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Smith, Harrison B.

California Institute of Technology

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Submitted to the Department of Naval Architecture and Marine Engineering on 16 May 1952 in partial fulfillment of the requirements for the degree of Naval Engineer.

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ABSTRACT

A STUDY OF THE EFFECT OF ENGINE SIZE ON HEAT REJECTION

by

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Submitted for the Degree of Naval Engineer in the Department of Naval Architecture and Marine Engineering on May 16, 1952.

A theory has been formulated which states that the rate of heat rejection from the cylinder gases of an internal combustion engine may be approximated by the equation

$$q = \text{CONSTANT}\left[\ell^2 \Delta T\, C_p (\rho_i e S)^n \left(\frac{\ell}{\mu}\right)^{n-1}\right] \qquad (1)$$

where n is an exponent less than unity. For a given engine operating at constant fuel air ratio and water jacket temperature equation (1) takes the form

$$q = \text{CONSTANT}\,(S \times \text{IMEP})^n \qquad (2)$$

The theory predicts that geometrically similar engines all have the same value of n. The relation between heat rejection and engine size for geometrically similar engines operating at the same piston speed and IMEP is given by

$$\frac{q}{\ell^2} = \text{CONSTANT}\,(\ell)^{n-1} \qquad (3)$$

The purpose of this thesis was to investigate the validity of the theory with respect to the effect of engine size on heat rejection. Three geometrically similar spark ignition internal combustion engines installed in Sloan Laboratory were used in this investigation.

The results of this study give, for the M.I.T. G.S.E., a value of .6 for the exponent n defined in equation (2) and a value of .9 for n in equation (3). By theory these two values should be the same. The results, therefore, indicate that as engine size increases, more heat per unit area is rejected than predicted by theory.

Since this study represents the first attempt to correlate engine size and heat rejection, it is recommended that further studies be conducted on the M.I.T. G.S.E. with particular emphasis on the effects of friction, spark advance, and thermal efficiency on heat rejection. It is further recommended that attempt be made to measure separately the heat rejected to the cylinder walls, cylinder head, lubricating oil and in the exhaust gas.

May 16, 1952

Professor J. C. Newell
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Professor Newell:

In compliance with the requirements for the degree of Naval Engineer from the Massachusetts Institute of Technology, we hereby submit a thesis entitled, "A Study of the Effect of Engine Size on Heat Rejection."

ACKNOWLEDGEMENT

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NOMENCLATURE

a,b,c,	Constant
BHP	Brake horsepower
BMEP	Brake mean effective pressure
BPSA	Best power spark advance
^{o}BTC	Degrees before top center
C_p	Specific heat at constant pressure
D	Diameter
e	Volumetric efficiency
E_c	Lower heating value of fuel
F	Fuel air ratio
FHP	Friction horsepower
FMEP	Friction mean effective pressure
G	Mass velocity
GSE	Geometrically similar engines
h	Brake scale absolute reading
h	Coefficient of heat transfer between fluid and surface
h_i	Local individual coefficient of heat transfer for the inner surface.
h_o	Local individual coefficient of heat transfer for the outer surface
IHP	Indicated horsepower
IMEP	Indicated mean effective pressure
k	Thermal conductivity of fluid
k_w	Thermal conductivity of cylinder wall
K	Overall dynometer constant
K	Orifice flow coefficient
ℓ	Characteristic dimension
m	Exponent

$\dot{M}_a$	Air flow rate
$\dot{M}_f$	Fuel flow rate
n	Exponent
N	Revolutions per unit time
P_a	Atmospheric pressure
P_e	Exhaust pressure
P_i	Inlet air pressure
ΔP	Pressure drop across orifice
psia	Absolute pressure
psig	Gage pressure
q	Rate of heat flow
$\dot{Q}$	Rate of heat flow
R_e	Reynolds number
RPM	Revolutions per minute
S	Piston speed ft/min
SA	Spark advance
ΔT	Mean gas temperature minus wall temperature
T_f	Fuel temperature
T_h	Cylinder head temperature
T_i	Inlet air temperature
T_j	Water jacket temperature
U	Local overall heat transfer coefficient
u	Local velocity
V_d	Displacement volume
X_w	Cylinder wall thickness
Y_1	Orifice espansion factor
η_b	Brake thermal efficiency
η_i	Indicated thermal efficiency

η_m Mechanical efficiency

ϕ Function

μ Absolute viscosity

ρ Density

INTRODUCTION

The principle of similitude has been used as a tool by the Naval Engineer and Naval Architect since the days of William Froude, in 1870, to predict the performance of ships and propellers from the results of small scale model tests. The concept of dimensional analysis and its application to controlled model experiment have become a powerful tool in producing a practical solution to design problems in many engineering applications.

In the past twenty years a considerable amount of theory has been developed [1] concerning the performance and behavior of geometrically similar internal combustion engines. The validity of certain relations such as weight, gravity and inertia stresses can be demonstrated by mathematical proof [2], but the more complex relationships of heat rejection, combustion and detonation, friction and wear cannot be predicted by theory alone nor proved mathematically. The M.I.T. Geometrically Similar Engines, described in Appendix A, have therefore been built as a means of attacking these and other complex problems through controlled experimentation on equipment which faithfully fulfills the conditions of similitude as they are presently understood.

Because of high cyclic temperatures existing inside the cylinder of an internal combustion engine while operating, it is necessary to remove heat from the cylinder and associated metal parts to prevent destruction.

Numbers in brackets refer to reference numbers.

This is universally accomplished by circulating a cooling fluid, usually water, oil, or air, in or around the cylinder walls and heads. In a closed loop water cooling system some means must be provided for removing the cylinder heat from the cooling fluid before it can be recirculated. In order to design such a means, it is desirable to be able to predict the amount of heat which must be removed from a given engine and how this amount of heat will vary as the size of the engine is varied. The purpose of this study is to determine by experimentation with the M.I.T. Geometrically Similar Engines the effect of engine size on heat rejection. Although the problem is here applied specifically to the internal combustion engine the broader implication to the problem of relating size and behavior of any power unit is obvious.

The general relation for the heat transferred from one fluid through a solid wall to another fluid was first expressed by Newton as

$$dq = U\,dA\,\Delta t \qquad (1)$$

For this case, considering no foreign material such as scale on either side of the solid wall, the local overall heat transfer coefficient U may be defined as follows:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{x_w}{k_w} + \frac{1}{h_o} \qquad (2)$$

where h_i and h_o are the "local individual coefficients of heat transfer"* for the inner and outer surfaces

---

*McAdams, W. H., Heat Transmission, McGraw-Hill, N.Y., 1942.

respectively, and x_w and k_w are the thickness and thermal conductivity of the solid wall. The terms of equation (2) may be considered as resistances to heat flow. Since the thermal conductivity of metals, k_w, is large compared to x_w in engines of usual size, the resistance $\frac{x_w}{k_w}$ is small. In the case of an engine cooling system, where heat must be transferred from a hot gas through the cylinder, to a coolant, the value of h_o may also be very large compared with h_i as long as the coolant in contact with the wall remains a liquid. Therefore, with h_o and k_w large, as is the case for an engine, h_i becomes the controlling factor in determining the overall coefficient U, and hence the amount of heat transferred. In order to determine a relationship for h_i, let us compare the geometry of an engine cylinder with the geometry of shapes for which heat transfer study has been made. The engine cylinder most closely approximates a pipe. For this case McAdams in reference [6] gives by dimensional analysis

$$\frac{hD}{k} = \varphi\left(\frac{DG}{\mu}, \frac{C_p\mu}{k}\right) \tag{3}$$

which may be rewritten as

$$\frac{hl}{k} = \varphi\left(\frac{\rho u l}{\mu}, \frac{C_p\mu}{k}\right) \tag{4}$$

Assuming a power function McAdams rewrites (4) in the following form

$$\frac{hl}{k} = \text{CONSTANT}\left(\frac{\rho u l}{\mu}\right)^n\left(\frac{C_p\mu}{k}\right)^m \tag{5}$$

Since from (1)

$$h = q/A\Delta T \tag{6}$$

$$\frac{q\,l}{k\,A\Delta T} = \text{Const.}\left(\frac{\rho u l}{\mu}\right)^n \left(\frac{C_p \mu}{k}\right)^m \tag{7}$$

or

$$q = a\,A\,\Delta T(\rho u)^n \frac{k}{l}\left(\frac{l}{\mu}\right)^n\left(\frac{C_p\mu}{k}\right)^m \tag{8}$$

For gases

$$\frac{C_p\mu}{k} = \text{CONSTANT} \quad , \quad k = (C_p\mu)\cdot\text{CONSTANT} \tag{9}$$

Then equation (8) may be written as

$$q = b\,A\,\Delta T\,C_p(\rho u)^n\left(\frac{l}{\mu}\right)^{n-1} \tag{10}$$

Applying equation (10) to the transfer of heat from gases in an internal combustion engine through the cylinder walls, if we take the density of the gases in the cylinder as represented by inlet density times volumetric efficiency, and the velocity as represented by piston speed, and the heat transfer area as proportional to l^2 we get

$$q = b(l^2)(\Delta T)C_p(\rho_i e S)^n\left(\frac{l}{\mu}\right)^{n-1} \tag{11}$$

where ΔT is some mean gas temperature minus the wall

temperature, ρ_i is the inlet density, e the volumetric efficiency, S the average piston speed, μ some mean gas viscosity, and l some characteristic dimension.

Equation (11) is an approximation at best and its application involves theoretical difficulties. The velocity of the gas during exhaust blowdown at the exhaust valve is equal to the speed of sound in the gas and is independent of the piston speed. ΔT, C_P, ρ, μ, u, k, and the speed of sound vary continuously during the cycle. Further, the temperatures of the surfaces in contact with the gases are not all the same. However, despite the limitations involved, experiments on individual engines [1] , where l the size factor is a constant, have shown that equation (11) gives a good approximation for the rate of heat transfer from cylinder gases. The authors propose to determine in this study whether or not equation (11) may be used to approximate the rate of heat transfer for an engine in which the size is varied. By using the three Geometrically Similar Engines, we have in effect, one engine whose size has been varied. Geometric similarity is confined to the engines themselves; however, associated equipment is such that similarity of operating conditions may be maintained.

To the best of the authors' knowledge these engines are the only set of geometrically similar engines in existence; as no prior work on heat rejection has been done using these engines, this study represents the

first attempt at direct experimental verification of equation (11) with size as a variable.

In order to obtain data which may be correlated with engine size, the effect of other variables appearing in equation (11) must be eliminated by maintaining them similar or constant for the three engines. This involves maintaining the specific heat C_p and the viscosity μ of the working gas, ΔT, the temperature difference between the working gas and the cylinder wall, and the product ($\rho e S$) constant.

C_p and μ are functions of the gas temperature. Therefore if the average gas temperature is constant, C_p and μ will be constant. The gas temperature is a function of the fuel air ratio, if the compression ratio and thermal efficiency are constant. With compression ratio and fuel-air ratio constant, the thermal efficiency is dependent on spark advance. The point of optimum spark advance (greatest output) varies with load and with speed. If spark advance is maintained constant, thermal efficiency will vary with load and speed. Varying thermal efficiency will vary the exhaust gas temperature, and hence vary the average gas temperature. However, theory predicts [1] and experiments have verified [3] that, at constant load and speed, similar engines have the same thermal efficiency at the same spark advance. Therefore, for similar conditions of operation, speed and load, thermal efficiency and average gas temperature will be constant, and therefore C_p and μ, the viscosity, will both be constant. In addition,

the results of other experiments indicate that the change in the rate of heat transfer is not large when the spark advance is changed a small amount [1] . In view of the above, it appears that the error introduced by using a constant spark advance for all the engines at all conditions of load and speed may be neglected.

As the size of the engine is varied, another difficulty is encountered. For geometrically similar cylinders of different sizes q varies as $\Delta T\, A(l)^{n-1}$ or $\Delta T(l)^{n+1}$, if all other conditions are constant. By dimensional analysis it may be shown that for heat flow through any solid body, in this case the cylinder wall,

$$q = k_w\, l\, \Delta T_w\, f(R_1, R_2, R_3, \cdots R_n) \qquad (12)$$

where R_1, R_2, R_3, are ratios describing the shape of the body, and ΔT_w is temperature drop in the cylinder wall.

If equations (11) and (12) are to give the same results, and assuming that the gas temperature is the same in each case, ΔT_w in equation (12) must be varied as l^n to keep ΔT in equation (11) constant. This means that in order to have the same temperature drop across the cylinder walls of similar engines, the temperature of the cooling fluid would have to be lowered as l is increased. This may be impracticable or undesirable when changes in size are made. However, if the thermal conductivity of the cylinder wall is large, the temperature drop through the wall will be

[illegible]

[illegible]

$$[illegible]$$

[illegible]

[illegible]

small if its thickness is a reasonable size. Since this is the case, the drop through the wall will be small in comparison with the total temperature drop from the gas to the cooling fluid. Therefore as an approximation the temperature difference ΔT in equation (11) may be considered constant for the three engines.

The product ($\rho_i e S$) in equation (11) is not dependent on size and must be held constant if correlation with size is to be obtained. At constant gas temperature and thermal efficiency, as has been assumed, the mean gas density represented by ($\rho_i e$) is proportional to the indicated mean effective pressure. Consequently, equation (11) may be written in the form

$$\frac{q}{\ell^2} = \text{CONST}(\text{IMEP} \times S)^n (\ell)^{n-1} \qquad (13)$$

The IMEP is equal to the sum of friction mean effective pressure, which may be measured, and the brake mean effective pressure which may be measured and controlled. The piston speed, S, may be measured and controlled, therefore the product (IMEP)x(S) may be measured and varied for each engine within the limits of its operating range. At any operating point where the product (IMEP)x(S) is the same for the three engines the effect of this variable will be constant and may be eliminated.

It is seen that the effect of all factors not dependent on size in equation (11) may be eliminated.

The equation in its simplest form relating engine size to heat rejection may, therefore, be written as

$$q/\ell^2 = \text{CONSTANT}(\ell)^{n-1} \qquad (14)$$

Equation (11) predicts that heat rejection, with ρ_i, e, S, ℓ, C_p, and μ constant, is dependent on ΔT. Neglecting the variation in the temperature drop through the cylinder wall as size is varied, ΔT, with the gas temperature constant, is dependent on the average jacket temperature. Therefore, if heat rejection is plotted against the average jacket temperature, this curve may be extrapolated to the point at which no heat will be rejected. The temperature of the jacket at this point should be the same as the average temperature of the working gas in the cylinder.

PROCEDURE

The experimental procedure used was to maintain similarity of operating conditions between the three engines. The data obtained were such that correlation between engine size and heat rejection could be obtained by eliminating the effects of non-related variables by maintaining their effects constant. If curves of heat rejection plotted against the product of the load and the speed factors ($\rho_i e S$) for each engine (ℓ = constant) are obtained, cross plots may be drawn at constant ($\rho_i e S$), over a range of values, yielding curves of heat rejection plotted against the size factor ℓ.

The procedure described above, yielded information relating heat rejection to the output of the engine. The same procedure also yielded information relating heat rejection to the input to the engine, which, at constant fuel air ratio may be measured by the amount of air taken by the engine.

Likewise, data were obtained by which heat rejection may be related to the jacket temperature or ΔT, the effect of other factors being eliminated as far as possible. This was achieved by maintaining speed, air consumption at constant fuel air ratio, and all other operating variables constant or similar for the three engines while the average jacket temperature was varied independently.

In order to preserve similarity between the engines, lubricating oils were used with varying viscosity so that

the ratio (μ/ℓ) at 250°F was constant for the three engines.

For all runs, data were taken only after conditions had stabilized. The length of time between readings and the number of readings taken were determined by the degree of stability attained. During all runs at least 2 observers were present to maintain precise control and to insure nearly simultaneous data observation. For most runs satisfactory stability was easily obtained and maintained.

The actual values chosen for each of the operating variables in each series of test runs and the measuring means are given below.

Friction runs:

FMEP was measured by motoring each engine. This was done with electric dynamometers, and a hydraulic scale. Piston speed was arbitrarily varied over a range of 600 ft/minute to 2000 ft/minute and FMEP was measured at constant exhaust pressure and three inlet pressures for each piston speed. Thus a family of curves was obtained for each engine from which the FMEP may be read for any speed or inlet pressure by interpolation or extrapolation.

Firing runs on the 2½" engine indicated that with an inlet air temperature of 150°F, which was used for friction runs on that engine, fuel evaporation was incomplete. Therefore a value of 160°F was chosen, and this value was used for the friction runs for the 6" engine. However, as the inlet air temperature has small effect on the pumping friction only, it was decided that the friction data obtained for the 2½" and 4" engines were adequate, for the purpose intended.

The dynamometer zero position was determined by taking an average of 20 steady state zero positions reached after displacement from rest. The dynamometer coupling was disconnected during this determination. After a zero reading was found, the scale was adjusted to read zero.

SUMMARY OF OPERATING VARIABLES

Variable	Value	Means of Measurement
Air consumption	Not controlled	Standard ASME square edged orifice. (ΔP air by water manometer)
Air inlet temperature	160°F	Thermocouple in inlet pipe
Main bearing temperature	Not controlled	Thermocouple
Air inlet pressure	28" Hg to 10" Hg (absolute)	Mercury manometer
Exhaust pressure	1" Hg	Mercury manometer
Oil sump temperature	150°F	Mercury thermometer
Main bearing oil supply temperature	Not controlled	Mercury thermometer
Oil pressure	50 psi	Bourdon pressure gauge
Piston speed	600-2000 ft./min.	Stroboscope and tachometer
Water circulation rate through jacket	.04 $lb/sec/in^2$	ASME standard orifice
Water jacket temperature	145°F	Mercury thermometer
FMEP	Variable	Dyn. & Hydraulic scale. Scale reading by mercury manometer.

<u>Heat Rejection Runs</u>:

The procedure for firing runs was similar to that for friction runs in that piston speed was arbitrarily varied over a range of 600 ft./min. to 2000 ft./min. and inlet pressures were varied at each speed to give a range of BMEPS. Operating conditions were controlled as far as possible so that the friction curves would be directly applicable to obtain the IMEP at which the engine was

operating during each run.

At low speeds some difficulty was experienced in maintaining stable conditions at low brake loads. This difficulty was partially eliminated by changing the size of the air orifice for low loads. The orifice sizes chosen were such that the pressure drop across the orifices were sufficient to insure accurate air measurements.

The cooling water system used was so designed that only a portion of the water being circulated was cooled. The cooled water was then mixed with the uncooled water and recirculated through the engine. This was done in order to obtain a large temperature difference and thus increase the accuracy of the heat rejection measurement. The portion of the water to be cooled was determined arbitrarily, but, was such that a large temperature drop across the cooler was obtained while maintaining constant circulation rate through the engine water jacket. The recirculating portion of the cooling water system was insulated to prevent excessive unaccountable heat losses. The fuel-air ratio chosen was .078 pounds fuel per pound of air. This value, being a rather rich mixture, was chosen so that unavoidable variations in fuel-air ratio would have a little effect as possible on heat rejection. Fuel-air ratio of .078 also makes it possible for later workers to enter these data on Hottel charts in studying fuel-air cycles of similar engines. Although a previous investigation [3] indicated that the fuel-air ratio for best power for these engines is .073, it was decided that

[illegible]

a richer mixture would probably insure more nearly similar conditions. It was also observed from brake reading that .073 is lower than the actual best power fuel-air ratio.

The gasoline used was a commercial unleaded automotive fuel. An unleaded fuel was used to prevent lead deposits on the heat transfer surfaces which would alter the heat transfer characteristics of the surfaces during the period of testing. Before taking any data, the cylinder heads were removed and cleaned to remove the lead deposits resulting from previous firing of the engines. The cooling system was flushed with detergent and refilled with a weak rust prevention solution (potassium chromate).

After investigating previous work on these engines [3] it was found that best power spark advance varied from twenty to forty degrees depending on bore, RPM, and F. It was decided that 25 degrees spark advance was a good mean value at which all engines would operate satisfactorily at all conditions planned; therefore this value was used for all runs.

The spark plugs assigned to the engines are as follows:

6" engine - Champion 7 or equivalent

4" engine - Champion J8 or equivalent

2½" engine - Champion Y-4A or equivalent

These plugs represent the medium heat range, being neither hot nor cold. Because the low density of charge at the higher vacuum runs makes wide spark gap desirable, a gap of .035" was selected for all engines and all runs.

SUMMARY OF OPERATING VARIABLES

Variable	Value	Means of Measurement
Air consumption	Measured variable	Standard ASME square edged orifice. (ΔP air by water manometer)
Air inlet temperature	160°F	Thermocouple
Air inlet pressure	As noted above	Mercury manometer
Main bearing temperature	Not controlled	Thermocouple
Exhaust pressure	1" Hg	Mercury manometer
Oil sump temperature	150°F	Mercury thermometer
Oil to bearings supply temperature	Not controlled	Mercury thermometer
Oil pressure	Approx. 50 psi	Bourden pressure gauge
Fuel - air ratio	.078	Calculated
Fuel flow rate	As required for const. F	Calibrated rotameter
Fuel temperature	Not controlled	Mercury thermometer
Piston speed	600 ft./min. 2000 ft./min.	Stroboscope
Water circulation rate through jacket	.04 lb/sec/in^2	ASME standard orifice
Water jacket temperature	145°F average	Mercury thermometer
Water circulation rate through cooler	Measured variable	Calibrated rotameter
Water temp. difference through cooler	Measured variable	Mercury thermometer
BMEP	Measured variable	Dynamometer and hydraulic scale. Scale readings by Mercury manometer.

<u>Heat Rejection at Varying Jacket Temperature</u>:

Data were taken at only one condition of speed and load. A piston speed of 1200 ft./min. and inlet pressure of 3" Hg. vacuum when the average jacket temperature was 145° were chosen as operating points. As the jacket temperature was varied from 145°, the air consumption

was maintained at the same value, rather than the inlet pressure. This was done to eliminate dependence on equal volumetric efficiencies for similarity between the three engines. Unequal volumetric efficiencies were possible because of the dissimilar air intake systems. Otherwise operating procedure was the same as for the heat rejection runs described above. Data for as wide a range of average jacket temperatures as was possible to obtain, with the equipment used, were taken.

RESULTS

Figure	I	Friction MEP (Motoring) versus Piston Speed, $2\frac{1}{2}$" G.S.E.
Figure	II	Friction MEP (Motoring) versus Piston Speed, 4" G.S.E.
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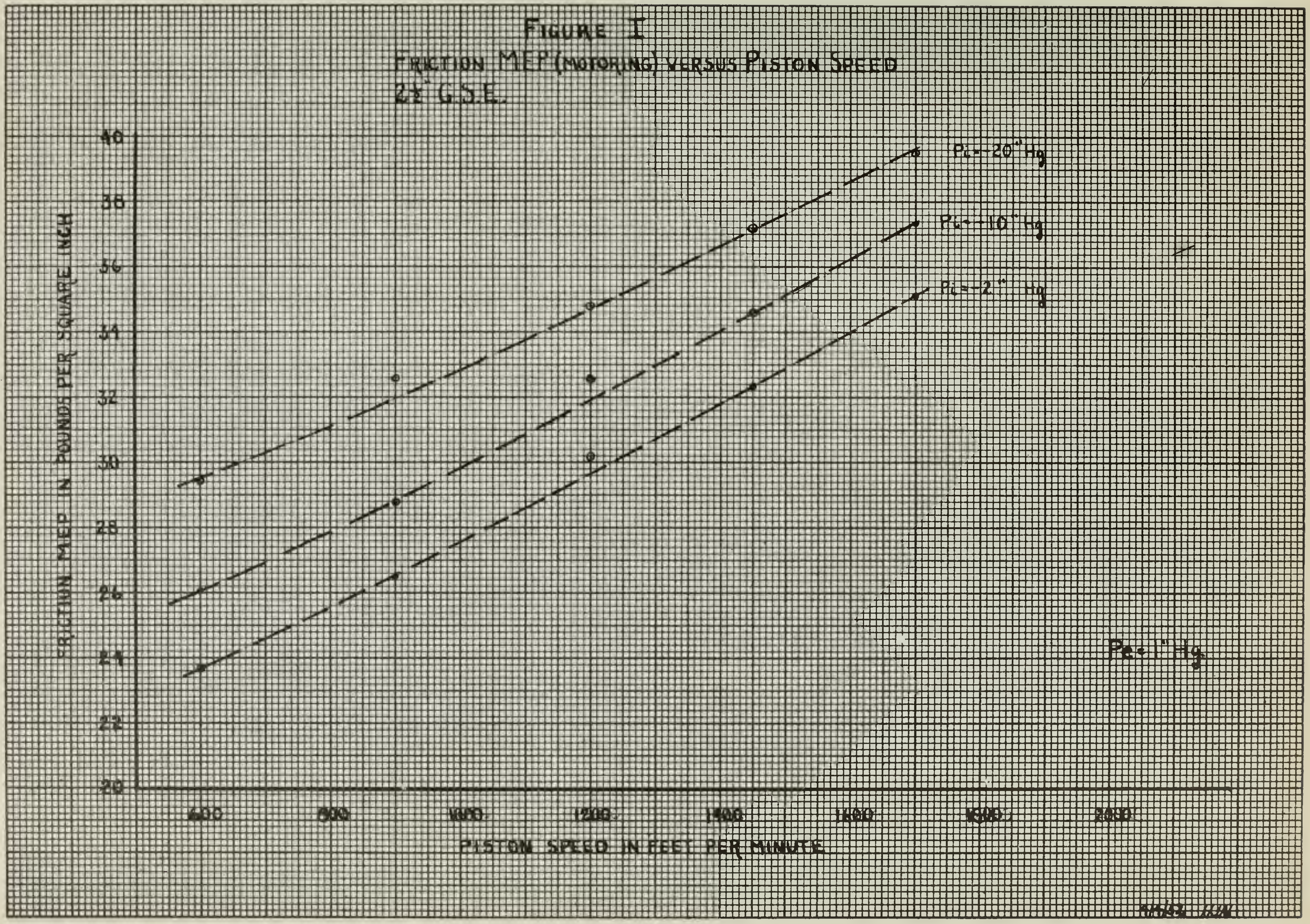
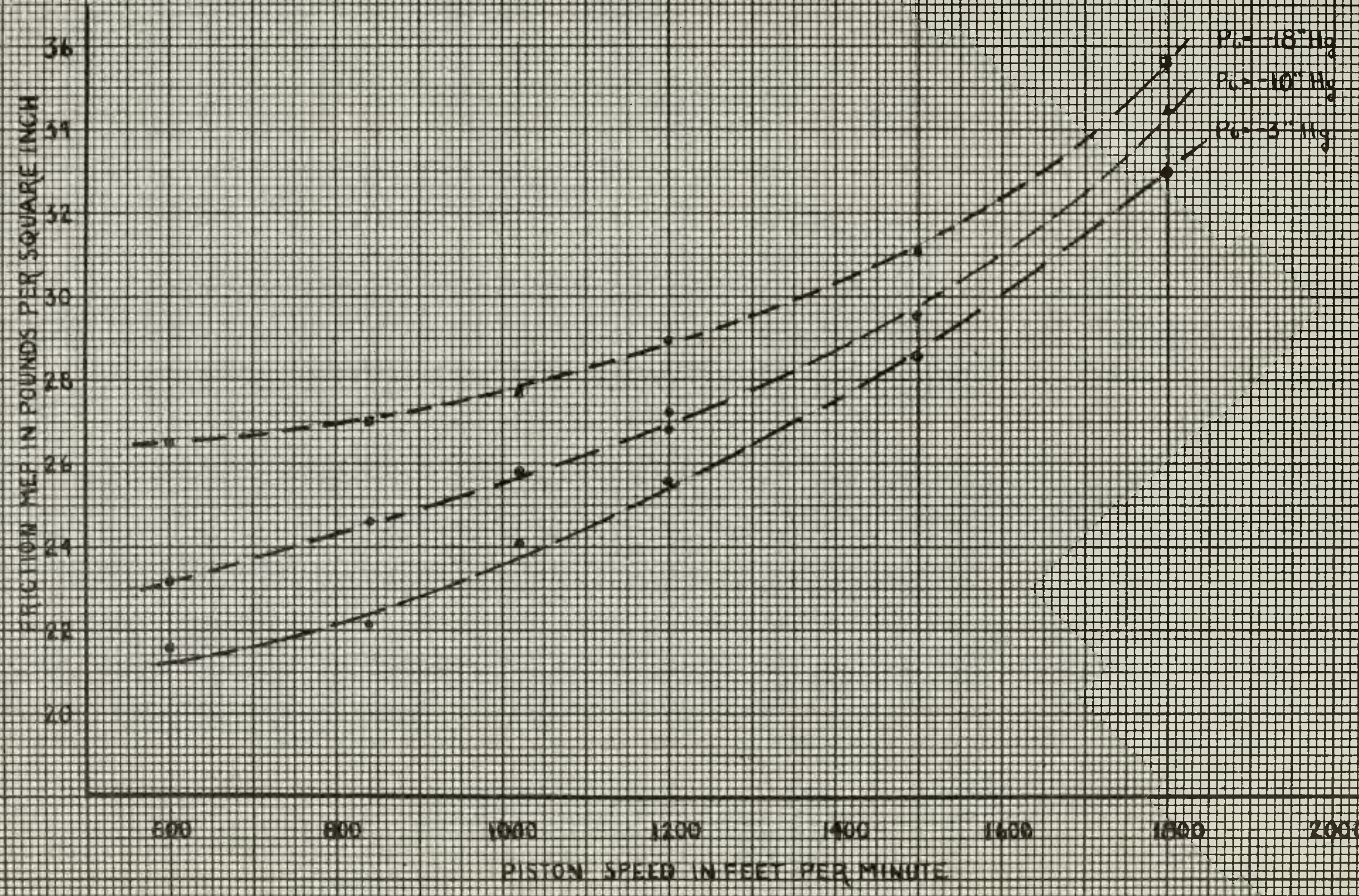


FIGURE I
FRICTION MEP (MOTORING) VERSUS PISTON SPEED
2½" G.S.E.
FRICTION MEP IN POUNDS PER SQUARE INCH
40
38
36
34
32
30
28
26
24
22
20
400
600
800
1000
1200
1400
1600
1800
2000
PISTON SPEED IN FEET PER MINUTE
Pi = -20" Hg
Pi = -10" Hg
Pi = -2" Hg
Pe = 1" Hg

FIGURE II
FRICTION MEP (MOTORING) VERSUS PISTON SPEED
4" G.S.E.



