INTERNAL COOLING APPLIED TO A 12 H. P. GAS ENGINE

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GRADUATE SCHOOL.

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Professor C. M. McKergow,

Mechanical Engineering.

Dear Professor McKergow,

I send you herewith the thesis submitted by Cedric Ryan for the degree of Master of Science, for which you and Professor Durley have been appointed examiners. When you have read it, please forward it to Professor Durley for his examination. I shall be grateful for your joint report as soon as possible.

I am,

Yours very truly,

Cyrus Macmillan

Secretary of the Committee
on Graduate Studies.

I think this paper is worth about 65% as the material is practically all his own

CMcKergow

INTERNAL COOLING APPLIED TO A I2 H.P. GAS ENGINE.

Introductory.

The following investigation was carried out with a view to determining the feasibility of internal cooling as applied to internal combustion engines. By internal cooling, is meant the injection of water into the cylinder, by means of a mechanically operated valve or nozzle, the several jets striking the cylinder walls, valves, head and piston, and absorbing heat from them.

This investigation was prompted by accounts of failures in large gas engines, (See the Mechanical Engineer Feb. 6th., Feb. I3/I4) of which the majority were due to seized pistons, caused by failure of the cooling system. When in large gas engines, (I2" bore or over) the cooling system of the piston fails, the latter expands and seizes in the cylinder, generally wrecking the whole engine. Should the cylinder cooling fail however, less damage would be done, as there would be no tendancy for the piston to seize, and the failure would only result in a scorred cylinder at worst. In the small engine class, the piston is not water cooled, but delivers its heat to the cylinder walls, the latter being jacketed. When cooling fails in an engine of this type, seizure of the piston practically never occurs, because the cylinder and piston being of the same material expand at the same rate.

The object of internal cooling as defined above then, is to cool piston and walls from the inside, and in case of failure of the system, little or no damage will be done.

The application of this system is expected to be of great benefit to owners and builders of large gas engines, by first cutting down the initial cost of casting a jacketed cylinder, and by avoiding the elaborate piston cooling mechanisms. Futhermore, this system leads to greater safety from breakdowns, makes water cooling tanks unnecessary, and the amount of cooling water used is reduced to less than one per cent of that needed in present systems.

The objects of this investigation were:-

(a) To determine the quantity of water required for the safe operation of the engine.

(b) To determine the effect of the variation of the quantity of water on the temperature of the walls, on the thermal and mechanical efficiencies, and on the form of the indicator card.

(c) To compare the efficiencies of the engine with internal and jacket cooling.

This report will be divided under the following headings:-

Theory appertaining to internal cooling.

Description of engine selected for trials.

Preliminary trials with jacket cooling.

Final trials with jacket cooling and complete analysis.

Analysis of average indicator card.

Apparatus fitted for internal cooling.

Results of internal cooling.

Theory appertaining to internal cooling.

The fundamental principle on which internal cooling depends, lies in the fact that heat will flow faster from a metal to a liquid, than from a gas to a liquid. As an example, compare the efficiencies of a boiler and a superheater. This case is not an exact parallel however since the heat in the superheater flows from a metal to a gas, but illustrates the difficulty with which a gas absorbs heat. The efficiency of the superheater is as a rule only about one third of the efficiency of the boiler, the reason being that the water absorbs the heat from the metal much easier than the gas does. In the case of internal cooling heat flows from the hot gas to the cooling water on the walls, but not as fast as it does from the walls to the water.

When we inject water into the cylinder of a gas engine, the object, then, is to make it flow over the surface of the metal, thus absorbing heat from the walls. Should the water be injected in the form of a spray however, the cooling effect on the walls will be lost, since the spray will be partly evaporated in passing through the hot gas. The effect of this, as was found in the experimental trials, is to cut down the peak of the card to such a degree that the area of the card is diminished. The proper point in the cycle to inject the cooling water is during ignition, since the heat returned from the walls to the working fluid will increase the working pressure toward the end of expansion. The exact crank angle for opening, and the time of opening is an experimental determination, although partly governed by the practical consideration of lubrication. It is essential that the streams of water have the shortest possible distance to travel through the hot gas; and in view of lubrication, that as little as possible of the working surfaces be touched by the streams. For this reason the injection valve is opened about 20 degrees before dead centre and closes not later than 20 degrees after.

Description of engine selected for trials.

The engine chosen for these tests was a horizontal, four cycle, I2 H.P. Northey Gas Engine, of 8I/2" bore by I2" stroke. Governing is of the hit-and-miss type, the exhaust valve being held open during a miss, and the automatic inlet being locked in the closed position. Ignition originally was by hot tube, but a jump-spark system had been installed with a timer of very broad range which gave better service for experimental work. The fuel used on the tests was city gas with an average calorific value of 523 B.T.U. per. cu. ft. The mixing valve was of the simplest type, being nothing more than a flat check valve covering air and gas ports; the only adjustment being a valve on the gas main. A large rubber bag was inserted in the gas line between the engine and the gas meter to steady the flow of gas through the meter, and also aid the engine in getting gas on the suction stroke. A Crosby Gas Engine Indicator was connected by means of a water cooled jacket. Thermometer plugs were placed close to the jacket on water inlet and outlet, and the cooling water was weighed in a tank on scales. The brake used was made up of narrow hard wood blocks with a strip of leather belting screwed to them, the ends being fastened in a frame which rested on scales. This form of brake was found to be much superior to the ordinary form of rope brake since it was possible to obtain lighter loads, as well as a steadier regulation. During all trials a Junkers Calorimeter was used with a connection to the gas main beyond the meter.

Preliminary Trials with Jacket Cooling.

Since no results of the best working conditions of the engine were obtainable, an exhaustive series was run to determine those conditions. The variable conditions were 150° and 180° jacket outlet temperature, and different strengths of mixture. As to the latter, the only way to ~~e~~numerate conditions was by different openings of the gas valve, which had a pointer and a graduated segment. In the following sets, #8 signifies the gas valve to be wide open, and #5 to be half open. This sequence of numbers however does not bear out with the cubic feet of gas used per hour, on account of the varying calorific value of the gas. The results of the preliminary trials showed that the more efficient jacket temperature was 180°. It is possible that a higher jacket temperature would be still more efficient but is not regarded as practical on an engine of this size and type. Fig. 2 shows the thermal efficiency on a base of valve setting, and clearly demonstrates the advantage of the 180° jacket temperature over the 150°. Since the calorific value of the fuel was found to vary so much from day to day, ~~that~~ a more common base was chosen; that of B.T.U.'s per cubic foot of mixture, and curves of thermal efficiency at full load are compared. (see Fig. 3). To calculate the points for this last curve a volumetric efficiency had to be assumed, and was taken as 85%. This value was known to be approximately correct under the full load conditions, from a few specimen cards, although light spring cards were not regularly taken during the preliminary trials. Having completed the preliminary trials, the best running conditions were chosen and the final trials with jacket cooling were carried out.

Final Trials with Jacket Cooling and complete analysis.

For these trials a few extra appliances were fitted; a pyrometer, to measure the temperature of the exhaust gases directly under the exhaust valve, (for position see Fig. 12) and a sampling tube on the exhaust pipe near the engine, to collect the exhaust gas for analysis. The governor was detached, and the speed was regulated by the brake and a tachometer driven from the main shaft. The object in detaching the governor was to get the maximum possible power and efficiency out of the engine at normal speed, and to give a more accurate reading of the temperatures of the exhaust gas; also, to make it possible to get a better sample of the exhaust gas for analysis. During these trials the Junkers Calorimeter was operated continually and about twelve readings were taken per hour. Four thirty minute trials were run over the range of gas valves settings to insure a proper mixture being obtained at one of them, and the jacket temperature was kept constant at 180°.

Light spring indicator cards were taken at intervals during each trial to determine the volumetric efficiency. Specimen cards are shown in Fig. 4. On the light spring cards it will be noticed that the exhaust drops very suddenly, and in some cases to the atmosphere line; while in others the drop is less severe. This wave was at first attributed to inertia in the indicator, but a spring of double the strength was tried and the diagrams showed the same wave. The greatest wave occurs after the violent explosions, and the milder form after the weaker explosions. The first card in Fig. 4 at a 200 scale shows the series of explosions, and the first, second, and third light spring cards correspond to the high, medium, and light pressures. The sudden drop of the exhaust pressure, and the wave in the exhaust line, is caused by inertia of the exhaust gases in the exhaust pipe. Results bearing out this explanation were found in a two-stroke cycle, motor-boat engine. When running at a constant load and about 650 R.P.M., the cut-out on the exhaust pipe gave an increase in power, but at 800 R.P.M. and over, maximum horse-power was obtained with the exhaust passing through the muffler, and the use of the cut-out proved a disadvantage. This would seem to show that waves were set up in the exhaust pipe and muffler and could be used to advantage at speeds which gave the proper wave. Conditions as stated above however can only be found by experiment, as so much depends upon the design of the valves, ports, and obstructions in the exhaust line.

The results of the exhaust analysis which was made during the final trials are as follows;-

CO_2	4.96% by volume.
O_2	12.15% " "
CO	.066% " " "
N_2	82.824% " "

An analysis of the city gas was obtained from the Montreal Light Heat and Power Company and is as follows:- Coal Gas, 70% by vol. and Carbureted Water Gas 30% by vol.

