, was a six-cylinder Chalmers engine commonly known to the public as the 3400 revolutions per minute engine. The engine is cast with the cylinders en-bloc. It is an L head engine with the valves on the right hand side of the cylinder casting.

The bore and stroke are 3 1/4 and 4 1/2 inches respectively, and at 3400 revolutions per minute the piston displacement volume is approximately twelve thousand five hundred cubic feet per hour.

Dynamometer:-

A Sprague one-hundred horsepower electric dynamometer and control switches regulate and absorbe the power generated by the engine. The dynamometer consisted of a one-hundred horsepower direct current inter-pole generator mounted upon

ball bearing trunnions in such a manner that the field is free to rotate about the armature axis. The torque of the engine is taken off of the generator by a link mechanism one end of which is fastened to the frame and the other end is connected to a spring scale and a Buffalo scale. The radius arm registering the torque is 1.75 feet radius so that the weight indicated on the scales multiplied by the factor $\frac{\text{R.P.M.}}{3000}$ gives the horsepower of the engine. The field of the generator can be seperately excited in order to have a proper field flux at any desired speed.

The control panel holds the control switches, field rheostat, circuit breaker, ammeter, voltmeter, and electric tachometer.

The current generated from the engine power is dissipated by a set of resistances mounted upon the north wall of the laboratory.

The dynamometer bed is so arranged that it may be adjusted to allow a wide difference of engine sizes to be tested.

Orifice Meter:-

The orifice meter consists of a cylindrical chamber 30 inches in diameter and 60 inches long with eight circular orifices attached to one end. On the other end is the connecting pipe leading the air to the carburetor. The orifice plates may be readily removed from their holders and other sizes of orifices may be substituted. Three sizes of orifices are available and are designated by the approximate diameter od the orifice. The designations and the actual diameters of the orifices are,

5/16"	orifice	-	.3044" diameter,
1"	orifice	-	.9570" diameter,
3"	orifice	-	3.0000" diameter.

A set of eight orifices of each size was made and any combination of orifices may be used in order to secure a pressure drop within the range of .75 inches of water to 3.00 inches of water; the preferable drop being 1.5 inches because the orifice constants were determined at this drop of pressure and are very

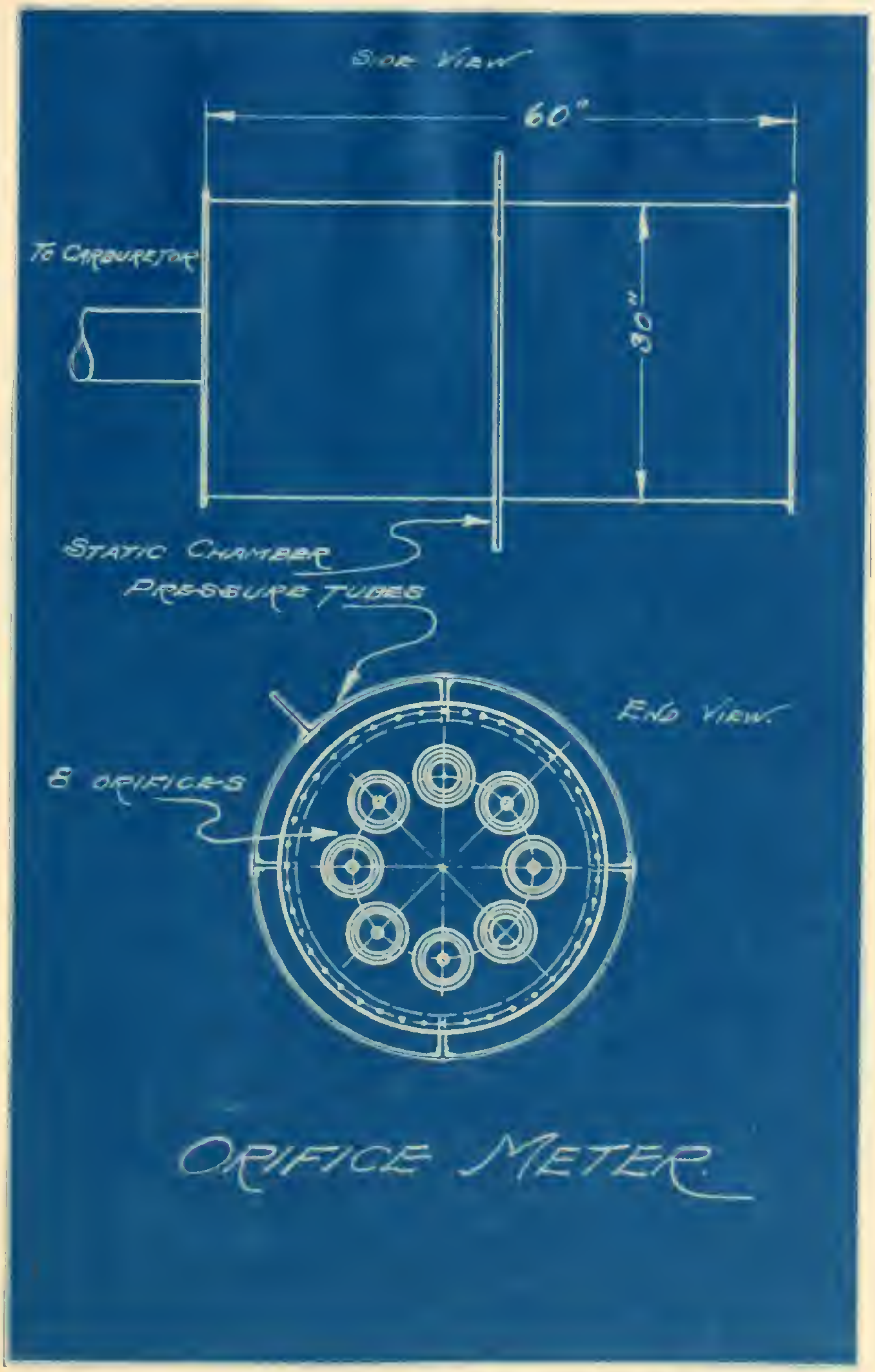

SIDE VIEW
60"
TO CARBURETOR
30"
STATIC CHAMBER
PRESSURE TUBES
END VIEW.
8 ORIFICES
ORIFICE METER

Figure 1.

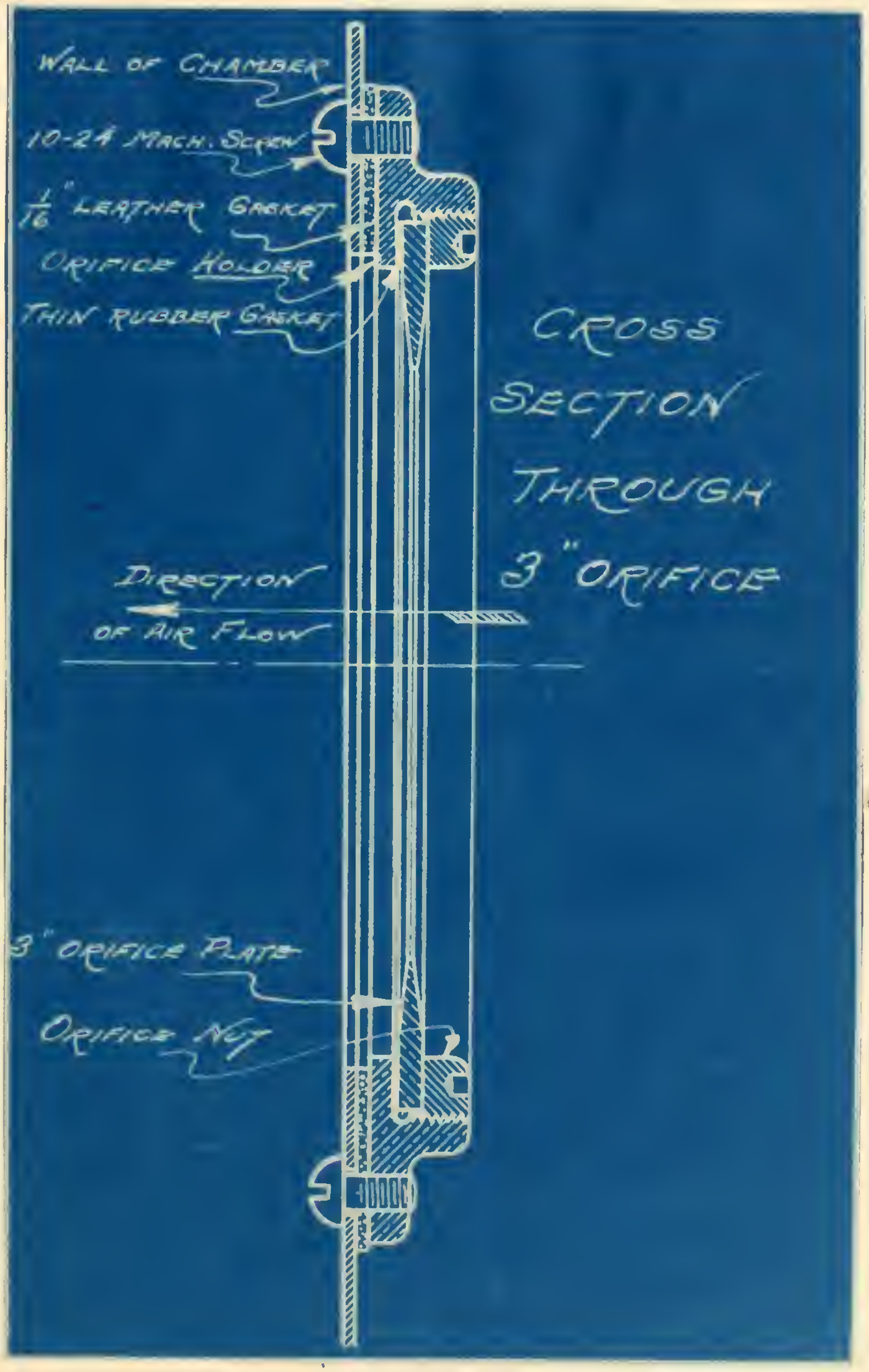

WALL OF CHAMBER
10-24 MACH. SCREW
1/16" LEATHER GASKET
ORIFICE HOLDER
THIN RUBBER GASKET
CROSS SECTION THROUGH 3" ORIFICE
DIRECTION OF AIR FLOW
3" ORIFICE PLATE
ORIFICE NUT