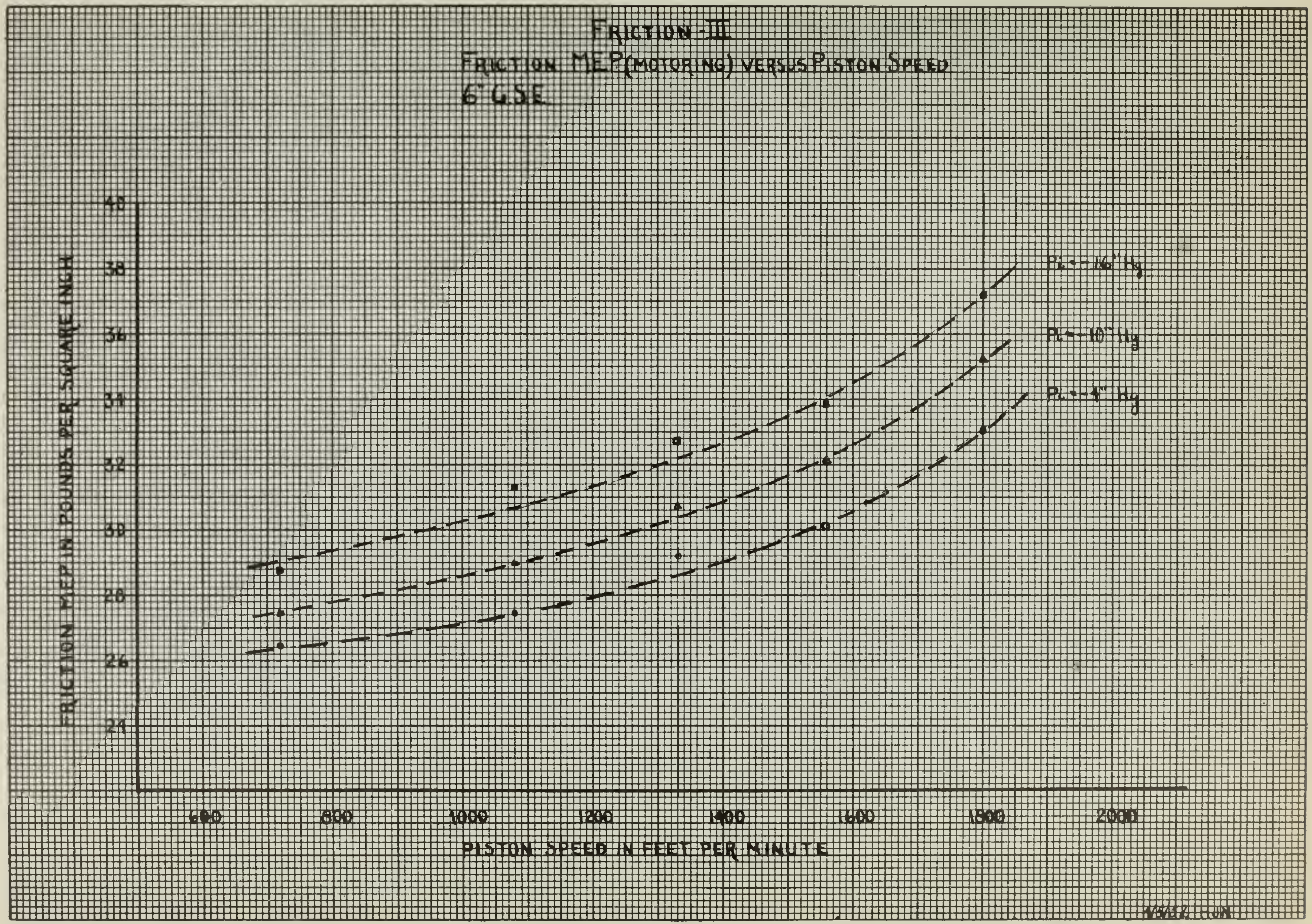
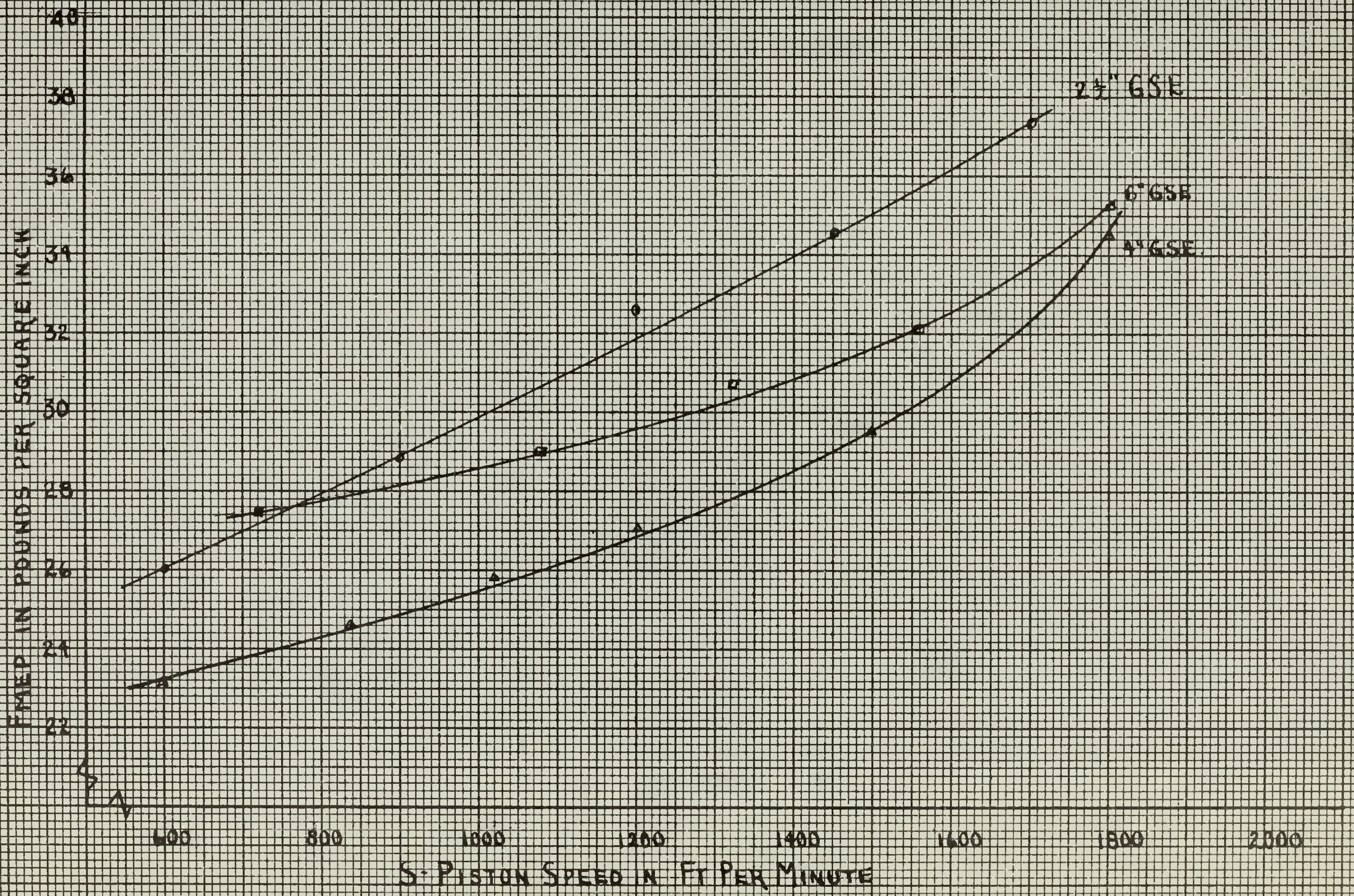

FRICTION - III
FRICTION MEP (MOTORING) VERSUS PISTON SPEED
6" G.S.E.
FRICTION MEP IN POUNDS PER SQUARE INCH
40
38
36
34
32
30
28
26
24
600
800
1000
1200
1400
1600
1800
2000
PISTON SPEED IN FEET PER MINUTE
Pi = -16" Hg
Pi = -10" Hg
Pi = -4" Hg

FIGURE - IV

FRICTION M.E.P. (MOTORING) VS PISTON SPEED

3 G.S.E. P_i = -10" Hg.



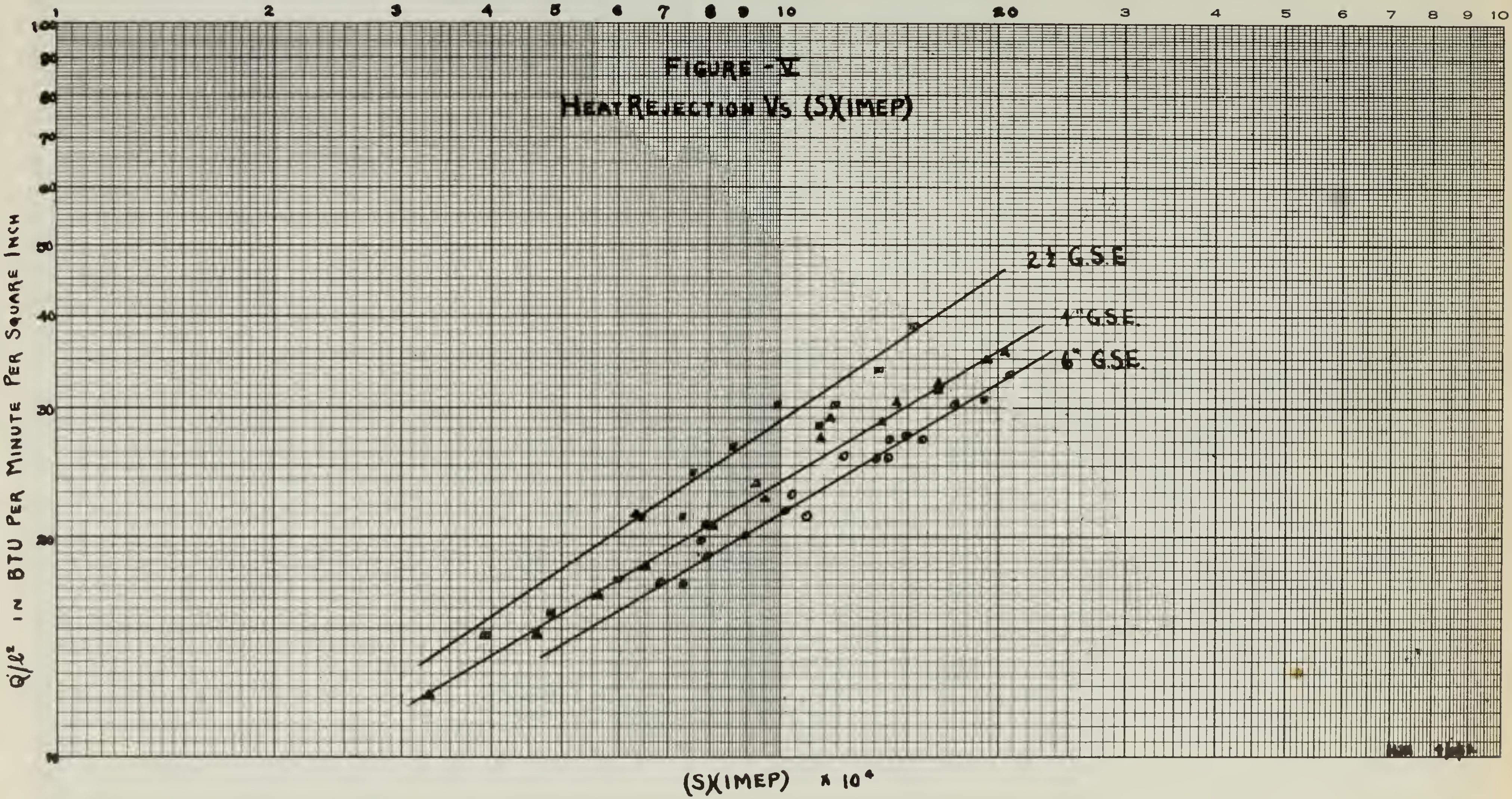


FIGURE - V
HEAT REJECTION VS (S)(IMEP)

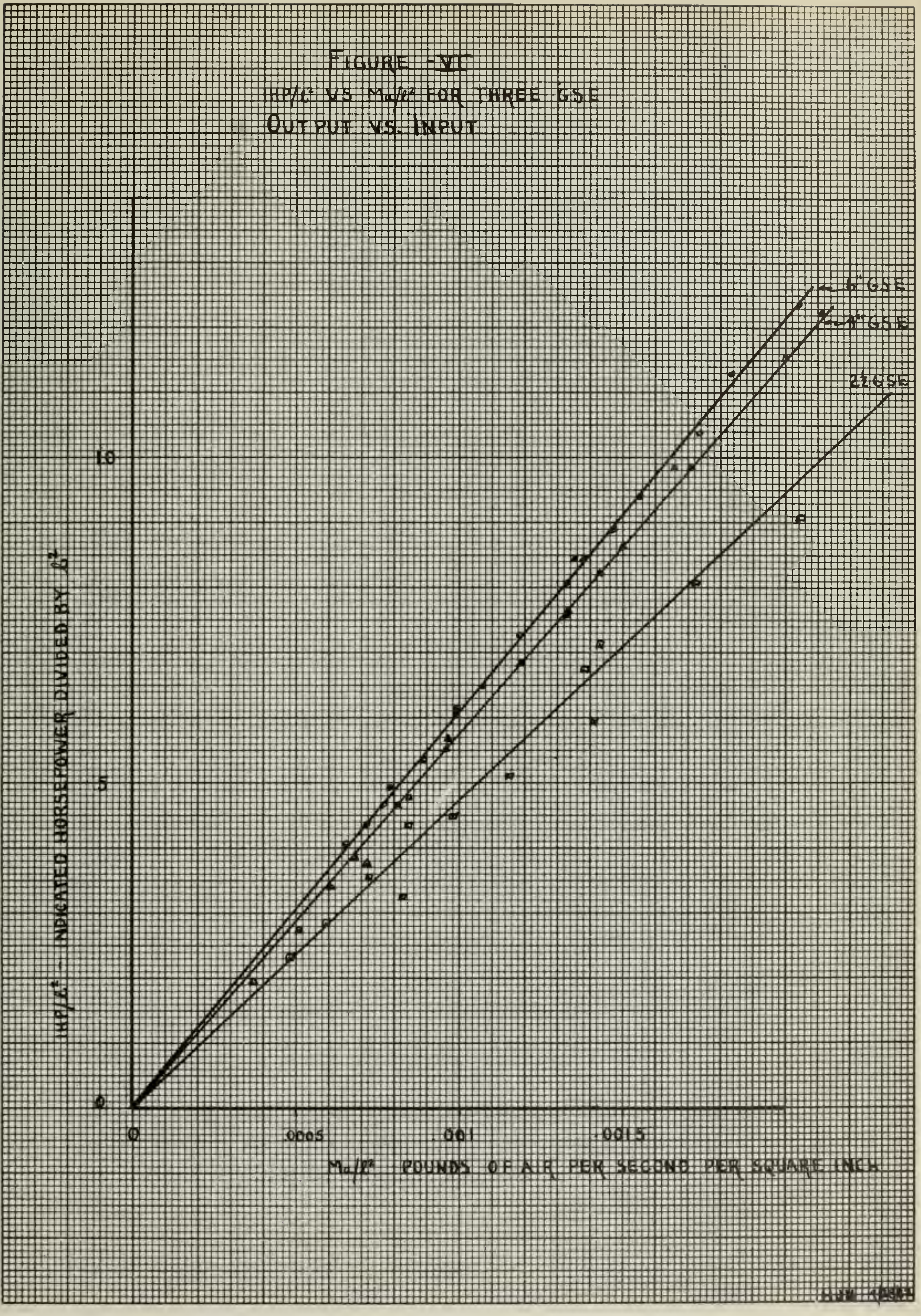


FIGURE - VI
IHP/ℓ² VS Ma/ℓ² FOR THREE GSE
OUTPUT VS. INPUT
6" GSE
4" GSE
2½ GSE
1.0
.5
0
0
.0005
.001
.0015
IHP/ℓ² - INDICATED HORSEPOWER DIVIDED BY ℓ²
Ma/ℓ² POUNDS OF AIR PER SECOND PER SQUARE INCH

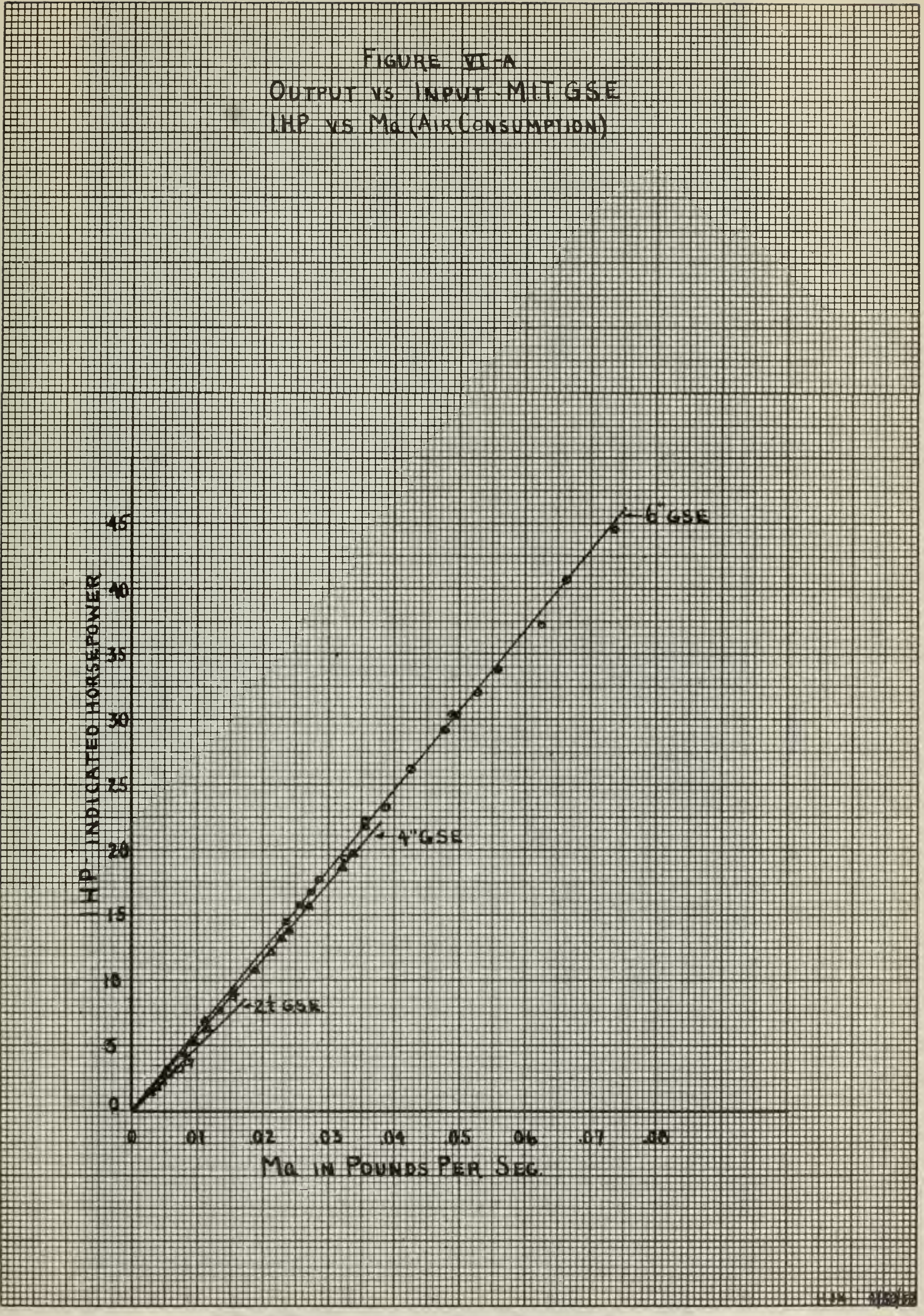


FIGURE VI-A
OUTPUT VS INPUT - M.I.T. GSE
IHP VS Ma (AIR CONSUMPTION)
6" GSE
4" GSE
2½" GSE
IHP - INDICATED HORSEPOWER
Ma IN POUNDS PER SEC.
0
5
10
15
20
25
30
35
40
45
.01
.02
.03
.04
.05
.06
.07
.08

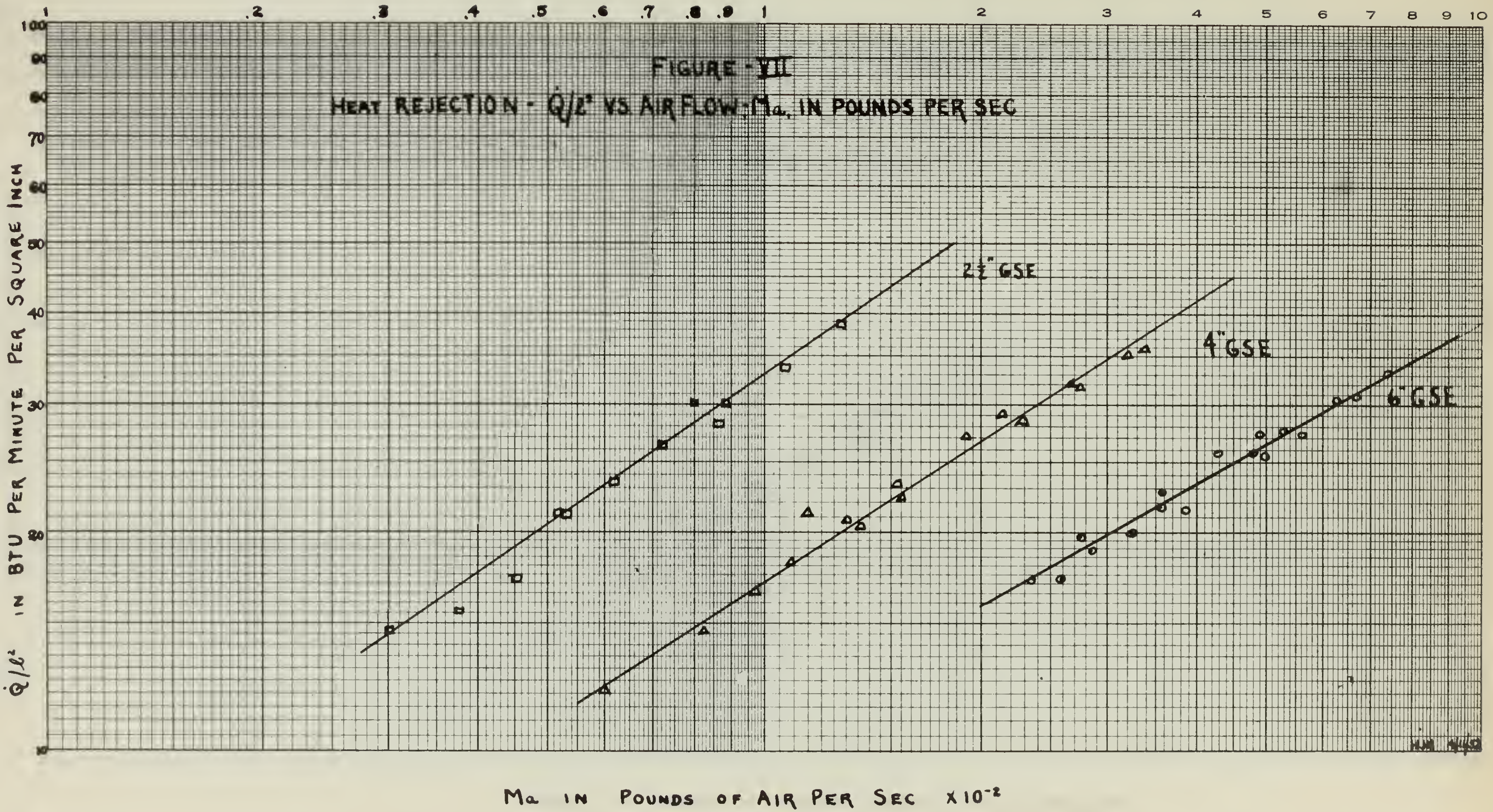


FIGURE - VII
HEAT REJECTION - $\dot{Q}/\ell^2$ VS AIR FLOW, M_a, IN POUNDS PER SEC
2½" GSE
4" GSE
6" GSE
$\dot{Q}/\ell^2$ IN BTU PER MINUTE PER SQUARE INCH
M_a IN POUNDS OF AIR PER SEC $\times 10^{-2}$

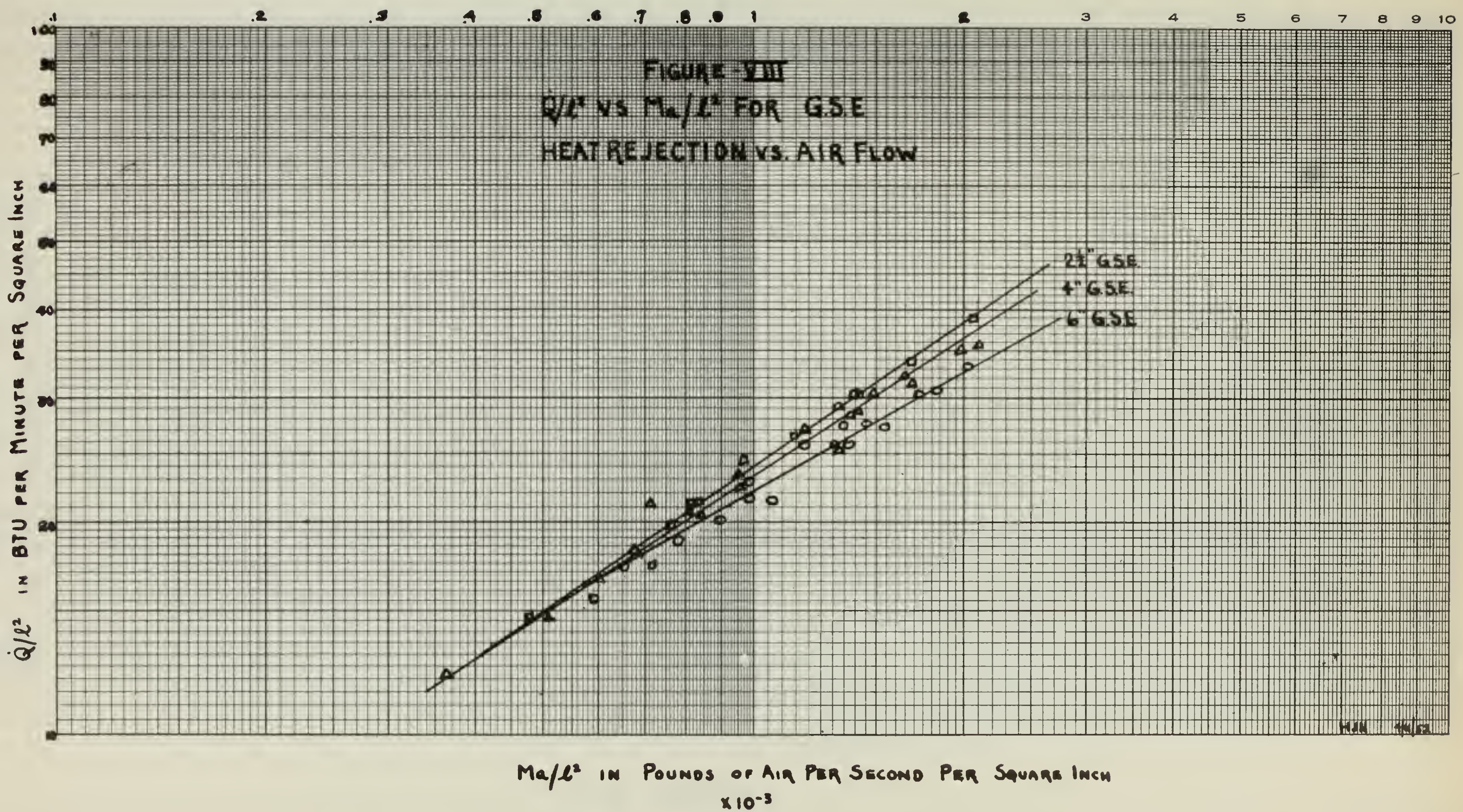


FIGURE - VIII
$\dot{Q}/\ell^2$ VS M_a/ℓ^2 FOR G.S.E
HEAT REJECTION VS. AIR FLOW
2½" G.S.E.
4" G.S.E.
6" G.S.E.
$\dot{Q}/\ell^2$ IN BTU PER MINUTE PER SQUARE INCH
M_a/ℓ^2 IN POUNDS OF AIR PER SECOND PER SQUARE INCH
$\times 10^{-3}$
HJN

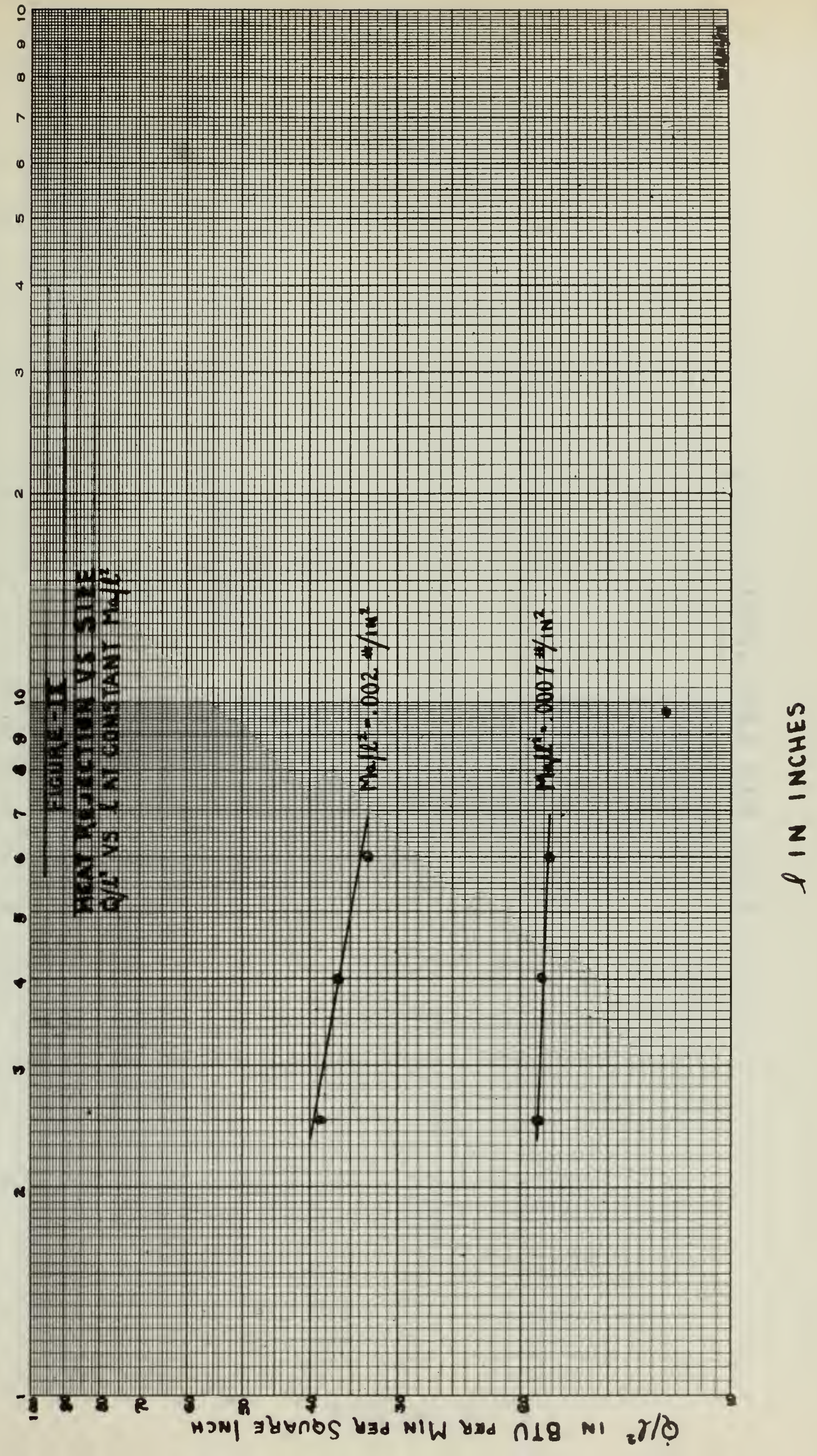


FIGURE-IX
HEAT REJECTION VS SIZE
Q/l² VS l AT CONSTANT M_eff/l²
M_eff/l² = .002 #/in²
M_eff/l² = .0007 #/in²
l IN INCHES
Q/l² IN BTU PER MIN PER SQUARE INCH

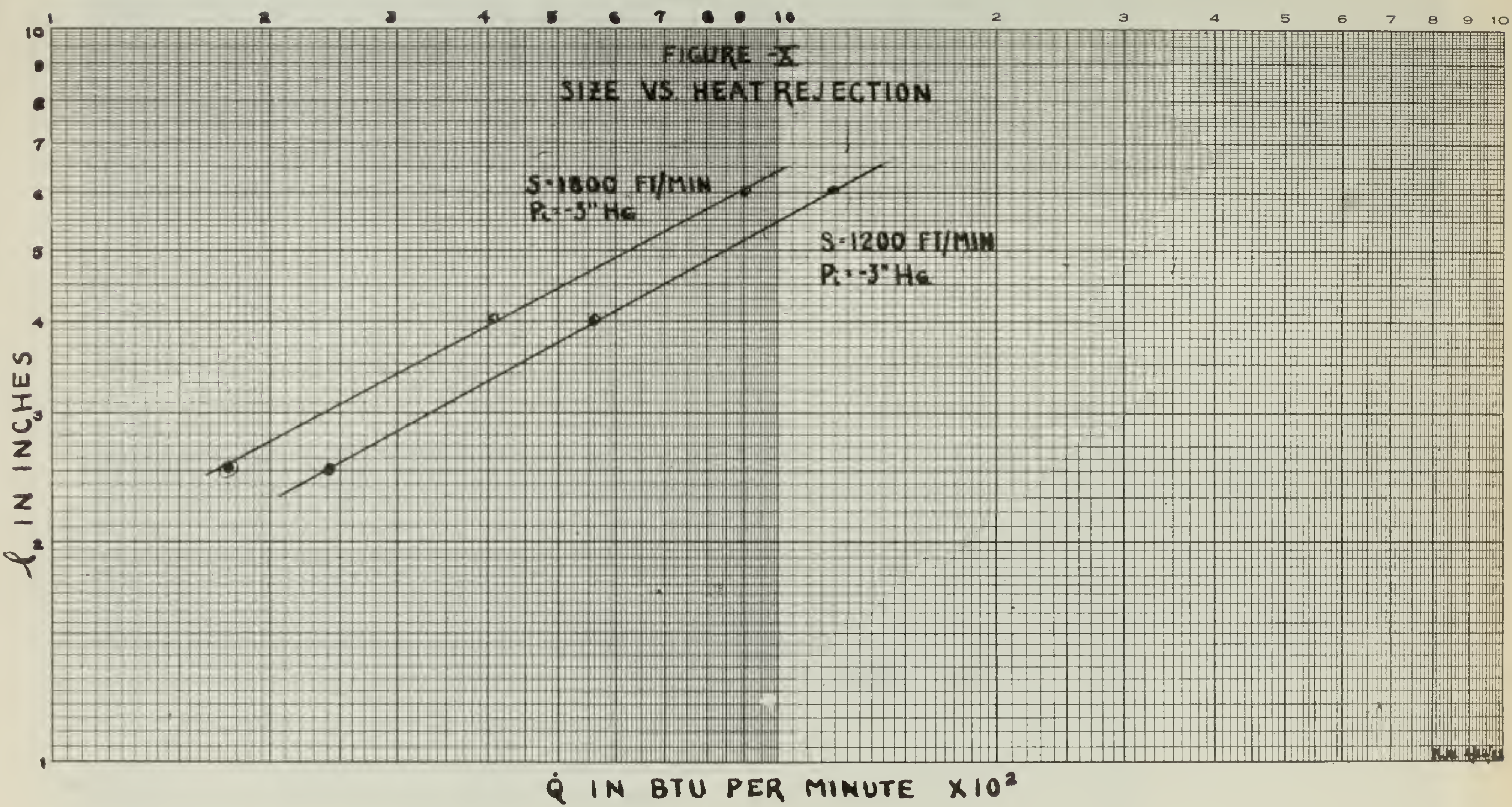


FIGURE -X
SIZE VS. HEAT REJECTION
S=1800 FT/MIN
P_t = -3" Hg
S=1200 FT/MIN
P_t = -3" Hg
l_2 IN INCHES
$\dot{Q}$ IN BTU PER MINUTE X10^2