	Coal Gas	Water Gas	Per. Cent of total
CO_2	2.6	5.4	3.45
C_2H_4	3.2	12.2	5.9
O_2	0.8	0.6	0.74
CO	8.0	27.4	13.83
CH_4	30.5	16.0	26.15
H_2	49.6	32.5	44.45
N_2	5.3	5.9	5.48

Gas analysis as a check on the excess air:-

Let a = % CO   By Burning:-

b = % CH_4 $2CO + O_2 = 2CO_2$

c = % C_2H_4 Volumes $I + I/2 = I$

d = % H_2 $CH_4 + 2O_2 = CO_2 + 2H_2O$

e = % O_2 " $I + 2 = I + 2$

f = % N_2 $C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$

g = % CO_2 " $I + 3 = 2 + 2$

$2H_2 + O_2 = 2H_2O$

" $I \quad I/2 \quad I.$

Therefore for a perfect mixture

Total $CO_2 = (a+b+2c+g) = A$

" $H_2O = (2b+2c+\frac{1}{2}d) = B$

A perfect mixture–additional oxygen $= (\frac{1}{2}a+2b+3c+\frac{1}{2}d-e) = C$

and nitrogen with additional oxygen $= 79/21\ C$

therefore total $N_2 = 79/21C+f$ ------ D

Let x excess air in cu.ft./cu.ft. of fuel gas.

Then total $O_2 = 0.21x$

" " $N_2 = D + 0.79x$ and exhaust gas in cu.ft./cu.ft. of fuel gas $= A + B + D + x$

Therefore in analysis of exhaust gas $CO_2 = \dfrac{A}{A+D+x}$ ---(I)

$$O_2 = \frac{0.21x}{A+D+x} \quad \text{---(2)}$$

$$N_2 = \frac{D+0.79x}{A+D+x} \quad \text{---(3)}$$

By substituting values from the above exhaust gas analysis the x of equations (I) gives 3.57 cu.ft./cu.ft. fuel gas.

"	x "	"	(2)	"	3.53	"	"	" "
"	x "	"	(3)	"	3.46	"	"	" "

Average value $= 3.52$. This value is probably the most accurate as it is practically the same as the result of equation (2), which is worked from the O_2 analysis of the exhaust gas.

The theoretical air needed for perfect combustion is obtained as follows:-

C_2H_4	5.9% by vol.,	Vols. req'd.=	3	vol. of	O_2 =	17.7	%
CO	13.83 "	" "	1/2	"	"	6.91	
CH_4	26.15 "	" "	2	"	"	52.3	
H_2	44.45 "	" "	1/2	"	"	22.23	

Total O_2 req'd. =99.i4

Then theoretical air $= \dfrac{99.14}{21} = 4.71$ cu.ft./cu.ft. fuel gas.

Then total air per stroke $= 4.71 + 3.52 = \underline{\underline{8.23}}$ cu.ft./cu.ft.gas.

By light spring diagram:-

Gas/hour = 220 cu.ft. explosions/minute = 104.5

temp. mixture entering cylinder = $67.2^\circ F$

Cu.ft. gas/exp. = 0.035 Specific vol. of gas = 30 cu.ft./lb. at 32°, or 32.15 cu.ft. at $67.2^\circ F$. Therefore weight of charge = 0.0019 lbs. Temp. of exhaust gases by pyrometer = $807^\circ F$

Vol. of exhaust gases at this temp. = clearance + 9.65% of displacement. = 0.1214 + 0.0388 = 0.1602 cu.ft.

Vol. of mixture taken in = Volumetric Efficiency × Displacement. = $0.8599 \times 0.4025 = 0.3460$ cu.ft. Hence vol. of air taken in = 0.3460 - 0.035 = 0.311 cu.ft.

Then total air ~~per. stroke~~ = $\underline{\underline{8.88}}$ cu.ft. per. cu.ft. of fuel gas.

The small ~~dis~~agreement in these figures may be accounted for by inability to measure the correct volumetric efficiency, either through personal error or error in the indicator. It is also doubtful as to the accuracy of the exhaust gas analysis. A D'Orsat apparatus was used and had been freshly charged, giving very constant results. The analysis of the fuel as furnished by the company was quite possibly an average value, as the calorific values are seen in the results of preliminary trials to vary from 622 B.T.U. to 498 in the final trials.

The calorific value by analysis is as follows:-

	% by vol.		B.T.U./cu.ft.		
C_2H_4	5.9	×	1600	=	9440
CO	13.83	×	339	=	4690
CH_4	26.15	×	960	=	25100
H_2	44.45	×	795	=	13100
					523.3 B.T.U./cu ft.

at atmospheric pressure and 32°F.

At 60° for comparison with values obtained with calorimeter,

calorific value $= 523.3 \times \frac{492}{520} = 495$ B.T.U./cu.ft.

Analysis of Average Indicator Card.

From the final trials with jacket cooling, twelve cards were selected at random, and the average ordinates plotted to an enlarged scale. The points of beginning of compression and beginning of suction were averaged similarly from light spring cards. The temperature at beginning of compression was calculated as follows:-

Inlet mixture temp.	67.2°F
Volumetric Efficiency	85.9%
Clearance volume	0.1214 cu.ft.
Displacement	0.4025 "
Gas per. hour	220 "
Explosions per. minute	104.5

Cu.ft. gas per. explosion = 0.035 cu.ft.

Specific vol. of gas at 67.2° = 32.15 " /lb.

Hence weight of gas per. suction stroke = 0.0019 lbs.

Average temp. of exhaust gas by pyrometer (for position see Fig. 12) = 807°F. Vol. of exhaust gas at this temp. = 0.1602 cu.ft.

Vol. of mixture taken in = Vol. Effy. × Displacement = 0.3460 cu.ft.

Sp. heat of exhaust gases taken as .24

Vol. of air taken in = 0.3460 − 0.035 = 0.3II cu.ft.

Hence lbs. of air taken in = 0.0235 lbs.

Assuming the exhaust gases to be of the same density as air their weight would then be $= \dfrac{520 \times 0.1602}{1267 \times 13.09} = 0.00503$ lbs.

Let W_m = weight of mixture including burnt gas trapped at beginning of compression.

W_1 = Wt. of fuel gas, and S_1 its specific heat.

W_2 = " " air, " S_2 " " "

W_3 = " " burnt gas, S_3 " " "

Then $W_m = W_1 + W_2 + W_3$. Let t_m, t_1, t_2, t_3 = corresponding temps. abs.

and $W_m\ t_m\ S_m = W_1 \times t_1 \times S_3 + W_2 \times t_2 \times S_2 + W_3 \times t_3 \times S_3$

To find S_m:- $W_m S_m = W_1 S_1 + W_2 S_2 + W_3 S_3$

therefore $S_m = 0.235$

Hence t_m <u>654.9°F abs.</u>

$$W_m = \begin{cases} 0.0019 \\ 0.0235 \\ 0.005 \end{cases}$$
$$\overline{0.0304}\ \text{lbs.}$$

By setting off the clearance volume at absolute pressure the distance from zero to minus fifteen may be taken to represent 650°F abs., and hence a temperature scale may be plotted. By drawing radial lines from absolute pressure and zero clearance to the actual pressure scale, actual temperature points will be found by dropping perpendiculars from a horizontal pressure line, to the radial line from the same pressure. As a comparison, and to determine the manner of heat flow, to, or from the walls, adiabatic curves of temperature and volume are shown on the same figure. These adiabatics are taken from the theoretical curves at variable specific heats shown in Fig. 1.

The maximum temperature obtained was 2170 ° Fah. absolute, and was found to drop during the expansion to 1800°F. abs. The corresponding pressure drop~~ped~~ starting at 160 lbs./sq.in. lies most of the way under the actual pressure drop, showing signs of after-burning. It is possible that a pressure line starting at a lower pressure, say 140 lbs./sq.in. would lie above theactual, as one tested from 170 lbs./sq.in. fell very much below the actual all of the way. The adiabatic pressure line plotted lies nearest to the actual. Of the compression curves, it will be noticed that the actual rises from the adiabatic very suddenly towards head end, and is explained by the fact that the cylinder head, clearance volume, and piston, were incrusted with carbon deposits which hindered the heat flow through the walls. Hence, when the piston was slowing up towards head end, heat was returned to the compressed mixture, raising its temperature and pressure.

A Temperature Entropy diagram was constructed from the Average Card to show also, the manner of heat flow during the different cycles. Owing to the number of stages of calculation and measurements necessary to obtain this diagram, it was not thought practicable to obtain from it numerical values for heat flow in the different stages. The diagram shows however, that during compression, (a to b) heat is being lost to the walls, or by leakage. This probably explains the actual temperature curve for compression in fig. 6, which it will be noticed, lies below the adiabatic. From c to d on the expansion line, heat is being supplied. This effect is shown clearly on both diagrams.

As a demonstration of the effect of variable specific heats, and to determine the value of $\underline{n}$ in the equation $PV^n = K$, the exponents of P and V were plotted from the actual expansion curve. (See Fig. 7)

$$P_1V_1^n = P_2V_2^n \text{ therefore } n = \frac{\log.P_1 - \log.P_2}{\log.V_2 - \log.V_1} = \text{slope of curve.}$$

It will be noticed that the slope of the curve becomes steeper as the volume increases, meaning that the value of $\underline{n}$ becomes less, and that the effect of the temperature on the specific heats is also decreasing. A dotted line is shown and represents approximately the average value for $\underline{n}$ during expansion.

Very low values for $\underline{n}$ were obtained, varying from 1.175 at the beginning of expansion, to 1.025 at the end of expansion. These low values are probably due to the large clearance volume, and low compression.

Apparatus fitted for Internal Cooling Trials.