Figure 2.

slightly in error at much greater or lesser pressure drops. The error involved in using the limiting pressure drops is very small and is neglected.

The above diameters of orifices were chosen such that each successive larger diameter of orifice has a flow exactly ten times the flow of the next smaller orifice for the same pressure drop. Thus the flow past the three inch orifice is ten times the flow past the one inch and one-hundred times the flow past the three-sixteenths inch orifice. This, together with the facts that there are eight orifices and that the limiting pressure drops allow a two-to-one ratio of flow, permits of good flexibility of measurement. The range of this meter extends from approximately two pounds of air per second down to about one one-thousandth of a pound per second. This range is sufficient for practically all cases of engine testing up to 500 horsepower.

The pressure drop is determined by a tilting manometer reading in inches of water. One leg of the

manometer is connected to the static pressure connection on the orifice chamber. The other leg of the manometer remains open under atmospheric pressure.

The air enters the orifice chamber through the set of orifices and passes through the connecting pipe to the carburetor chamber, an air-tight chamber surrounding the carburetor and through which the carburetor controls project. The air from here follows its normal course through the carburetor and manifolding into the cylinders.

Fuel Weighing Apparatus:-

The fuel tank is mounted upon Buffalo scales to the arm of which is attached an electric contact which rings a bell when the arm is down. The test run is started with the arm up but just ready to fall. When it falls and rings the bell time is taken. The rider on the scale arm is moved to indicate a lower weight of fuel and the arm raises. As soon as the fuel indicated by the

movement of the rider is consumed the arm again falls and rings the bell. The time between the first and second ringing of the bell is the time required to cunsume the known amount of fuel.

Water Cooling:-

Cold water is mixed in a tank with the hot water from the engine and the overflow of hot water is led to waste. By careful regulation of the amount of cold water entering the tank, a definite inlet or outlet temperature may be maintained.

Engine Speed Counting:-

On the opposite end of the dynamometer to which the engine is connected are mounted two revolution counters which are connected to the armature shaft through electro-magnetic clutches.These clutches are operated by two push switches, and the counters are started and stopped almost instantaneously.

Manifold Depression:-

The carburetion manifold is

drilled and tapped near its connection with the cylinders and a copper tube is fitted. The other end of this small copper tube connects with a vacuum gauge which indicates the depression in the manifold due to the throttling by the butter-fly valve in the carburetor. The pressure drop across the orifices causes as much as two-tenths of an inch of mercury at the highest.

*** Part III ***

Theory and Operation of the Orifice Meter.

Theory of flow through an orifice

Effect of humidity upon the flow

Selection of orifice diameters and constants

Operation of Orifice meter

Calculation of air flow through orifice meter.

THEORY OF FLOW THROUGH AN ORIFICE.

The theory of the orifice meter is based upon the consideration of an elastic fluid flowing through a short tube in which the length is but a small fraction of the diameter of the tube. Mechanical difficulties prohibit the reduction of the length of the tube to an infinitesimal amount, but for practical purposes this length is so small that the friction of the fluid may be considered as zero and it is neglected in the present discussion.

Figure 3. diagramatically shows an air chamber, A, in which the pressure has been reduced by suction on the outlet pipe, B. The outlet pipe is connected to the carburetor air inlet of an internal combustion engine and the normal operation of the engine causes this suction.

The atmospheric pressure, p_1, exists on the outside of the chamber and p_2 designates the pressure within the chamber. The difference of pressure, p_1-p_2, between the interior and the exterior of the chamber causes a flow of the fluid into the chamber to take

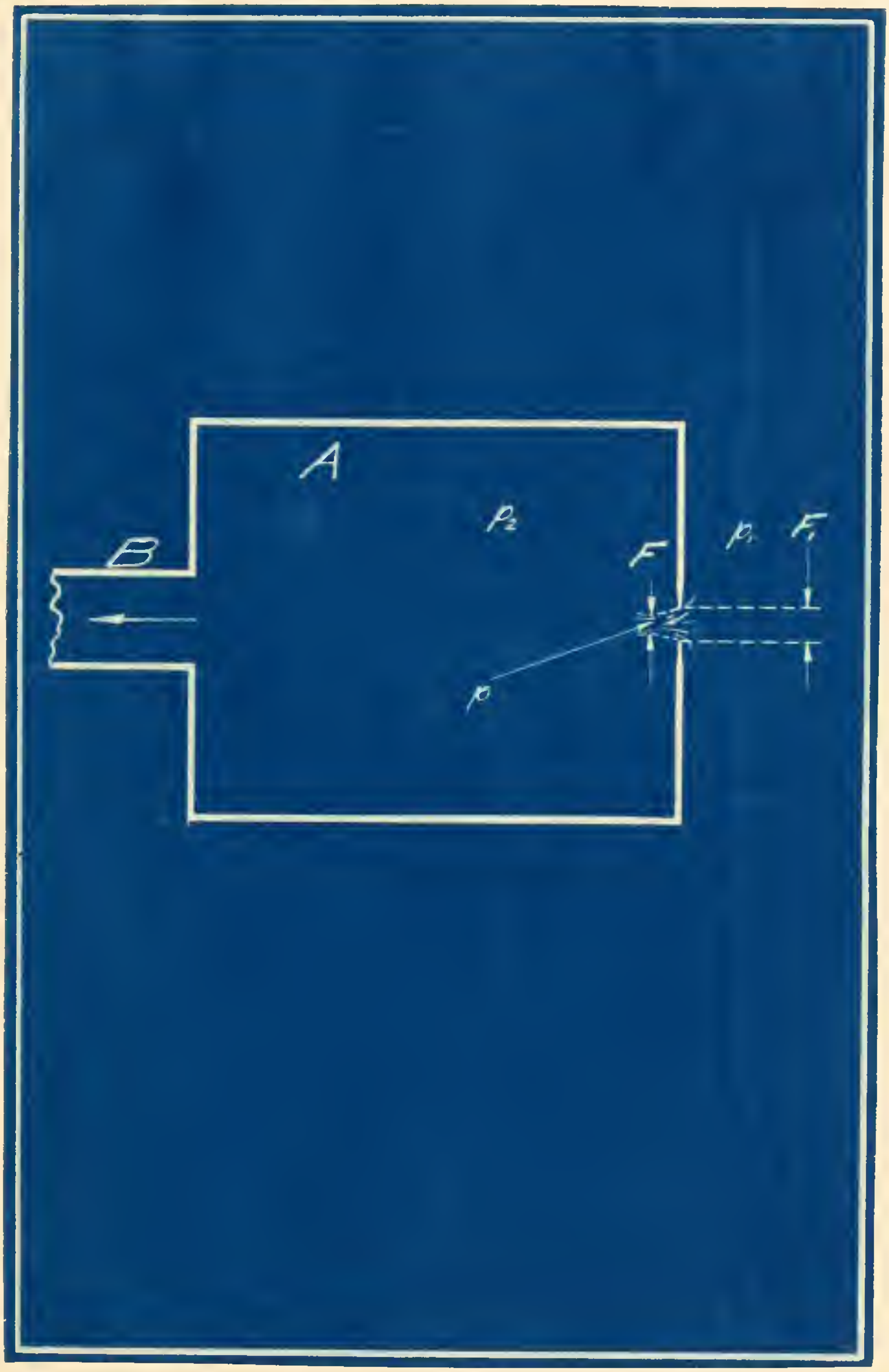


Figure 3.

place. Assuming this flow to be frictionless and adiabatic, we have, since $w_1 = 0$,

$$\frac{w^2}{2g} = \int_{p}^{p_1} v\,dp, \qquad (1.)$$

where,

w is the velocity of flow through orifice,

v is the specific volume of the fluid,

p is the pressure of the fluid.

The law of adiabatic expansion for the fluid is

$$p_1 v_1^n = p v^n, \qquad (2.)$$

where,

n is the ratio of the specific heats of the fluid.