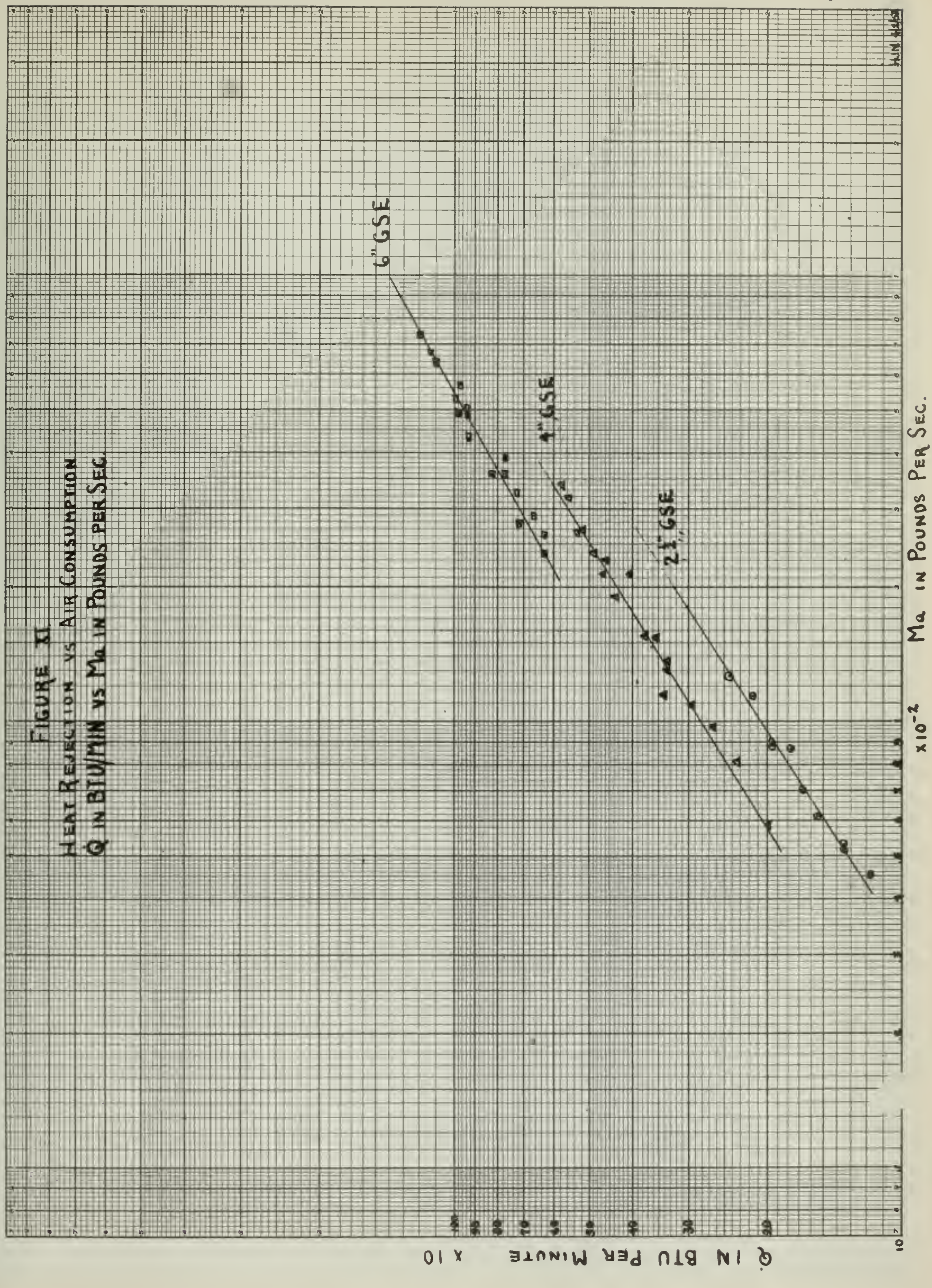


FIGURE XI
HEAT REJECTION VS AIR CONSUMPTION
Q IN BTU/MIN VS Ma IN POUNDS PER SEC.
6" GSE
4" GSE
2½" GSE
Ma IN POUNDS PER SEC.
x10^-2
Q IN BTU PER MINUTE x 10

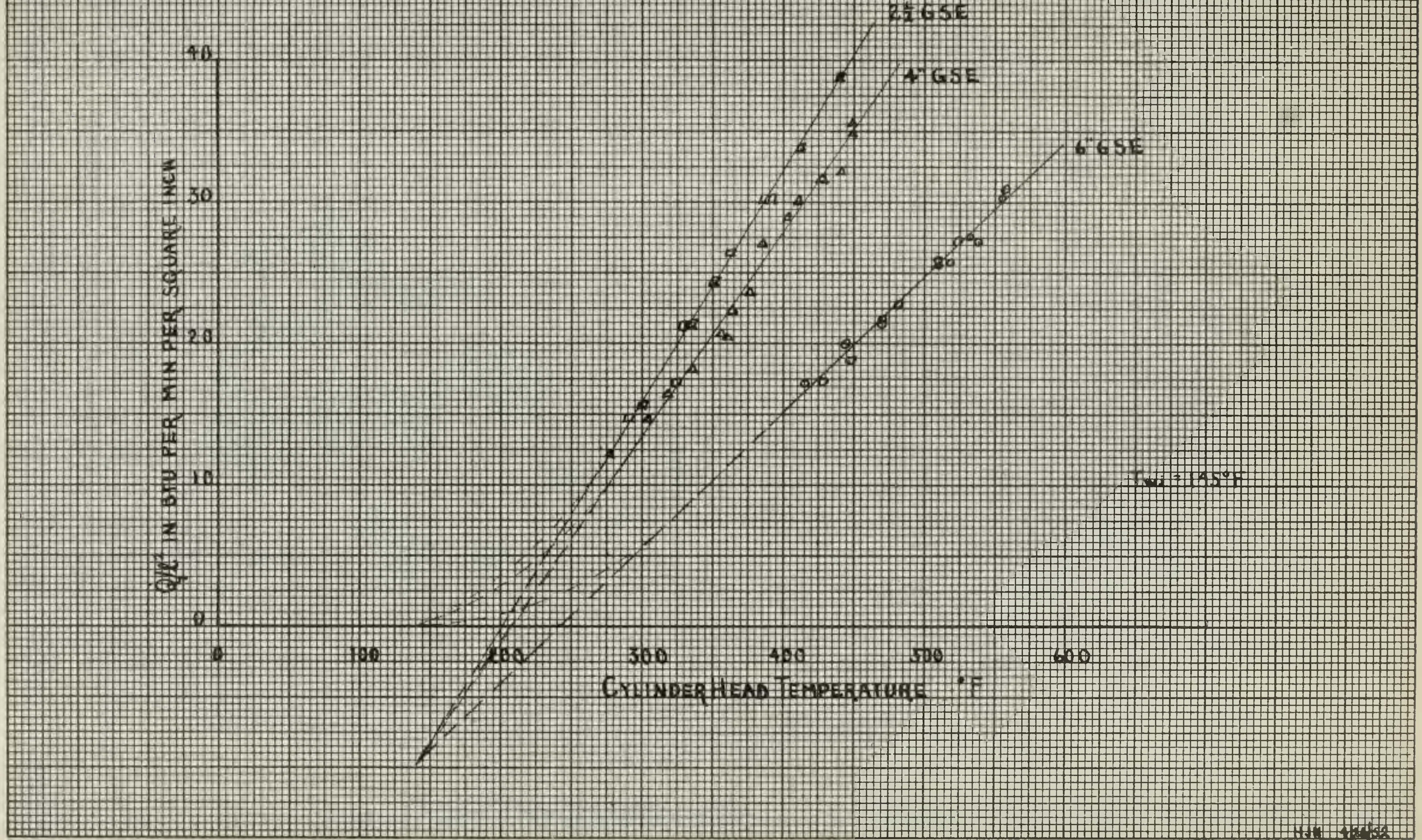
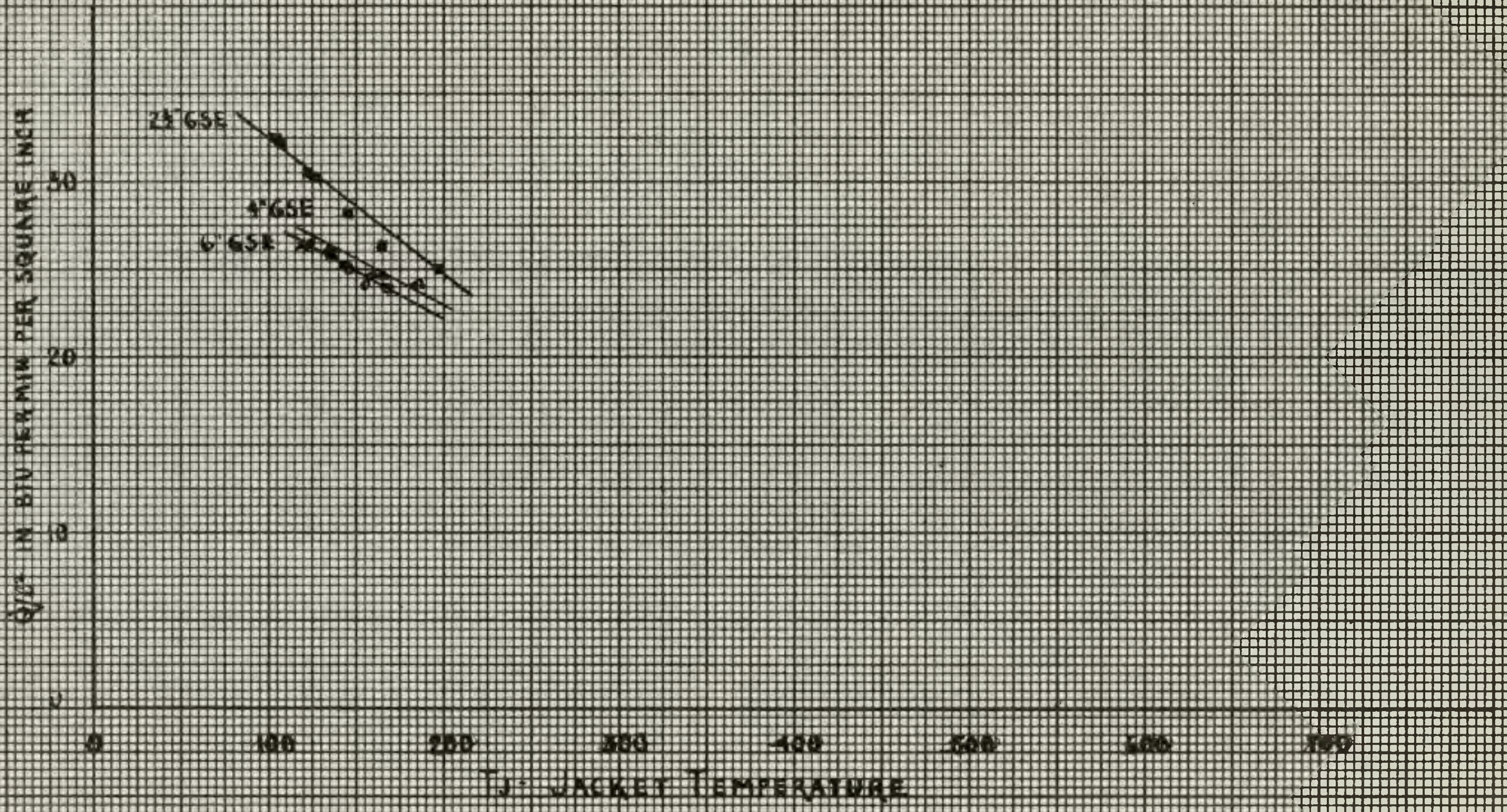


FIGURE XII
HEAT REJECTION VS CYLINDER HEAD TEMPERATURE
WITH CONSTANT WATER JACKET TEMPERATURE

Figure XIII
Heat Rejection Vs Water Jacket Temperature
Q/L^2 vs. TJ For G.S.E.



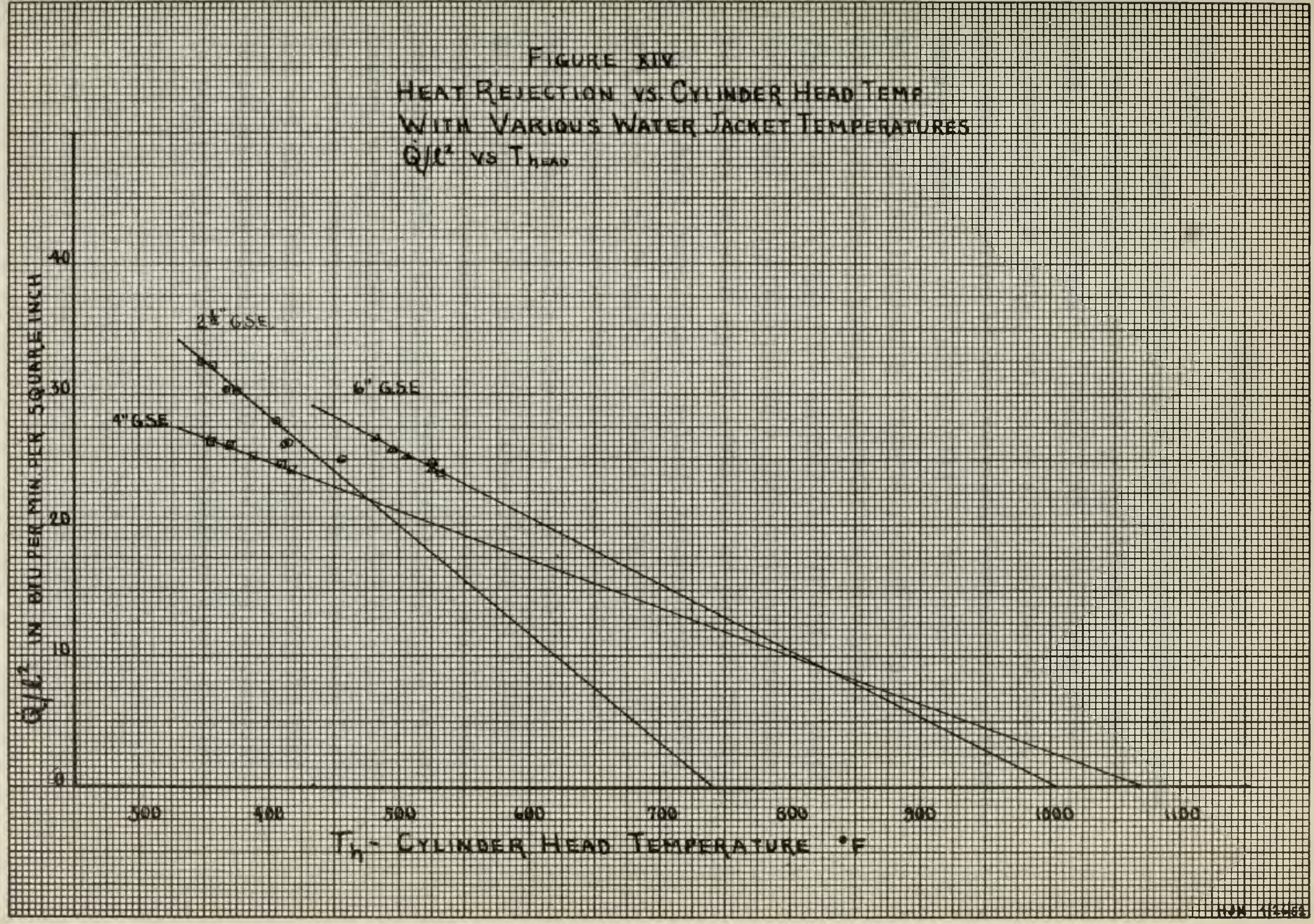


FIGURE XIV
HEAT REJECTION VS. CYLINDER HEAD TEMP
WITH VARIOUS WATER JACKET TEMPERATURES
Q/ℓ^2 VS T_{HEAD}

DISCUSSION

Friction

In order to ascertain IMEP for the heat rejection runs it was decided to measure BMEP by the dynamometer and add FMEP read from curves to obtain IMEP. The curves were constructed from data obtained by motoring the engines, and fair correlation of the individual engines was obtained. However, when the data are plotted for the three engines at constant inlet pressure there is poor correlation of results. Theory states that FMEP due to viscous friction will be equal for similar engines when the ratio of the lubricant viscosity to a characteristic dimension is the same for each engine. FMEP due to coulomb friction is considered proportional to load, represented by IMEP, and should also be equal for similar engines.

Determination of FMEP firing from motoring test data is a questionable procedure. Such factors as lower gas pressure in the cylinder, lower piston and cylinder temperature, cleaner and cooler oil on the cylinder walls and the different pressure differential during the exhaust process make a correlation of motoring and firing friction uncertain. However, the errors introduced by these factors are not all in the same direction and their effects tend to cancel. Therefore, in many cases, it may be justified to take motoring friction as approximately equal to firing friction.

The results of the friction runs are shown in Figures I, II, and III. Figure IV shows a comparison of the FMEP

for each engine at the same conditions. This comparison clearly shows that the friction for the three engines was not in accordance with theory. This fact has previously been reported by Gaboury et al. in reference [3] , and is not entirely unexpected considering the complex nature of friction and the difficulty in maintaining the ratio of oil viscosity to the linear dimension constant.

Heat Rejection

Figure V is a plot of equation (13) for each engine, ℓ a constant. This curve clearly shows the effect of size on heat rejection. As theory predicts, the rate of heat rejection from the smaller engines per unit area is greater than that for the larger engine. One of the variables of this curve is IMEP. There is some doubt as to the accuracy of the measurement of this quantity, because of the inherent inaccuracies present in taking motoring friction as a measure of firing friction.

As a check on the accuracy of the measurement of FMEP, Figures VI and VI-A were plotted. These are plots of input versus output as measured. If, as theory predicts, indicated efficiencies for similar engines are equal at similar conditions, these plots would result in a single line for the three engines. In the actual plot of Figure VI there was a marked difference in output for a given input with change in size. If the three engines do not have equal efficiencies for similar conditions, the deviation from a single straight line could be explained on this basis. The deviation could also be explained if

it were to be determined that the values of IMEP used are actually in error. This latter explanation is probably the true one. Other investigators [3] have reported differences in motoring friction and firing friction. The assumption that motoring friction would be sufficiently accurate was apparently unjustified in this case.

Therefore, rather than attempting to correlate heat rejection for a given output, the measurement of which is doubtful, a plot which shows the relation between heat rejection from the engines and inputs to the engines was made. This plot is shown in Figure VII. At constant thermal efficiency, which may properly be assumed when conditions are similar, output is proportional to input. In this Figure VII, input is measured by the air consumption of the engines. In order to obtain correlation between the various engines the plot shown in Figure VIII was made. This plot which eliminates the uncertainties due to output measurement, provides a valid basis for determining the effect of size on heat rejection.

Figure VIII is similar in form to Figure V, though the curves for the three engines are more closely grouped. For a given engine, ℓ constant, equation (13) may be reduced to

$$\frac{q}{\ell^2} = \text{const.}\left[(\text{IMEP})(S)\right]^n \qquad (15)$$

Theory predicts that the value of n in equation (15)

[illegible]

is the same for similar engines. In Figure VIII, equation (15) plots as a straight line for each engine. The slopes of the curves give values of .688, .651, and .578 for the exponent n for the $2\frac{1}{2}$", 4" and 6" G.S.E. respectively. The difference in values of n may be explained by considering the variation of thermal efficiencies caused by using constant spark advance. A lower thermal efficiency, at constant fuel air ratio, means a higher exhaust temperature. A higher exhaust temperature raises the average ΔT . Since the rate of heat flow is proportional to the temperature difference, a lowering of thermal efficiency would result in a greater heat flow. With a spark advance of 25^{0} used in this study, it is likely that for a given engine the thermal efficiency at low speed would be greater than at high speeds. Considering this, it appears that a higher rate of heat flow was actually present at high speed than would have existed had spark advance been adjusted with speed to maintain constant thermal efficiency. The order of magnitude of the error introduced by neglecting this consideration is believed to be small; however the tendency of compensating for this error would be to rotate each line in a clockwise direction about the lower end. If theory is correct in predicting a greater rate of heat rejection from the smaller engine, the curve for the $2\frac{1}{2}$" engine would be rotated a little more than the curves for the larger engines. This would tend to make the exponents n more nearly equal for the three engines. Therefore, it is concluded that similar

engines do have a constant value for the exponent n in equation (15). This value is approximately .6 for the M.I.T. G.S.E.

Figure XI shows the relation of heat rejection to input on the basis of total rather than heat rejection per unit area as is shown in Figure VII. The orientation of the curves with respect to each other is different from Figure VII because of the difference in total heat transfer surfaces of the three engines. The values of n obtained from these curves are the same as those read from Figures VII and VIII.

Figure VIII may be used as a basis for determining the effect of size with other conditions similar or constant. This may be done by making a cross plot at constant rate of air flow per square inch. The results of such a cross plot are shown in Figure IX. The two curves plotted were taken at different points of constant input. The slope of these curves should be the exponent (n - 1) in equation (14). The fact that the slopes are different may be explained by considering the effects which made the various n's different in Figure VIII. The correction to Figure VIII discussed above would tend to make the slopes of the curves in Figure IX more nearly equal. The actual values of n found from the cross plots vary from .96 to .832. These values are in variance with the value predicted by theory. Theory predicts that the value of n should be the same as those obtained from plotting equation (15).

In order to check the conclusions which may be based on the results shown in Figure IX, Figure X was plotted. Figure X is a plot of actual measured rate of heat rejection from each engine at constant piston speed and inlet pressure versus engine size. Under these conditions, the thermal efficiency of each engine is constant. With constant F, all factors except size in equation (11) are constant for the three engines. The values of n so obtained from curves in Figure X are .90 and .805. The difference between the two values so obtained may be explained by inherent inaccuracies in experimental measurement. These values of n obtained by this method are substantially in agreement with those obtained by the cross plots of Figure IX and both are in variance of the value .6 predicted by theory. This study indicates that the theory is correct in predicting that a large engine will reject less heat per unit area of heat transfer surface than a smaller engine, but that the theory is in error in predicting the order of magnitude of the effect of engine size.

The results of this study are not in complete accordance with theory. As pointed out in the introduction, the basic equation of heat transfer was applied to an internal combustion engine in spite of the formidable theoretical difficulties involved in so doing. If the cyclic operation of an internal combustion engine and the other complications involved do in truth prevent the application of an equation derived for flow in pipes to

[illegible]

[illegible]

an engine, then there is reason to expect results in variance with theory.

Before concluding on the basis of experimental results that the theory is inadequate, the complete applicability of the data must be established. There are sources of error for which the data collected do not account. One of these errors is due to the fact that a portion of the heat rejection measured was not from the working gas but rather from mechanical friction in the cylinder. The heat rejection was not measured during the motoring friction runs except for the 6" engine. Within the limitations of experimental accuracy, data collected for the 6" engine motoring indicated that the heat rejection from the cylinder varied from about 36 BTU/min at lowest speed to 250 BTU/min at highest speed. A portion of this heat was due to the incoming air being at a higher temperature than the jacket water. To separate heat from this source from the heat due to friction is impossible. In any case, the applicability of the motoring data to the firing runs is subject to question. However, if a correction were made for friction heat, the tendency would be to decrease the heat flow at higher speeds. Another source of error is due to some heat from the working gas going to the lubricating oil and some by conduction to the engine foundation. These quantities of heat were not measured. If this correction were applied the result would be opposite in sense to the correction due to friction heat.

Whether the two corrections would cancel each other is not known. For lack of better information, it may be assumed that the results shown by this study are approximately correct as far as these two errors are concerned.

The largest source of heat loss in an internal combustion engine is the heat in the exhaust gases. No attempt was made to measure the temperature of the exhaust gases of the engine nor to account for the additional heat transfer due to the high velocities existing during the exhaust process. The theory however, also neglects these considerations.

The most probable source of error is inherent in the assumption that since the fuel air ratio was constant at all times, the gas temperature was constant. The theory as advanced acknowledges the difference in ΔT due to variation in cylinder wall thickness. Therefore the results should not be corrected for this difference in the various engines. Allowing this approximation, ΔT is dependent on the jacket temperature and the gas temperature. Jacket temperature was accurately controlled and was in fact maintained constant. Fuel-air ratio was accurately controlled within the degree of accuracy of the equipment used, and may be considered constant. However, the fuel supplied was proportional to the inlet air only. The inlet air on the suction stroke, is mixed with the residual gases in the cylinder. The conditions of similarity require that exhaust pressure be

maintained constant, which was accurately done. As inlet pressure is varied, the gases in the cylinder are composed of varying proportions of inlet air and residual gas. At very low inlet pressures, the proportion of inlet air to residual gas is less that at higher inlet pressures. Considering this, it may be concluded that the gas temperature is indeed less at a low inlet pressure than at a high inlet pressure. The effect of this variation of gas temperature on heat transfer would depend on the order of magnitude of the temperature variation. Neither theory nor experimental results account for this variation. However, the fact remains that this condition does exist in any actual engine. Therefore, if the theory is applicable, no correction to experimental results is required.

In view of the above discussion it is considered that the experimental data obtained is applicable and the experimental results may be compared with predicted results. Therefore the conclusion that engine size does not have the magnitude of effect predicted by theory is justified.

Cylinder Head Temperatures

Theory of heat flow states that as the amount of heat flow across a body increases, the temperature drop across the path of flow must increase. Figure XII, a plot of cylinder head temperature versus heat flow per unit area, for each engine, clearly shows the proportionality of heat flow and temperature drop. A size effect is clearly

indicated though the correlation is not clear.

Theoretically the curves should intersect the axis at the jacket temperature, the point at which the temperature difference and consequently the heat flow are zero. The curves do not intersect at 145°F at the axis for two reasons; first, the ordinate is total heat per unit area through the cylinder walls and head rather than through the head alone, and second, the ordinate also contains heat due to piston friction. If the ordinate of the curves are reduced by the amount of heat flow through the walls the curves would be lowered and the slope decreased. If a correction for friction is made, since there is more friction at high speeds, the slopes would be decreased. Therefore, the combined effect of these two corrections would be to lower and rotate the curves clockwise, tending to bring their intersection closer to the predicted point.

Heat rejection at various jacket temperatures

In order to justify the assumption that a constant overall temperature difference existed in the three engines, runs at constant IMEP and piston speed were made while varying average jacket temperature. Theoretically, if the curve of heat rejection versus jacket temperature is extrapolated to zero heat rejection, the jacket temperature at this point will be equal to the mean gas temperature in the cylinder.

The results of these runs are shown in Figures XIII and XIV. While fairly good curves were obtained for

each engine, and a size effect was clearly indicated, the correlation between engines was obscure. When extrapolation is attempted in Figure XIII, no common point of intersection is found. Figure XIV, in which cylinder head temperature is plotted against heat rejection gives no better results. This curve would not be expected to give pertinent information unless the ordinate were heat rejection through the head alone rather than total heat rejection.

The difficulty in obtaining a common point of intersection in Figure XIII may be due to the fact that either extrapolation is not justified, because of the narrow range of temperatures which could be used, or the data on which the curves are based are unreliable. At the higher jacket temperature runs, boiling of the cooling water in the jacket was detected. Accurate water flow measurement under such conditions was difficult.

Considering the above, conclusions based on information in Figure XIII are not justified. If the range of possible jacket temperatures were extended by the use of a cooling fluid with a higher boiling point, more reliable data could be obtained.

CONCLUSIONS

1. Heat transfer theory, as applied to internal combustion engines, in the form of equation (11), may be used to approximate heat rejection from the cylinder gases.
2. Similar engines, considered independently, act as theory predicts. The value of n in equation (15) is approximately .6 for the M.I.T. G.S.E.
3. Large engines reject less heat per unit area than small engines.
4. The effect of engine size on heat rejection per unit area is less than predicted by theory. The value of n in equation (14) is approximately .9 for the M.I.T. G.S.E.
5. Motoring friction may not be taken as a measure of firing friction for the M.I.T. G.S.E.

RECOMMENDATIONS

In order to augment the information resulting from this study and to provide a basis for future studies using the M.I.T. G.S.E., it is recommended that:

1. An attempt be made to correlate heat rejection data obtained in this study with IMEP measured by indicator cards rather than by motoring friction and BMEP.
2. Heat rejection due to friction be measured during motoring.
3. A cooling circuit be engineered whereby heat rejection to the oil in the crankcase can be measured.
4. An attempt be made to measure separately the heat rejection through the cylinder walls and through the cylinder head.
5. The effect of the use of varying spark advance with load and speed to obtain constant efficiency, as opposed to the use of constant spark advance to maintain similar areas exposed to the burning gases, be studied.
6. Heat rejected in the exhaust gas be measured.
7. Additional thermocouples be installed in the cylinder heads and walls in order to better approximate the temperature drop across them.

8. The range of possible jacket temperatures be extended by the use of a coolant with a high boiling point.
9. The validity of selecting lubricating oils with a single temperature used as a basis for selecting equal $\frac{\mu}{\ell}$ ratios be investigated and compared with the feasibility of having a series of lubricating oils with equal ratios at several different temperatures or using a lubricant, perhaps one of the silicones, in which temperature has less effect on viscosity.
10. The use of contact oil seals in these engines be discontinued in order to attempt to eliminate uncertain performance in friction forces. It is suggested that a non-contact labyrinth type seal be investigated as a possible replacement.
11. Future studies be made, with the heat rejection cooling system installed by the authors, in order to supplement the experimental data in this study relating heat rejection and engine size.

APPENDIX

APPENDIX A

DESCRIPTION OF ENGINES & ASSOCIATED EQUIPMENT

[illegible]

[illegible]

[illegible]
[illegible]
[illegible]
[illegible]
[illegible]
[illegible]
[illegible]
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DESCRIPTION OF ENGINES & ASSOCIATED EQUIPMENT

The three geometrically similar engines used in this study were designed and built under the supervision of the Mechanical Engineering Department of The Massachusetts Institute of Technology. They were built to provide a means of study of the effect of size on engine characteristics; to verify a rapidly growing theory of similar engines, and to evolve a more rational approach to engine design.

Geometrically similar engines are defined as engines of different sizes, whose corresponding dimensions bear the same ratio as some characteristic dimension and whose corresponding parts are constructed of the same material. In the case of the M.I.T. engines all linear dimensions are in ratio of the bore diameters, which is the dimension used to identify the individual engines. See Table A-I for detail dimensions. Great care was taken in design to preserve geometric similarity even down to screw thread sizes. C. F. Taylor in Reference (2) describes these engines in considerable detail. It is believed that these are the only completely geometrically similar research engines in existence.

The engines under study, Figures A-I, A-II, and A-III are single cylinder, four stroke cycle, spark ignited internal combustion engines. They are mounted on spring supported bed plates located in individual test cells and are equipped with suitable operating controls and

[illegible]

instrumentation.

Associated equipment includes a double shell vaporizing tank for mixing fuel and air at any desired temperature; a water jacketed exhaust tank with a valve for controlling exhaust pressure; a circulating oil pump provides high velocity circulation through the crank-case and oil heater-cooler, a second pressure pump is installed to service the main bearings and cylinder. The four and six inch engines are provided with remote control electrically operated throttle valves, and the two and a half inch engine with a manually operated valve.

Brake measurements are made by rheostat controlled dynamometers equipped with hydraulic scales. Figures A-IV and A-V show schematically the engine and dynamometer set up and the hydraulic scale installation respectively. See also Figures A-VII, A-VIII, and A-IX. The 4" engine is provided with a conventional direct current dynamometer while the $2\frac{1}{2}$" and the 6" engines are equipped with alternating current dynamometers with magnetic speed control clutches.

The cooling water circuit redesigned and installed by the authors was suggested by Professor Rogowski. It is shown schematically in Figure A-VI and photographically in Figures A-VII, A-VIII and A-IX. This redesigned cooling system provides a large flow, high velocity, small temperature rise main circuit through the cylinder, and a small flow, large temperature drop secondary parallel circuit through the rotameter and heat exchanger. This system ensures that jacket surfaces are adequately cooled and

scrubbed free of gas bubbles and also provides a large temperature difference secondary circuit. The heat rejected is measured in the secondary circuit by accurately measuring the flow through the rotameter and the temperature difference between the water to the rotameter and the water from the heat exchanger. (See Figure A-V). This large temperature difference, of the order of fifty degrees, and an accurately measured nominally large flow provides a reasonably accurate method of determining the heat rejected. This system eliminates the uncertainties involved in measuring the heat rejected to a stream having a small temperature rise as was the case with the original cooling system. This rise, eight to ten degrees, through the cylinder was considered to yield questionable data and therefore the new circuit was designed and installed.