Owing to the design of the engine used for the trials, it was found to be difficult to design the apparatus necessary for internal cooling. Had a cam shaft along the side of the cylinder been available, and the design of the head similar to moderh design, the problem would have been much simplified, since it would have been possible to install a more compact apparatus. Under the prevailing conditions however, a small water pump (See Fig. 10) was fastened to the frame of the engine near the main bearings, and driven by the exhaust cam. The pump discharged into an accumulator, which was fitted with a water glass and pressure gauge. (See Fig.11) The accumulator fed directly to theinjector valve, which was placed in the head of the cylinder. (Fig.8) The injector valve was placed through the water jacket of the head and held in place by a shoulder butting against the inner wall, and a follow-nut in the outer wall. The operating lever was held in a bracket fastened under the cylinder-head bolts. The timing of the injector valve was accomplished by the mechanism shown in Fig.9. A pin placed in the exhaust rod, operated through a finger, the contact lug. In the figure, the contact lug is shown in the release position. The exhaust valve is being held open by the governor, and the pin on the exhaust rod holds the lug in such a position, that the cam lever will not operate it. When the exhaust valve closes previous to an explosion, the contact lug is thrown into position, and is hit by the cam-lever. A steel wire connects the contact lug and the operating arm of the injector valve. The cam which operates the valve is placed on the face of the exhaust cam, and can be adjusted for any angle of opening. A small blow-off valve is placed in the bottom of the accumulator and serves as an adjustment to the water pressure of the accumulator.

Results of Internal Cooling Trials.

All parts of the apparatus fitted for the internal cooling trials were made in the University Shops, and were not completed until late in March. These parts were assembled, and the engine was first started on March 28th. It then being so late in the term time was not available to procure a complete set of trials which would fulfillthe objects of the investigation as stated in the introductory. In the trials which were run, however, the engine worked well uhder all loads, and it was found to be easy to regulate the temperature of the walls.

The part of the apparatus from which most trouble was expected ~~from~~ was the spray nozzle. The first nozzle was made in the form of a cap, with a hollow ring into which the water flowed after passing under the valve. Small holes, 10 in number, were drilled into the ring with a no. 60 drill, and positioned in such a manner, that three jets would hit the cylinder head, three the piston, and four the barrel. One of the latter was directed into the exhaust port and hit the wall just above the exhaust valve.

The temperature was at first regulated by keeping the water pressure in the accumulater at about 180 lbs./sq. in and adjusting the amount of opening of the injector valve. A better manner of adjustment was found later by throttling the water supply to the injector valve, and keeping pressure and valve opening constant. With this latter method less water consumption was noticed, and at the same time the indicator cards increased in size. The explanation of this fact was found after the trial, when the cylinder head was backed off several inches and the action of the jets could be seen. By throttling the water supply, the jets were noticed to be less violent, and with a proper degree of throttling could be made to just reach the walls without splashing back from them. When the flow was open, however, the jets were very fast, and would break into spray after hitting the walls. In this way, the greater part of the water injected was being evaporated by the gas directly, and not absorbing heat from the walls, while with the throttling adjustment, the spray was cut down and the water would run down the sides of the walls, having a greater cooling effect. As a remedy for the "too-vicious" jets, a plug was placed in the water line with a small hole drilled in it. The size of the hole was found by experiment, and gave the same effect as throttling, allowing a range of pressure in the accumulator from 15 to 100lbs./sq.in. A new spray nozzle was then fitted, having only seven holes in it instead of ten. Two jets were directed into the cylinder head quite high up and two on the piston, while the remaining three hit the barrel and exhaust port. During all of these trials, the cam governing the time of admission was set to open on dead centre. Finding that the engine worked well under these conditions, and that a full opening of the valve continued for 40°, it was thought advisable to advance the operating cam to give an opening 20° ahead of dead centre; thereby shortening the average distance for streams to travel to cool the piston. With this final adjustment the following ~~tria~~ results were obtained.

Cubic Feet of Gas per. hour ----375

Calcrific Value of Gas ----------523.5 (B.T.U./cu.ft.)

Ignitions per. Minute -----------104.2

Mean Effective Pressure ---------74.3 lbs./sq.in.

Indicated Horse Power -----------13.4

Brake Load -----------------------100 lbs.

Brake Horse Power ----------------10.1

Mechanical Efficiency -----------75.5%

Thermal Efficiency (I.H.P.) ----17.4%

" " (B.H.P.) ----13.2%

Cooling Water/I.H.P./hour -------2.85 lbs.

Temperature of Cylinder Head ---275°F (1/8" from inner wall)

Specimen cards for this trial are shown in Fig. 13. The increased power developed under full load conditions is explained by the light spring cards, which show a marked increase in volumetric efficiency. Developing maximum horse power with jacket cooling and with the governor in operation the volumetric efficiency averaged 85%, while with internal cooling an efficiency of 96% was obtained. The overall thermal efficiency is only bettered in one instance in the preliminary trials which makes it seem possible that higher efficiencies might be obtained in a complete series of tests.

As to the conditions of the engine after being cooled by this system it was found that the cylinder was kept very clean; in fact all deposits which had been on the face of the piston and clearance area at beginning of trials absolutely disappeared after a short run. As a test of the effect of this system on the condition of the barrel and clearance area, the engine was given an excess of oil and run without water for a few minutes, after which the surfaces were found to be encrusted with a carbon deposit. On starting up again the jet was turned on and after a short run the head was again removed and all traces of deposit had disappeared. The working surface of the barrel, however, was still smeared with oil showing that lubrication was not affected in the least.

The temperature of the cylinder walls could be governed easily by altering the ammount of opening of the injector valve. Under light loads the temperature could be kept as low as 150°F, but full load could not be obtained with a temperature less than 275°F. Temperatures as high as 350°F were obtained and had no bad effects on the working conditions of the engine. No trials were taken at this temperature however as the paint was beginning to burn off the cylinder. It was also noticed that no preeignitions occured when the spark was cut out and the gas left on.

On starting the engine, the injector valve should not be in operation, as very little water will stop it when cold. When above 275°F however, excess water had no effect on the burning of the mixture. No trouble was experienced with water on the spark plug since it was placed at the bottom of a deep port in the cylinder head and was not in the direction of a jet.

Photographs are included in this report which show the engine and its equipment.

As far as it was possible to carry this investigation, and with the results obtained, it would seem that a very valuable field had been opened for further experiment. Up to this time, very little work has been done on internal cooling, and the only trials of a similar nature were carried out by Professor Hopkinson at Cambridge, England. His experiments have been so successful that an engine with a jacketless cylinder is now in operation at Cambridge. (A description of this engine will be found in "Internal Combustion Engineering", March 1914). When this investigation was commenced, a very incomplete report of Professor Hopkinsons trials had just been published in the "Mechanical Engineer", and the only figures given were for water consumption. The values given were 2.4 lbs./H.P./hour. This value was made use of in designing the apparatus used.

As a plan for further investigation, I would recomend a complete series of tests to be run, to determine the most efficient temperatures and adjustments. It would also be possible to experiment with a new cycle of operation that of having a firing stroke followed by a stroke in which steam generated by the heat of the walls was the working fluid This system could probably be developed to advantage in a two-stroke cycle engine. A series of such tests would be of great advantage to the engineering world and the possibilities in this direction are great.

Results of Preliminary and Final Trials.

Index No.	
1	Trial Mark
2	Valve Setting
3	Duration of trial (minutes)
4	Gas per. hour (cu.ft.)
5	Revolutions per. min.
6	Jacket Temp, inlet (F^{0})
7	" " outlet "
8	Explosions per. min.
9	Mean Effective Pressure (lbs./sq.in.)
10	Indicated Horse Power
11	Effective Brake Load (lbs.)
12	Brake Horse Power
13	Mechanical Efficiency (%)
14	Gas per. B.H.P. per. hour (cu.ft.)
15	Calorific Value of Gas (B.T.U's/cu.ft.)
16	Thermal Efficiency, on the B.H.P. (%)
17	" " " " I.H.P. "
18	Efficiency Ratio (%)
19	Temp. of Inlet Gas
20	" " " Air
21	" " Exhaust Gas
22	Volumetric Efficiency (%)
23	Weight of Cooling Water (lbs.)

Index No.

1	P,1							
2	8							
3	30							
4	325	264	201	73.4	288	226	180	66
5	205.8	205.8	207.5	206.8	204.4	206.7	207.6	206
6	53.5	54	54	56.3	53.8	54.7	59	61.4
7	158	153	151	151	179	182	185	177
8	95.8	76.1	56.2	16.5	85.8	67.8	50.5	14.8
9	73.1	72.4	76.6	85.5	76.1	81.7	77	82.3
10	12.1	9.53	7.44	2.44	11.3	9.58	6.73	2.1
11	100	75	50	0	100	75	50	0
12	9.05	6.79	4.56	0	9.00	6.82	4.56	0
13	74.7	71.2	61.3	0	79.7	71.1	67.7	0
14	35.9	39.9	44.1	0	32.0	33.1	39.5	0
15	622							
16	11.3	10.5	9.29	0	12.8	12.3	10.4	0
17	15.2	14.7	15.1	13.6	16.0	17.3	15.3	13.0
18	13.5	12.5	11.0		12.8	12.3	10.4	
23	768	482	436	143	388	324	266	84