Substituting (2.) in (1.) and integrating, we have

$$\frac{w^2}{2g} = \frac{n}{n - 1}\, p_1 v_1 \left[1 - \left(\frac{p}{p_1}\right)^{\frac{n-1}{n}}\right] \qquad (3.)$$

From the law of continuity of flow of fluids, $Mv = Fw$, we can eliminate w from (3.) and obtain the equation

$$M = \frac{F}{v}\sqrt{2gp_1v_1\frac{n}{n-1}\left[1-\left(\frac{p}{p_1}\right)^{\frac{n-1}{n}}\right]} \qquad (4.)$$

where,

M is the weight of fluid flow per second,

F is the area of the orifice.

From (2.) we have

$$v = v_1\left(\frac{p_1}{p}\right)^{\frac{1}{n}},$$

which can be substituted in (4.) to give

$$M = F\sqrt{2g\frac{n}{n-1}\frac{p_1}{v_1}\left[\left(\frac{p}{p_1}\right)^{\frac{2}{n}}-\left(\frac{p}{p_1}\right)^{\frac{n+1}{n}}\right]} \qquad (5.)$$

Since the ratio of p_1 to p_2 is kept very close to unity at all times in this discussion, the question of critical pressures of flow does not enter and p may justly be assumed to be equal to p_2. Equation (5) then becomes

$$M = F\sqrt{2g\frac{n}{n-1}\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}}-\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]} \qquad (6.)$$

Equation (6.) may be simplified by substuting in it the proper values of g, F, and n for any particular investigation but there still remains an equation which contains fractional exponents which are tedious

and difficult to evaluate.

Further assumptions and approximations are necessary in order to bring out an equation of a more simple form and one which can be readily evaluated. These assumptions are,

(1) That no difference in temperature occurs during the flow,

(2) That the specific volume, v, remains constant,

(3) That the velocity of flow is constant over the cross section.

In other words, the fluid is considered as an inelastic fluid during its flow through the orifice.

The velocity of flow may then be expressed by the equation

$$w = \sqrt{2gh},$$

where h is the head of fluid.

Where i is the pressure drop across the orifice in inches of water, the head, h, is given by the equation

$$h = \frac{62.4}{12}\,iv,$$

$= 5.20$ iv, which may be substituted in (7.)

to get

$$w = \sqrt{2g \cdot 5.2\ iv}\ . \qquad (8.)$$

From the law of the continuity of flow and from (8.) we have

$$M = \frac{F}{v}\sqrt{2 \cdot 5.2\ giv}$$

$$= F\sqrt{\frac{2 \cdot 5.2\ gi}{v}} \qquad (9.)$$

But, for dry air,

$$v = \frac{53.35\ T}{p} \qquad (10.)$$

where T is the absolute temperature in degrees Fahrenheit. Therefore,

$$M = F\sqrt{\frac{2 \cdot 5.2\ gip}{53.35\ T}} \qquad (11.)$$

The values of this equation, (11.) and those of the more complicated one, (6.) show practically no differences for small values of i. For the small pressure drops used in this discussion, under three inches of water, the differences are so minute that the accuracy is not impaired. This indicates that the previous assumptions were justifiable. Since the pressure drops used in this investigation are main-

tained at a low value the form of equation as given in (11.) is suitable and sufficiently accurate.

Experiments with both elastic and inelastic fluids show that, due to the contraction of the flow after passing through the orifice, the theoretical equations must be modified to include this contraction effect. This is done by the introduction of a contraction coefficient, C, which may be considered to be the ratio of the cross-sectional area of the jet after passing the orifice to the area of the orifice. These areas, using air, cannot readily be obtained by direct measurement and C may best be found by experimentally determining the flow and calculating the contraction. For sharp orifices, similiar to the ones used in this case, the value of C is close to six-tenths. With a change in diameter of the orifice or with a pressure drop change the value of C varies according to no definite law.

Taking the contraction into consideration, the formula for the amount of flow becomes

$$M = CF\sqrt{\frac{2\cdot 5.2\ gip}{53.35\ T}} \,, \qquad (12.)$$

The values of C used in this investigation were experimentally determined for a pressure drop of one and one-half inches of water, and in order to prevent a material variation from this experimental value the proposed range of pressure drop is from three-quarters of an inch to three inches of water.

The accompanying nomograph is based upon the above equation, (12.), with the proper values of C and F inserted and applies to dry air.

EFFECT OF HUMIDITY UPON THE FLOW.

In determining the amount of oxygen supplied to an internal combustion engine, the moisture contained in the atmosphere is inert and complicates the matter of computation. Not only is the composition of the mixture of air and moisture changed but the density is different from that of dry air alone under the same conditions.

In the question of the volumetric efficiency of an engine, the subject of mixture volume enters with no reference to the amount of air in that mixture. Formula (9.) of the theory of flow shows how the density of the mixture affects the flow. Equation (9.) is

$$M = F\sqrt{\frac{2 \cdot 5.2\ ig}{v}},$$

and may be modified to deal with volume as follows,

$$Q = Mv = Fv\sqrt{\frac{2 \cdot 5.2\ ig}{v}}$$

$$= F\sqrt{2 \cdot 5.2\ igv} \qquad (13.)$$

where

Q is the volume of flow in cubic feet per second.

To show the effect of humidity an extreme case is taken where the temperature is one hundred and twenty degrees Fahrenheit and the air is saturated, and this condition is compared with one at the same temperature with no moisture in the air. For the dry air v is found to be 14.62 cubic feet per pound, and for the saturated air v is 15.26 cubic feet per pound.

Taking dry air as a standard and making the original calculations for dry air with a correction factor for humidity, the dry air volume would have to be multiplied by $\sqrt{\frac{15.26}{14.62}}$ or 1.022 in order to obtain the volume of moist air flowing. The volume of saturated air then is 2.2 per cent greater than that flowing when dry under these assumed conditions. The per cent of increase would be approximately proportional to the humidity for a given temperature, and a humidity of fifty per cent at one hundred and twenty degrees would therefore would have a correction factor of 1.011. These correction factors apply only for the volume of mixture and are given for different

temperatures and humidities on Chart #1. Table #1 gives the calculated correction factors for saturation and the in laying out the chart the partially saturated conditions were considered as having a correction proportional to their per centage of saturation.

Under the conditions usually met in testing, this correction will not be greater than one-half of one per cent. The effect of humidity in testing for the volumetric efficiency of an engine may then be neglected, but it can not be neglected in testing for the amount of oxygen or dry air entering the engine.

Considering the dry air content of the mixture, the velocity head of flow is due to the mixture density but only a part of this mixture is dry air. If R represents the proportion of dry air in the mixture, then formula (9.) takes the form

$$M = FR\sqrt{\frac{2 \cdot 5.2\ ig}{v_m}} \quad , \qquad (14.)$$

where v_m is the specific volume of the mixture.

The correction factor, where v is the volume of dry air, is then

$$\frac{\frac{R}{\sqrt{v_m}}}{\frac{1}{\sqrt{v}}} = R\sqrt{\frac{v}{v_m}} \quad , \qquad (15.)$$

The value of R may be found by dividing the weight of dry air in a saturated mixture by the weight of that saturated mixture. Both of these may be found in tables of the properties of saturated air. For one hundred and twenty degrees and saturation R is $\frac{60.60}{65.52}$ or .926, $v = 14.62$, $v_m = 15.26$, and the factor then works out as .906. The weight of oxygen in this mixture is but .906 of what it would be were no moisture present.

Table #2 gives the calculated correction factors for saturation at different temperatures and Chart #2 shows the factors for varying humidity. These are to be applied in questions involving the weight of dry air in a mixture of air and moisture.

SELECTION OF ORIFICE DIAMETERS AND CONSTANTS.

Since the value of the discharge coefficient is dependent upon experiment entirely, it was necessary either to calibrate new orifices or to simulate the conditions which existed during the tests carried on by other experimenters and employ their results.

Accurate calibration, while seemingly simple, involved the use of apparatus which was not available at the time, and it was therefore decided that, if possible, the results of carefully made tests should be used.

Search through the published technical literature revealed the results of but one test of the the flow of air through orifices upon which any reliance could be placed. This was the work of R. J. Durley, " On the Measurement of Air Flowing into the Atmosphere Through Circular Orifices in Thin Plates and under Small Differences of Pressure", which appeared in the American Society of Mechanical Engineers Transactions, No. 1098, page 193, 1906.

Durley's experiments were carried out with orifices up to five inches in diameter and with pressures as high as five inches of water. His available apparatus limited him to these capacities but he exterpolated the data to give the coefficients for six inches diameter of orifice and six inches of water pressure drop across the orifices. In Table #3 are given the experimental discharge coefficients which Durley obtained for orifices in plates .0571 inches thick.

Durley's conclusions from his work are as follows:

(1.) The coefficient for small orifices increases as the head increases, but at a lesser rate the larger the orifices, until for a 2" orifice it is almost constant. For orifices larger than 2" it decreases as the head increases and at a greater rate the larger the orifice.

(2.) The coefficient decreases as the diameter of the orifice increases and at a greater rate the higher the head.

(3.) The coefficient does not change appreciably with the temperature ($40°$ to $100°$F.)