TABLE A-I

DESCRIPTION OF ENGINES

Engine	6"	4"	$2\frac{1}{2}$"
Bore (in)	6.0	4.0	2.5
Stroke (in)	7.2	4.8	3.0
Piston area (in^2)	28.27	12.57	4.91
V_d (in^3)	203.5	60.35	14.71
Compression ratio	5.74	5.74	5.74
Inlet valve clearance cold (in)	.012	.008	.005
Exh. valve clear. cold (in)	.015	.012	.006
Piston speed per rpm (ft/min)	1.2	.8	.5
Spark plug, Champion	J7-18mm	J8	Y-4A
Valve overlap (deg.)	30	30	30
Dynamometer			
Scale piston diam. (in)	2.795	1.614	.932
Dyn. torque arm (in)	21.008	15.765	12.605
Overall dyn. const. K	1000	4000	15000
Scale force for 1" Hg. (lb)	3	1	.333
Bmep	3.89h	3.28h	3.59h

$$BMEP = \frac{792{,}000\ h}{K\ \ V}$$

$$BHP = \frac{N\ h}{K}$$

Figure A-I
M.I.T. Geometrically Similar Engines Disassembled

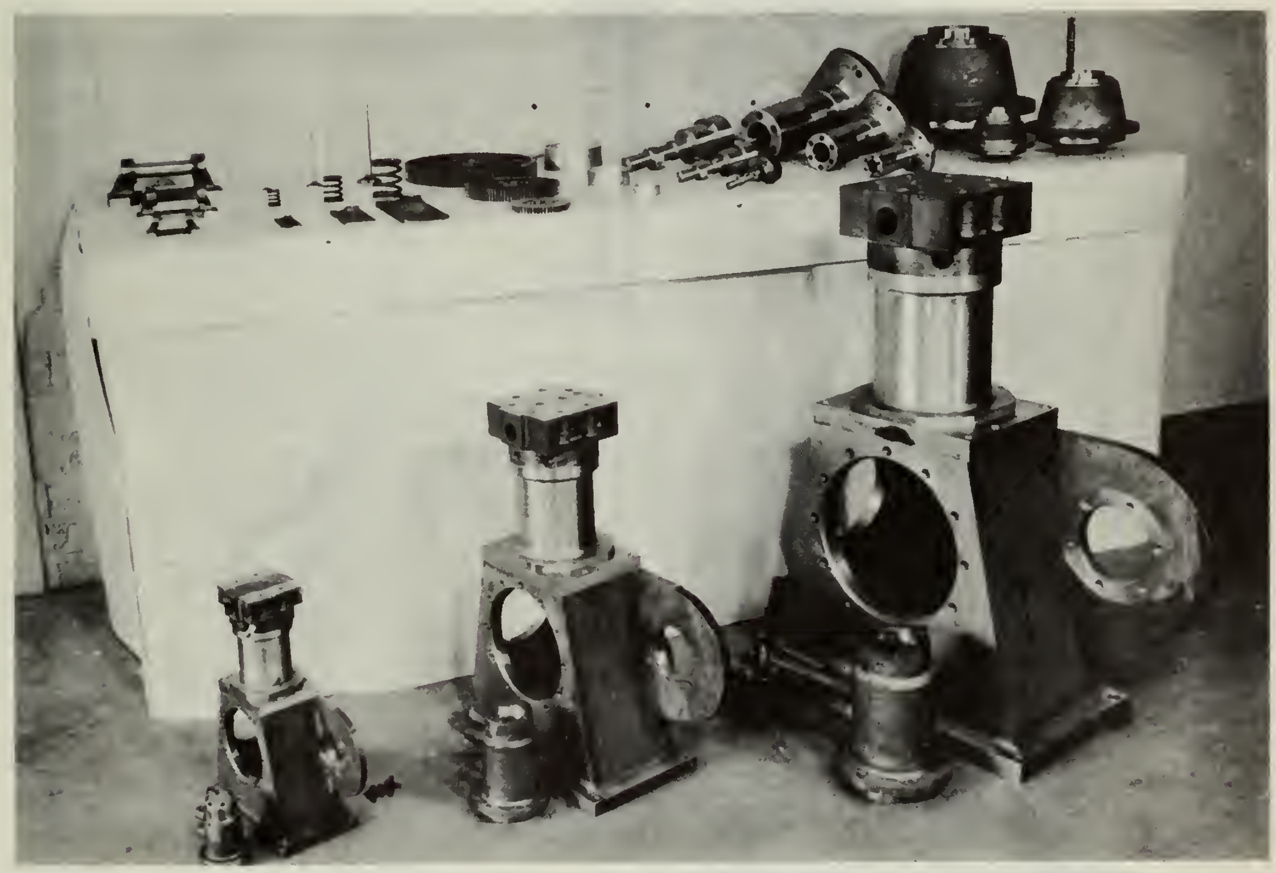


FIGURE A-II

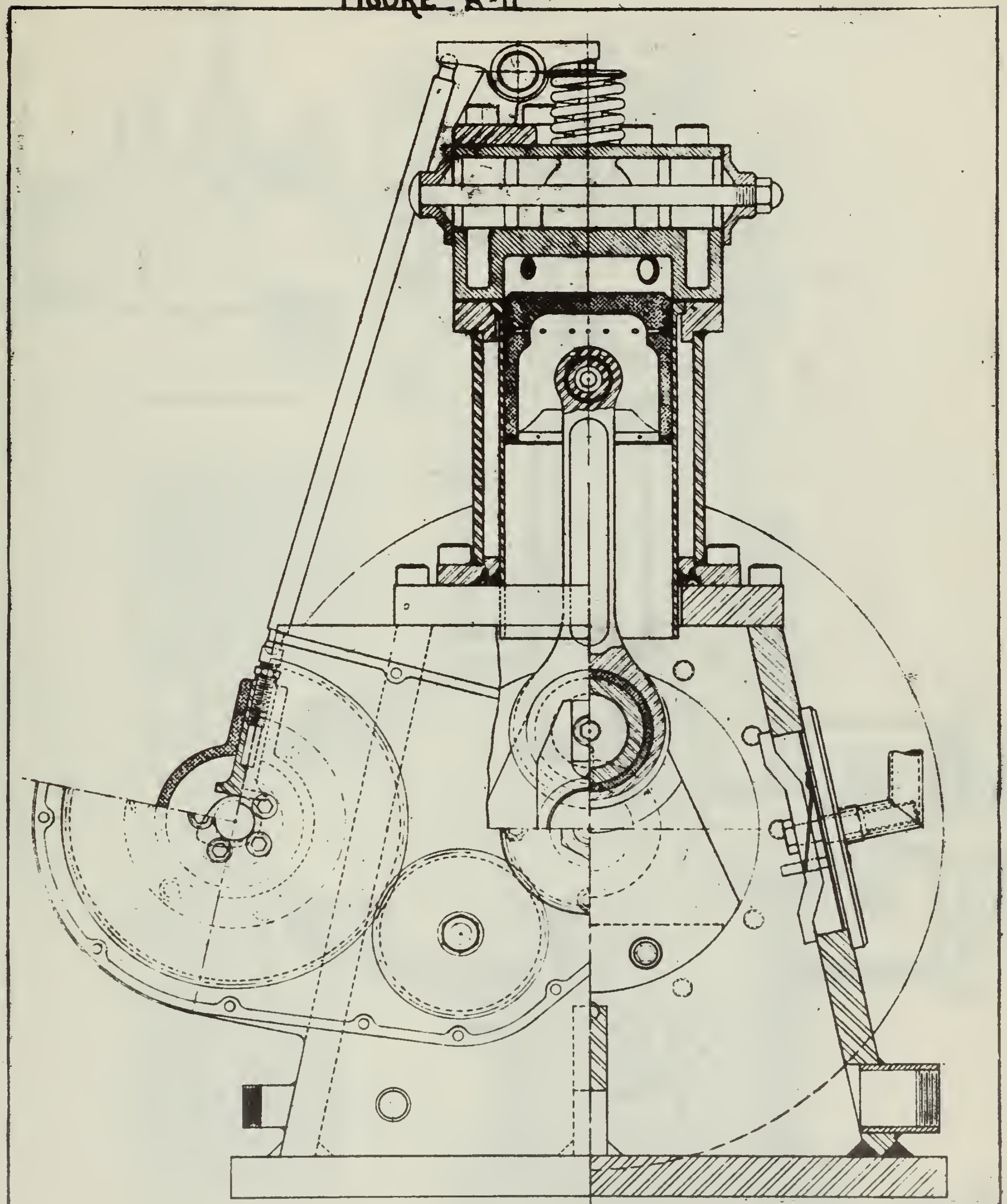

INTERNAL COMBUSTION LABORATORY
Viewed from Timing-Gear End
MATERIAL
036-700A

FIGURE A III

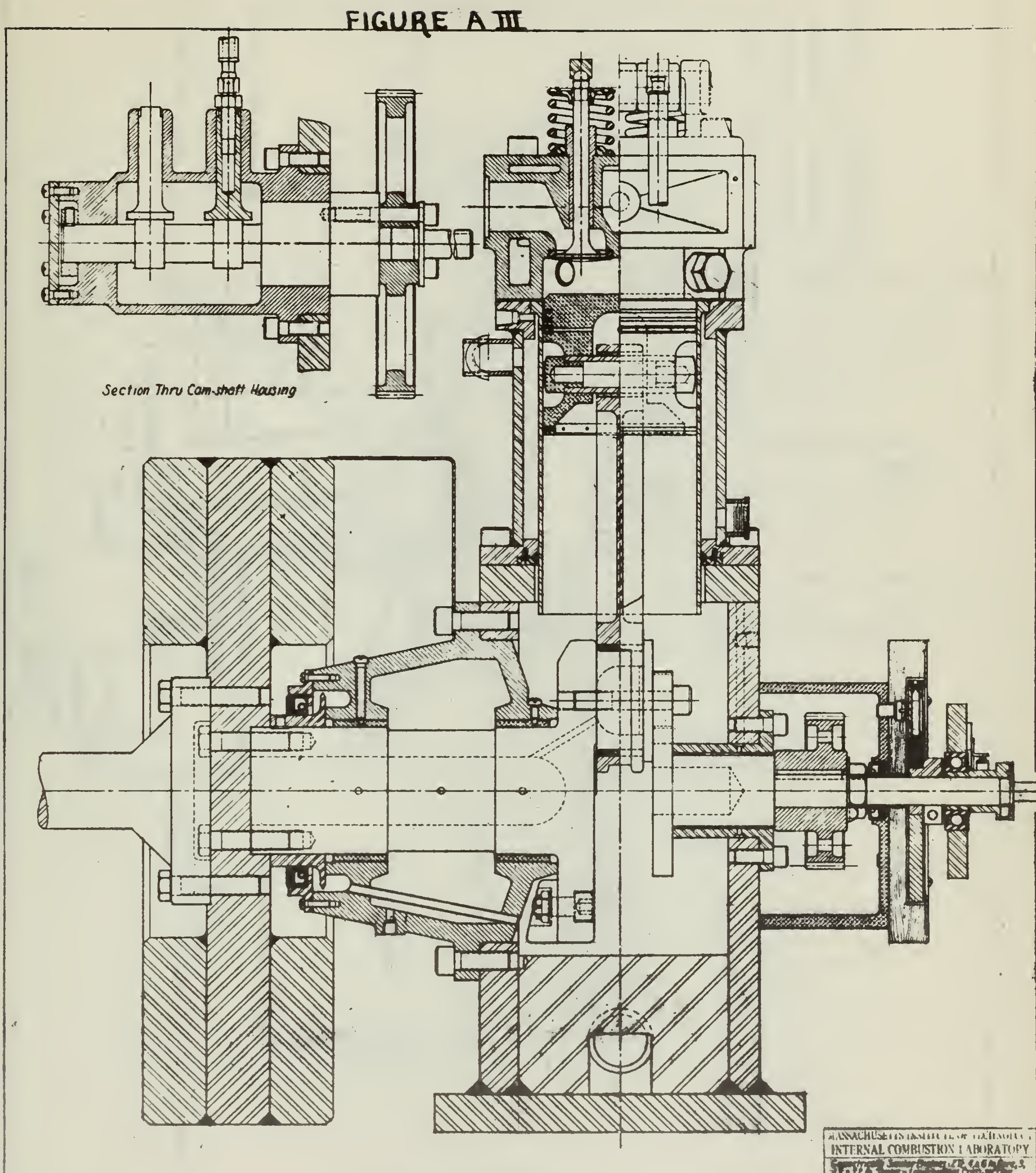
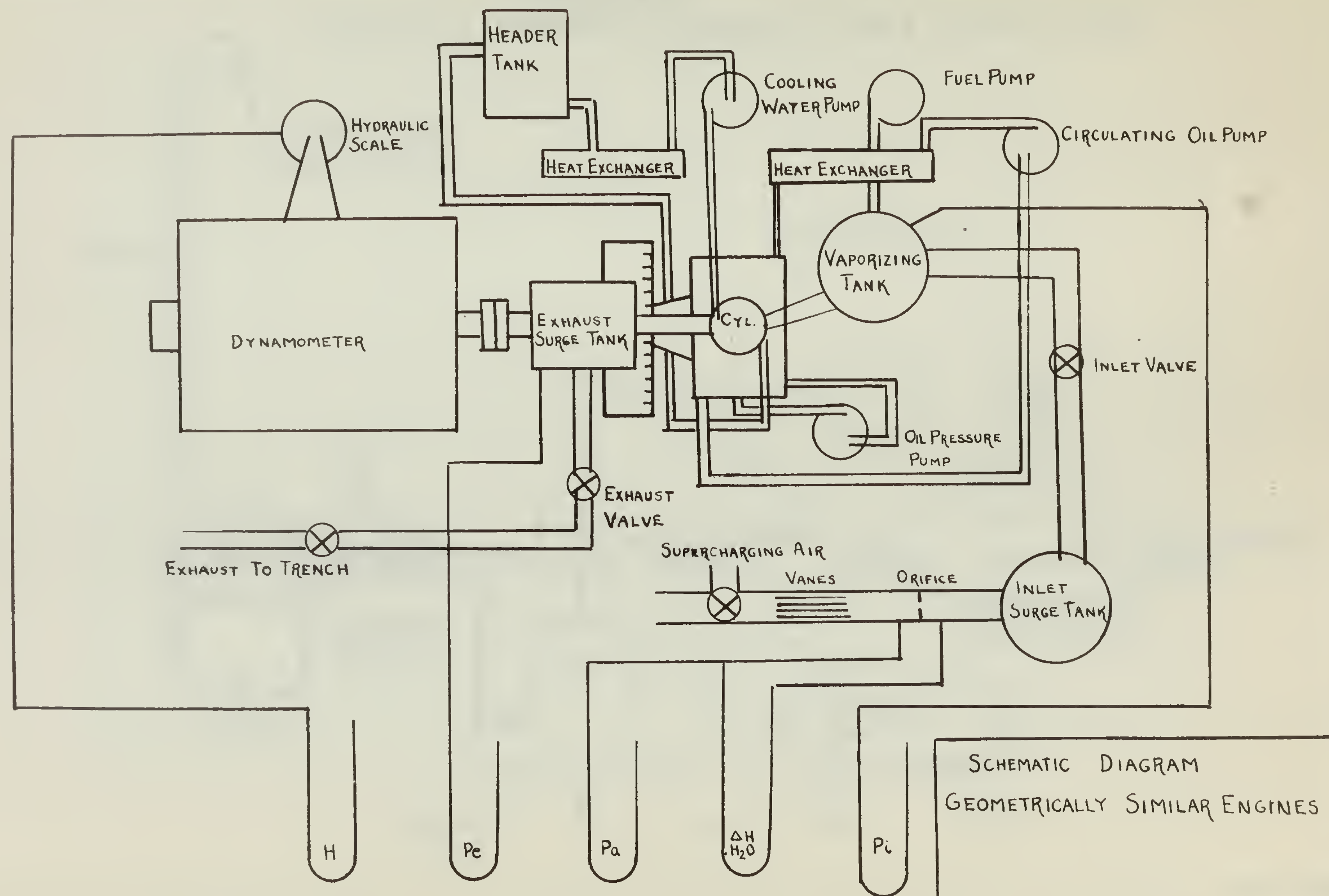

Section Thru Cam-shaft Housing
INTERNAL COMBUSTION LABORATORY

FIGURE A-IV



SCHEMATIC DIAGRAM
GEOMETRICALLY SIMILAR ENGINES

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FIGURE A V

SCHEMATIC DIAGRAM OF HYDRAULIC SCALE INSTALLATION

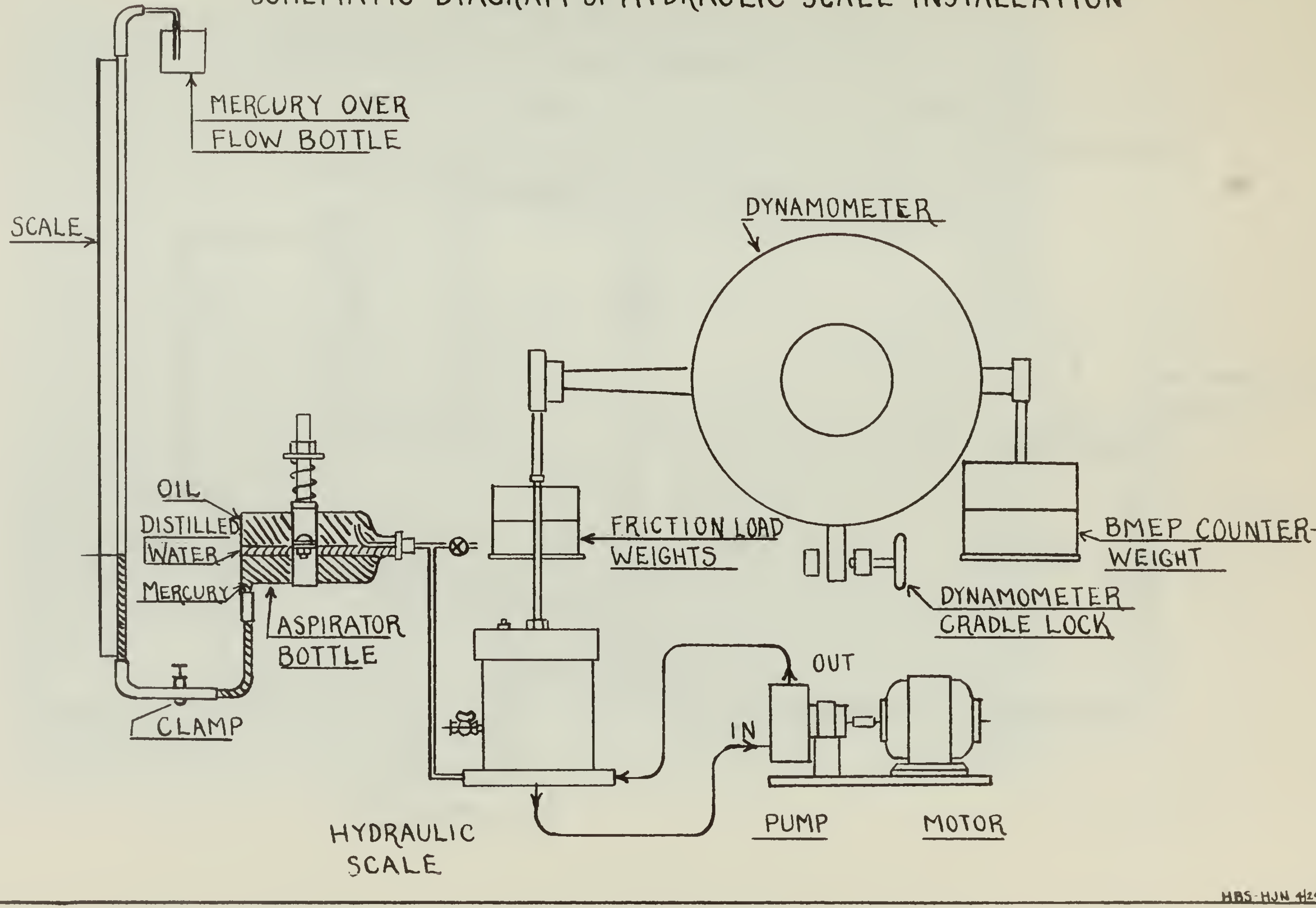


FIGURE A-VI

SCHEMATIC DIAGRAM - HEAT REJECTION CIRCUIT
GEOMETRICALLY SIMILAR ENGINES

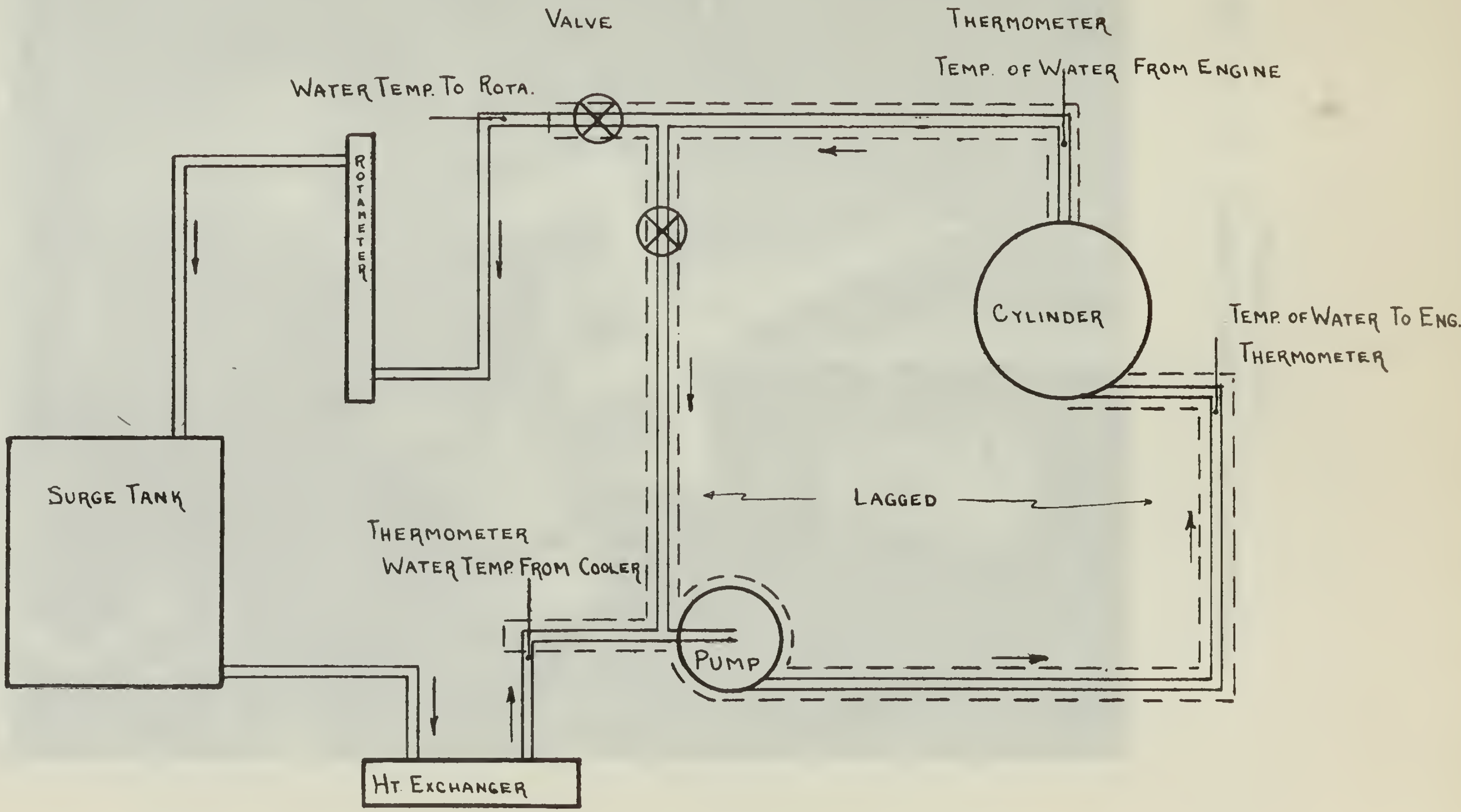


Figure A-VII
$2\frac{1}{2}$" M.I.T. Geometrically Similar Engine

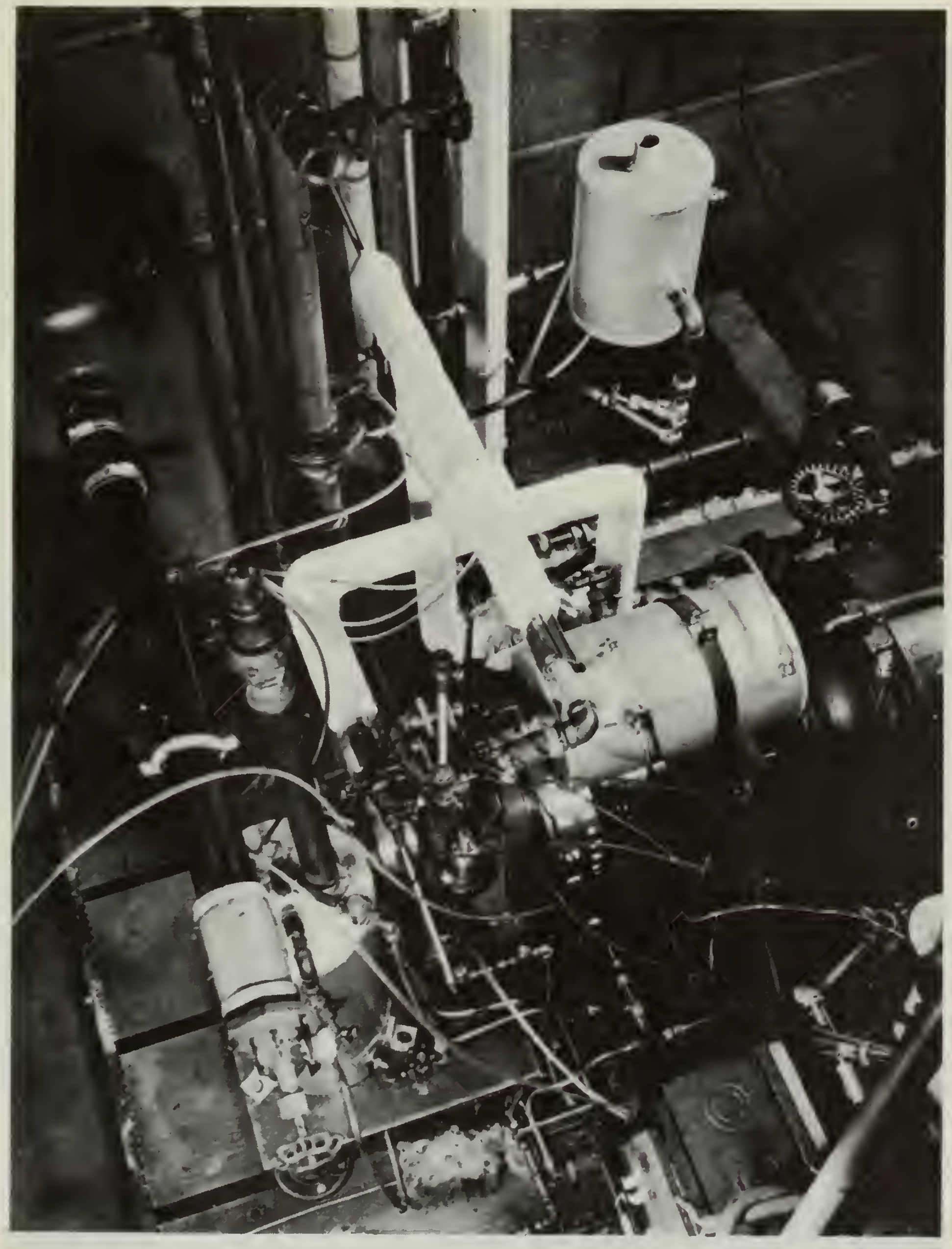


Figure A-VIII
4" M.I.T. Geometrically Similar Engine

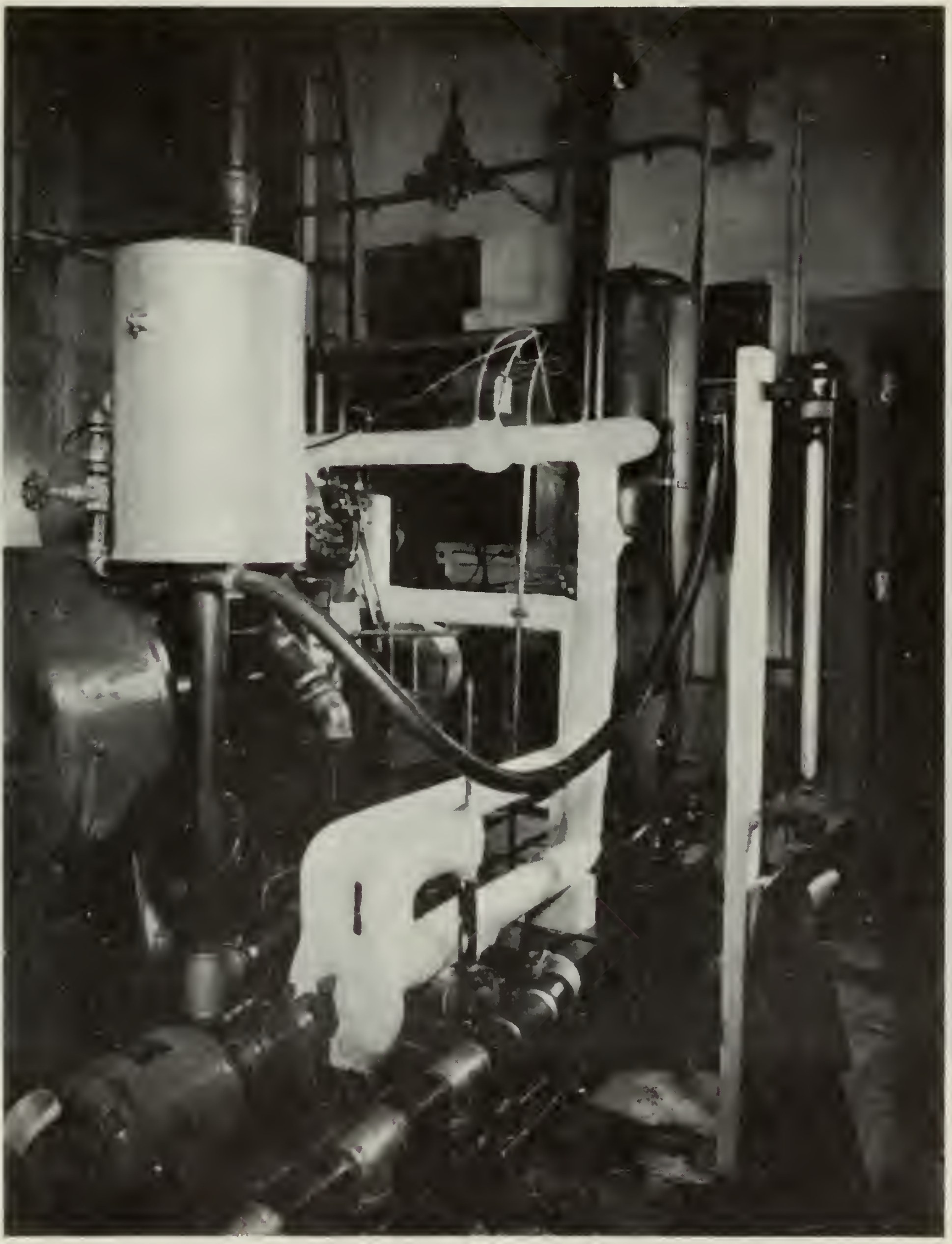
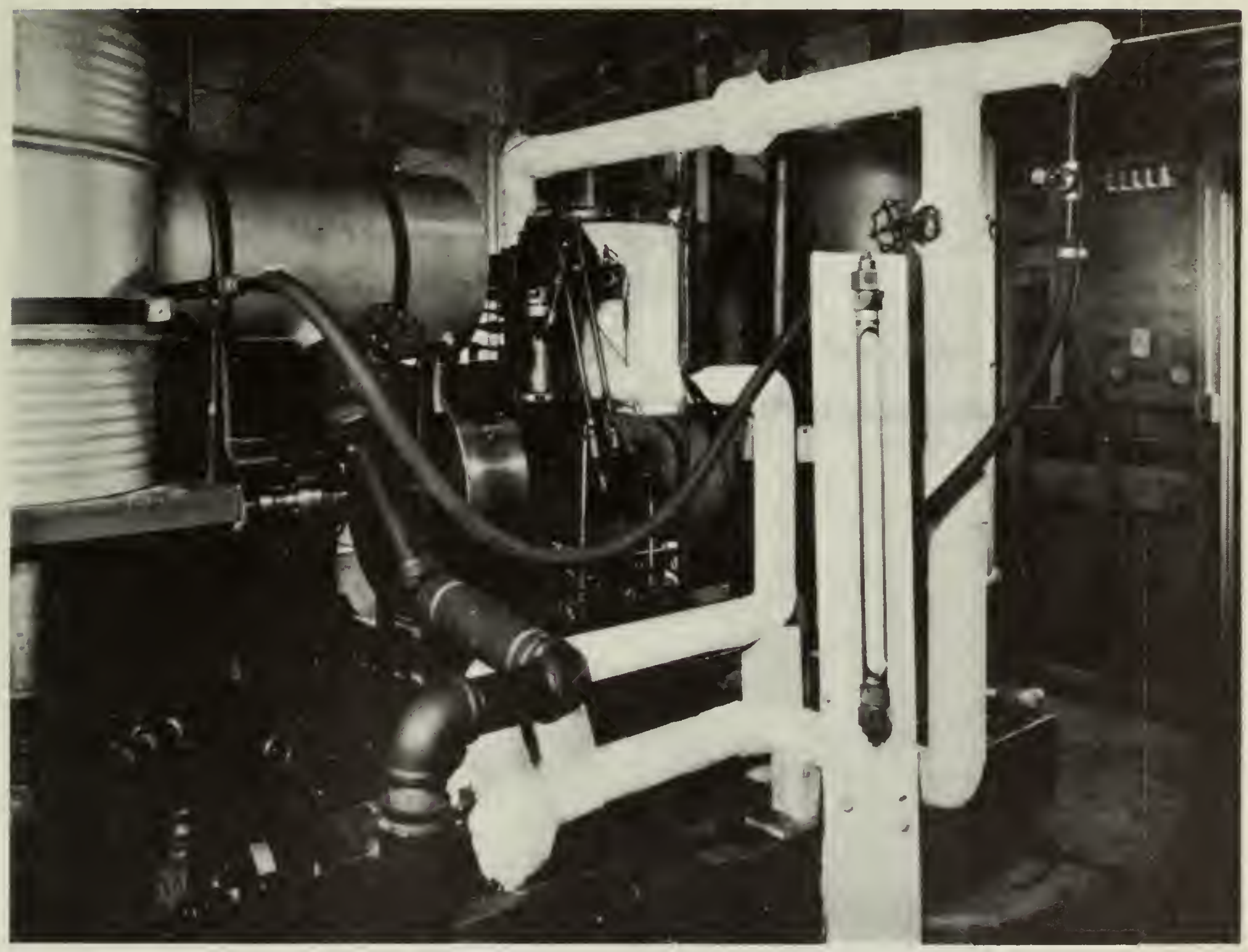


Figure A-IX
6" M.I.T. Geometrically Similar Engine



APPENDIX B

WATER FLOW MEASUREMENT

APPENDIX B

WATER FLOW MEASUREMENT

Water circulating through the engines was maintained at equal velocities and measured by means of A.S.M.E. standard water orifices. Water to and from the cooler and surge tank was measured by Fischer and Porter Rotameters. Calibration data was taken at 150°F. which was the average temperature through the rotameter during runs. Flow was measured by electrical timer scale for the lower rates and by beam balance and stop watch for higher rates. The floats used had an average zero suppression of fifty percent of maximum flow.