Index No.								
1	P,2							
2	7							
3	30							
4	289	227	171	90	260	208	180	66
5	207	205	206	209	204	205	206	205
6	54	54	54	54	54.7	52.7	52.4	53
7	146	147	150	153	178	180	182	178
8	86.8	67.5	49.7	21.3	89.6	72.9	53.9	16.5
9	74.5	77.2	79.5	82.5	66.6	68.6	70.7	77.5
10	11.17	9.00	6.84	3.04	10.3	8.65	6.58	2.21
11	100	75	50	0	100	75	50	0
12	9.1	6.76	4.54	0	8.87	6.71	4.50	0
13	81.6	75.1	66.4	0	97.3	78.4	68.5	0
14	31.8	33.6	37.7	0	28.9	30.7	39.8	0
15	575							
16	12.8	12.2	11.0	0	15.3	14.4	11.0	0
17	15.8	16.2	16.5	13.8	17.5	18.4	16.1	14.8
18	15.2	14.5	13.0		15.5	14.3	12.3	
23	564	469	366	304	298	320	223	37

Index No.								
1	P,3							
2	6							
3	30							
4	300	246	192	93	288	225	175	63
5	201	203	204	205	205	203	205	205
6	55.9	54.9	54.9	56.4	55.5	56.7	57.8	63.9
7	150	150	150	150	181	173	179	174
8	95	79.2	60	24.8	93.7	72.8	54.2	14.2
9	62.5	64.6	65.8	73.7	77.8	70.3	71.8	81
10	10.2	8.8	6.8	3.1	12.5	8.8	6.7	1.9
11	100	75	50	0	100	75	50	0
12	8.8	6.7	4.5	0	9.0	6.7	4.5	0
13	86.5	75.8	65.8	0	71.8	75.8	66.9	0
14	33.8	36.7	42.6	0	31.9	33.5	38.8	0
15	575	575	575	579	550	550	550	550
16	13.0	12.0	10.3	0	14.0	13.8	11.9	0
17	15.1	15.9	15.7	15.0	19.5	18.2	17.8	14.6
18	15.5	14.3	12.3		16.7	16.4	14.2	
23	549	458	382	216	340	259	300	228

Index No.

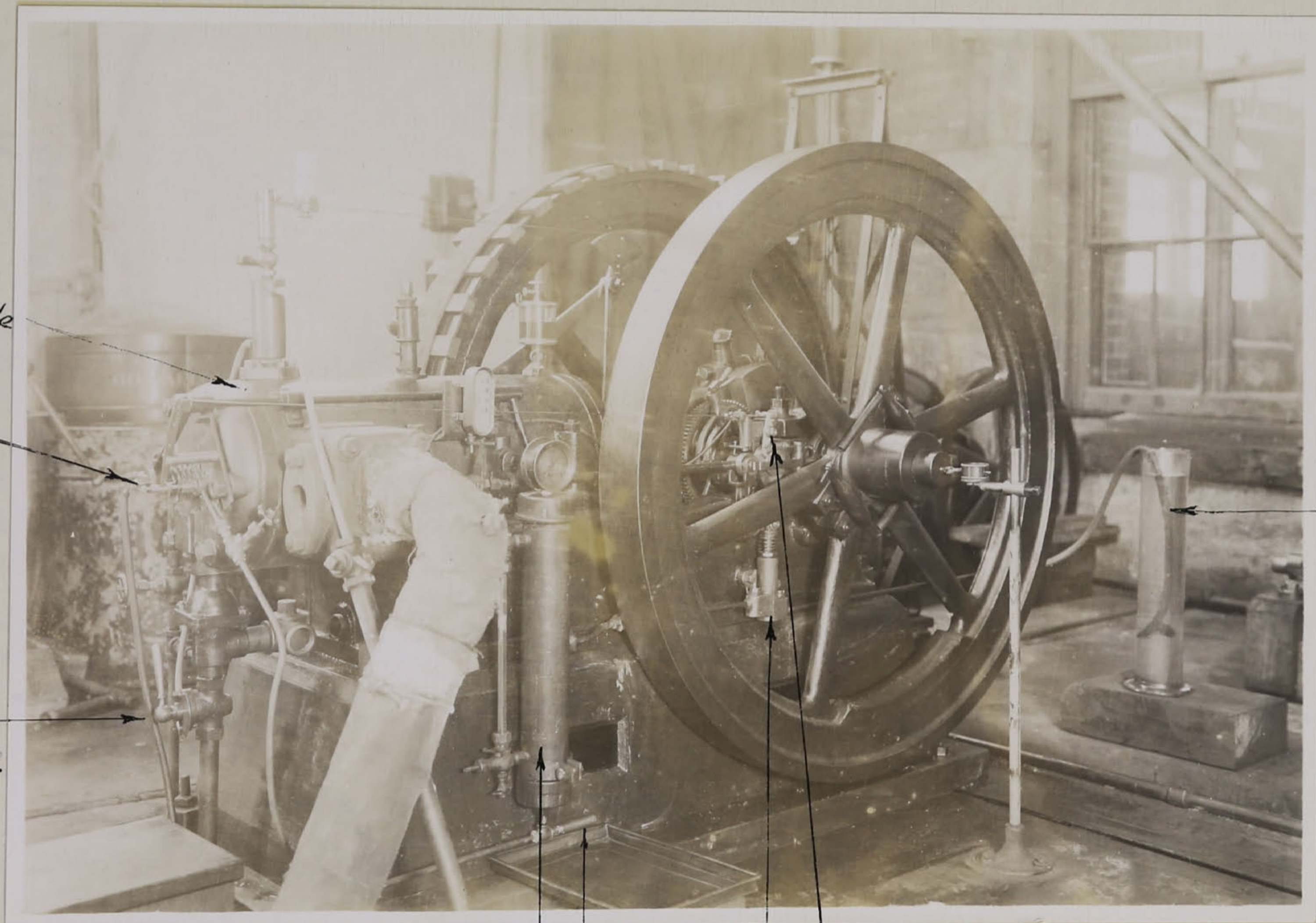
1	P,4							
2	5							
3	30							
4	295	225	172	78	274	233	185	64.8
5	195	205	205	205	204	204	205	206
6	52	52	52	55	62.4	52	54	57
7	150	150	150	150	179	177	176	175
8	94.6	79.1	61.7	22.2	95.6	79.1	62.6	19.4
9	59.3	63.8	62.5	68.7	65	62.5	64.4	69
10	9.7	8.73	6.65	2.63	10.7	8.55	6.94	2.31
11	100	75	50	0	100	75	50	0
12	8.6	6.76	4.51	0	8.97	5.83	4.52	0
13	88.7	77.3	67.8	0	83.5	68.2	65.1	0
14	34.3	33.3	38.3	0	30.5	39.9	41.0	0
15	555	555	555	555	584	584	584	584
16	13.4	13.9	12.0	0	14.3	10.9	10.6	0
17	15.1	17.9	17.7	17.2	17.1	16.0	16.3	15.5
18	15.9	16.5	14.2		16.9	12.2	12.6	
23	390	362	229	112	408	335	218	84

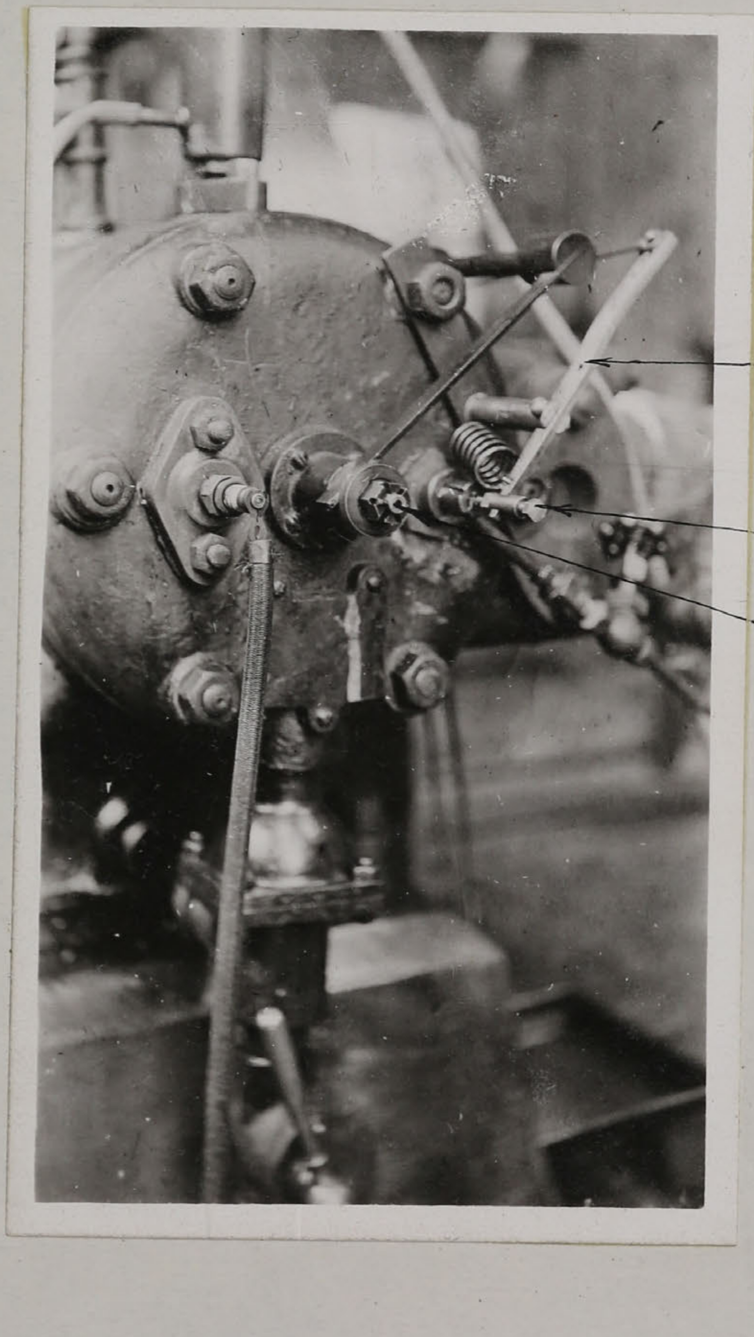
Index No.