(4.) At heads under 6" of water the coefficient is not affected by box size if the ratio of areas is greater than 20 to 1.

In the discussion of his work Durley brought out the fact that the position of the static tube should not be too near the inlet or the outlet of the chamber and the connection should be flush with the sides.

A.O.Muller (Forschungs-Arbeiten No.49) gives a value of .597 for the coefficient for sharp edged orifices bevelled on the upstream side. He used pressure drops of 5 to 50 millimeters of water and his determinations were made with great care but all of the details are not obtainable.

Through the courtesy of Mr. H. A. Sedgwick, Superintendent of The Cutler-Hammer Manufacturing Company of Milwaukee, Wisconsin; Mr. Packard, head of the Thomas meter department of the same company; and Mr. R. H. Earle, also of the Thomas meter department, the writers were given access to the results of extensive experiments upon the flow of air through orifices as carried out by this company in order to test and calibrate the Thomas meter as manufactured

by them. Numerous experiments were carried out to determine the flow through three sizes of orifices mounted in a holder which permitted changing the orifice plate to allow for different rates of flow. All of the calibration work was carried out under a constant pressure drop of one and one-half inches of water and the air density was corrected for humidity.

The smaller diameter, .315 inches in diameter, was tested against the flow through a 1/10 cubic foot container which had been checked by the Bureau of Standards, tested by various pitot tube traverses, and tested against the flow into gas holders of known volume. The coefficient determined for this diameter and pressure was 0.6128.

The next size tested was a three inch orifice which was tested against the .315inch orifice and in a similiar manner to the testing of the smaller one. The coefficient for the three inch orifice was determined as 0.6305. The ratio of these

two orifice capacities is about one hundred to one; and an intermediate size of orifice was made by trial to give a flow of exactly one-tenth of the flow through the three inch size. The probable error as stated is one-half of one per cent.

On account of the character of these Cutler-Hammer experiments and the care taken in their execution, it was decided to adopt the methods and results of these experiments in preference to the results of the experiments carried out by Durley.

A comparison of the results of the Cutler-Hammer and the Durley experiments show reasonable agreement considering the difference caused by the interference of the holders in the case of the Cutler-Hammer experiments. The Cutler-Hammer experiments show a somewhat higher coefficient than those given by Durley and this is accounted for by the interference of the orifice plate-

holders which give the air somewhat of an initial streamline before entering the orifice. All of the Cutler-Hammer experiments were carried out with the holders on the upstream side of the flow. The greatest difference in the coefficients is in the case of the larger diameter where the streamlining would be expected to show the greatest effect.

The diameters of the three orifices tested, together with their respective discharge coefficients, are as follows:

Nominal Diameter	Actual Diameter	Discharge Coefficient
5/16"	.315"	.6128
1"	.957"	.6196
3"	3.000"	.6305

The rates of flow through the three inch and the one inch orifices are in the exact ratio of ten-to-one, and it was desired to make the necessary changes in the five-sixteeth inch orifice in order to have its rate of flow exactly one one-hundredth of the rate of flow through the three inch orifice.

For a given rate of flow through two orifices with the same pressure drop, the product of the area and the discharge coefficient of the one must be equal to the product of the area and the discharge coefficient of the other. Then,

$$C_3 \cdot A_3 = 100 \cdot C_x . A_x \qquad (16.)$$

where

C_3 is, from experiment, 0.6305,

A_3 is 0.04909 square feet,

C_x is the new discharge coefficient for the new diameter of orifice,

A_x is the area of the new orifice.

As a first approximation the value of C_x can be taken as .6128, the coefficient for the 5/16 inch orifice since the diameter will not be far from .315 inches. Solving then for A_x, or rather, for the diameter corresponding to the area A_x, we obtain the value of .3044 inches as the diameter of the orifice which will give a flow of one-one hundredth the flow of the three inch

orifice. This is under the assumption that the discharge coefficient remains constant for this slight change of diameter. In order to check this coefficient the experimental values were plotted against the diameters of the orifices and a smooth curve was drawn connecting these three points. From the graph it was impossible to distinguish the difference between the two coefficients for the these two diameters. A tangent to the curve at this point was drawn and it showed that the coefficient at this point decreased at the rate of .0182 per inch decrease in diameter. The decrease in diameter amounts to .0106 inch. Therefore the new coefficient of discharge is .6128 - .0182×.0106 or .6126.

Substituting this new coefficient for the value of .6128 back in the expression used in determining the area of this small orifice produces an immaterial change in the diameter.

The diameters and discharge coefficients of the three orifices employed in this investigation are then as follows:

Nominal Diameter	Actual Diameter	Discharge Coefficient
5/16"	.3044"	.6126
1"	.9570"	.6196
3"	3.0000"	.6305

The flow through each is in the ratio of 1 : 10 : 100 in the order as given, and combinations of the different sizes can be readily handled in the computation of the total flow through the meter. Thus, two - 3", one - 1", and five -5/16" orifices in parallel would have the equivalent flow of 215 of the 5/16" size. The number of the 3" orifices is in the hundreds place, the number of 1" orifices is in the tens place, and the number of 5/16" orifices is in the units place.

OPERATION OF ORIFICE METER.

A test run of an internal combustion engine is usually made at a predetermined speed or at a set of predetermined speeds and the approximate volume of air which the engine requires may be estimated from the volume of the piston displacement and the engine speed. The orifices must then be arranged in such a combination that they will deliver the required volume of air and still keep the pressure drop across the orifices between the limits of 3/4" and 3" of water. The volume required per hour should be expressed in terms of the different orifices.

At a pressure drop of 1 1/2" the flow may be considered as follows:

Orifice	Flow in cubic feet per hour
5/16"	100
1"	1000
3"	10000

This serves as an approximation and permits of about forty percent variation either way on account

of the range of pressure drop. Thus an engine requiring 28,000 cubic feet of air per hour would draw it through two-3" and six-1" orifices. Since this combination gives 26,000 cubic feet at 1 1/2" of water, the pressure drop would be expected to be somewhat greater than 1 1/2".

An engine does not draw in its full volume of charge and the estimated volume should take this fact into consideration.

Should the arrangement as estimated prove to be beyond the requirements the orifices may be changed during the run. In assembling the orifices care should be taken that no leakage can occur. A glycerine sealed joint is preferred to one merely screwed up tight.

The engine on which the test is made should be of the multi-cylinder type in order to have a steady flow instead of a pulsating one. The head of water varies with the square of the velocity and a mean head does not indicate a

mean velocity.

The air chamber acts in a manner to aid in damping out the pulsations but it will not rectify large ranges of velocity.

CALCULATION OF AIR FLOW THROUGH ORIFICE METER

The weight of air passing through the meter may be calculated from equation (12.) with the proper values of C and F substituted. For the 5/16" orifice this equation can be simplified and to read in pounds per hour it becomes

$$\text{Weight/Hour} = 23.477\sqrt{\frac{pi}{459.6+t}} \qquad (17.)$$

where

p is the barometric pressure in inches of mercury

t is the air temperature in degrees Fahrenheit

i is the inches of water pressure drop.

For the 1" orifice the decimal point in the above equation is moved one place to the right, and two places for the 3" orifice.

In order to reduce the labor required in making repeated solutions of the above equation the accompanying nomograph was devised for rapid

graphical solutions. This nomograph is based upon equation (17.) in each of its three forms.

In using the nomograph a line is drawn connecting the observed pressure drop and barometric pressure. The reference line is cut by this line at some point. Through the point where the reference line is cut and the air temperature another line is drawn extending to the air flow scale. The point at which the line cuts the air flow scale indicates the weight of dry air passing the different orifices. The amounts passing each orifice have to be totalled , and in the case of fuel ratios, the dry air weight must be multiplied by the correction factor for humidity, obtained from Chart #2.

Should more accurate results be desired than those obtainable from the nomograph, direct calculation from equation (17.) can be made; or, the discharge coefficient may be inserted in

equation (9.) and the value of v be calculated with respect to the humidity by means of the vapor pressure and Dalton's law.

*** PART IV ***

Appendix

Drawings,Data, and Charts.

TABLE #1

CORRECTION FOR VOLUME OF MIXTURE.

Temperature Degrees Fahrenheit.	Correction Factor. (100% Humidity)
0	1.000
10	1.000
20	1.001
30	1.001
40	1.001
50	1.002
60	1.003
70	1.005
80	1.007
90	1.010
100	1.013
110	1.017
120	1.022

TABLE #2

CORRECTION FOR WEIGHT OF DRY AIR.

Temperature Degrees Fahrenheit.	Correction Factor (100% Humidity)
0	.999
10	.998
20	.996
30	.995
40	.993
50	.990
60	.986
70	.980
80	.972
90	.961
100	.948
110	.930
120	.906

TABLE #3

VALUES OF DISCHARGE COEFFICIENTS FOR SHARP ORIFICES IN .0571 INCH PLATE AS FOUND BY R. J. DURLEY.