Repeat calibration data was taken at other temperatures to find the effect of variation in viscosity and density. When data taken at 72°F. was plotted it showed a regular plus error of about three percent compared to the 150° calibration curve.

Calibration data was also taken at higher temperatures but in the range from 150° to 200° it was found that the combined effects of viscosity, specific gravity, and cavitation gave erratic results, although the error in the temperature range in which the runs were made at no time was in excess of four percent. It is suggested that this condition might have been improved by maintaining the rotameter under pressure by throttling at the surge tank. Water Orifice and Water Rotameter calibration curves are appended as Figures B-I and B-II respectively.

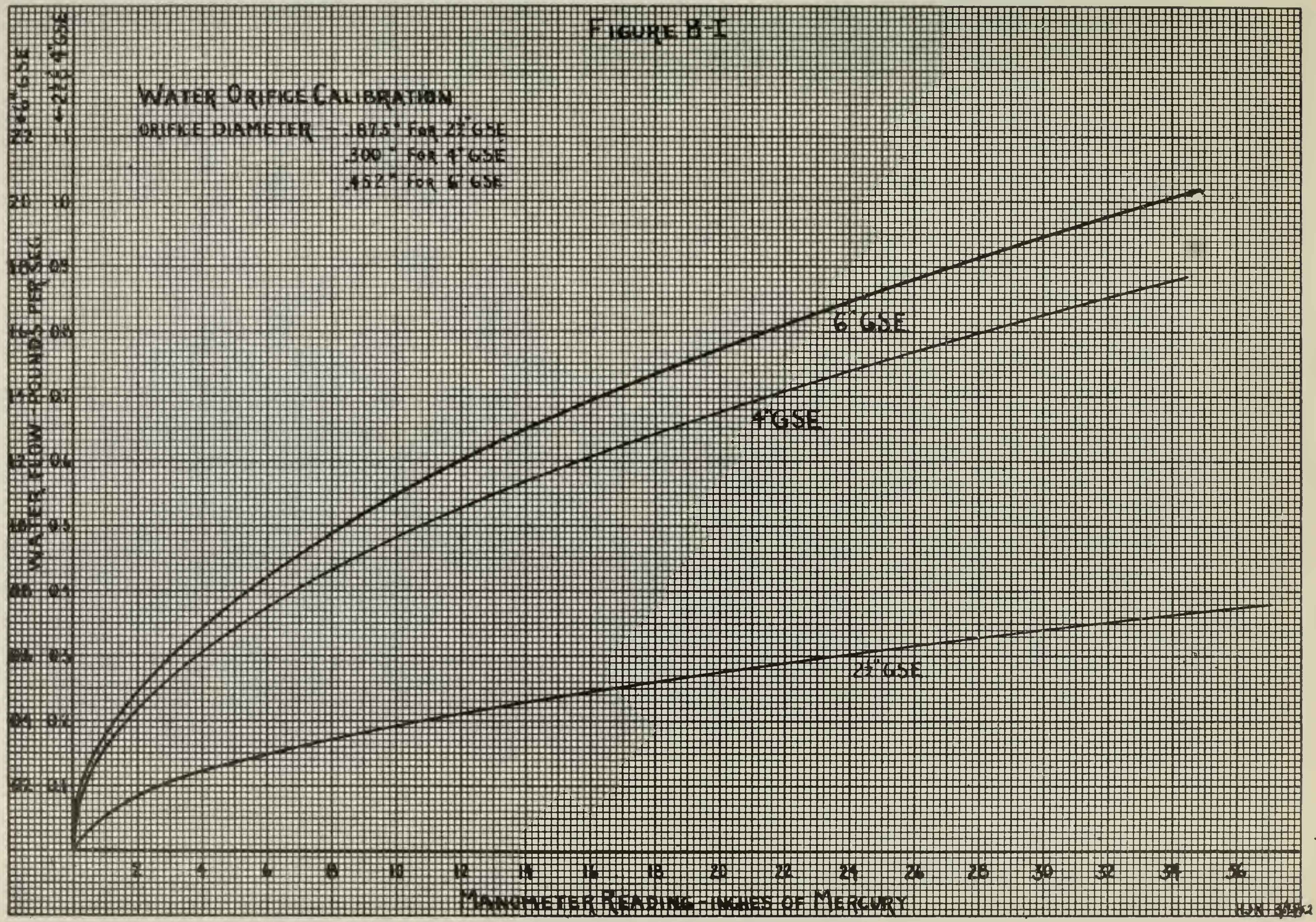


FIGURE B-I
WATER ORIFICE CALIBRATION
ORIFICE DIAMETER - .1875" FOR 2½" GSE
.300" FOR 4" GSE
.452" FOR 6" GSE
6" GSE
4" GSE
2½" GSE
WATER FLOW - POUNDS PER SEC.
MANOMETER READING - INCHES OF MERCURY

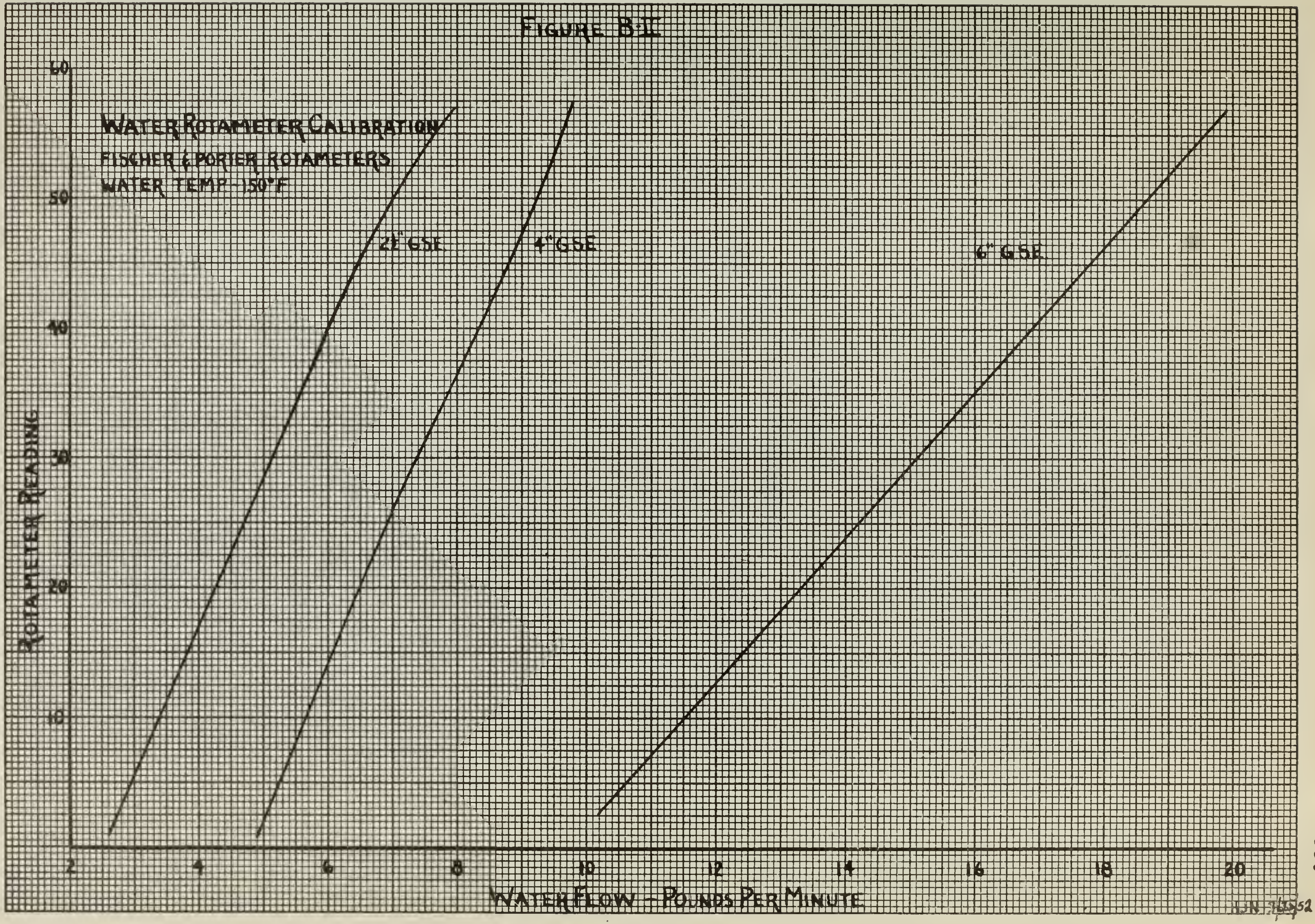

WATER ROTAMETER CALIBRATION
FISCHER & PORTER ROTAMETERS
WATER TEMP. 150°F
2½" GSE
4" GSE
6" GSE
ROTAMETER READING
WATER FLOW – POUNDS PER MINUTE
0
10
20
30
40
50
60
2
4
6
8
10
12
14
16
18
20

FIGURE B II

APPENDIX C

AIR FLOW MEASUREMENT

APPENDIX C

AIR FLOW MEASUREMENT

Air flow to all engines was measured by ASME square edged orifices with flange taps located one inch from the upstream and downstream faces of the orifice. The equation for flow through this type orifice is

$$W = .1145\, D_2^2\, K\, Y \sqrt{\frac{P_1}{T_1}\, G\, Y\, \Delta P} \qquad \text{Reference (7)}$$

where

W = flow rate, lb. mass/second.

D_2 = orifice diameter, inches.

K = flow coefficient, dimensionless, Table 6, Reference (7).

Y = expansion factor, dimensionless, Table 37, Reference (7).

P_1 = static pressure before orifice, inches Hg. abs.

T_1 = temperature before orifice, °F absolute.

G = specific gravity of gas (1.00 for air).

y = super compressibility factor, dimensionless, Figure 11, Reference (7).

ΔP = pressure drop across orifice, inches H_2O.

Flow coefficient K

This coefficient combines the discharge coefficient $C = \frac{\text{actual mass rate of flow}}{\text{theoretical mass rate of flow}}$ and the velocity of approach factor $\frac{1}{\sqrt{1-\left(\frac{D_2}{D_1}\right)^4}}$ where D_1 is the diameter of the pipe.

Expansion factor, Y .

This factor takes into account uncontrolled expansion of the gas after the orifice due to reduced pressure in that region. A table values of Y has been

[illegible]

determined experimentally, Reference (7), and found to fit the following empirical formula.

$$Y = 1 - \left[0.41 \quad 0.35 \left(\frac{D_2}{D_1}\right)^4 \left(\frac{\Delta P}{P_1} \cdot \frac{1}{K}\right)\right]$$

where $K = \frac{C_p}{C_v}$.

<u>Pressure drop across orifice, ΔP .</u>

ΔP was measured by water manometers. No readings were taken at less than 3" of water. For the lower air flows orifice plates were replaced with plates of smaller diameter.

<u>Super-compressibility factor y .</u>

This factor corrects for departure from perfect gas conditions

$$y = \frac{\text{actual density}}{\text{theoretical density}}$$

<u>Correction curve.</u>

A flow curve was plotted for standard conditions and mean Reynold's number. This curve was then corrected for the particular Reynold's number at each flow rate.

<u>Reynold's number.</u>

$$Re = \frac{\rho u D_2}{12\mu}$$

ρ - fluid density before orifice, $\frac{\#\ \text{mass}}{\text{ft}^3}$

u - velocity before orifice, ft/sec

μ - fluid viscosity before orifice, $\frac{\#\ \text{mass}}{\text{ft/sec}}$

<u>Precision.</u>

For flows measured, accuracy is considered to be within $\pm 1.5\%$. Calculations were made for average conditions in the engine cells. Errors due to departures from temperature, pressure, and humidity in the laboratory were not considered to be significant.

[illegible]

Results.

Curves of air flow vs. manometer reading are shown, for the various engines, in Figures C-I, C-II, and C-III.

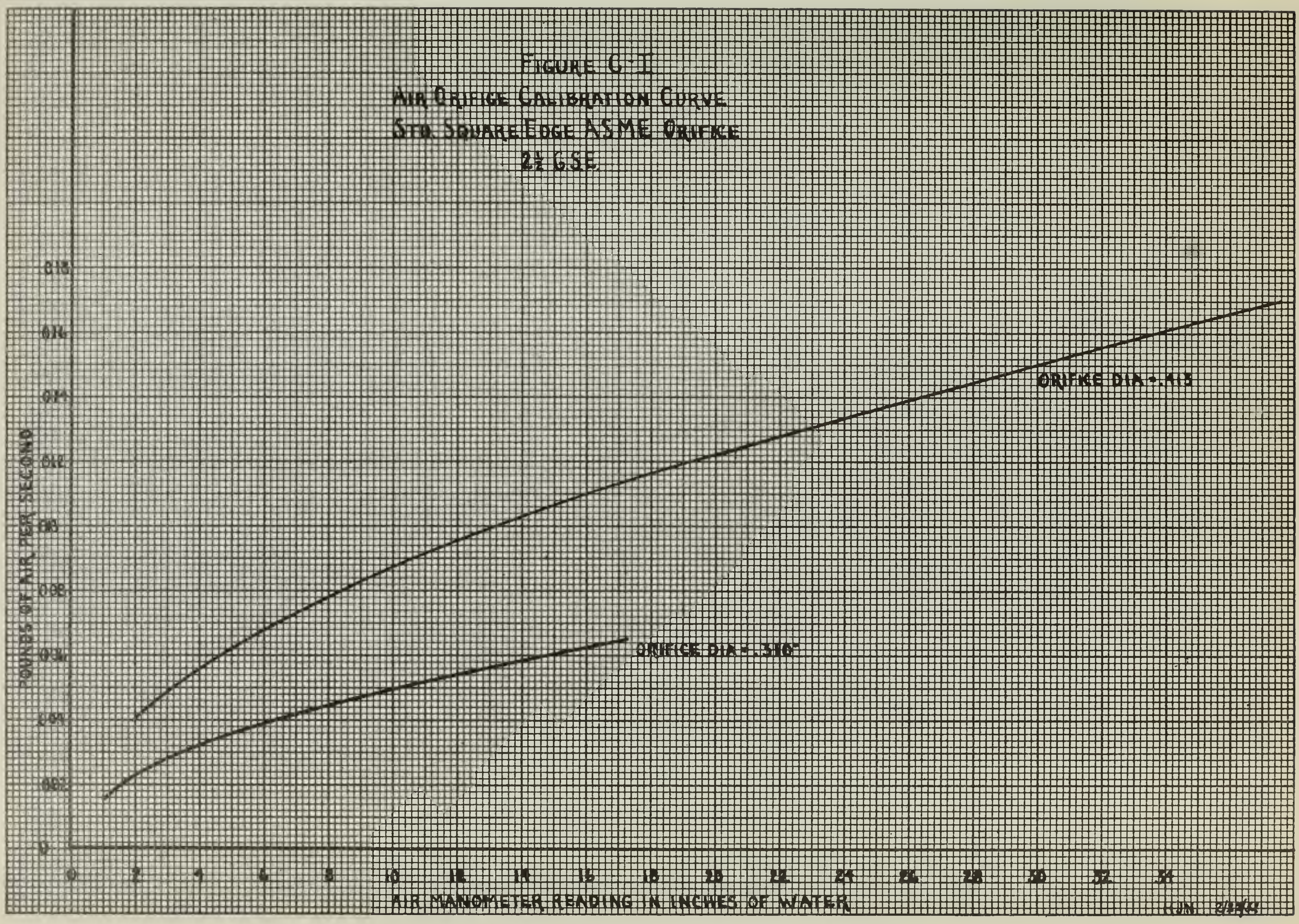


FIGURE C-I
AIR ORIFICE CALIBRATION CURVE
STD. SQUARE EDGE ASME ORIFICE
2½ GSE
ORIFICE DIA = .413
ORIFICE DIA = .310"
POUNDS OF AIR PER SECOND
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
AIR MANOMETER READING IN INCHES OF WATER

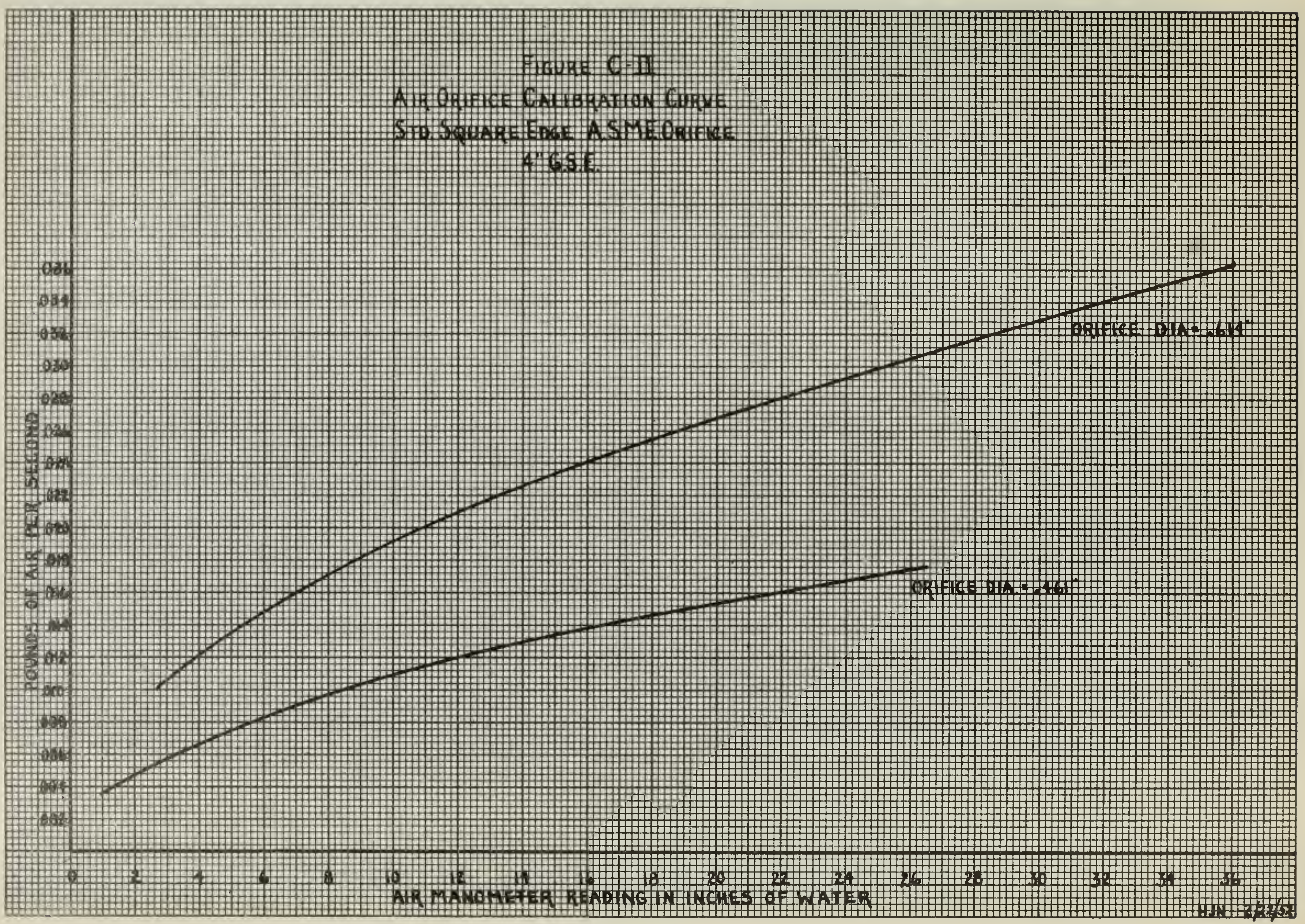


FIGURE C-II
AIR ORIFICE CALIBRATION CURVE
STD. SQUARE EDGE A.S.M.E. ORIFICE
4" G.S.E.
ORIFICE DIA = .614"
ORIFICE DIA = .461"
POUNDS OF AIR PER SECOND
.036
.034
.032
.030
.028
.012
.010
.008
.006
.004
.002
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36
AIR MANOMETER READING IN INCHES OF WATER
HJN

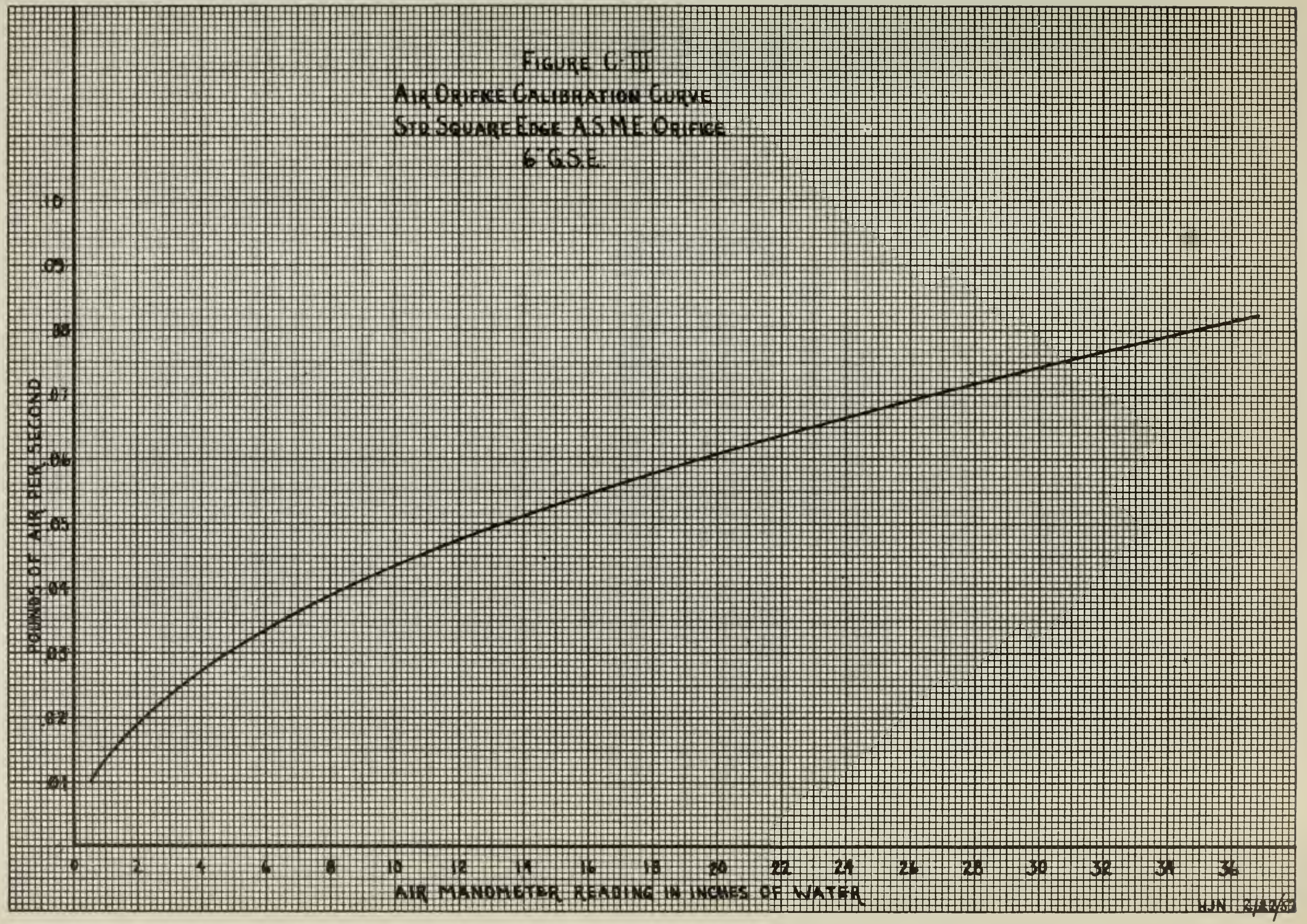


FIGURE C-III
AIR ORIFICE CALIBRATION CURVE
STD SQUARE EDGE A.S.M.E. ORIFICE
6" G.S.E.
POUNDS OF AIR PER SECOND
.01
.02
.03
.04
.05
.06
.07
.08
.09
.10
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36
AIR MANOMETER READING IN INCHES OF WATER
HJN

APPENDIX D

FUEL FLOW MEASUREMENT

APPENDIX D

FUEL FLOW MEASUREMENT

Fuel was measured with Fischer and Porter Rotameters. The Rotameters were calibrated with fuel at room temperature, and curves of fuel flow versus Rotameter reading plotted. From the air calibration curve and the fuel calibration curve cross curves of pressure drop across air orifice in inches of water versus Rotameter reading for constant fuel air ratio of .078 were plotted. Errors due to departure from pressure and temperature conditions for which curves were plotted were found to be insignificant. These curves permitted quick and accurate adjustment of fuel air ratio. Fuel Rotameter curves are shown in Figure D-I, and cross curves for constant fuel air ratio are shown in Figures D-II, D-III, and D-IV.

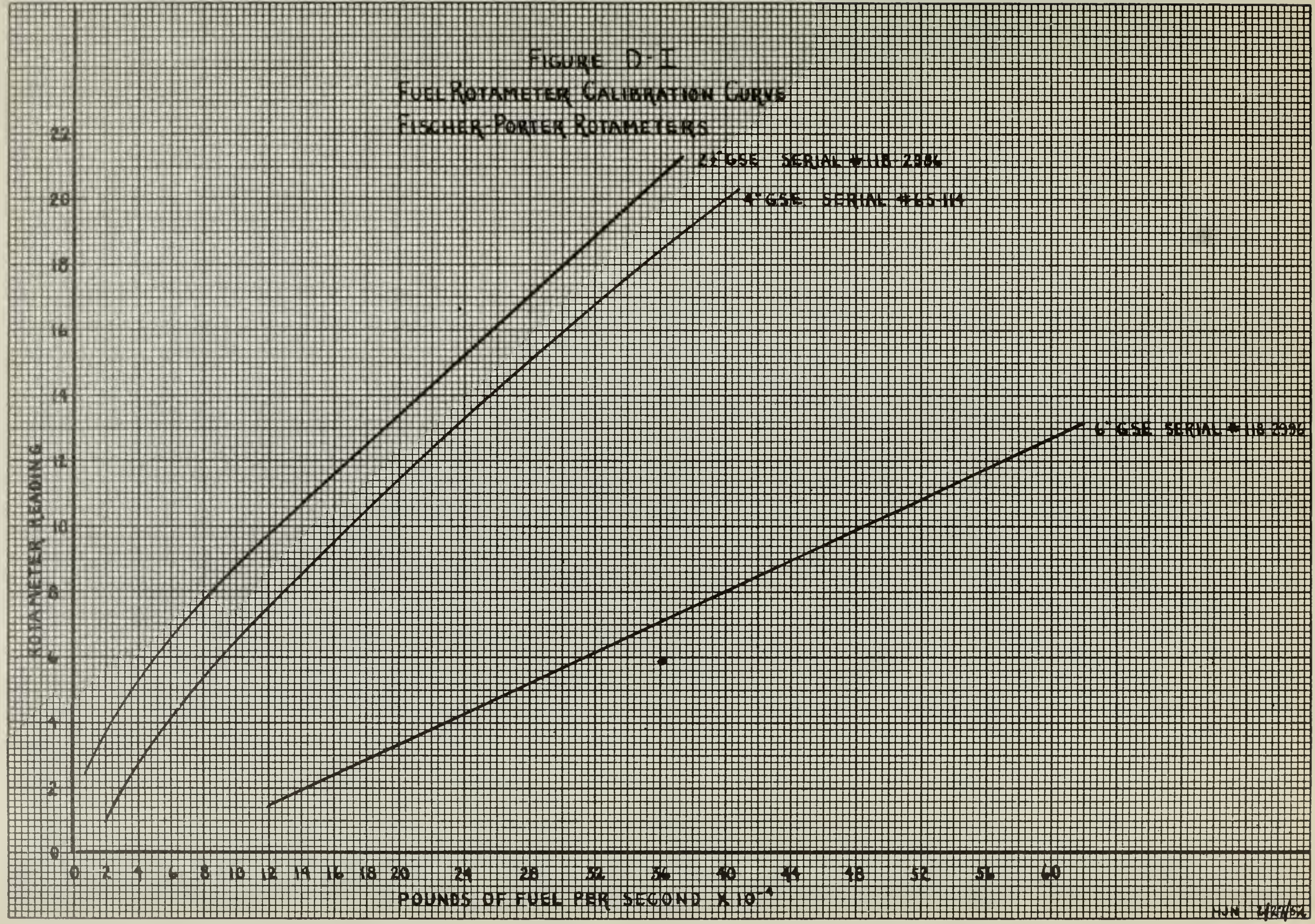


FIGURE D-I
FUEL ROTAMETER CALIBRATION CURVE
FISCHER-PORTER ROTAMETERS
2½" GSE SERIAL #118 2386
4" GSE SERIAL #65-114
6" GSE SERIAL #118 2996
ROTAMETER READING
0
2
4
6
8
10
12
14
16
18
20
22
0 2 4 6 8 10 12 14 16 18 20 24 28 32 36 40 44 48 52 56 60
POUNDS OF FUEL PER SECOND X 10⁻⁴
HJN 2/27/56

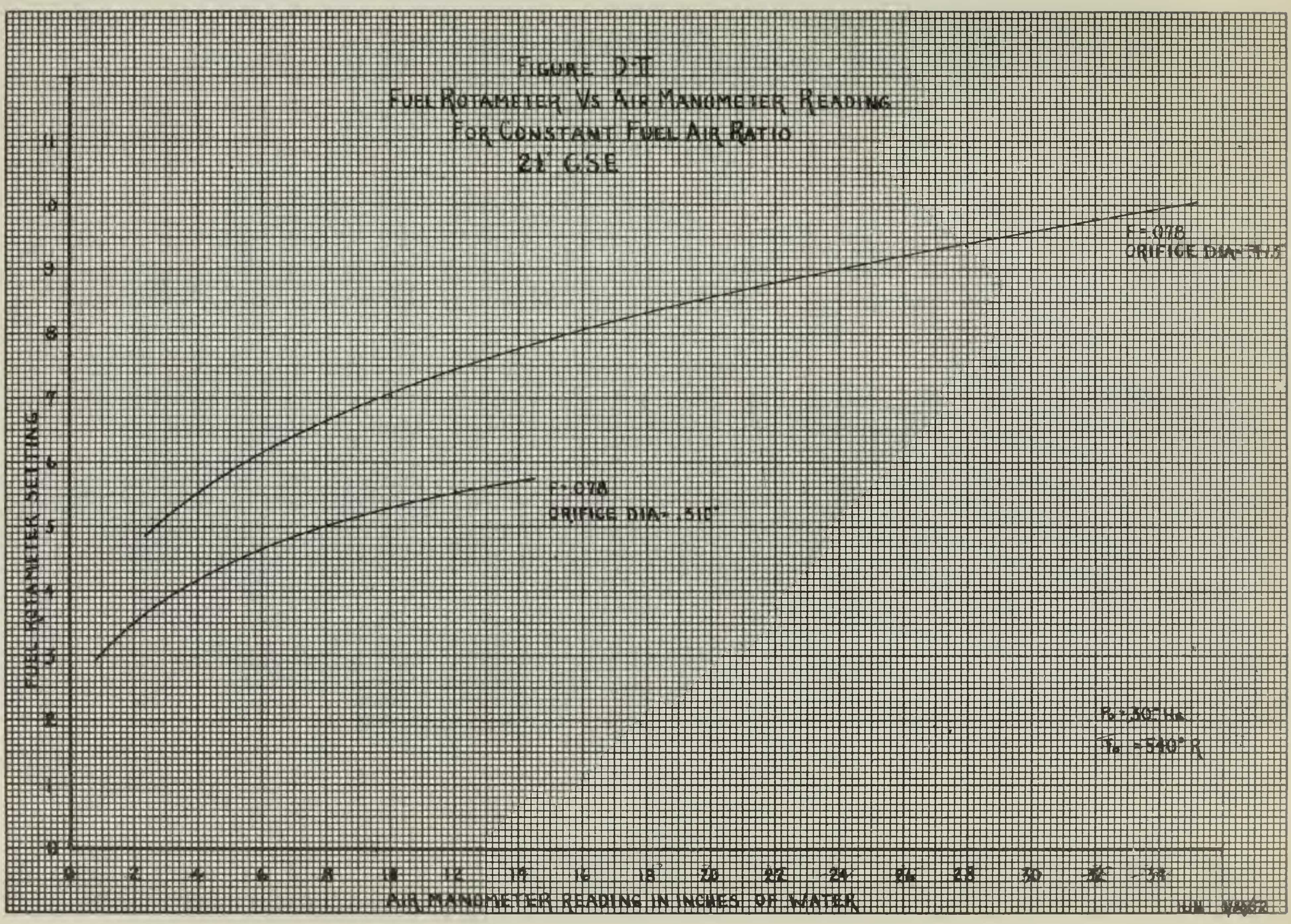


FIGURE D-II
FUEL ROTAMETER VS AIR MANOMETER READING
FOR CONSTANT FUEL AIR RATIO
2½" C.S.E.
F = .078
ORIFICE DIA = .510"
F = .078
FUEL ROTAMETER SETTING
AIR MANOMETER READING IN INCHES OF WATER
Po = 30" Hg
To = 540° R