1	F			
2	8	7	6	5
3	30	30	30	30
4	230	230	220	224
5	195	193.5	209	193.5
6	56.8	51.8	51.3	51.2
7	180	180	180	180
8	94	97	104.5	96.7
9	45.6	47.1	47.7	47.0
10	7.4	7.89	8.63	7.86
11	75.8	76.6	75.5	73.5
12	6.52	6.52	6.94	6.26
13	88.2	82.6	80.4	79.6
14	36.8	36.8	31.7	35.8
15	498.1	498.1	498.1	498.1
16	14.5	14.45	16.1	14.3
17	16.45	17.5	20	17.95
18	17.2	17.15	19.1	16.95
19	69.4	68	65.3	63
20	72.4	72.7	69.2	67.8
21	822	812	807	804
22	84.9	79.2	85.9	83.6
23	230	230	180	196

Cylinder Constant 0.001729

Brake Constant 0.0044


Thermometer hole
Injector Valve.
Fuel Gas
Regulating Valve.
Water Supply.
Accumulator
Blow Off Valve.
Water Pump.
Timing Mechanism

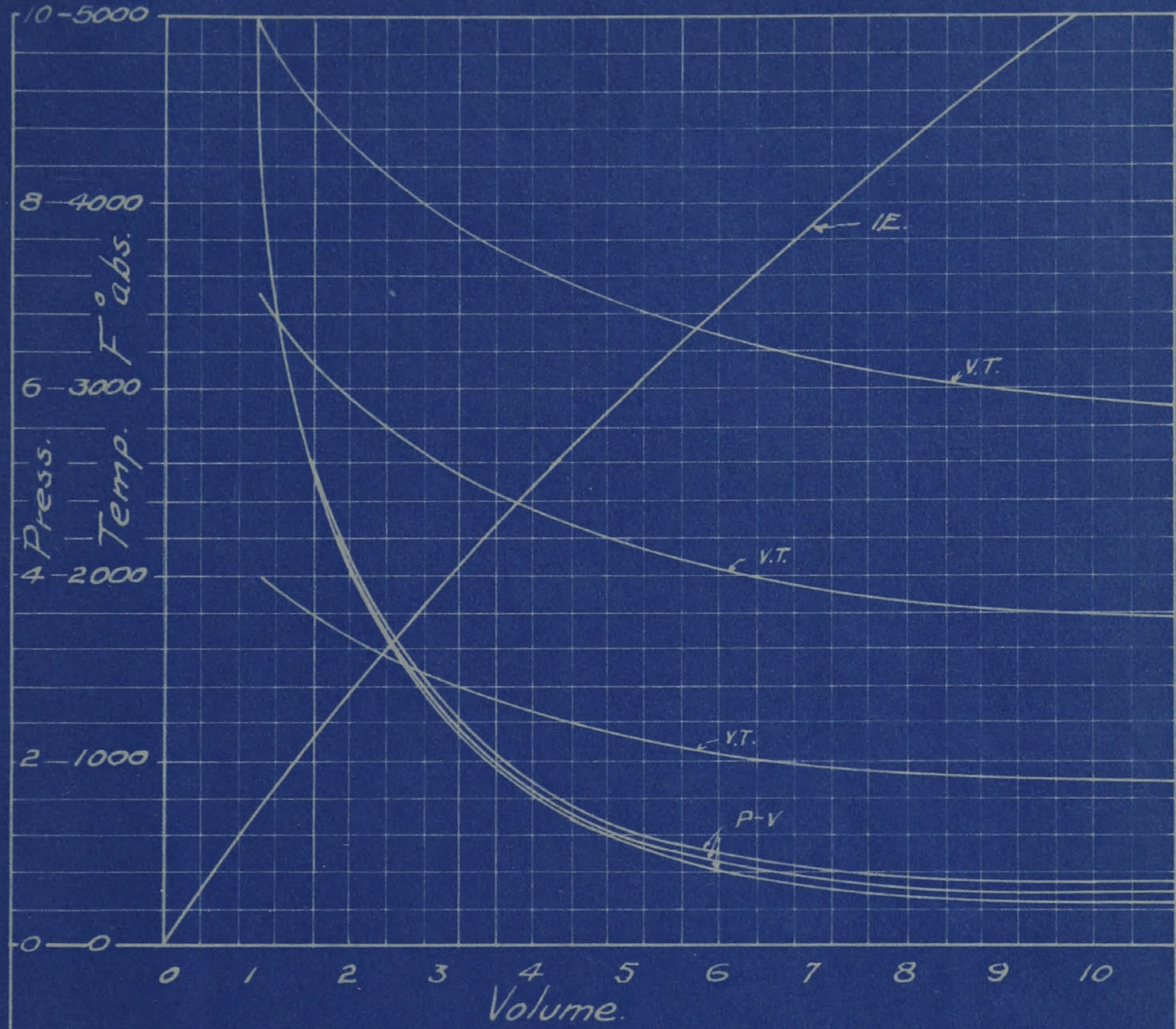


Operating Arm.

Injector Valve

Automatic Gas Inlet

View of Cylinder Head



Curves of Adiabatic Expansion and Internal Energy
for
Products of Combustion of Gas-Engine Mixture.
Curves drawn for Vol.=1 and Press.=1 at
5000°, 3500° and 2000° Farh. absolute.

$K_p = 182.0 + .0281T$ ft. lbs./lb.

$K_v = 126.5 + .0281T$ " "

$R = 55.5$ Mol. Wt. $= 27.65$

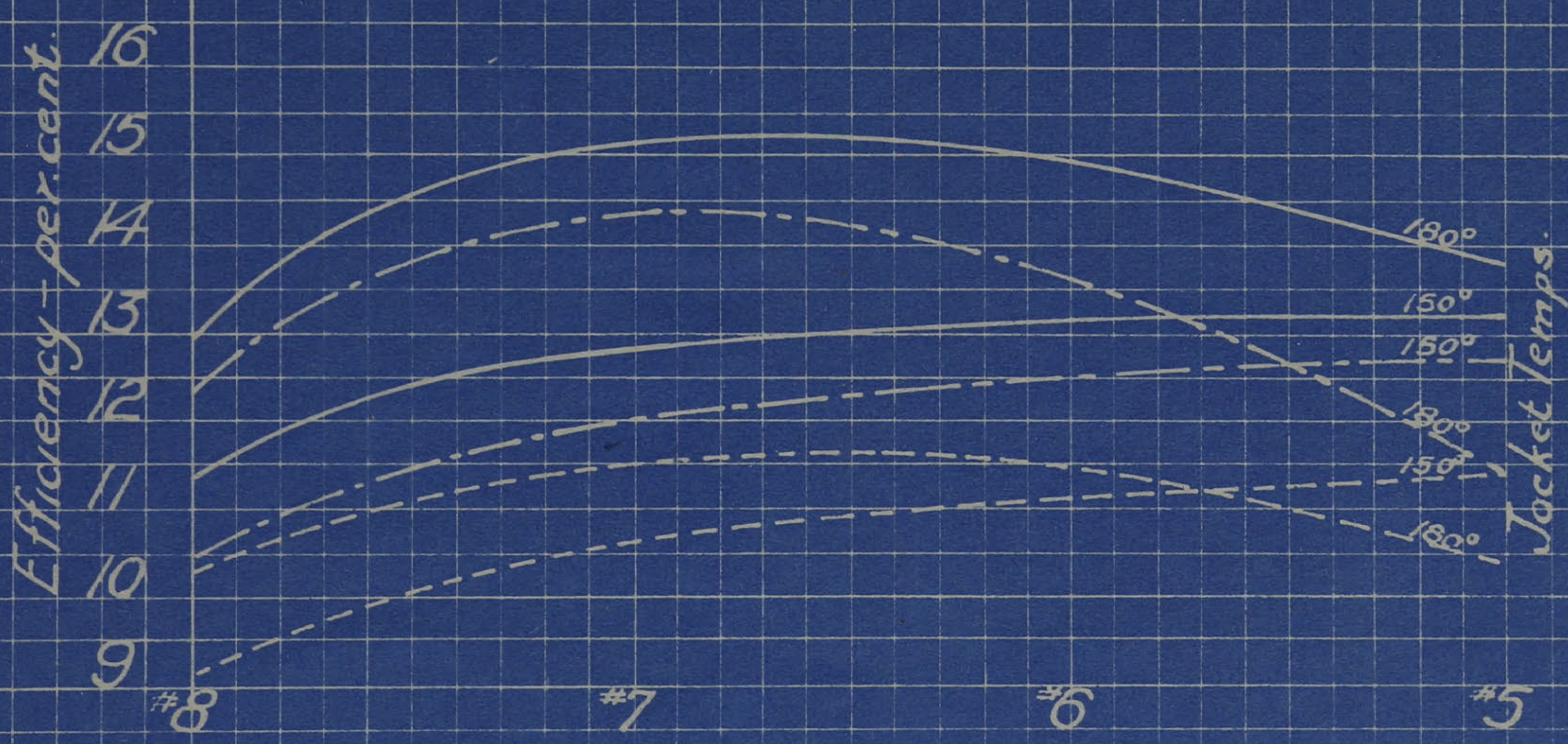
Equations:– $V = \text{Antilog.}[K - (2.28 \log T + .00022T)]$

$$\frac{PV}{T} = \text{Constant.}$$

Note:- To read Internal Energy multiply scale at bottom by 100,000 ft. lbs./lb.