Diameter in Inches	Pressure Drop in Inches of Water 1	2	3	4	5
5/16	.603	.606	.610	.613	.616
1/2	.602	.605	.608	.610	.613
1	.601	.603	.605	.606	.607
1 1/2	.601	.601	.602	.603	.603
2	.600	.600	.600	.600	.600
2 1/2	.599	.599	.599	.598	.598
3	.599	.598	.597	.596	.596
3 1/2	.599	.597	.596	.595	.594
4	.598	.597	.595	.594	.593
4 1/2	.598	.596	.594	.593	.592

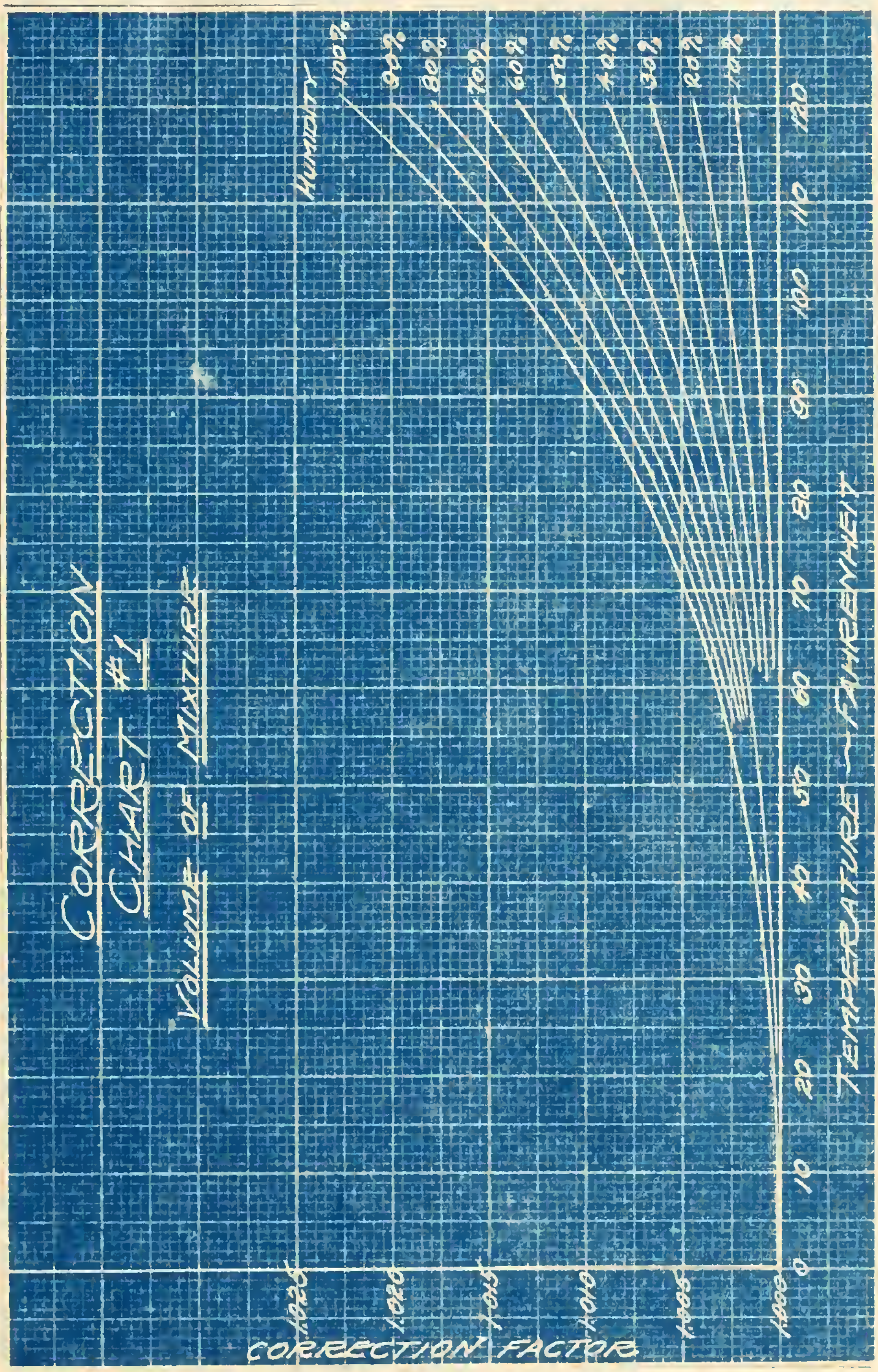

CORRECTION
CHART #1
VOLUME OF MIXTURE
HUMIDITY
100%
90%
80%
70%
60%
50%
40%
30%
20%
10%
1.025
1.020
1.015
1.010
1.005
1.000
0
10
20
30
40
50
60
70
80
90
100
110
120
TEMPERATURE ~ FAHRENHEIT
CORRECTION FACTOR

Figure 4.

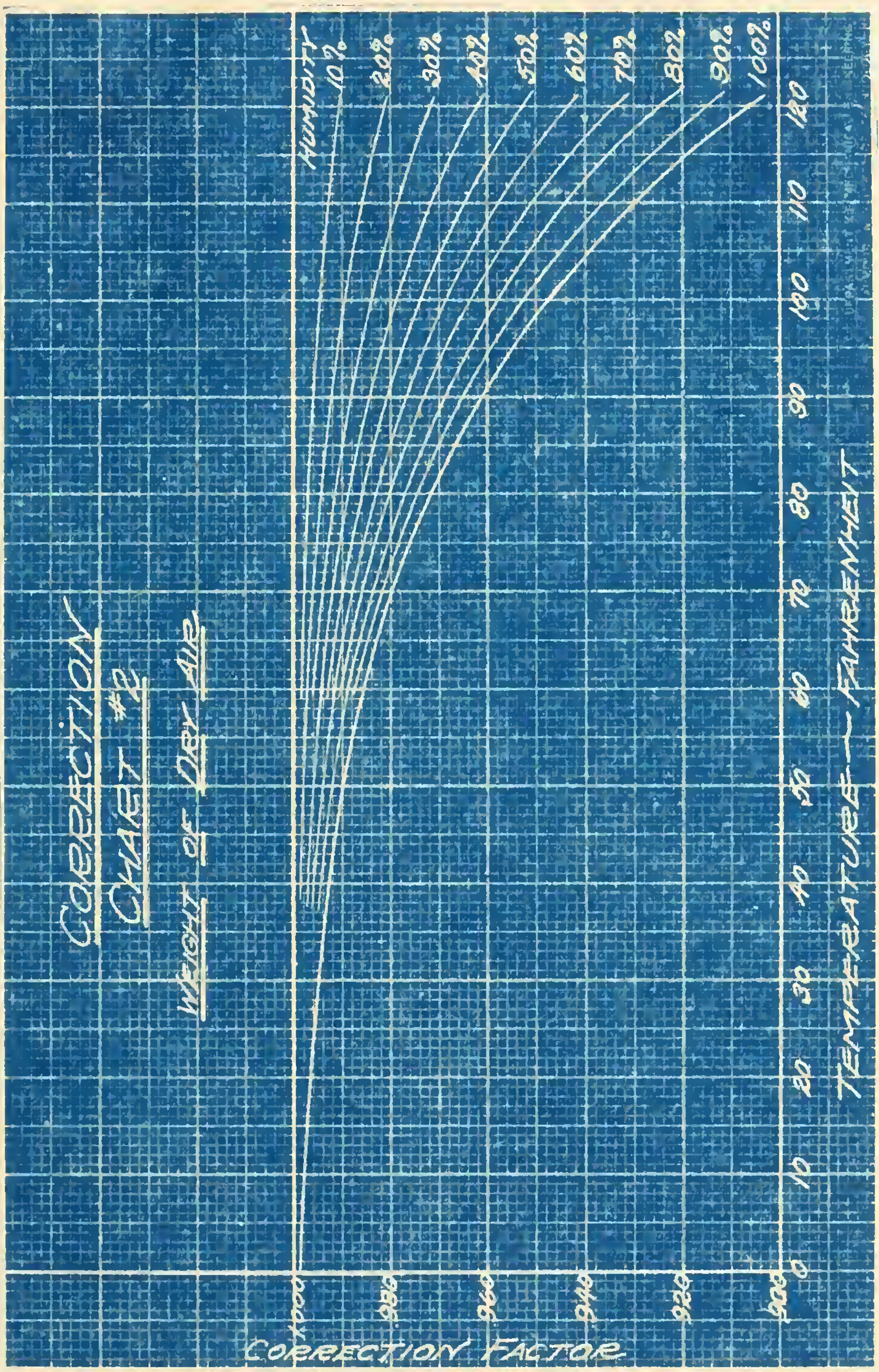

CORRECTION CHART #2
WEIGHT OF DRY AIR
HUMIDITY
10%
20%
30%
40%
50%
60%
70%
80%
90%
100%
0
10
20
30
40
50
60
70
80
90
100
110
120
TEMPERATURE — FAHRENHEIT
CORRECTION FACTOR

Figure 5.