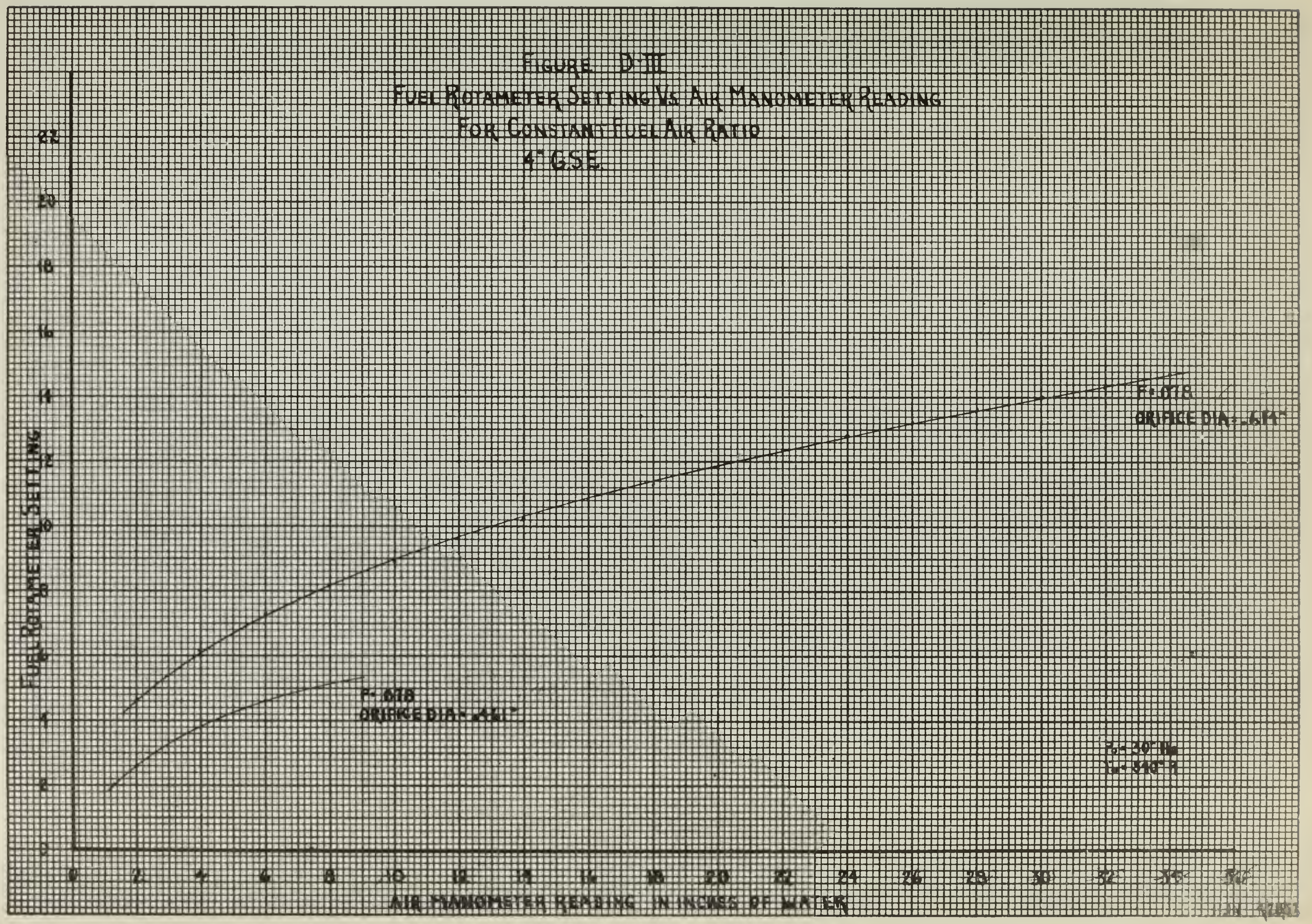


FIGURE D-III
FUEL ROTAMETER SETTING VS. AIR MANOMETER READING
FOR CONSTANT FUEL AIR RATIO
4" G.S.E.
FUEL ROTAMETER SETTING
AIR MANOMETER READING IN INCHES OF WATER
F = .078
F = .078
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36

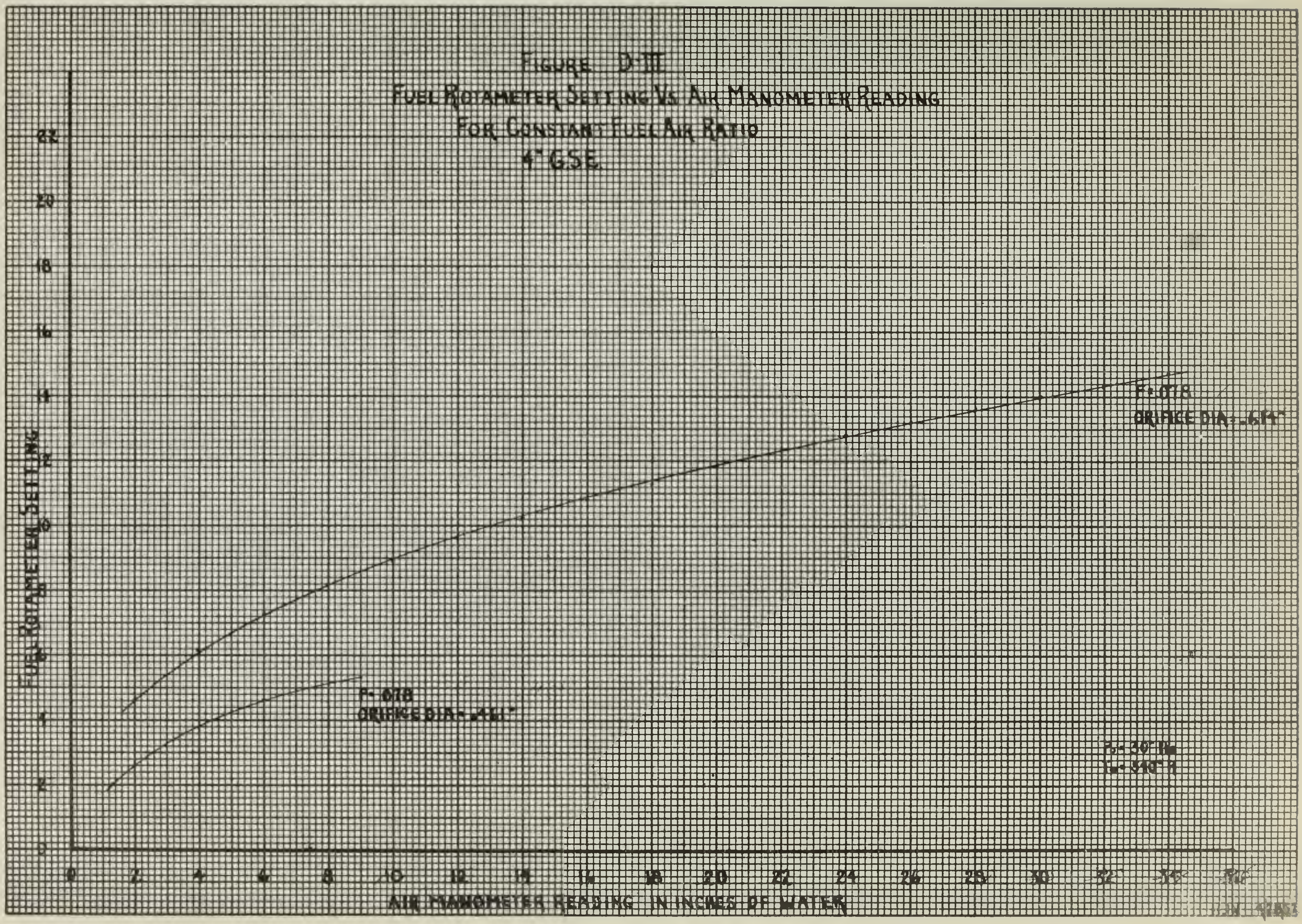


FIGURE D-III
FUEL ROTAMETER SETTING VS AIR MANOMETER READING
FOR CONSTANT FUEL AIR RATIO
4" G.S.E.
F = .078
ORIFICE DIA = .614"
FUEL ROTAMETER SETTING
AIR MANOMETER READING IN INCHES OF WATER
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36

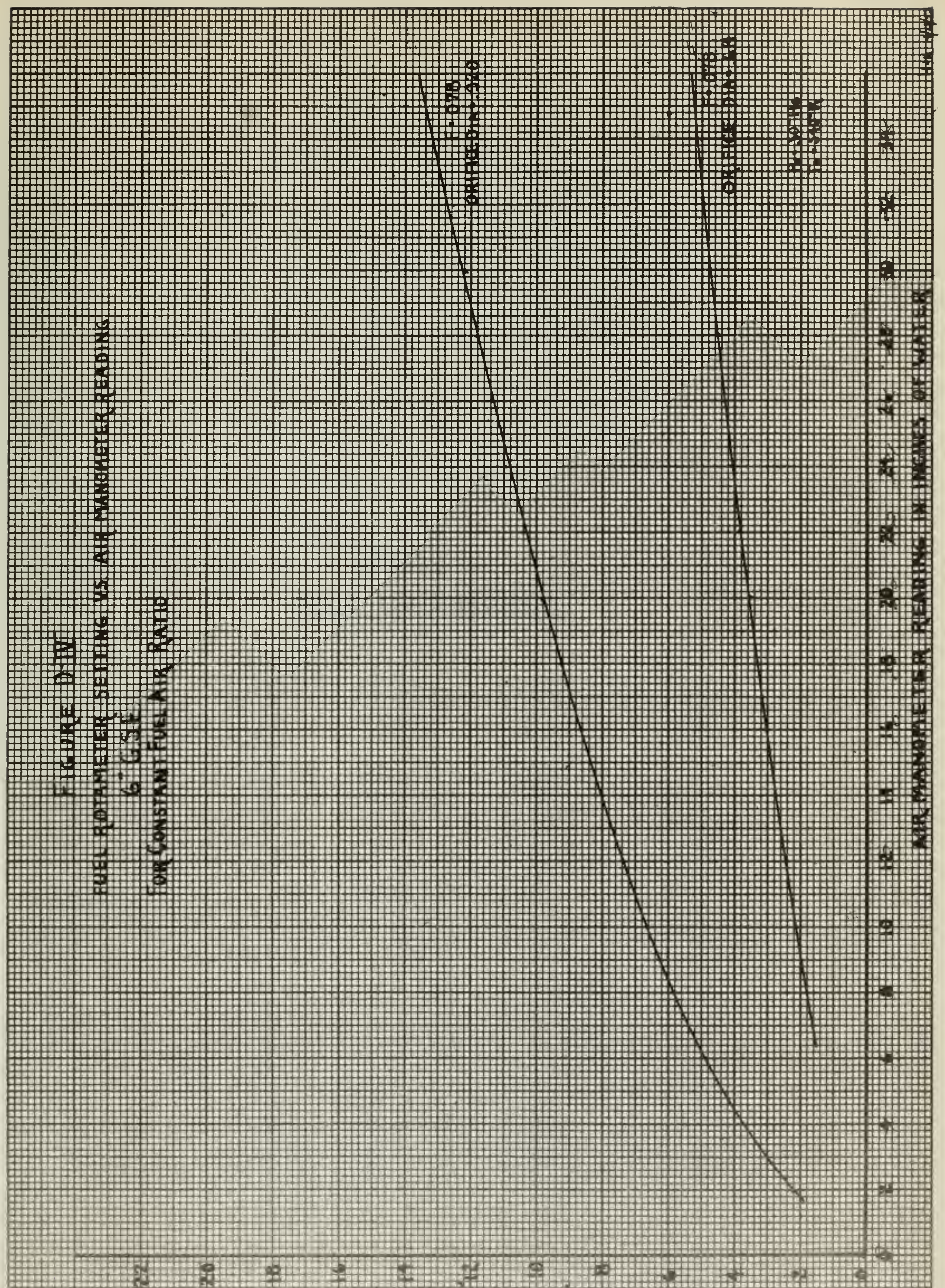


FIGURE D-IV
FUEL ROTAMETER SETTING VS. AIR MANOMETER READING
6" C.S.E.
FOR CONSTANT FUEL AIR RATIO
FUEL ROTAMETER SETTING
AIR MANOMETER READING IN INCHES OF WATER

APPENDIX E

FUELS AND LUBRICANTS

APPENDIX E

FUELS AND LUBRICANTS

In all runs a premium grade unleaded gasoline was used. The octane number was 91.5 to 92 by research method and 80.5 to 81.5 by motor method.

Similitude with respect to lubricating oils was achieved by mixing SAE 20 and SAE 60 as follows:

2½"	100% URSA P 20 (Texaco Symbol)
4"	54% URSA P 60, 46% P 20
6"	95% URSA P 60, 5% P 20

properties of the mixtures are listed in the following table,

Viscosity S.U.S.

Engine	100°F	130°F	210°F	Gravity
2½"	349	160.3	52.8	.88
4"	828	339.0	76.2	.89
6"	1665	627.0	114.1	.90

Oils used were straight mineral, parafin base, distilled with no additives.

It was determined by Gaboury et al. reference [3] that the above oils had the same $\frac{\mu}{\rho}$ at 250°F. This temperature was selected arbitrarily to attempt to satisfy lubrication requirements regarding viscous friction. A crankcase inlet temperature of 150° was selected and maintained throughout the runs.

APPENDIX F

ORIGINAL DATA AND SUMMARY OF CALCULATIONS

TABLE F-I

Motoring Friction Summary

2½" G.S.E.

Run	Piston Speed	Dyn. Reading	Dyn. Load	FMEP
		P_i = -2" Hg		
I	600	3.40	6.60	23.70
VI	900	2.63	7.37	26.45
VII	1200	1.60	8.40	30.19
XII	1450	.97	9.03	32.40
XIII	1700	.21	9.79	35.10
		P_i = -10" Hg		
II	600	2.73	7.27	26.10
V	900	1.98	8.02	28.80
VIII	1200	.94	9.06	32.55
XI	1450	.35	9.65	34.60
XIV	1700	-.40	10.40	37.35
		P_i = -20" Hg		
III	600	1.77	8.23	29.55
IV	900	.90	9.10	32.62
IX	1200	.30	9.70	34.80
X	1450	-.35	10.35	37.20
XVI	1700	-.98	10.98	39.40

4" G.S.E.

Run	Piston Speed	Dyn Reading	Dyn Load	FMEP
		P_i = -3" Hg		
IX	600	3.42	6.58	21.60
XIII	840	3.25	6.75	22.15
XIV	1020	2.65	7.35	24.10
XV	1200	2.20	7.80	25.59
XVII	1500	1.27	8.73	28.60
		P_i = -10" Hg		
XIX	600	2.93	7.07	23.20
XII	840	2.52	7.48	24.58
XI	1020	2.14	7.86	25.80
XVI	1200	1.70	8.30	27.20
VII	1500	1.00	9.00	29.55
XVIII	1800	-.50	10.50	34.42
		P_i = -18" Hg		
X	600	1.93	8.07	26.50
XI	840	1.77	8.23	27.00
III	1020	1.56	8.44	27.65
V	1200	1.17	8.83	28.98
VIII	1500	.52	9.48	31.10
VI	1800	-.85	10.85	35.60

6" G.S.E.

Run	Piston Speed	Dyn Reading	Dyn Load	FMEP
		P_i = -4" Hg		
I	720	3.20	6.80	26.45
VI	1080	3.00	7.05	27.20
VII	1320	2.50	7.50	29.20
XIII	1560	2.26	7.74	30.10
XII	1800	1.50	8.50	33.05
		P_i = -10" Hg		
II	720	2.95	7.05	27.42
V	1080	2.55	7.45	29.00
VIII	1320	2.10	7.90	30.72
XIV	1560	1.75	8.25	32.10
XI	1800	.95	9.05	35.20
		P_i = -16" Hg		
III	720	2.61	7.39	28.78
IV	1080	1.95	8.05	31.32
IX	1320	1.60	8.40	32.70
XV	1560	1.30	8.70	33.85
X	1800	.45	9.55	37.20

TABLE F-II

FRICTION RUNS $2\frac{1}{2}$" GSE

	Time	Run	RPM	Piston Speed FT/MIN	Cyl. Head (Temperatures in °F)	Inlet Air	H_2O Exit	Cyl. Wall	Mn. Brng. Oil Inlet	Mn. Brng	Oil From Sump	Dyn. Reading	(ΔP) Air "H_2O	P_i Intake (Pressure in in. Hg.)	P_e Exhaust	P_o Orifice	(ΔP) H_2O	H_2O From Engine °F	H_2O To Rotameter °F	H_2O From Cooler °F
	P_i = -20" Hg																			
IX	1243	32	2400	1200	159	150	145	155	141	180	152	0.30	0.60	-19.9	1.05	0	16.6	147	147	146
	1300	33	2400	1200	159	150	145	155	139	180	149	0.30	0.60	-20.0	1.00	0	16.6	146	146	146
	1310	34	2400	1200	159	150	149	154	139	180	149	0.30	0.58	-20.0	1.00	0	16.6	146	146	145
	P_i = -20" Hg																			
X	1341	35	2900	1450	163	151	145	157	140	189	150	-0.35	0.70	-20.0	1.10	0	16.6	146	146	144
	1347	36	2900	1450	163	150	145	157	140	190	150	-0.35	0.70	-20.0	1.00	0	16.6	146	146	149
	1358	37	2900	1450	163.5	150	144	157	138	190	150	-0.35	0.70	-20.0	1.00	0	16.6	146	146	144
	P_i = -10" Hg																			
XI	1400	38	2900	1450	165	150	144	156	135	190	150	+0.35	6.50	-10.0	1.00	0	16.6	146	146	144
	1405	39	2900	1450	166	150	145	157	132	190	149	+0.35	6.50	-10.0	1.00	0	16.6	145	145	143
	1410	40	2900	1450	166	150	145	157	131	190	150	+0.35	6.50	-10.0	1.00	0	16.6	144	144	143
	P_i = -2" Hg																			
XII	1422	41	2900	1450	165	150	144	156	128	190	152	+0.96	15.95	-2.0	1.00	0	16.6	145	145	144
	1428	42	2900	1450	166	150	145	157	120	190	150	+0.98	15.85	-2.0	1.00	0	16.6	146	146	144
	1435	43	2900	1450	166	150	145	157	116	190	150	+0.98	15.85	-2.0	1.00	0	16.6	146	146	145
	P_i = -2" Hg																			
XIII	1605	44	3400	1700	170	150	143	159	138	199	150	0.21	19.85	-2.0	1.00	0	16.7	145	145	140
	1610	45	3400	1700	169	150	143	158	138	198	150	0.21	19.83	-2.0	1.00	0	16.5	144	144	141
	1615	46	3400	1700	169	150	143	158	139	198	150	0.22	19.83	-2.0	1.00	0	16.6	144	144	142
	P_i = -10" Hg																			
XIV	1622	47	3400	1700	170	150	143	159	140	198	151	-0.35	7.80	-10.0	1.00	0	16.6	145	145	142
	1627	48	3400	1700	170	150	144	159	139	198	150	-0.45	7.80	-10.0	1.00	0	16.6	146	146	143
	1635	49	3400	1700	170	150	144	159	140	198	150	-0.40	7.80	-10.0	1.00	0	16.6	147	147	144
	P_i = -20" Hg																			
XV	1640	50	3400	1700	168	153	144	160	140	197	150	-0.75	0.70	-20.0	1.0	0	16.6	146	146	144
		51	3400	1700	167	150	144	160	140	197	150	-0.65	0.70	-20.0	1.0	0	16.6	146	146	144

25 MARCH 1952

BAROMETER 763.3 mm Hg. TEMP. 19°C.

	Time	Run	RPM	Piston Speed FT/MIN	Cyl. Head	Inlet Air	H_2O Exit	Cyl. Wall	Mn. Brng. Oil Inlet	Mn. Brng	Oil From Sump	Dyn. Reading	(ΔP) Air "H_2O	P_i Intake	P_e Exhaust	P_o Orifice	(ΔP) H_2O	H_2O From Engine °F	H_2O To Rotameter °F	H_2O From Cooler °F
	P_i = -20" Hg																			
XVI	1018	52	3400	1700	166	150	142	158	140	195	150	-1.00	0.75	-20.0	1.01	0	16.6	145	145	143
	1028	53	3400	1700	166	150	143	158	140	195	150	-1.00	0.70	-20.0	1.00	0	16.6	145	145	142
	1035	54	3400	1700	166	150	143	158	141	195	150	-0.95	0.70	-20.0	1.00	0	16.7	145	145	142

19 March, 1952
OBSERVERS: Smith, Bohner
Barometer 768.2 mm Hg
Temp. 22.7°C ; 74°F = Rm Temp.

TABLE F-III

FRICTION RUNS $2\frac{1}{2}$" G.S.E.

Dynamometer Zero Reading: 10" Hg

	Time	Run	RPM	Piston Speed Ft/Min	Temperatures in °F: Cyl. Head	Inlet Air	H_2O Exit	Cyl. Wall	Mn. Brng Inlet Oil	Mn. Brng	Oil From Sump	Dyn. Reading	$(\Delta P)_{Air}$ " H_2O	Pressures in inches of Hg: P_i Intake	P_e Exhaust	P_o Orifice	$(\Delta P)_{H_2O}$	H_2O From Engine °F	H_2O To Rotameter °F	H_2O From Cooler °F
	P_i = -2.0" Hg																			
I	1555	1	1200	600	152	148	149	145	127	150	150	3.4	2.9	-2.0	1.0	0	16.5	146	146	147
	1610	2	1200	600	151	150	143	143	127	150	150	3.4	2.85	-2.0	1.0	0	16.4	146	146	144
	1615	3	1200	600	150	149	142	142	128	150	150	3.4	2.85	-2.0	1.0	0	16.4	144	144	144
	P_i = -10.0" Hg																			
II	1625	4	1200	600	150	144	142	142	128	151	150	2.72	1.25	-10.00	1.0	0	16.45	145	144	146
	1632	5	1200	600	151	150	143	143	128	151	150	2.73	1.23	-10.00	1.0	0	16.45	145	145	145
	1640	6	1200	600	151	150	143	143	127	151	150	2.73	1.20	-10.00	1.0	0	16.60	146	146	145
	1645	7	1200	600	150	150	141	141	127	151	150	2.73	1.20	-10.00	1.0	0	16.60	144	144	141

21 March 1952

Barometer 763 mm Hg
Temp. 21.9°C
Rm. Temp. = 79°F

	Time	Run	RPM	Piston Speed Ft/Min	Cyl. Head	Inlet Air	H_2O Exit	Cyl. Wall	Mn. Brng Inlet Oil	Mn. Brng	Oil From Sump	Dyn. Reading	$(\Delta P)_{Air}$ " H_2O	P_i Intake	P_e Exhaust	P_o Orifice	$(\Delta P)_{H_2O}$	H_2O From Engine °F	H_2O To Rotameter °F	H_2O From Cooler °F
	P_i = -20.0" Hg																			
III	1312	8	1200	600	153	150	145	150	141	149	150	1.78	0.15	-20.0	1.0	0	16.6	148	148	148
	1317	9	1200	600	154	150	146	146	140	150	150	1.78	0.15	-20.0	1.0	0	16.6	148	148	147
	1325	10	1200	600	151	150	143	143	141	150	150	1.76	0.15	-20.0	1.0	0	16.6	145	145	145
	P_i = -20.0" Hg																			
IV	1340	11	1800	900	155	151	143	143	140	163	150	0.80	0.30	-20.0	1.0	0	16.6	145	145	144
	1345	12	1800	900	156	151	144	144	142	164	150	0.90	0.30	-20.0	1.0	0	16.6	146	146	146
	1352	13	1800	900	156	153	144	144	141	164	150	0.90	0.30	-20.0	1.0	0	16.6	146	146	145
	1357	14	1800	900	155	148	144	144	141	164	150	0.90	0.30	-20.0	1.0	0	16.6	146	146	145
	P_i = -10.0" Hg																			
V	1406	15	1800	900	155	150	143	149	141	164	150	1.90	2.70	-10.0	1.0	0	16.6	145	145	145
	1415	16	1800	900	155	150	142	150	142	165	150	1.93	2.70	-10.0	1.0	0	16.6	145	145	145
	1422	17	1800	900	155	147	143	149	142	164	150	1.95	2.70	-10.0	1.0	0	16.6	144	144	145
	1427	18	1800	900	155	150	142	149	142	164	150	1.98	2.70	-10.0	1.0	0	16.6	144	144	144
	1433	19	1800	900	155	150	143	143	142	165	150	1.98	2.70	-10.0	1.0	0	16.6	146	146	145
	1445	20	1800	900	157	150	146	146	142	165	150	2.02	2.65	-10.0	1.0	0	16.6	148	148	147
	P_i = -2.0" Hg																			
VI	1455	21	1800	900	157	149	145	146	142	165	150	2.65	6.55	-2.0	1.0	0	16.6	147	147	146
	1501	22	1800	900	156	150	144	144	142	165	150	2.63	6.55	-2.0	1.0	0	16.6	146	146	145
	1508	23	1800	900	156	149	144	144	142	165	150	2.62	6.55	-2.0	1.0	0	16.6	146	146	145
	P_i = -2.0" Hg																			
VII	1525	24	2400	1200	160	146	143	155	141	178	150	1.6	11.5	-2.0	1.0	0	16.6	146	146	145
	1532	25	2400	1200	160	151	143	154	142	179	150	1.6	11.45	-2.0	1.0	0	16.6	145	145	144
	1537	26	2400	1200	160	150	143	154	142	179	150	1.6	11.45	-2.0	1.0	0	16.6	145	145	144

24 March 1952

Barom: 761.2 mm Temp. 23.5°C

	Time	Run	RPM	Piston Speed Ft/Min	Cyl. Head	Inlet Air	H_2O Exit	Cyl. Wall	Mn. Brng Inlet Oil	Mn. Brng	Oil From Sump	Dyn. Reading	$(\Delta P)_{Air}$ " H_2O	P_i Intake	P_e Exhaust	P_o Orifice	$(\Delta P)_{H_2O}$	H_2O From Engine °F	H_2O To Rotameter °F	H_2O From Cooler °F
	P_i = -10" Hg																			
VIII	1155	27	2400	1200	159	150	143	151	140	178	148	0.85	4.75	-10.0	1.0	0	16.6	146	146	145
	1218	28	2400	1200	160	151	143	153	140	178	150	0.86	4.75	-9.9	1.04	0	16.7	145	145	144
	1225	29	2400	1200	160	150	143	153	140	179	150	1.00	4.80	-9.9	1.05	0	16.6	146	146	144
	1235	30	2400	1200	161	150	145	155	140	180	150	1.00	4.80	-10.0	1.00	0	16.6	148	148	147
		31	2400	1200	160	150	145	155	140	180	150	1.00	4.75	-10.0	1.00	0	16.6	147	147	146

21 MARCH 1952

OBSERVERS: MITCHELL, NAROONE

TABLE F-IV

FRICTION RUNS 4" G.S.E.

DYNAMOMETER ZERO READING: 10" Hg.

CELL TEMP. 79°F

BAROMETER 763 mm Hg 21.9°C

	Time	Run	Piston Speed ft/min	Temperature in °F: Cyl. Head	Inlet Air	H_2O Entering Eng.	Cyl. Wall	Mn. Brng. Inlet Oil	Mn. Brng	Oil From Sump	Dyn. Reading	(ΔP) Air "H_2O	Pressure in inches Hg: P_i Intake	P_e Exhaust	P_o Orifice	$(\Delta P)_{H_2O}$	Temperatures: H_2O From Engine °F	H_2O To Rotameter °F	H_2O From Cooler °F	RPM	Water Rotameter Reading
	$P_i = -2.0$" Hg																				
I	1250	1	1020	164	151	150		146	175	148	2.58	11.2	-2.0	1.0	0	18.05		150.0	144	1275	
	1315	2	1020	169	150.5	150		150	175	150	2.58	11.2	-2.0	1.0	0	18.02		150	148	1275	
	1321	3	1020	164	150	150		150	175	150	2.60	11.2	-2.0	1.0	0	18.00		150	148	1275	
	1346	4	1020	164	150	150		148	175	146	2.60	11.2	-2.0	1.0	0	18.00		150	148	1275	
	1352	5	1020	164	150	150		148	175	148	2.60	11.2	-2.0	1.0	0	18.00		150	148	1275	
	$P_i = -10$" Hg																				
II	1425	1	1020	165	151	150		150	177	150	2.1	4.7	-10.0	1.0	0	18.0		150	148	1275	
	1430	2	1020	164	151	150		151	177	151	2.09	4.7	-10.0	1.0	0	18.0		150	148	1275	
	1438	3	1020	164	150	150		150	177	150	2.12	4.7	-10.0	1.0	0	18.0		150	148	1275	
	1530	4	1020	165	151.5	150		153	177.5	151	2.15	4.65	-10.0	1.0	0	18.0		150	148	1275	
	1535	5	1020	162	151	149		150	177	151	2.19	4.65	-10.0	1.0	0	18.0		150	147	1275	
	1642	6	1020	164	150	150		150	177	150	2.19	4.65	-10.0	1.0	0	18.0		150	148	1275	
	24 MARCH 1952		MITCHELL & WILKINSON																		
	$P_i = -18$" Hg																				
III	1033	2	1020	164	152	150		150	174	150	1.45	1.03	-18.15	1.0	0	18.0		150	146	1275	
	1045	3	1020	164	150	150		150	174	150	1.60	1.03	-18.10	1.0	0	18.0		150	146	1275	
	1054	4	1020	168	151	150		150	175	150	1.61	1.15	-18.10	1.05	0	18.0		150	146	1275	
	1058	5	1020	165	151	150		150	175	150	1.58	1.15	-18.10	1.0	0	18.0		150	146	1275	
	$P_i = -2$" Hg																				
IV	1115	1	1200	167	150	150		150	181	150	2.45	15.2	-2.0	1.0	0	18.0		150	146	1500	
	1120	2	1200	167	149	150		150	181	150	2.48	15.25	-2.0	1.0	0	18.0		150	146	1500	
	1125	3	1200	167	149	150		150	181	150	2.45	15.3	-2.0	1.0	0	18.0		150	146	1500	
	1130	4	1200	167	150	150		150	181	150	2.43	15.3	-2.0	1.0	0	18.0		150	146	1500	
	$P_i = -18$" Hg																				
V	1215	1	1200	168	150	150		150	187	150	1.0	1.55	-18.0	1.0	0	18.0		150	146	1500	
	1240	2	1200	168	150	150		150	185	150	1.25	1.45	-18.10	1.0	0	18.0		150	146	1500	
	1247	3	1200	168	149	150		150	185	150	1.25	1.50	-18.05	1.0	0	18.0		150	146	1500	
	1252	4	1200	168	149	150		150	185	150	1.15	1.50	-18.00	1.0	0	18.0		150	146	1500	
	1255	5	1200	168	150	150		150	185	150	1.20	1.50	-18.00	1.0	0	18.0		150	146	1500	
	$P_i = -18$" Hg																				
VI	1515	1	1800	184	151	150		150	210	150	-.85	2.15	-18.0	1.00	0	18.0	150	150	148	2250	59
	1520	2	1800	184	150	150		150	210	150	-.80	2.20	-18.0	1.00	0	18.0	150	150	148	2250	59
	1525	3	1800	184	150	150		150	210	150	-.85	2.20	-18.0	1.00	0	18.0	150	150	148	2250	59
	$P_i = -10$" Hg																				
VII	1542	1	1500	176	151	150		150	200	150	1.00	9.1	-10.0	1.0	0	-18.0	150	150	148	1875	58
	1545	2	1500	177	150	150		150	200	150	1.00	9.1	-10.0	1.0	0	-18.0	150	150	148	1875	57.8
	1552	3	1500	178	152	150		150	200	150	1.00	9.1	-10.0	1.0	0	-18.0	150	150	148	1875	58
	$P_i = -18$" Hg																				
VIII	1600	1	1500	178	151	150		150	197	150	.50	2.0	-18.0	1.0	0	-18.0	150	150	148	1875	58.5
	1605	2	1500	175	150	150		150	197	150	.55	2.0	-18.0	1.0	0	-18.0	150	150	148	1875	58.5
	1610	3	1500	175	148	150		150	197	150	.50	2.0	-18.0	1.0	0	-18.0	150	150	148	1875	58.5

TABLE F·V

FRICTION 4" GSE

	Time	Run	RPM	Piston Speed ft/min	Temperatures in °F: Cyl. Head	Inlet Air	H_2O Entering Eng.	Cyl Wall	Mn Brng Oil Inlet	Mn Brng.	Oil From Sump	Dyn. Reading "Hg	(ΔP) Air "H_2O	Pressure in "Hg: Pi Intake	Pe Exhaust	Po Orifice	$(\Delta P)_{H_2O}$	Temperature: H_2O From Engine °F	H_2O To Rotameter °F	H_2O From Cooler °F	Water Rotameter Reading
	Pi = -3" Hg											24 MARCH 1952 (CONTINUED)									
IX	1020	1	750	600	157	151	150		149	145		3.4	3.4	-3.0	1.0	0	18	150	150	148	58
	1030	2	750	600	158	150	150		150	146		3.42	3.4	-3.0	1.0	0	18	150	150	149	58
	1040	3	750	600	158	149	150		150	147		3.44	3.4	-3.0	1.0	0	18	150	150	149	58
										25 MARCH 1952 (NARDONE, WILKINSON)											
	Pi = -18" Hg																				
X		1	750	600	157	152	150		150	149	150	1.90	.40	-18.0	1.0	0	18	150	150	148	56
		2	750	600	157	150	150		150	149	150	1.93	.40	-18.0	1.0	0	18	150	150	148	56
		3	750	600	157	150	150		150	150	150	1.95	.40	-18.0	1.0	0	18	150	150	148	56
XI	1445	4	1050	840	160	150	150		150	163	150	1.80	.70	-18.0	1.0	0	18	150	150	148	56
	1500.	5	1050	840	161	151	150		150	164	150	1.75	.75	-18.0	1.0	0	18	150	150	148	56
	Pi = -10" Hg																				
XII	1515	1	1050	840	160	151	150		150	164	150	2.52	3.0	-10.0	1.0	0	18	150	150	148	56
	1525	2	1050	840	160	150	150		150	164	150	2.51	3.05	-10.0	1.0	0	18	150	150	148	56
	Pi = -3" Hg																				
XIII	1530	1	1050	840	160	151	150		150	164	150	3.25	6.7	-3.0	1.0	0	18.0	150	150	148	56
	1540	2	1050	840	160	150	150		150	164	150	3.25	6.7	-3.0	1.0	0	18.0	150	150	148	56
	Pi = -3" Hg																				
XIV	1550	1	1275	1020	165	152	150		150	172	150	2.6	9.9	-3.0	1.0	0	18.0	150	150	148	56
	1600	2	1275	1020	165	150	150		150	172	150	2.7	10.0	-3.0	1.0	0	18.0	150	150	148	56
	Pi = -3" Hg																				
XV	1610	1	1500	1200	167	152	150		150	178	150	2.2	13.7	-3.0	1.0	0	18.0	150	150	148	56
	1620	2	1500	1200	167	152	150		150	178	150	2.2	13.7	-3.0	1.0	0	18.0	150	150	148	56
	Pi = -10" Hg																				
XVI	1625	1	1500	1200	167	152	150		150	180	150	1.7	6.4	-10.0	1.0	0	18.0	150	150	148	56
	1635	2	1500	1200	167	151	150		150	180	150	1.7	6.4	-10.0	1.0	0	18.0	150	150	148	56
	Pi = -3" Hg																				
XVII	1650	1	1875	1500	175	149	150		150	193	150	1.27	20.8	-3.0	1.0	0	18.0	150	150	148	56
	1655	2	1875	1500	175	150	150		150	195	150	1.27	20.8	-3.0	1.0	0	18.0	150	150	148	56
	Pi = -10" Hg																				
XVIII	1700	1	2250	1800	175	150	150		150	205	150	-0.50	11.1	-10.0	1.0	0	18.0	150	150	148	56
	1710	2	2250	1800	175	150	150		150	205	150	-0.50	11.1	-10.0	1.0	0	18.0	150	150	148	56
	Pi = -10"																				
XIX			750	600	157	152	150		150	149	150	2.90	1.6	-10.0	1.0	0	18.0	150	150	148	56
			~~750~~	~~600~~	~~157~~	~~150~~	~~150~~		~~150~~	~~149~~	~~150~~	~~1.93~~	~~.4~~	~~-10.0~~	~~1.0~~	~~0~~	~~18.0~~	~~150~~	~~150~~	~~148~~	~~56~~ HJN
			750	600	157	150	150		150	150	150	2.95	1.6	-10.0	1.0	0	18.0	150	150	148	56

26 MARCH, 1952
OBSERVERS: SMITH, BOHNER
BAROMETER 765.2 mm HG
TEMP. 24°C ;

TABLE F-VI

DYNAMOMETER ZERO READING: 10" HG

FRICTION RUNS 6" G.S.E.