Fig. 1

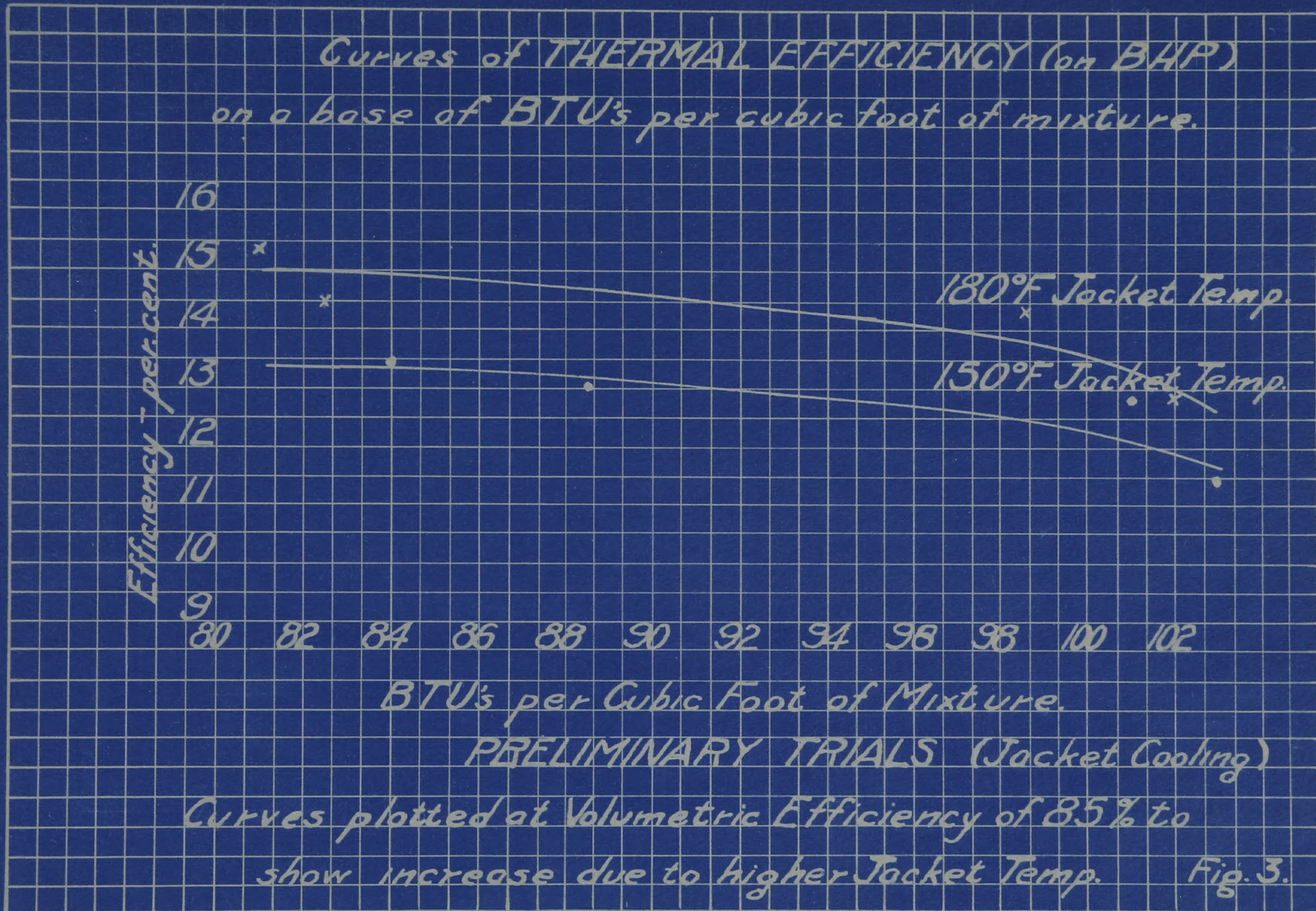
Curves of THERMAL EFFICIENCY (on the BHP) at different settings of gas regulator valve. Number 8 being wide open and number 5 being half open.



PRELIMINARY TRIALS (Jacket Cooling)

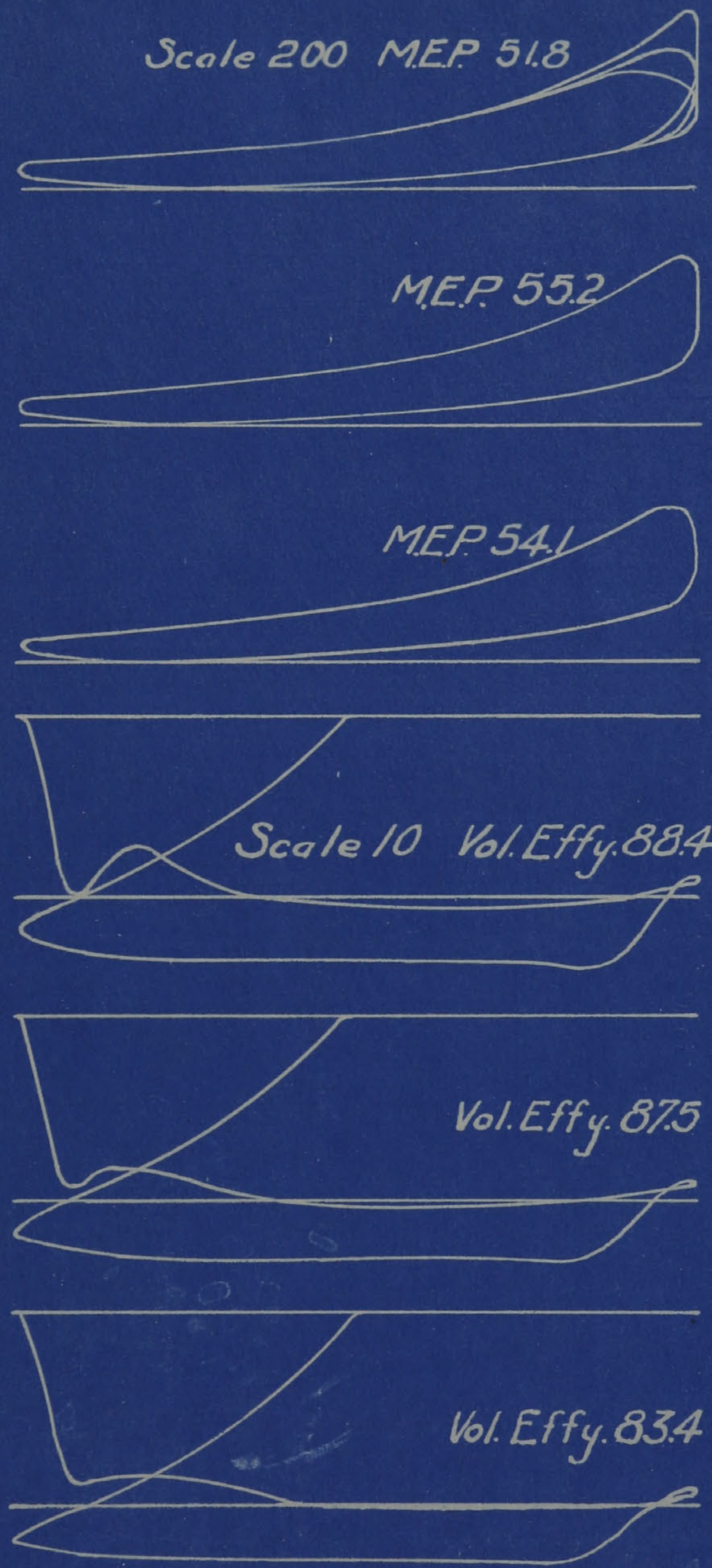
Curves
——— full load.
—·—·— three quarter load.
- - - - - half load.

Fig. 2.