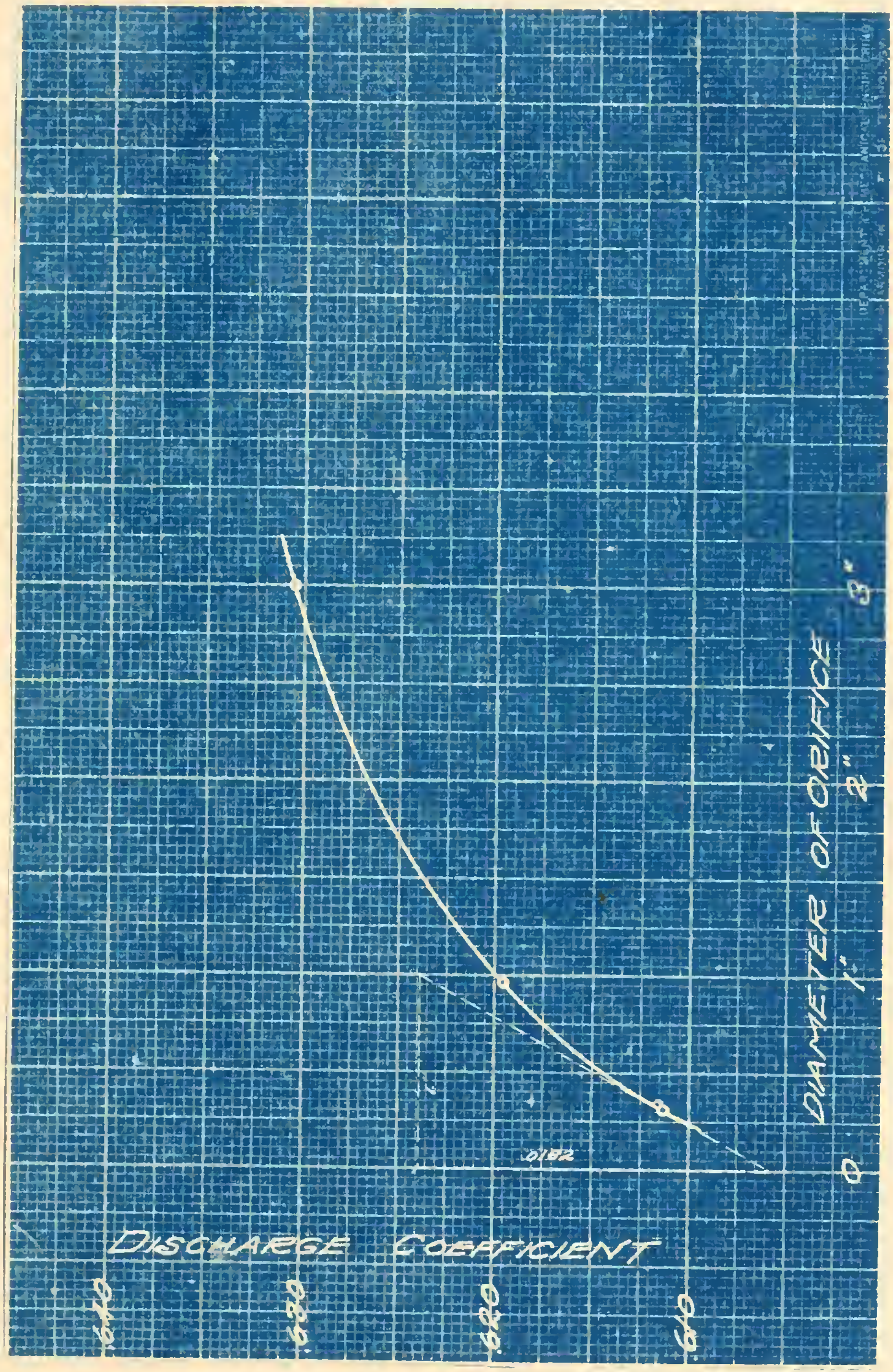

DIAMETER OF ORIFICE
0
1"
2"
3"
.0182
DISCHARGE COEFFICIENT
.640
.630
.620
.610

Figure 6.

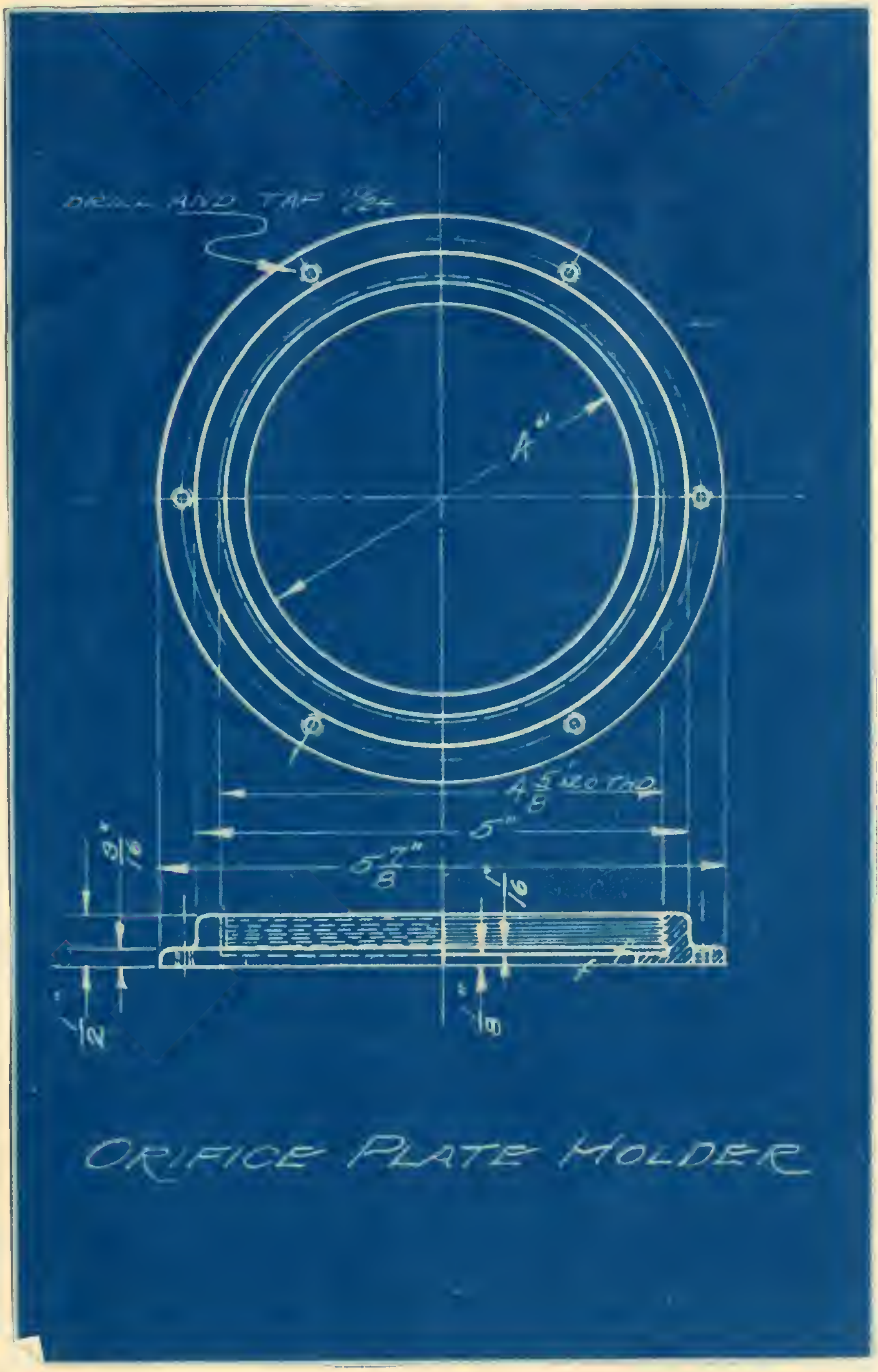


Figure 7.