	TIME	RUN	RPM	PISTON SPEED FT/MIN	TEMPERATURES IN °F: CYL. HEAD	INLET AIR	MN. BRNG. INLET OIL	MN. BRNG.	OIL FROM SUMP	DYN. READING " HG	$(\Delta P)_{AIR}$ " H_2O	PRESSURES IN " HG: P_i INTAKE	P_e EXHAUST	$(\Delta P)_{H_2O}$	TEMPERATURES IN °F: H_2O TO ENGINE	H_2O FROM ENGINE	H_2O TO ROTAMETER	H_2O FROM COOLER	H_2O ROTA.
	P_i = −4.0" HG																		
	1035	1	600	720	160	160	150	165	151	2.81	3.9	−4	1	17.5	145	145	145	145	26.5
	1060	2	600	720	160	160	150	160	151	3.1	3.9	−4	1	17.4	145	145	145	144	27.4
I	1055	3	600	720	160	160	150	160	150	3.2	3.9	−4	1	17.4	145	145	145	144	27.3
	1100	4	600	720	160	160	150	162	150	3.3	3.8	−4	1	17.4	145	145	145	143	27.5
	1105	5	600	720	160	160	150	162	150	3.3	3.9	−4	1	17.4	145	145	145	143	27.6
	P_i = −10.0" HG																		
	1115	6	600	720	159	160	150	166	150	2.9	1.65	−10	1	17.4	145	145	145	142	27.6
II	1120	7	600	720	159	165	151	166	150	3.0	1.56	−10	1	17.4	145	145	145	142	27.5
	1125	8	600	720	159	160	151	167	150	3.0	1.55	−10	1	17.4	145	146	145	142	27.5
	1132	9	600	720	159	159	152	167	150	2.9	1.6	−10	1	17.4	145	145	145	142	27.7
	P_i = −16" HG																		
	1140	10	600	720	157	164	152	168	150	2.6	.35	−16	1	17.4	145	145	145	142	27.6
III	1150	11	600	720	157	159	152	168	150	2.63	.35	−16	1	17.4	145	145	145	142	27.5
	1157	12	600	720	157	159	152	169	150	2.6	.35	−16	1	17.4	145	145	145	142	31.5
	P_i = −16" HG																		
	1210	13	900	1080	164	163	151	185	150	1.95	1.2	−16	1	17.4	145	145	145	140	31.6
IV	1215	14	900	1080	164	160	151	186	150	1.95	1.2	−16	1	17.4	145	145	145	140	31.4
	1223	15	900	1080	164	160	151	187	150	1.95	1.25	−16	1	17.4	145	145	145	140	31.4
	P_i = −10" HG																		
	1232	16	900	1080	164	160	151	187	150	2.5	4.4	−10	1	17.4	145	145	146	139	31.6
V	1240	17	900	1080	165	161	151	188	150	2.6	4.3	−10	1	17.4	145	145	145	139	31.5
	1245	18	900	1080	165	159	151	188	150	2.55	4.3	−10	1	17.4	145	145	145	139	31.6
	P_i = −4" HG																		
	1300	19	900	1080	166	160	151	188	150	3.0	9.6	−4	1	17.4	145	145	145	139	31.5
VI	1305	20	900	1080	166	158	151	188	150	3.0	9.6	−4	1	17.4	145	145	145	139	31.5
	1310	21	900	1080	166	159	151	188	150	3.0	9.6	−4	1	17.4	145	145	146	139	31.5
	P_i = −4" HG																		
	1320	22	1100	1320	173	160	150	199	149	2.5	14.5	−4	1	17.4	144	145	145	137	31.6
VII	1330	23	1100	1320	173	160	151	199	151	2.6	14.5	−4	1	17.4	145	145	145	138	31.6
	1340	24	1100	1320	173	160	151	200	151	2.5	14.5	−4	1	17.4	144	145	145	137	31.8
	P_i = −10" HG																		
	1350	25	1100	1320	171	160	151	201	151	2.1	6.7	−10	1	17.4	145	145	145	137	31.8
VIII	1355	26	1100	1320	171	160	151	200	151	2.11	6.7	−10	1	17.4	144	145	145	137	31.9
	1400	27	1100	1320	171	160	151	200	151	2.1	6.7	−10	1	17.4	144	146	145	138	31.8
	P_i = −16" HG																		
	1410	28	1100	1320	169	161	150	201	150	1.6	2.05	−16	1	17.4	144	146	145	138	31.7
IX	1415	29	1100	1320	170	161	150	201	150	1.6	2.1	−16	1	17.4	144	146	145	138	31.8
	1420	30	1100	1320	170	161	150	201	150	1.6	2.1	−16	1	17.4	144	146	145	138	32.0

26 MARCH, 1952
OBSERVERS: SMITH, BOHNER
BAROMETER 765.2 mm HG
TEMP. 24 °C

TABLE F-VIII

DYNAMOMETER ZERO READING: 10" HG

FRICTION RUNS 6" G.S.E.

	TIME	RUN	RPM	PISTON SPEED FT/MIN	TEMPERATURES IN °F: CYL. HEAD	INLET AIR	MN. BRNG. INLET OIL	MN. BRNG.	OIL FROM SUMP	DYN. READING	$(\Delta P)_{AIR}$ "H_2O	PRESSURES IN "HG: P_i INTAKE	P_e EXHAUST	$(\Delta P)_{H_2O}$	TEMPERATURES IN °F: H_2O TO ENGINE	H_2O FROM ENGINE	H_2O TO ROTA-METER	H_2O FROM COOLER	H_2O ROTA.
	P_i = -16" HG																		
X	1630	31	1500	1800	184	161	151	210	151	0.4	3.55	-16	1	17.5	144	146	146	128	16.9
	1635	32	1500	1800	184	160	151	210	150	0.45	3.5	-16	1	17.5	144	146	146	130	16.8
	1640	33	1500	1800	184	160	150	210	150	0.45	3.6	-16	1	17.5	144	146	146	128	16.8
	P_i = -10" HG																		
XI	1645	34	1500	1800	186	160	150	212	150	0.9	11.1	-10	1	17.5	144	146	146	128	14.5
	1650	35	1500	1800	186	160	150	212	150	1.0	11.1	-10	1	17.5	144	146	146	128	15.0

1 APRIL, 1952

OBSERVERS: SMITH, NARDONE, BOHNER
BAROMETER 763.6 mm HG
TEMP. 21.3 °C

	TIME	RUN	RPM	PISTON SPEED FT/MIN	CYL. HEAD	INLET AIR	MN. BRNG. INLET OIL	MN. BRNG.	OIL FROM SUMP	DYN. READING	$(\Delta P)_{AIR}$ "H_2O	P_i INTAKE	P_e EXHAUST	$(\Delta P)_{H_2O}$	H_2O TO ENGINE	H_2O FROM ENGINE	H_2O TO ROTA-METER	H_2O FROM COOLER	H_2O ROTA.
	P_i = -4" HG																		
	1215	36	1500	1800	187	160	150	212	150	1.15	23.6	-4	1	17.4	144	146	146	118	11.5
	1220	37	1500	1800	187	160	150	212	150	1.15	23.6	-4	1	17.4	148	150	150	120	11.5
XII	1228	38	1500	1800	187	160	150	214	150	1.5	23.6	-4	1	17.4	144	147	147	123	11.5
	1234	39	1500	1800	187	160	150	214	150	1.5	23.6	-4	1	17.4	144	146	146	124	11.4
	1240	40	1500	1800	187	160	150	214	150	1.5	23.6	-4	1	17.4	144	146	146	125	11.5
	P_i = -4" HG																		
	1245	41	1300	1560	184	161	150	210	150	2.3	19.6	-4	1	17.4	146	146	146	142	19.6
XIII	1250	42	1300	1560	185	160	150	210	150	2.2	19.6	-4	1	17.4	150	150	150	142	18.0
	1255	43	1300	1560	185	160	150	210	150	2.3	19.6	-4	1	17.4	150	150	150	142	18.0
	P_i = -10" HG																		
	1300	44	1300	1560	183	160	150	210	150	1.75	8.8	-10	1	17.5	146	146	146	131	15.8
XIV	1305	45	1300	1560	183	160	150	210	150	1.75	8.8	-10	1	17.5	146	146	146	130	15.2
	1310	46	1300	1560	182	160	150	210	150	1.75	8.8	-10	1	17.5	144	144	144	130	16.1
	P_i = -16" HG																		
	1315	47	1300	1560	180	164	150	210	150	1.3	2.6	-16	1	17.4	146	146	146	138	16.5
XV	1320	48	1300	1560	180	160	150	210	150	1.3	2.6	-16	1	17.4	146	146	146	138	16.5
	1325	49	1300	1560	180	160	150	210	150	1.3	2.6	-16	1	17.4	146	146	146	138	17.5
	P_i = -4" HG																		
XVI	1405	50	500	600	133	150	151	130	151	3.7	2.4	-4.0	1	17.5	147	148	148	140	37.0
	1415	51	500	600	133	143	150	137	150	3.75	2.3	-3.9	1	17.5	146	146	146	144	30.7

TABLE F-VIII

$L^2 = 6.25$

HEAT REJECTION CALCULATIONS 2½" G.S.E.

RUN	$(\Delta T)_{H_2O}$ °F	W_{H_2O} #/MIN	$\dot{Q}$ BTU/MIN.	h_{brake} "Hg	BMEP P.S.I. 3.59h	$h_{frict.}$ ← FROM CURVES	FMEP FROM CURVES → 3.59h	$h_{indicated}$ $h_f + h_b$	IMEP PSI. 3.59h	IHP $\frac{Nh}{15000}$	S FT/MIN	S×IMEP	$\frac{\dot{Q}}{L^2}$	Ma #AIR/SEC.	$\frac{Ma}{L^2}$
A-1	32	4.180	134.0	11.32	40.6	7.85	28.2	19.17	68.8	1.15	900	62,000	21.43	~~.00625~~	
A-2	29.5	6.445	190.0	20.8	74.7	8.25	29.6	29.05	104.3	2.325	1200	125,300	30.40		
A-3	32.9	4.188	138.0	10.45	37.5	8.87	31.8	19.32	69.4	1.545	1200	83,300	22.07		
A-4	43.5	2.908	126.5	3.2	11.5	9.2	33.0	12.4	44.6	.992	1200	53,500	20.25		
A-5	28	4.750	133.0	21.4	76.8	6.9	24.8	28.3	101.6	2.717	720	73,100	21.30	.00525	.000840
A-6	28	4.750	133.0	3.5	12.6	9.2	33.0	12.7	45.6	2.030	1200	54,700	21.30	.00513	.000821
A-7	32	4.780	153.0	8.67	31.1	8.87	31.8	17.54	63.0	2.810	1200	76,600	24.50	.00612	.000980
A-8	39	4.819	188.0	19.53	70.2	8.25	29.6	27.78	99.7	4.445	1200	119,600	30.1	.00891	.00143
A-9	38.6	6.290	242.7	13.5	48.5	10.15	36.4	23.65	85.0	5.676	1800	153,000	38.85	.01274	.00204
A-10	45	4.175	188.0	4.65	16.7	10.72	38.5	15.37	55.2	3.694	1800	99,450	30.1	.00881	.00141
A-11	49	4.315	211.5	17.25	62	9.0	32.3	26.25	94.3	5.038	1440	135,800	33.85	.01063	.001702
A-12	41	4.032	165.4	7.02	25.2	9.6	34.5	16.62	59.7	3.188	1440	86,000	26.46	.00720	.00115
A-13	20	4.58	91.6	10.8	38.8	7.35	26.4	18.15	65.2	1.453	600	39,100	14.67	.00300	.00048
A-14	23	4.23	97.3	15.45	55.5	6.9	24.8	22.35	80.3	1.787	600	48,100	15.58	.003725	.000596
A-15	35.9	3.02	108.4	21.2	76.1	6.64	23.8	27.84	99.9	2.224	600	59,940	17.35	.0045	.000721
Ai	50	3.51	175.5	17.9	64.25	8.35	30.0	26.25	94.25	2.100	1200	113,200	28.1	.00864	.00138

NOTE: RUNS A-1 THRU A-4 NOT USED DUE TO INCOMPLETE VAPORIZATION OF FUEL CAUSED BY LOW VAPORIZING TANK TEMPERATURE.

TABLE F IX

HEAT REJECTION CALCULATIONS - 4" GSE

$\ell^2 = 16$

RUN	ΔT_{H_2O} °F	ROTAMETER READING	W_{H_2O} #/MIN	$\dot{Q}$ BTU/MIN	h_{BRAKE} "Hg	BMEP	$h_{FRICTION}$	FMEP	$h_{INDICATED}$	IMEP	IHP	S	(S)(IMEP)	$\dot{Q}/\ell^2$	Ma #AIR/SEC	Ma/ℓ^2
B 1	60	32.5	7.615	457	27.46	90.00	7.65	25.10	35.1	115.10	13.16	1200	138200	28.6	.02295	.001432
2	51	30.2	7.390	376.5	15.25	50.00	8.25	27.05	23.5	77.0	8.81	1200	92500	23.55	.01534	.000960
3	38.5	42.5	8.525	328.2	27.40	89.90	6.55	21.50	33.95	111.40	7.64	720	80250	20.50	.01355	.000846
4	68	30.8	7.455	505.5	24.93	81.75	8.75	28.70	33.68	110.45	15.84	1500	165600	31.60	.02730	.001706
5	55.5	34.7	7.815	434.5	14.25	46.75	9.06	29.70	23.31	76.45	10.92	1500	114 700	27.10	.01900	.001188
6	43.7	34.5	7.800	341.0	3.47	11.4	9.45	31.0	12.92	42.40	6.06	1500	63600	21.30	.0115	.000719
7	53.5	50.0	9.135	488.5	14.20	46.6	10.40	34.1	24.6	80.70	13.84	1800	145200	30.50	.0240	.001500
8	59.3	44.3	8.675	514	24.87	81.7	8.74	28.7	33.61	110.40	15.75	1500	165800	32.10	.02658	.001660
9	38.0	33.6	7.715	293	26.75	87.8	6.30	20.65	33.05	108.45	6.20	600	65150	18.30	.0109	.000681
10	44.0	32.5	7.615	335	26.875	88.2	6.50	21.35	33.375	109.55	7.51	720	79,000	20.90	.0130	.000812
11	63.0	46.9	8.885	560	22.85	75	10.0	32.8	32.85	107.8	18.48	1800	193800	35.00	.0320	.00200
12	61.5	51.75	9.280	571	20.10	66	11.2	36.7	31.30	102.7	19.56	2000	205400	35.70	.03387	.002118
13	52.0	48.05	8.975	466.5	11.15	36.6	10.4	34.1	21.55	70.8	12.12	1800	127500	29.10	.02127	.01330
14	46.0	39.50	7.80	359	25.35	83.2	7.0	22.95	32.35	106.1	9.09	900	95,500	22.40	.01546	.000966
15	30	34.85	7.832	235	16.60	54.4	6.86	22.50	23.46	76.95	4.4	600	46170	14.70	.0082	.000512
Bi	68.5	12.76	5.89	404	24.7	81.0	7.75	25.45	32.45	106.4	12.17	1200	127300	25.25	.02130	.001330
16	26.5	29.95	7.37	195.3	9.2	30.2	7.55	24.75	16.75	54.94	3.14	600	32964	12.20	.00596	.000366
17	36	30.35	7.40	266.5	21.90	71.8	6.57	21.53	28.47	93.40	5.34	600	56040	16.65	.00975	.000609

TABLE F-X

HEAT REJECTION CALCULATIONS - 6" G.S.E.

$\ell^2 = 36$

RUN		ΔT_{H_2O}	ROTAMETER READING	W_{H_2O}	$\dot{Q}$	h_{BRAKE}	BMEP	$h_{FRICTION}$	FMEP	$h_{INDICATED}$	IMEP	IHP	S	(S)(IMEP)	$\dot{Q}/\ell^2$	Ma	Ma/ℓ^2
		°F		#/MIN	BTU/MIN	"Hg	$3.89 h_B$	← FROM CURVES →	$3.89 h_F$	$h_F + h_B$	$3.89 h_{IND}$	$\frac{NH}{1000}$	FT/MIN			#AIR/SEC	
C	1	44	31.6	15.42	679	22.8	88.10	6.63	25.8	29.43	110.6	17.7	720	79500	18.89	.0284	.000789
	2	44	43.5	17.59	771.5	22.18	86.10	6.78	26.4	28.96	112.45	23.2	960	108000	21.45	.0387	.001075
	3	52	44.3	17.70	920	23.26	90.50	6.98	27.2	30.24	113.70	30.24	1200	141,100	25.60	.0496	.001378
	4	49	44.6	17.75	780	14.25	55.50	7.50	29.2	21.75	84.70	21.75	1200	101,700	21.70	.0356	.000990
	5	40	45.2	17.85	714	8.90	34.60	7.80	30.3	16.70	64.90	16.70	1200	77,850	19.85	.0275	.000764
	6	58	39.5	16.89	977	23.53	91.50	7.20	28.0	30.73	118.50	33.80	1320	157,600	27.15	.0558	.001550
	7	55	39.5	16.84	926	19.10	74.40	7.40	28.8	26.50	102.10	29.10	1320	136,200	25.70	.0478	.001328
	8	49	39.5	16.84	825	12.40	48.25	7.81	30.4	20.21	78.65	22.20	1320	109,000	22.90	.0356	.000987
	9	56	38.0	16.57	927	11.80	45.95	8.30	32.3	20.10	78.25	26.10	1560	122,000	25.75	.0425	.001178
	10	60	38.0	16.57	993	16.70	65.00	7.92	30.8	24.62	95.80	32.0	1560	149,400	27.60	.05275	.001461
	11	67	38.0	16.57	1109	23.80	92.50	7.58	29.5	31.38	122.00	40.7	1560	190,700	30.80	.0662	.001835
	12	71	39.0	16.75	1190	21.40	83.25	8.33	32.4	29.73	115.65	44.5	1800	208,000	33.01	.0736	.002090
	13	67	37.0	16.38	1096	16.30	63.45	8.61	33.5	24.91	96.95	37.3	1800	174,000	30.40	.0624	.001730
	14	60	37.0	16.38	982	11.20	43.55	9.10	35.4	20.30	78.95	30.4	1800	142000	27.25	.0487	.001339
	15	32	54	19.40	621	17.45	67.90	6.84	26.6	24.29	94.50	14.6	720	68000	17.25	.02351	.000653
	16	32	54.5	19.48	623	12.30	47.80	7.30	28.4	19.60	76.20	15.7	960	73100	17.30	.02589	.000718
	17	37	54.5	19.48	720	16.95	65.90	7.04	27.4	23.99	93.30	19.2	960	89600	20.00	.0322	.000893

OBSERVERS: WILKINSON & MITCHELL

TABLE F-XI

ASME STD. SQ. EDGE
ORIFICE DIA = .413"

SPARK ADVANCE 25° BTC.

HEAT REJECTION RUNS 2½ G.S.E.

TIME	RUN	RPM	PISTON SPEED FT/MIN	TEMPERATURES: CYL HEAD °F	INLET AIR °F	WATER TO ENGINE °F	MN BRNG °F	DYN. READ. "Hg	PRESSURES: (ΔP) AIR "H_2O	FUEL ROTA. READING	P_i INTAKE "Hg	P_e EXHAUST "Hg	P_o ORIFICE	$(\Delta P)_{H_2O}$ "Hg	TEMPERATURES: WATER FROM ENGINE °F	OIL FROM SUMP °F	MN BRNG INLET OIL °F	WATER TO ROTAMETER °F	WATER FROM COOLER °F	WATER ROTAMETER
1042-0	1	1440	720	335	161	140	154	21.2	3.7	5.4	-2.0	1.0	0	16.6	150	149	140	150	122	25.3
+10	A-5 2	1440	720	331	163	140	155	21.4	3.7	5.4	-2.0	1.0	0	16.6	150	152	142	150	122	25.4
+20	3	1440	720	335	160	140	155	21.4	3.7	5.4	-2.0	1.0	0	16.6	150	150	142	150	122	25.4
+30	4	1440	720	335	161	140	155	21.4	3.7	5.4	-2.0	1.0	0	16.6	150	150	142	149	121	25.6
1132-0	1	2400	1200	331	160	140	177	3.2	3.0	5.15	-13.90	1.0	0	16.6	150	149	140	150	122	25.3
+10	A-6 2	2400	1200	330	165	140	178	3.5	3.0	5.15	-13.80	1.0	0	16.6	150	152	142	150	122	25.4
+20	3	2400	1200	328	160	140	179	3.5	3.0	5.15	-13.90	1.0	0	16.6	150	150	142	150	122	25.3
+30	4	2400	1200	327	160	140	178	3.5	3.0	5.15	-13.90	1.0	0	16.6	149	150	142	149	121	25.6
1253-0	1	2400	1200	352	161	141	178	8.6	4.95	5.8	-10.0	1.0	0	16.6	151	149	141	151	119	26.0
+10	A-7 2	2400	1200	348	161	139	179	8.65	4.95	5.8	-10.0	1.0	0	16.6	149	152	143	149	117	25.7
+20	3	2400	1200	353	161	139	179	8.70	4.95	5.8	-10.0	1.0	0	16.6	150	153	144	150	118	25.8
+30	4	2400	1200	352	161	139	179	8.70	4.95	5.8	-10.0	1.0	0	16.6	150	151	142	150	118	25.75
1335-0	1	3600	1800	441	160	137	199	13.50	22.85	8.90	-2.10	1.0	0	16.6	153	152	145	153	114	42.8
+10	A-9 2	3600	1800	438	160	137	200	13.50	22.85	8.90	-2.10	1.0	0	16.6	150	149	143	154	115	42.6
+20	3	3600	1800	435	160	137	201	13.50	22.85	8.90	-2.10	1.0	0	16.6	153	153	127	153	115	43.0
1415-0	1	2400	1200	392	163	138	179	19.50	10.60	7.15	-2.0	1.0	0	16.6	151	151	143	151	111	25.8
+10	A-8 2	2400	1200	393	162	138	179	19.60	10.65	7.17	-2.0	1.0	0	16.6	151	151	143	151	112	26.2
+20	3	2400	1200	392	162	138	179	19.50	10.65	7.17	-2.0	1.0	0	16.6	151	151	143	151	112	26.2
1520-0	1	3600	1800	389	159	138	199	4.60	10.45	7.12	-10.0	1.0	0	16.6	151	150	120	151	106	18.5
+10	A-10 2	3600	1800	388	160	139	200	4.65	10.40	7.10	-10.0	1.0	0	16.6	152	151	124	152	107	18.6
+20	3	3600	1800	387	160	139	200	4.70	10.40	7.10	-10.0	1.0	0	16.6	152	150	123	152	107	18.6
1600-0	1	2880	1440	410	161	137	185	17.3	15.3	7.99	-2.0	1.0	0	16.6	151	146	137	151	102	20.0
+10	A-11 2	2880	1440	412	161	138	187	17.2	15.3	7.99	-2.0	1.0	0	16.6	152	154	143	152	103	20.2
+20	3	2880	1440	412	161	138	185	17.25	15.3	7.99	-2.0	1.0	0	16.6	152	150	142	152	103	20.2

1 APRIL 1952 WILKINSON & MITCHELL

TIME	RUN	RPM	PISTON SPEED	CYL HEAD	INLET AIR	WATER TO ENGINE	MN BRNG	DYN. READ.	(ΔP) AIR	FUEL ROTA.	P_i	P_e	P_o	$(\Delta P)_{H_2O}$	WATER FROM ENGINE	OIL FROM SUMP	MN BRNG INLET OIL	WATER TO ROTAMETER	WATER FROM COOLER	WATER ROTAMETER
1245-0	1	2880	1440	365	161	139	188	6.95	6.90	7.37	-10.0	1.0	0	16.6	150	150	142	150	109	17.0
+10	A-12 2	2880	1440	362	162	139	187	7.05	6.90	7.37	-10.0	1.0	0	16.6	150	149	141	150	109	16.9
+20	3	2880	1440	361	163	139	187	7.05	6.90	7.37	-10.0	1.0	0	16.6	150	147	140	150	109	17.0

ASME STD SQ EDGE ORIFICE - DIA = .310"

9 APRIL 1952 BAROMETER 753 MM Hg T = 22°C MITCHELL & WILKINSON.

TIME	RUN	RPM	PISTON SPEED	CYL HEAD	INLET AIR	WATER TO ENGINE	MN BRNG	DYN. READ.	(ΔP) AIR	FUEL ROTA.	P_i	P_e	P_o	$(\Delta P)_{H_2O}$	WATER FROM ENGINE	OIL FROM SUMP	MN BRNG INLET OIL	WATER TO ROTAMETER	WATER FROM COOLER	WATER ROTAMETER
1015-0	1	1200	600	288	160	142	151	10.8	3.7	4.15	-10.25	1.0	0	16.6	148	150	140	148	128	23.4
+10	A-13 2	1200	600	290	160	142	151	10.8	3.7	4.15	-10.30	1.0	0	16.6	148	150	140	148	128	23.3
+20	3	1200	600	290	160	142	151	10.8	3.7	4.15	-10.35	1.0	0	16.6	148	150	140	148	128	23.35
1302-0	4	1200	600	300	160	142	153	15.45	5.7	4.60	-6.45	1.0	0	16.6	148	151	142	148	124	17.5
+10	A-14 5	1200	600	300	160	142	152	15.45	5.7	4.60	-6.45	1.0	0	16.6	148	151	142	148	126	22.2
+20	6	1200	600	300	160	142	153	15.45	5.7	4.60	-6.45	1.0	0	16.6	148	152	142	148	125	19.2
+30	7	1200	600	300	160	142	153	15.45	5.7	4.60	-6.45	1.0	0	16.6	142	151	142	148	125	19.2
1420-0	8	1200	600	326	160	141	150	21.0	8.4	5.0	-2.15	1.0	0	16.6	141	148	140	149	112	3.1
+10	9	1200	600	324	160	141	150	21.2	8.4	5.0	-2.15	1.0	0	16.6	141	150	141	149	113	5.5
+20	10	1200	600	324	160	141	150	21.2	8.4	5.0	-2.15	1.0	0	16.6	141	150	141	149	113.3	5.6
+30	11	1200	600	324	160	141	150	21.2	8.4	5.0	-2.15	1.0	0	16.6	141	150	141	149	113	5.6

27 MARCH 1952
OBSERVERS: SMITH, BOHNER
BAROMETER 764.3 mm Hg
TEMP. 19.8°C

TABLE F-XII

HEAT REJECTION RUNS 4" GSE.

DYNAMOMETER IS 20 + READING LISTED

AIR ORIFICE ASME STD. D=.614"

	TIME	RUN	R.P.M.	PISTON SPEED FT/MIN.	TEMPERATURES °F: CYL. HEAD	TEMPERATURES °F: INLET AIR	TEMPERATURES °F: MAIN BRG.	DYN. READING " Hg	(ΔP) AIR " H_2O	PRESSURES IN INCHES Hg: P_i INTAKE	PRESSURES IN INCHES Hg: Pe EXHAUST	PRESSURES IN INCHES Hg: Po ORIFICE	PRESSURES IN INCHES Hg: ΔP H_2O	TEMPERATURES °F: H_2O INLET	TEMPERATURES °F: MAIN BRG INLET	TEMPERATURES °F: OIL FROM SUMP	TEMPERATURES °F: H_2O TO ROTAMETER	TEMPERATURES °F: H_2O FROM COOLER	ROTAMETER READING
	1153	1	1500	1200	438	151	200	+7.24	14.55	-2	1.0	0	8	142	152	150	153	88	32.6
	1158	2	1500	1200	440	152	201	+7.24	14.3	-2	1.0	0	8	144	152	150	153	87	32.8
	1205	3	1500	1200	435	151	203	+7.28	14.4	-2	1.0	0	8	140	152	150	150	89	32.6
B-1	1210	4	1500	1200	436	150	203	+7.3	14.6	-2	1.0	0	8	139	151	150	149	89	32.7
	1215	5	1500	1200	436	150	203	+7.4	14.6	-2	1.0	0	8.05	140	151	150	150	92	32.6
	1220	6	1500	1200	435	151	203	+7.5	14.6	-2	1.0	0	8.	140	151	150	150	90	32.5
	1230	7	1500	1200	434	150	203	+7.45	14.6	-2	1.0	0	8	139.5	151	150	150	90	32.4
	1320	8	1500	1200	377	150	199	-4.68	6.5	-10	1.0	0	8.05	140	150	150	150	102	30.2
	1347	9	1500	1200	376	150	199	-4.70	6.5	-10	1.0	0	8.05	142	150	150	151	98	30.2
	1356	10	1500	1200	374	151	198	-4.62	6.5	-10	1.0	0	8.05	140	150	150	150	99	30.1
B-2	1400	11	1500	1200	374	150	198	-4.72	6.5	-10	1.0	0	8.05	140	150	150	150	100	30.3
	1405	12	1500	1200	376	149	198	-4.8	6.5	-10	1.0	0	8.0	140	150	150	150	102	30.2
	1417	13	1500	1200	378	151	198	-4.82	6.5	-10	1.0	0	8.05	141	150	150	149.5	105	30.0
	1429	14	1500	1200	377	150	198	-4.89	6.5	-10	1.0	0	8.05	140	150	150	149.5	99	29.7
	1513	15	900	720	358	149	178	+7.4	5.0	-2	1.0	0	8.0	141	150	150	149.5	100	29.9
B-3	1550	16	900	720	360	148	177	+7.5	5.0	-2	1.0	0	8.0	141	150	150	149	112	43.1
	1605	17	900	720	359	150	178	+7.45	5.0	-2	1.0	0	7.98	141	150	150	149	111	43
	1610	18	900	720	360	150	178	+7.4	5.0	-2	1.0	0	8.0	141.5	150	150	149.5	111	42.5
	1618	19	900	720	359	151	177	+7.4	5.0	-2	1.0	0	8.0	141.5	150	150	149.5	111	42.5

2 APRIL 1952
OBSERVERS: MITCHELL, WILKINSON

	TIME	RUN	R.P.M.	PISTON SPEED FT/MIN.	CYL. HEAD	INLET AIR	MAIN BRG.	DYN. READING " Hg	(ΔP) AIR " H_2O	P_i INTAKE	Pe EXHAUST	Po ORIFICE	ΔP H_2O	H_2O INLET	MAIN BRG INLET	OIL FROM SUMP	H_2O TO ROTAMETER	H_2O FROM COOLER	ROTAMETER READING
	1022	20	1875	1500	421	160	210	+4.8	20.6	-3	1.0	0	18.0	139	148	148	152	85	30.7
	1032	21	1875	1500	421	160	213	+4.95	20.6	-3	1.0	0	18.0	139	152	152	151	83	31.0
B-4	1042	22	1875	1500	429	161	215	+5.0	20.65	-3	1.0	0	18.0	139	152	152	151	83	31.0
	1052	23	1875	1500	429	161	215	+4.95	20.65	-3	1.0	0	18.0	139	152	152	151	83	30.7
	1062	24	1875	1500	430	161	215	+4.95	20.65	-3	1.0	0	18.0	139	153	153	151	83	30.7
	1340	25	1875	1500	385	161	213	-5.80	10.00	-10	1.0	0	18.0	140	150	148	150	94.5	34.6
	1350	26	1875	1500	388	160	214	-5.70	10.00	-10	1.0	0	18.0	140	150	148	150	95	34.7
B-5	1400	27	1875	1500	385	160	214	-5.75	10.00	-10	1.0	0	18.0	140	150	150	150	94	34.7
	1415	28	1875	1500	386	160	214	-5.75	10.00	-10	1.0	0	18.0	140	152	150	160	94.5	34.8

TABLE F XIII

2 APRIL 1952
(CONT'D.)

HEAT REJECTION RUNS 4" G.S.E.