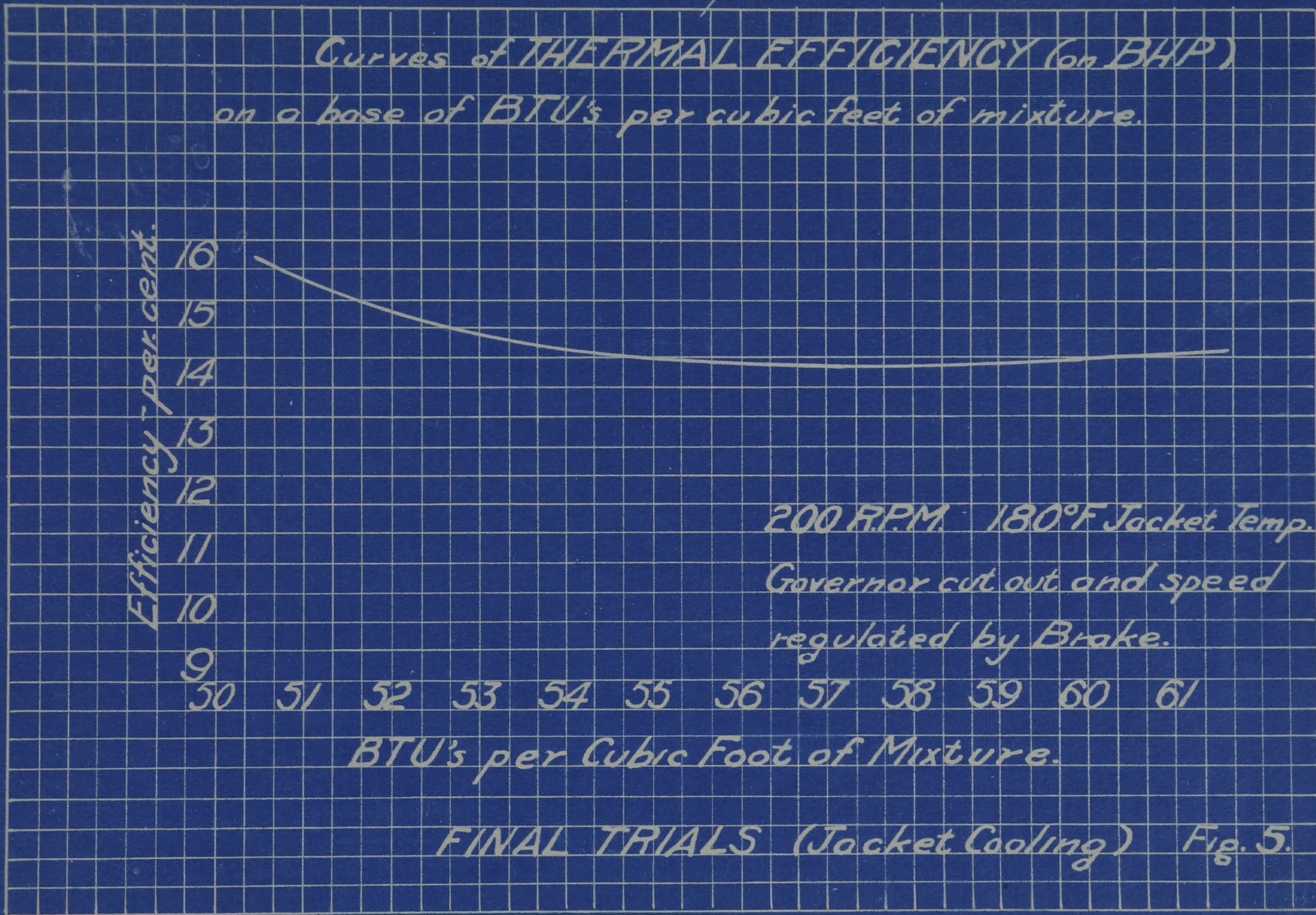


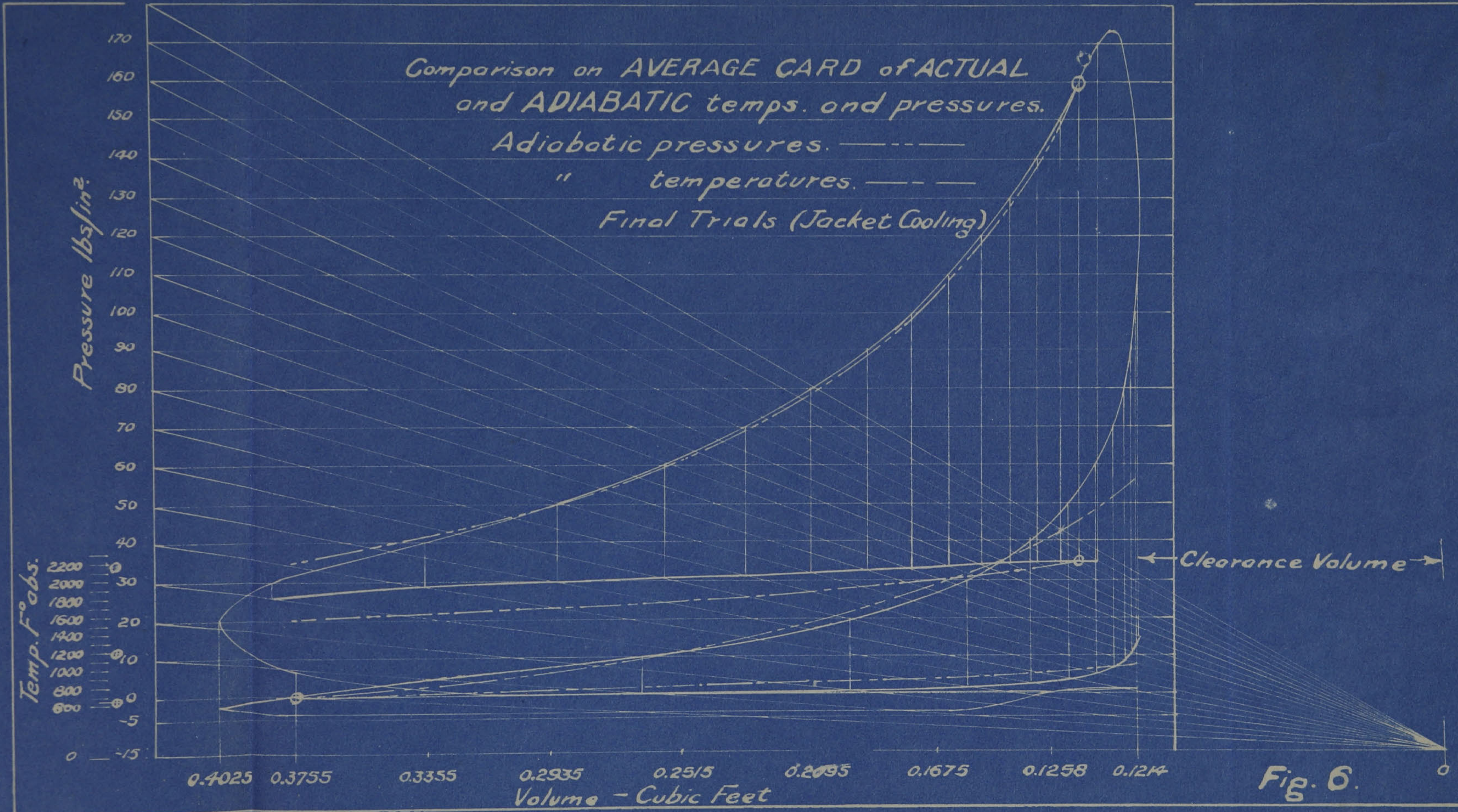
Curves of THERMAL EFFICIENCY (on BHP)
on a base of BTU's per cubic foot of mixture.
Efficiency - per cent.
16
15
14
13
12
11
10
9
80
82
84
86
88
90
92
94
96
98
100
102
180°F Jacket Temp.
150°F Jacket Temp.
BTU's per Cubic Foot of Mixture.
PRELIMINARY TRIALS (Jacket Cooling)
Curves plotted at Volumetric Efficiency of 85% to
show increase due to higher Jacket Temp.

Fig. 3.



AVERAGE INDICATOR CARDS Fig. 4


Curves of THERMAL EFFICIENCY (on BHP)
on a base of BTU's per cubic feet of mixture.
Efficiency - per cent.
16
15
14
13
12
11
10
9
50 51 52 53 54 55 56 57 58 59 60 61
BTU's per Cubic Foot of Mixture.
200 R.P.M. 180°F Jacket Temp.
Governor cut out and speed
regulated by Brake.
FINAL TRIALS (Jacket Cooling)
Fig. 5.



Comparison on AVERAGE CARD of ACTUAL
and ADIABATIC temps. and pressures.
Adiabatic pressures.
" temperatures.
Final Trials (Jacket Cooling)
Pressure lbs/in²
Temp. F° abs.
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
-5
-15
2200
2000
1800
1600
1400
1200
1000
800
600
0
0.4025
0.3755
0.3355
0.2935
0.2515
0.2095
0.1675
0.1258
0.1214
0
Volume - Cubic Feet
Clearance Volume

Fig. 6.

TEMPERATURE-ENTROPY diagram from AVERAGE CARD in Fig 6.