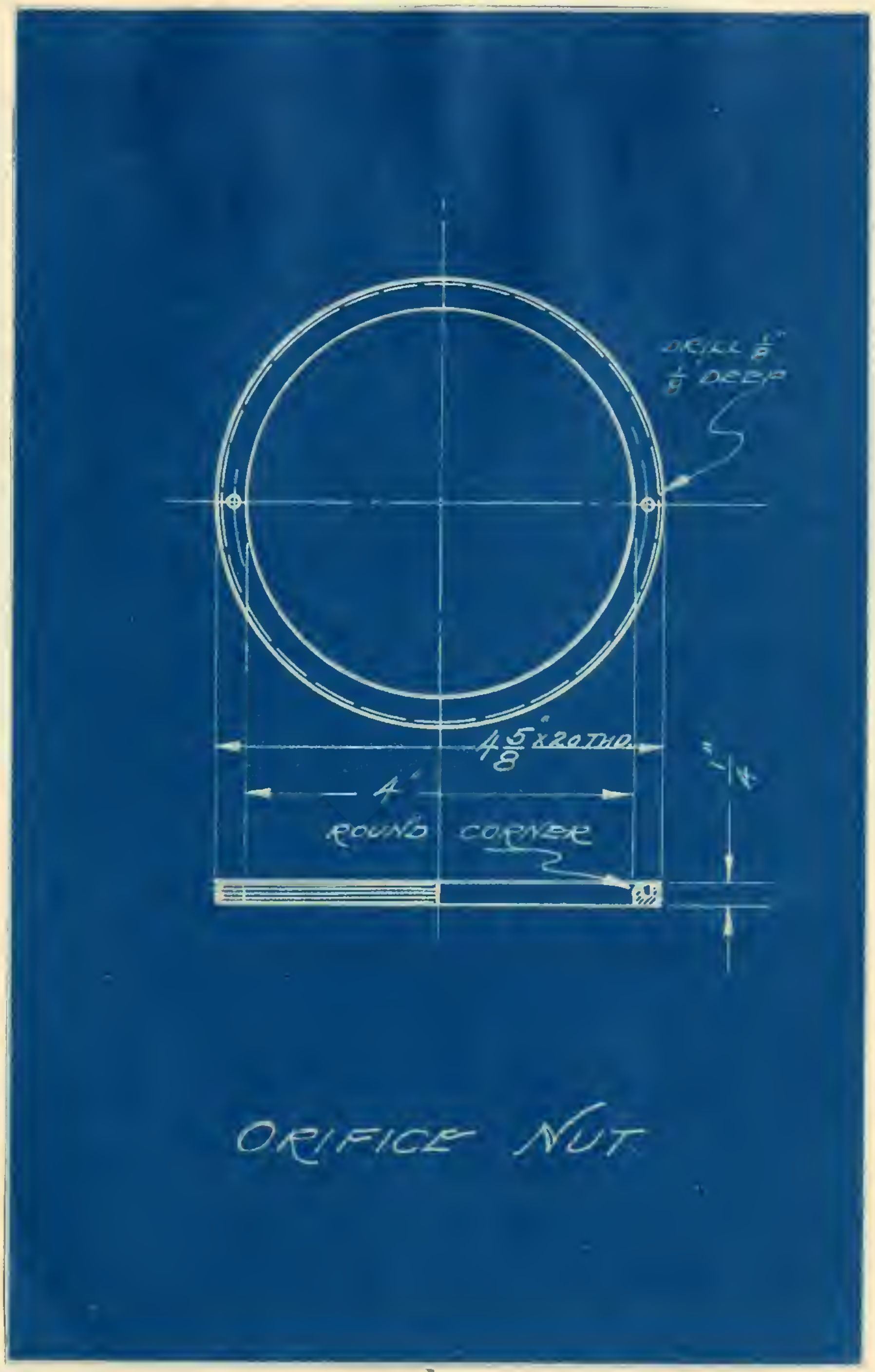


Figure 8.

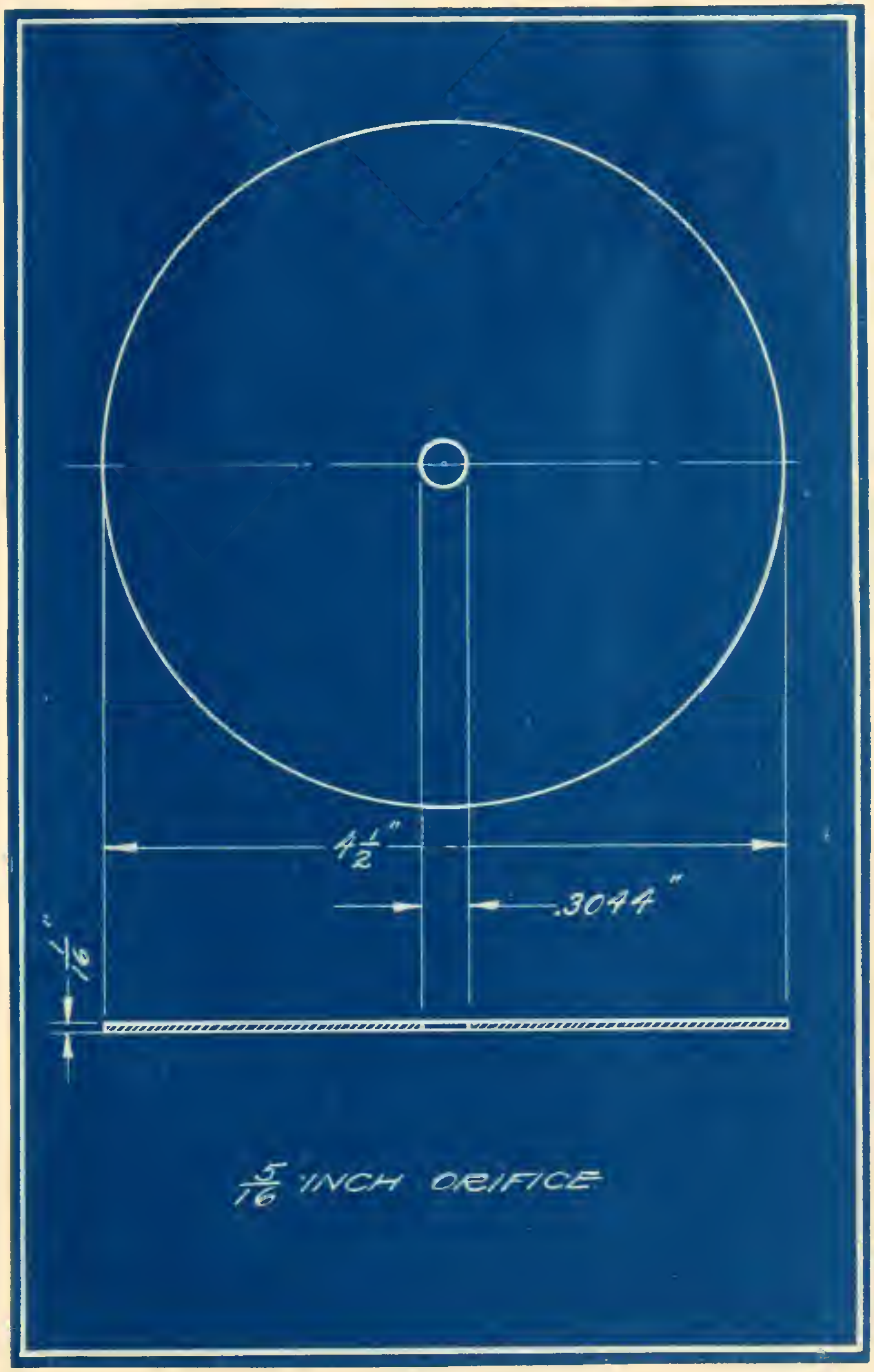


Figure 9.