DYNAMOMETER ZERO AT 0" Hg

AIR ORIFICE ASME STD. DIA. = .614"

	TIME	RUN	RPM	PISTON SPEED FT/MIN.	TEMPERATURES °F: CYL. HEAD	INLET AIR	MAIN BRG	DYN. "Hg	(ΔP)AIR "H2O	FUEL ROTAMETER READING	PRESSURES INCHES Hg: Pi INTAKE	Pe EXHAUST	Po ORIFICE	(ΔP)H2O	TEMPERATURES °F: H2O INLET	MAIN BRG INLET OIL	OIL FROM SUMP	H2O TO ROTAMETER	H2O FROM COOLER	H2O ROTAMETER READING
	1540	29	1875	1500	336	162	210	3.5	3.6	5.9	-17.1	1.0	0	18	141	150	150	149	105	34.5
B-6	1550	30	1875	1500	325	162	210	3.45	3.5	5.85	-17.1	1.0	0	18	141	150	150	149	105	34.6
	1600	31	1875	1500	336	161	210	3.45	3.5	5.85	-17.0	1.0	0	18	141	150	150	149	106	34.4
	1625	32	2250	1800	412	160	225	14.25	15.85	10.8	-8.5	1.0	0	18	139	150	150	150	96	50
B-7	1635	33	2250	1800	412	160	225	14.2	16.0	10.8	-8.5	1.0	0	18	139	150	150	150	97	50
	1645	34	2250	1800	412	160	225	14.1	16.0	10.8	-8.5	1.0	0	18	139	150	150	151	97.2	50
	1026	35	1875	1500	438	160	215	24.8	19.6	11.75	-3.1	1.0	0	18	139	152	151	152	92	44.1
B-8	1038	36	1875	1500	440	160	215	24.9	19.6	11.75	-3.1	1.0	0	18	139	149	147	152	93	44.3
	1050	37	1875	1500	440	160	215	24.9	19.6	11.75	-3.1	1.0	0	18	139	150	149	152	93	44.6
B-9	1125	38	750	600	336	160	172	26.75	3.15	5.6	-1.9	1.0	0	18	142	150	150	149	111	33.6
	1140	39	750	600	336	160	172	26.75	3.15	5.6	-1.9	1.0	0	18	142	150	150	149	111	33.6
B-10	1226	41	900	720	356	160	177	26.85	4.60	6.53	-2.08	1.0	0	18	141	150	150	149	105	32.5
	1240	42	900	720	356	160	177	26.9	4.60	6.53	-2.08	1.0	0	18	141	150	150	149	105	32.5
	1322	43	2250	1800	450	160	225	22.85	28.2	13.64	-2.95	1.0	0	18	138	150	150	151	88	46.8
B-11	1332	44	2250	1800	450	160	225	22.85	28.2	13.64	-2.95	1.0	0	18	138	150	150	151	88	47.2
	1342	45	2250	1800	450	160	225	22.85	28.2	13.64	-2.95	1.0	0	18	138	150	150	151	88	46.7
B-12	1405	46	2500	2000	450	160	228	20.1	31.55	14.3	-3.3	1.0	0	18	138	151	149	152	90.5	57.7
	1415	47	2500	2000	450	160	228	20.1	31.55	14.3	-3.3	1.0	0	18	138	150	150	152	90.5	57.8
	1450	49	2250	1800	406	161	212	11.1	12.50	9.8	-10.2	1.0	0	18	139	152	152	150	98	47.9
B-13	1500	50	2250	1800	404	161	212	11.15	12.50	9.8	-10.2	1.0	0	18	139	150	150	150	98	48.2
	1515	51	2250	1800	404	161	212	11.20	12.50	9.8	-10.2	1.0	0	18	139	151	151	150	98	48.0
	1630	52	1125	900	364	160	178	25.3	6.65	7.6	-3.0	1.0	0	18	140.5	150	150	149	103	34.5
B-14	1640	53	1125	900	364	160	178	25.35	6.65	7.6	-3.0	1.0	0	18	140.5	150	150	149	103	34.4
	1650	54	1125	900	364	160	178	25.35	6.65	7.6	-3.0	1.0	0	18	140.5	150	150	149	103	34.5

4 APRIL 1952
OBSERVERS: MITCHELL, WILKINSON
BAROMETER 767.0 mm Hg
TEMP. 21.25 °C

OBSERVERS: MITCHELL WILKINSON

TABLE F-XIV

HEAT REJECTION RUNS 4" G.S.E.

DYNAMOMETER ZERO AT 0" Hg

AIR ORIFICE ASME STD. DIA. .461"

	TIME	RUN	RPM	PISTON SPEED FT/MIN	TEMPERATURES °F: CYL HEAD	TEMPERATURES °F: AIR INLET	TEMPERATURES °F: MAIN BRG	DYN "Hg	(ΔP)AIR "H_2O	PRESSURES INCHES Hg: P_i INTAKE	PRESSURES INCHES Hg: P_e EXHAUST	PRESSURES INCHES Hg: P_o ORIFICE	(ΔP)H_2O	TEMPERATURES °F: H_2O INLET	TEMPERATURES °F: MAIN BRG INLET	TEMPERATURES °F: OIL FROM SUMP	TEMPERATURES °F: H_2O TO ROTA	TEMPERATURES °F: H_2O FROM COOLER	H_2O ROTAMETER READING	FUEL ROTAMETER READING
B-15	1035	55	750	600	305	160	165	16.65	5.85	-8.0	1.0	0	18.0	143.0	149	150	148	118	34.85	4.52
	1050	56	750	600	304	160	165	16.6	5.85	-8.0	1.0	0	18.0	143.0	149	150	148	118	34.85	4.52
	1105	57	750	600	278	160	162	9.2	3.0	-13.35	1.0	0	18.0	142.5	149	150	147	119.5	30.35	3.29
B-16	1115	58	750	600	277	160	162	9.2	3.0	-13.35	1.0	0	18.0	143	148	149	147	120.5	29.95	3.29
	1130	59	750	600	277	160	162	9.2	3.0	-13.35	1.0	0	18.0	143	149	150	147	120.5	29.95	3.29
B-17	1210	60	750	600	320	160	166	21.9	8.25	-4.5	1.0	0	18.0	142	149	150	148	112	30.5	5.2
	1225	61	750	600	317	160	166	21.9	8.25	-4.5	1.0	0	18.0	142	149	150	148	112	30.5	5.2

4 APRIL, 1952
OBSERVERS: SMITH, BOHNER
BAROMETER 766.8 mm HG
TEMP. 21.5 °C

TABLE F-XV

ASME STD. SQUARE EDGED ORIFACE DIAM. .920"
DYNAMOMETER ZERO READING: 0" HG

HEAT REJECTION RUNS 6" G.S.E.

TIME	RUN	RPM	PISTON SPEED FT/MIN	TEMPERATURES IN °F: CYL. HEAD	INLET AIR	MN. BRNG. INLET OIL	MN. BRNG.	OIL FROM SUMP	DYN. READING	$(\Delta P)_{AIR}$ " H_2O	PRESSURES IN " HG: P_i INTAKE	P_e EXHAUST	$(\Delta P)_{H_2O}$	TEMPERATURES IN °F: H_2O TO ENGINE	H_2O FROM ENGINE	H_2O TO ROTA-METER	H_2O FROM COOLER	H_2O ROTA.	FUEL TEMP. °F	FUEL ROTA-METER	
1400	13	1300	1560	534	160	151	208	150	16.7	15.1	-6.5	1	17.4	140	150	150	90	38.0	86	8.35	
1405	14	1300	1560	533	161	151	208	160	16.7	15.1	-6.5	1	17.4	140	150	150	90	38.0	86	8.35	C 10
1410	15	1300	1560	533	160	151	208	150	16.7	15.1	-6.5	1	17.4	140	150	150	90	38.0	86	8.35	
1425	16	1300	1560	554	159	151	207	150	23.8	23.6	-2.2	1	17.4	139	151	151	84	38.0	86	10.8	
1430	17	1300	1560	554	159	151	207	150	23.8	23.6	-2.2	1	17.4	139	151	151	84	38.0	86	10.8	C 11
1435	18	1300	1560	559	160	151	207	150	23.8	23.6	-2.2	1	17.4	139	151	151	84	38.0	86	10.8	
1505	19	1500	1800	off scale	160	151	213	150	21.4	29.1	-2.7	1	17.4	138	152	152	81	39.0	86	12.1	
1510	20	1500	1800	off scale	160	151	213	150	21.4	29.2	-2.7	1	17.4	138	152	152	81	39.0	86	12.1	C 12
1515	21	1500	1800	off scale	160	151	213	150	21.3	29.2	-2.7	1	17.4	138	152	152	81	39.0	86	12.1	
1540	22	1500	1800	554	160	151	213	150	16.3	20.9	-6	1	17.4	139	151	151	84	37.0	86	10.1	
1545	23	1500	1800	556	160	151	213	150	16.2	21.0	-6	1	17.4	139	151	151	84	37.0	86	10.1	C 13
1550	24	1500	1800	556	159	151	213	150	16.4	21.0	-6	1	17.4	139	151	151	84	37.0	86	10.1	
1610	25	1500	1800	529	158	151	212	150	11.2	13.0	-10	1	17.4	140	150	150	90	37.0	86	7.6	
1615	26	1500	1800	524	158	151	212	150	11.2	12.9	-10	1	17.4	140	150	150	90	37.0	86	7.6	C 14
1620	27	1500	1800	524	160	151	213	150	11.2	12.9	-10	1	17.4	140	150	150	90	37.0	86	7.6	

11 APRIL, 1952
OBSERVERS: SMITH, BOHNER
BAROMETER 766.6 mm HG
TEMP. 23.3 °C

ASME STD. SQUARE EDGED ORIFACE DIAM. .614"

TIME	RUN	RPM	PISTON SPEED FT/MIN	CYL. HEAD	INLET AIR	MN. BRNG. INLET OIL	MN. BRNG.	OIL FROM SUMP	DYN. READING	$(\Delta P)_{AIR}$ " H_2O	P_i INTAKE	P_e EXHAUST	$(\Delta P)_{H_2O}$	H_2O TO ENGINE	H_2O FROM ENGINE	H_2O TO ROTA-METER	H_2O FROM COOLER	H_2O ROTA.	FUEL TEMP. °F	FUEL ROTA-METER	
1455	1	600	720	414	160	151	169	150	17.45	15.5	-6	1	17.4	142	148	148	116	54.0	81	2.95	
1500	2	600	720	415	159	151	170	150	17.45	15.4	-6	1	17.4	142	148	148	116	54.0	81	2.95	C 15
1505	3	600	720	415	159	151	170	150	17.45	15.4	-6	1	17.4	142	148	148	116	54.0	82	2.95	
1525	4	800	960	426	159	151	179	150	12.5	18.6	-9.9	1	17.4	142	148	148	116	54.5	82	3.4	
1530	5	800	960	428	160	151	180	150	12.3	18.6	-9.9	1	17.4	142	148	148	116	54.5	82	3.4	C 16
1535	6	800	960	428	160	151	180	150	12.3	18.6	-9.9	1	17.4	142	148	148	116	54.5	82	3.4	
1540	7	800	960	428	160	151	180	150	12.3	18.6	-9.9	1	17.4	142	148	148	116	54.5	82	3.4	
1600	8	800	960	444	161	161	180	150	17.0	28.6	-6.3	1	17.4	141	149	149	112	54.5	82	4.6	
1605	9	800	960	445	161	151	180	150	16.9	28.6	-6.3	1	17.4	141	149	149	112	54.5	82	4.6	C 17
1610	10	800	960	446	160	151	180	150	17.0	28.7	-6.3	1	17.4	141	149	149	112	54.5	83	4.6	
1615	11	800	960	445	160	151	180	150	16.9	28.6	-6.3	1	17.4	141	149	149	112	54.5	83	4.6	

2 APRIL, 1952
OBSERVERS: SMITH, BOHNER
BAROMETER 757.6 mm HG
TEMP. 22 °C

TABLE F-XVI

ASME STD. SQUARE EDGED ORIFACE DIAM. .920"
DYNAMOMETER ZERO READING: 0" HG

HEAT REJECTION RUNS 6" G.S.E.

TIME	RUN	RPM	PISTON SPEED FT/MIN	TEMPERATURES IN °F: CYL. HEAD	INLET AIR	MN. BRNG INLET OIL	MN. BRNG	OIL FROM SUMP	DYN. READING	$(\Delta P)_{AIR}$ "H_2O	PRESSURES IN "HG: P_i INTAKE	P_e EXHAUST	$(\Delta P)_{H_2O}$	TEMPERATURES IN °F: H_2O TO ENGINE	H_2O FROM ENGINE	H_2O TO ROTA-METER	H_2O FROM COOLER	H_2O ROTA.	FUEL TEMP. °F	FUEL ROTA-METER	
1005	1	600	720	446	160	151	168	150	22.4	4.2	-2	1	17.4	141	149	149	105	31.6	79	4.0	
1010	2	600	720	446	160	151	168	150	22.45	4.4	-2	1	17.4	141	149	149	105	31.7	79	4.0	
1015	3	600	720	445	159	151	168	150	22.8	4.3	-2	1	17.4	141	149	149	105	31.7	79	4.0	C 1
1020	4	600	720	446	160	151	168	150	22.75	4.4	-2	1	17.4	141	149	149	105	31.5	79	4.0	
1030	5	600	720	446	160	151	168	150	22.85	4.4	-2	1	17.4	141	149	149	105	31.7	79	4.0	
1300	6	800	960	468	160	151	180	150	22.1	8.05	-2	1	17.4	141	149	149	105	43.6	79	5.85	
1310	7	800	960	468	160	151	180	150	22.15	8.0	-2	1	17.4	141	149	149	105	43.5	80	5.85	C 2
1315	8	800	960	470	160	151	181	150	22.23	8.1	-2	1	17.4	141	149	149	105	43.6	80	5.85	
1325	9	800	960	470	160	151	181	150	22.25	8.1	-2	1	17.4	141	149	149	105	43.5	80	5.85	
1423	10	1000	1200	509	159	151	192	150	23.8	13.4	-2	1	17.4	140	150	150	98	44.0	80	7.8	
1429	11	1000	1200	511	160	151	192	150	23.1	13.3	-2	1	17.4	140	150	150	98	44.2	81	7.8	
1434	12	1000	1200	511	159	151	193	150	23.2	13.5	-2	1	17.4	140	150	150	98	44.5	83	7.8	C 3
1439	13	1000	1200	510	161	151	193	150	23.1	13.4	-2	1	17.4	140	150	150	98	44.5	83	7.8	
1445	14	1000	1200	509	161	151	193	168	23.1	13.4	-2	1	17.4	140	150	150	98	44.5	83	7.8	
1505	15	1000	1200	470	163	151	194	150	14.3	6.8	-8	1	17.4	141	149	149	105	43.5	83	5.3	
1510	16	1000	1200	471	160	151	194	150	14.3	6.8	-8	1	17.4	141	149	149	105	44.8	82	5.3	C 4
1530	17	1000	1200	470	160	151	194	150	14.2	6.85	-8	1	17.4	141	149	149	105	44.7	83	5.3	
1535	18	1000	1200	470	160	151	194	150	14.2	6.85	-8	1	17.4	141	149	149	105	44.4	83	5.3	
1605	19	1000	1200	441	160	151	195	150	8.9	3.9	-12	1	17.4	141	149	149	109	45.3	82	3.7	
1615	20	1000	1200	444	160	151	195	150	8.9	4.0	-12	1	17.4	141	149	149	109	45.3	82	3.7	C 5
1620	21	1000	1200	442	160	151	195	150	8.9	4.0	-12	1	17.4	141	149	149	109	45.1	83	3.7	
1625	22	1000	1200	443	160	151	195	150	8.9	4.0	-12	1	17.4	141	149	149	109	45.1	83	3.7	

4 APRIL, 1952
OBSERVERS: SMITH, BOHNER
BAROMETER 766.8 mm HG
TEMP. 21.5°C

TIME	RUN	RPM	PISTON SPEED FT/MIN	CYL. HEAD	INLET AIR	MN. BRNG INLET OIL	MN. BRNG	OIL FROM SUMP	DYN. READING	$(\Delta P)_{AIR}$ "H_2O	P_i INTAKE	P_e EXHAUST	$(\Delta P)_{H_2O}$	H_2O TO ENGINE	H_2O FROM ENGINE	H_2O TO ROTA-METER	H_2O FROM COOLER	H_2O ROTA.	FUEL TEMP. °F	FUEL ROTA-METER	
1125	1	1100	1320	539	158	151	193	150	23.4	17.0	-2	1	17.4	140	150	150	92	39.5	81	8.9	
1140	2	1100	1320	537	161	151	196	150	23.6	16.8	-2	1	17.4	140	150	150	92	39.5	81	8.9	C 6
1145	3	1100	1320	538	161	151	196	150	23.6	16.8	-2	1	17.4	140	150	150	92	39.5	82	8.9	
1205	4	1100	1320	516	160	151	197	150	19.1	12.4	-5	1	17.4	140	150	150	95	39.5	82	7.4	
1215	5	1100	1320	517	160	151	197	150	19.1	12.4	-5	1	17.4	140	150	150	95	39.5	83	7.4	C 7
1220	6	1100	1320	517	160	151	197	150	19.1	12.4	-5	1	17.4	140	150	150	95	39.5	83	7.4	
1240	7	1100	1320	480	159	151	200	150	12.4	6.8	-10	1	17.4	140	150	150	101	39.5	80	5.3	
1245	8	1100	1320	481	160	151	200	150	12.4	6.8	-10	1	17.4	140	150	150	101	39.5	80	5.3	C 8
1250	9	1100	1320	481	160	151	200	150	12.4	6.8	-10	1	17.4	140	150	150	101	39.5	81	5.3	
1320	10	1300	1560	507	159	151	209	150	11.8	9.85	-10	1	17.4	140	150	150	94	38.0	84	6.5	
1325	11	1300	1560	507	160	151	209	150	11.8	9.85	-10	1	17.4	140	150	150	94	38.0	84	6.5	C 9
1335	12	1300	1560	507	160	151	209	150	11.8	9.85	-10	1	17.4	140	150	150	94	38.0	84	6.5	

TABLE-F XVII

SUMMARY

HEAT REJECTION AT VARIOUS JACKET TEMPERATURES

2½" G.S.E.

RUN	ΔT °F	ROTA	W #/MIN	T_j °F	Q BTU/MIN	Q/L^2
1	50	11.2	3.51	145.5	175.5	28.05
2	43.5	20.75	4.36	120.2	189.8	30.35
3	29	48.8	6.95	100.5	201.5	32.2
4	51	7.65	3.2	163.6	163.2	26.1
5	31.85	30.1	5.14	170.5	164.0	26.2
6	29	33.5	5.39	196.5	156.1	25.0
7	29	45.2	6.54	125	189.9	30.32
8	27.75	51.8	7.3	100.5	202.5	32.4

4" G.S.E

RUN	ΔT °F	ROTA	W #/MIN	T_j °F	Q BTU/MIN	Q/L^2
1	68.5	12.76	5.89	145	404	25.25
2	68	11.75	5.81	165	395	24.7
3	59	20.6	6.54	185	386	24.1
4	52	36.75	8.01	134	416	26.0
5	43	58.55	9.83	119	423	26.4

6" G.S.E.

RUN	ΔT °F	ROTA	W #/MIN	T_j °F	Q BTU/MIN	Q/L^2
1	46	55.5	19.66	145	905	25.15
2	50	49.0	18.52	135	925	25.7
3	51	50.3	18.75	123.5	956	26.55
4	44	56.0	19.72	155	868	24.1
5	52	38.5	16.65	167	865	24.0
6	52	41.5	17.2	159	894	24.8

9 APRIL, 1952
OBSERVERS: WILKINSON, MITCHELL

ASME STD. SQ. EDGE ORIFACE DIAM. .413"
DYN. ZERO READING 0" HG

TABLE F - XVIII

HEAT REJECTION AT VARIOUS JACKET TEMPERATURES $2\frac{1}{2}$" G.S.E.

TIME	RUN	RPM	PISTON SPEED FT/MIN	TEMPERATURES IN °F: CYL. HEAD	INLET AIR	WATER TO ENGINE	WATER FROM ENGINE	MN. BRNG.	DYN. READING	$(\Delta P)_{AIR}$ " H_2O	PRESSURES IN " HG: P_i INTAKE	P_e EXHAUST	P_o ORIFACE	$(\Delta P)_{H_2O}$	TEMPERATURES IN °F: OIL FROM SUMP	MN. BRNG INLET OIL	WATER TO ROTAMETER	WATER FROM COOLER	WATER ROTAMETER	FUEL ROTAMETER
1530	1	2400	1200	405	160	139	152	175	17.9	10.0	-3	1.0	0	16.6	150	138	152	102	11.2	7.02
1545	2	2400	1200	405	160	139	152	175	17.9	10.0	-3	1.0	0	16.6	150	139	152	102	11.2	7.02
	3	2400	1200	375	160	114.5	127	174	17.95	10.0	-3	1.0	0	16.6	148	139	127	83	20.75	7.02
	4	2400	1200	375	160	114.0	126.5	174	17.95	10.0	-3	1.0	0	16.6	148	139	126.5	83	20.75	7.02
	5	2400	1200	375	160	114.0	126.5	174	17.95	10.0	-3	1.0	0	16.6	150	139	126.5	83	20.75	7.02
	6	2400	1200	362	160	94.5	107.5	175	18.0	10.0	-3	1.0	0	16.6	146	138	107.5	78	48.8	7.02
	7	2400	1200	362	160	94.0	107.0	174	18.0	10.0	-3	1.0	0	16.6	149	139	107	78	48.8	7.02
	8	2400	1200	362	160	94.0	107.0	174	18.0	10.0	-3	1.0	0	16.6	151	140	107	78	48.7	7.02

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OBSERVERS: WILKINSON, MITCHELL

TIME	RUN	RPM	PISTON SPEED FT/MIN	CYL. HEAD	INLET AIR	WATER TO ENGINE	WATER FROM ENGINE	MN. BRNG.	DYN. READING	$(\Delta P)_{AIR}$ " H_2O	P_i INTAKE	P_e EXHAUST	P_o ORIFACE	$(\Delta P)_{H_2O}$	OIL FROM SUMP	MN. BRNG INLET OIL	WATER TO ROTAMETER	WATER FROM COOLER	WATER ROTAMETER	FUEL ROTAMETER
	9	2400	1200	410	160	156	168	177	18.0	9.6	-3	1.0	0	16.6	148	140	168	116	7.5	6.95
	10	2400	1200	412	160	158	169.5	178	18.4	10.0	-2.65	1.0	0	16.6	148	140	169.5	118	7.65	7.02
	11	2400	1200	410	160	158	169	179	18.4	10.0	-2.65	1.0	0	16.6	151	141	169.0	119	7.65	7.02
	12	2400	1200	410	160	158	169.5	179	18.4	10.0	-2.65	1.0	0	16.6	150	142	169.5	118	7.65	7.02
	13	2400	1200	456	160	191	202	179	19.2	10.0	-2.3	1.0	0	16.6	150	193	202	173	33.0	7.02
	14	2400	1200	455	160	191	202	180	19.2	10.0	-2.3	1.0	0	16.6	150	193	202	173	34.0	7.02
	15	2400	1200	415	160	164	176	180	19.0	10.0	-2.55	1.0	0	16.6	150	142	176	143	29.85	7.02
	16	2400	1200	411	160	165	177	180	19.0	10.0	-2.55	1.0	0	16.6	150	142	177	145	30.05	7.02
	17	2400	1200	413	160	164.5	176.5	180	19.0	10.0	-2.55	1.0	0	16.6	150	142	176.5	145	30.2	7.02
	18	2400	1200	413	160	164.5	176.5	180	19.0	10.0	-2.55	1.0	0	16.6	150	142	176.5	144	30.3	7.02
	19	2400	1200	365	160	118	130	177	18.8	10.0	-3.0	1.0	0	16.6	148	140	130	101	45.0	7.02
	20	2400	1200	366	160	119	131	180	18.8	10.0	-3.0	1.0	0	16.6	150	141	131	103	45.1	7.02
	21	2400	1200	368	160	119	131	178	18.6	10.0	-3.0	1.0	0	16.6	150	142	131	102	45.2	7.02
	22	2400	1200	356	160	94.5	107.5	175	18.2	10.0	-3.25	1.0	0	16.6	150	142	107.5	80	51.7	7.02
	23	2400	1200	349	160	95.0	108	175	18.6	10.0	-3.20	1.0	0	16.6	150	142	108	80	52.0	7.02
	24	2400	1200	346	160	94	107	176	18.3	10.0	-3.1	1.0	0	16.6	148	140	107	79.5	51.8	7.02
	25	2400	1200	348	160	94	107	176	18.3	10.0	-3.1	1.0	0	16.6	149	141	107	79.0	51.8	7.02

OBSERVERS:
MITCHELL, WILKINSON

TABLE F-XIX

ASME STD. ORIFICE .614" **EFFECT OF JACKET TEMPERATURE ON HEAT REJECTION 4" G.S.E.** DYN. ZERO AT 0" Hg

	TIME	RUN	RPM	PISTON SPEED FT/MIN	CYL. HEAD	AIR INLET	MAIN BRNG	DYN "Hg	$(\Delta P)_{AIR}$	FUEL ROTAMETER READING	P_i INTAKE	P_e EXHAUST	P_o ORIFICE	$(\Delta P)_{H_2O}$	H_2O INLET	MAIN BRNG INLET	OIL FROM SUMP	H_2O TO ROTAMETER	H_2O FROM COOLER	H_2O ROTAMETER READING
	1104	1	1500	1200	388	160	184	24.7	12.55	9.81	-3.0	1.0	0	18	140	145	144	150	81.5	12.8
B-i	1114	2	1500	1200	390	160	185	24.7	12.55	9.81	-3.0	1.0	0	18	140	150	150	150	82.0	12.8
	1124	3	1500	1200	390	160	185	24.7	12.55	9.81	-3.0	1.0	0	18	140	150	150	150	81.0	12.7
B-ii	1146	4	1500	1200	408	160	190	24.9	12.50	9.80	-3.0	1.0	0	18	160	150	150	170	102	11.8
	1201	5	1500	1200	409	160	190	24.9	12.50	9.80	-3.0	1.0	0	18	160	150	150	170	102	11.7
B-iii	1246	7	1500	1200	416	160	190	25.1	12.40	9.78	-3.0	1.0	0	18	181	150	150	190	131	20.7
	1301	8	1500	1200	416	160	190	25.1	12.40	9.78	-3.0	1.0	0	18	180	150	150	189	130	20.5
	1345	11	1500	1200	369	160	188	24.65	12.48	9.80	-3.0	1.0	0	18	133	149	149	143	90.5	36.9
B-v	1400	12	1500	1200	366	160	188	24.5	12.40	9.78	-3.0	1.0	0	18	131	150	150	140.5	89	36.9
	1410	13	1500	1200	369	160	188	24.5	12.40	9.78	-3.0	1.0	0	18	129	150	151	139	87	36.7
	1420	14	1500	1200	369	160	188	24.5	12.40	9.78	-3.0	1.0	0	18	129	150	150	139	87	36.8
B-vi	1441	15	1500	1200	356	160	188	24.2	12.40	9.78	-3.0	1.0	0	18	114	150	150	124	81	58.5
	1456	16	1500	1200	356	160	188	24.2	12.40	9.78	-3.0	1.0	0	18	114	150	150	124	81	58.6

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OBSERVERS:
SMITH
BOHNER

TABLE F-XX

HEAT REJECTION AT VARIOUS JACKET TEMPERATURES 6" G.S.E.

TIME	R.P.M.	PISTON SPEED FT./MIN.	TEMPERATURE °F: FUEL	TEMPERATURE °F: INLET AIR	TEMPERATURE °F: MN. BR. INLET OIL	TEMPERATURE °F: CYL. HEAD	PRESSURE: DYN. READ. "HG	PRESSURE: (ΔP) AIR "H_2O	PRESSURE: P_i INTAKE "HG	PRESSURE: P_e EXHAUST "HG	PRESSURE: (ΔP) H_2O "HG	TEMPERATURE °F: OIL FROM SUMP	TEMPERATURE °F: OIL TO BEAR.	TEMPERATURE °F: H_2O ROTO. READ	TEMPERATURE °F: WATER TO ROTO	TEMPERATURE °F: WATER FROM COOLER	TEMPERATURE °F: WATER TO ENG.	TEMPERATURE °F: WATER FROM ENG.	FUEL ROTO	AVG. JACK. °F
1125	1000	1200	80	160	185	506	21.6	12.3	-3.0	1.0	17.4	151	151	55.5	149	103	140	149	7.4	144.5
1130	1000	1200	80	160	186	507	21.6	12.4	-3.0	1.0	17.4	150	151	55.5	149	103	139	149	7.4	144.0
1140	1000	1200	80	160	187	506	21.5	12.35	-3.0	1.0	17.4	150	150	55.5	149	104	140	149	7.4	144.5
1145	1000	1200	80	161	187	505	21.6	12.35	-3.0	1.0	17.4	151	150	55.5	150	104	140	150	7.4	145
1150	1000	1200	80	161	188	505	21.6	12.35	-3.0	1.0	17.4	151	151	55.5	150	104	140	150	7.4	145
1200	1000	1200	81	160	189	500	21.5	12.35	-3.1	1.0	17.4	148	148	49.5	142	94	132	142	7.4	137
1210	1000	1200	81	159	188	498	21.5	12.35	-3.1	1.0	17.4	150	150	49.0	141	91	131	141	7.4	136
1225	1000	1200	82	160	189	496	21.6	12.35	-3.0	1.0	17.4	152	152	49.0	140	90	130	140	7.4	135
1230	1000	1200	82	160	189	496	21.6	12.35	-3.0	1.0	17.4	150	150	49.0	140	90	130	140	7.4	135
1235	1000	1200	82	160	189	496	21.6	12.35	-3.0	1.0	17.4	150	150	49.0	140	90	130	140	7.4	135
1250	1000	1200	81	160	189	485	21.5	12.35	-3.0	1.0	17.4	150	150	50.3	129	78	118	129	7.4	123.5
1255	1000	1200	81	159	189	484	21.6	12.35	-3.0	1.0	17.4	150	150	50.3	129	78	118	129	7.4	123.5
1300	1000	1200	82	160	189	483	21.6	12.35	-3.0	1.0	17.4	150	150	50.3	129	78	118	129	7.4	123.5
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1115	1000	1200	78	160	186	526	21.0	12.35	-3.0	1.0	17.4	150	151	56	160	116	150	160	7.4	155
1120	1000	1200	78	160	186	525	21.1	12.35	-3.0	1.0	17.4	150	151	56	160	116	150	160	7.4	155
1125	1000	1200	78	159	187	523	21.1	12.35	-3.05	1.0	17.4	150	151	56	160	116	150	160	7.4	155
1130	1000	1200	79	159	188	524	21.0	12.35	-3.1	1.0	17.4	150	151	56	160	116	150	160	7.4	155
1210	1000	1200	82	160	190	535	22.0	12.35	-3.0	1.0	17.4	150	151	38.5	172	120	162	172	7.4	167
1215	1000	1200	82	160	190	532	22.0	12.35	-3.0	1.0	17.4	150	151	38.5	172	120	162	172	7.4	167
1220	1000	1200	83	160	192	534	22.0	12.35	-3.0	1.0	17.4	150	151	38.5	172	120	162	172	7.4	167
1245	1000	1200	83	162	191	525	22.0	12.35	-3.0	1.0	17.4	150	151	41.5	164	112	154	164	7.4	159
1255	1000	1200	83	161	191	525	22.1	12.35	-3.0	1.0	17.4	150	151	41.5	164	112	154	164	7.4	159
1300	1000	1200	83	160	191	525	22.1	12.35	-3.0	1.0	17.4	150	151	41.5	164	112	154	164	7.4	159

APPENDIX G

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