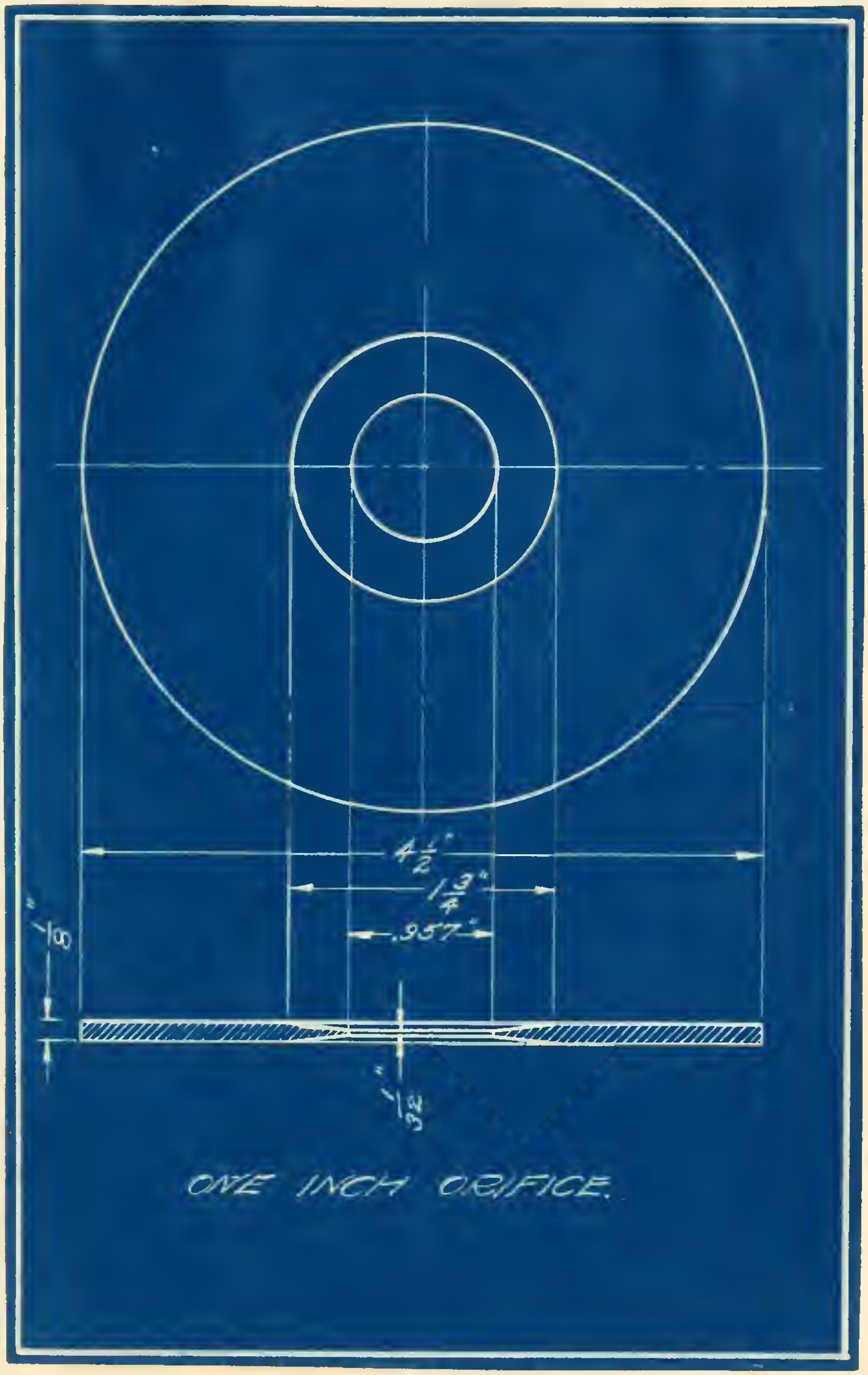


Figure 10

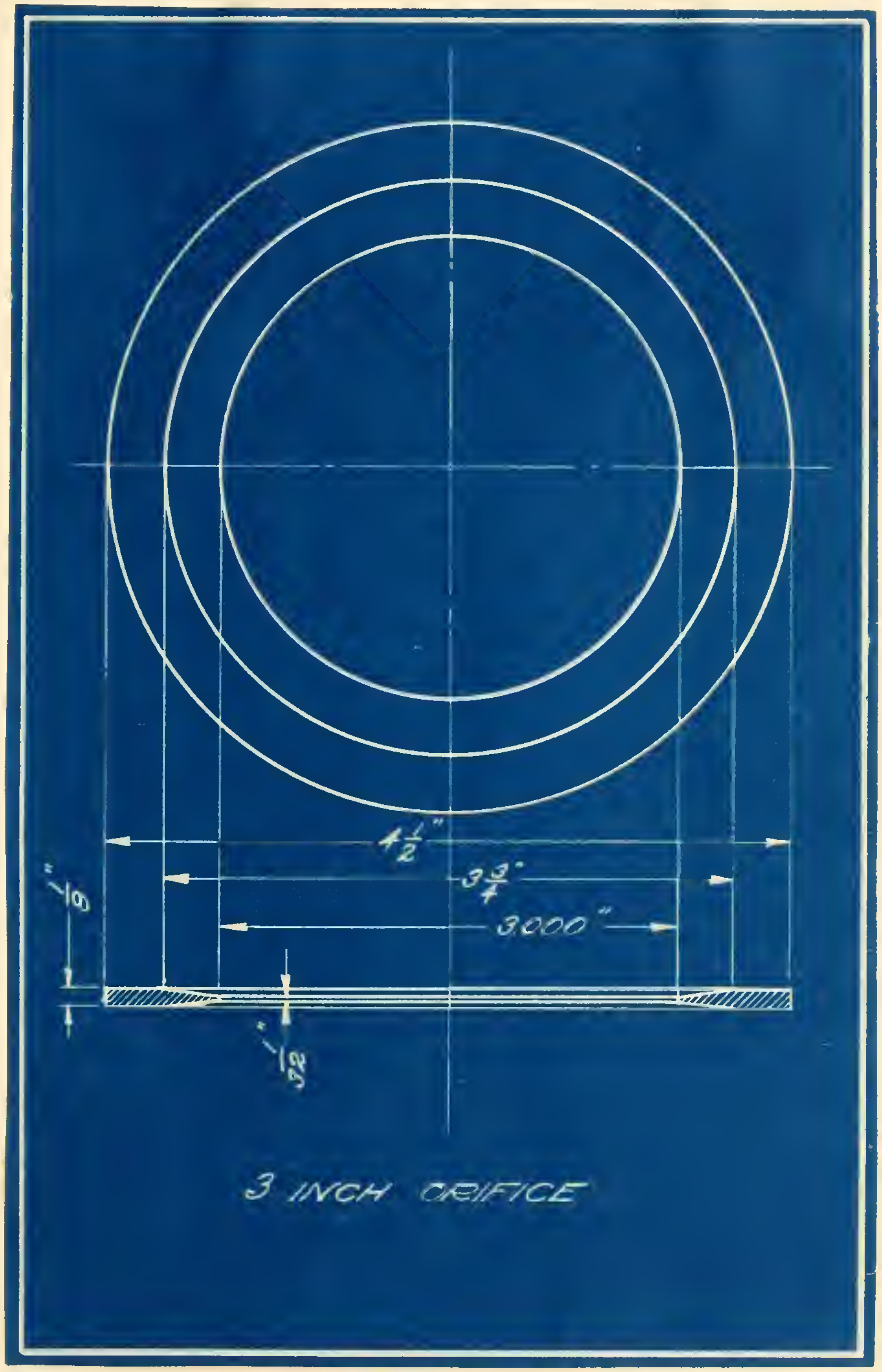
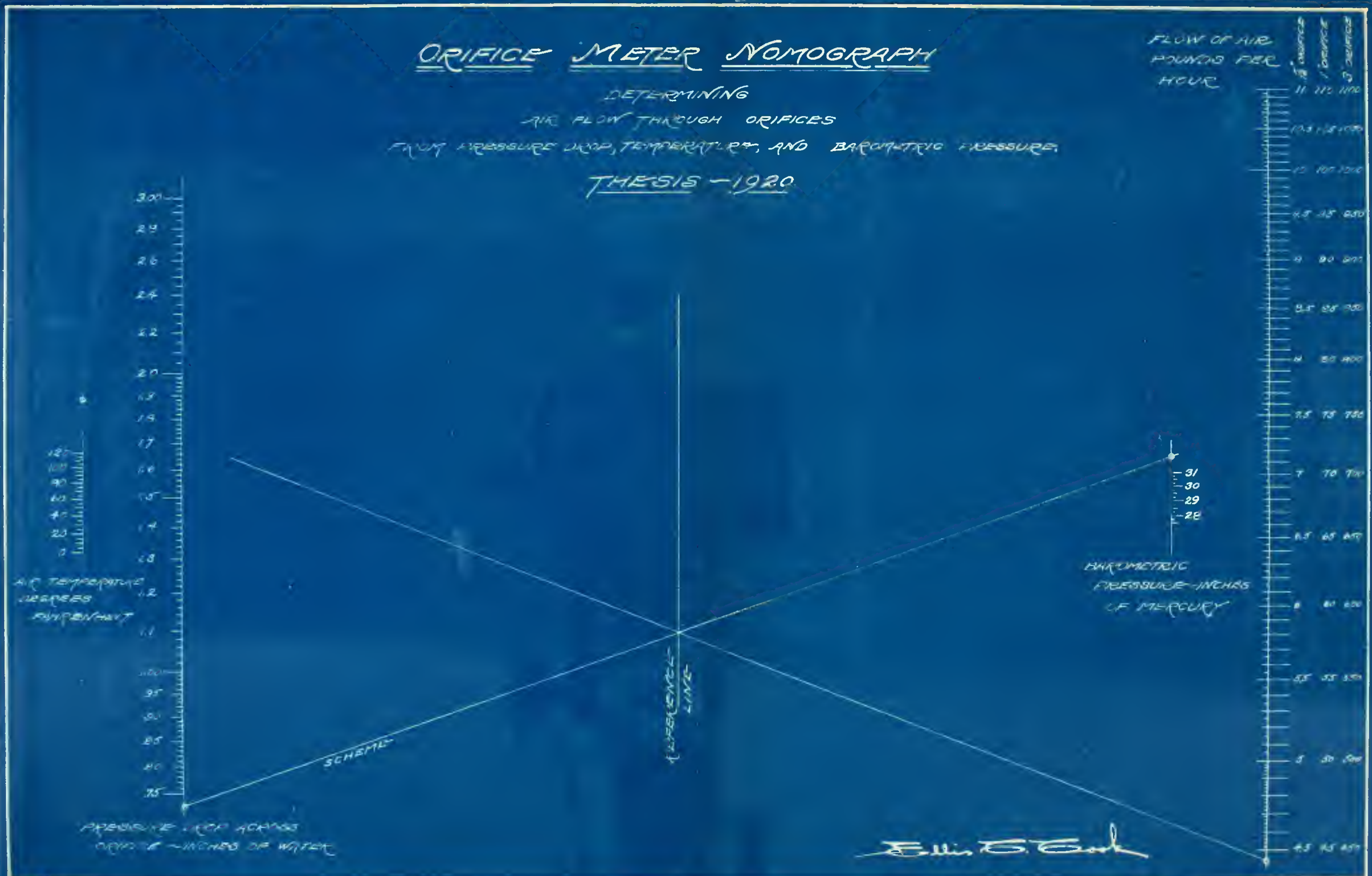


Figure 11.


ORIFICE METER NOMOGRAPH
DETERMINING
AIR FLOW THROUGH ORIFICES
FROM PRESSURE DROP, TEMPERATURE, AND BAROMETRIC PRESSURE.
THESIS - 1920
FLOW OF AIR POUNDS PER HOUR
AIR TEMPERATURE DEGREES FAHRENHEIT
PRESSURE DROP ACROSS ORIFICE - INCHES OF WATER
BAROMETRIC PRESSURE - INCHES OF MERCURY
31
30
29
28
REFERENCE LINE
SCHEME