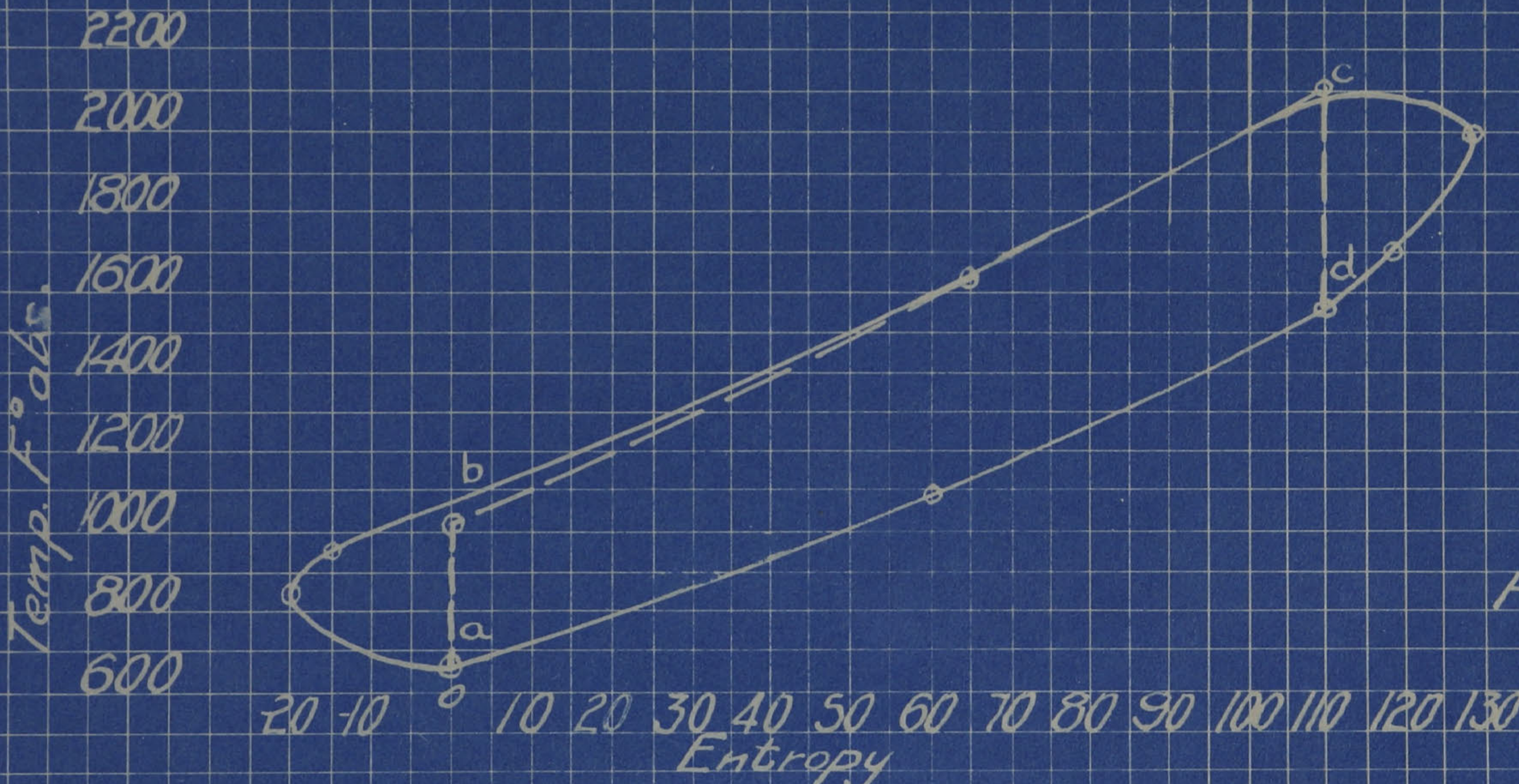
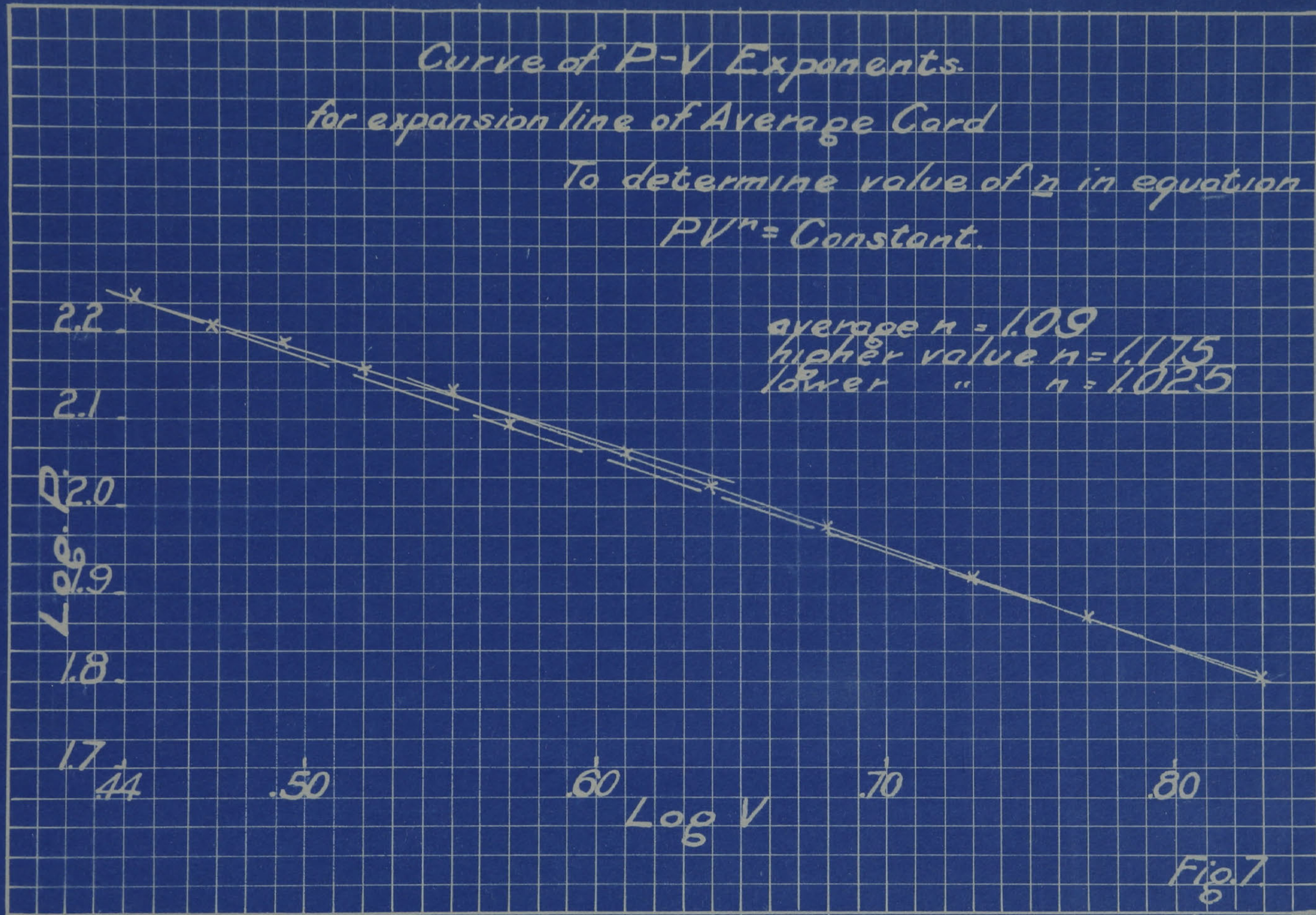


Fig. 6a.

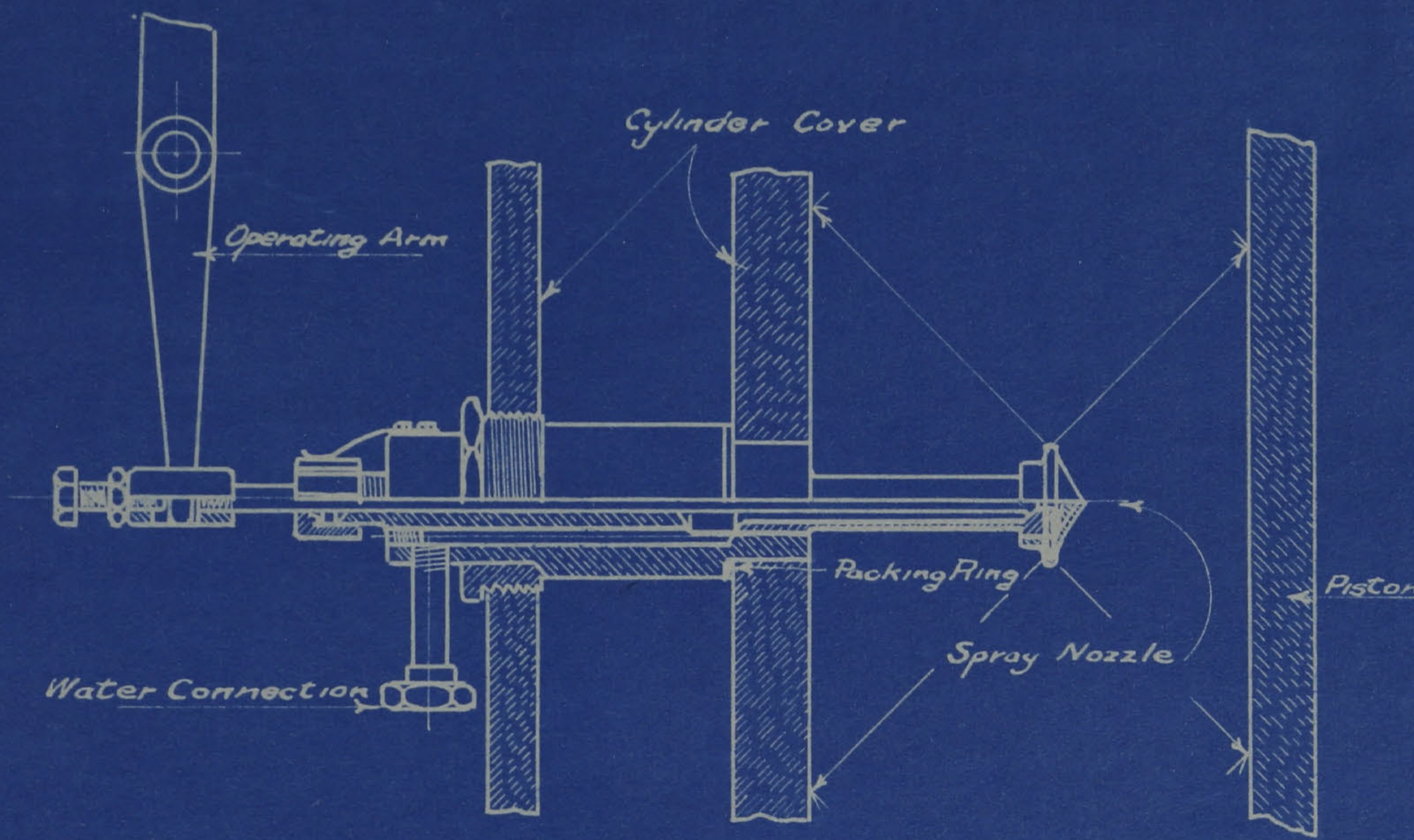
Dotted curve shows ADIABATIC with variable specific heats.

Equations $= \varphi_1 - \varphi_0 = a.\log_{\epsilon} \frac{P_1}{P_0} + b.\log_{\epsilon} \frac{V_1}{V_0} + \frac{s}{R}(P_1V_1 - P_0V_0)$.



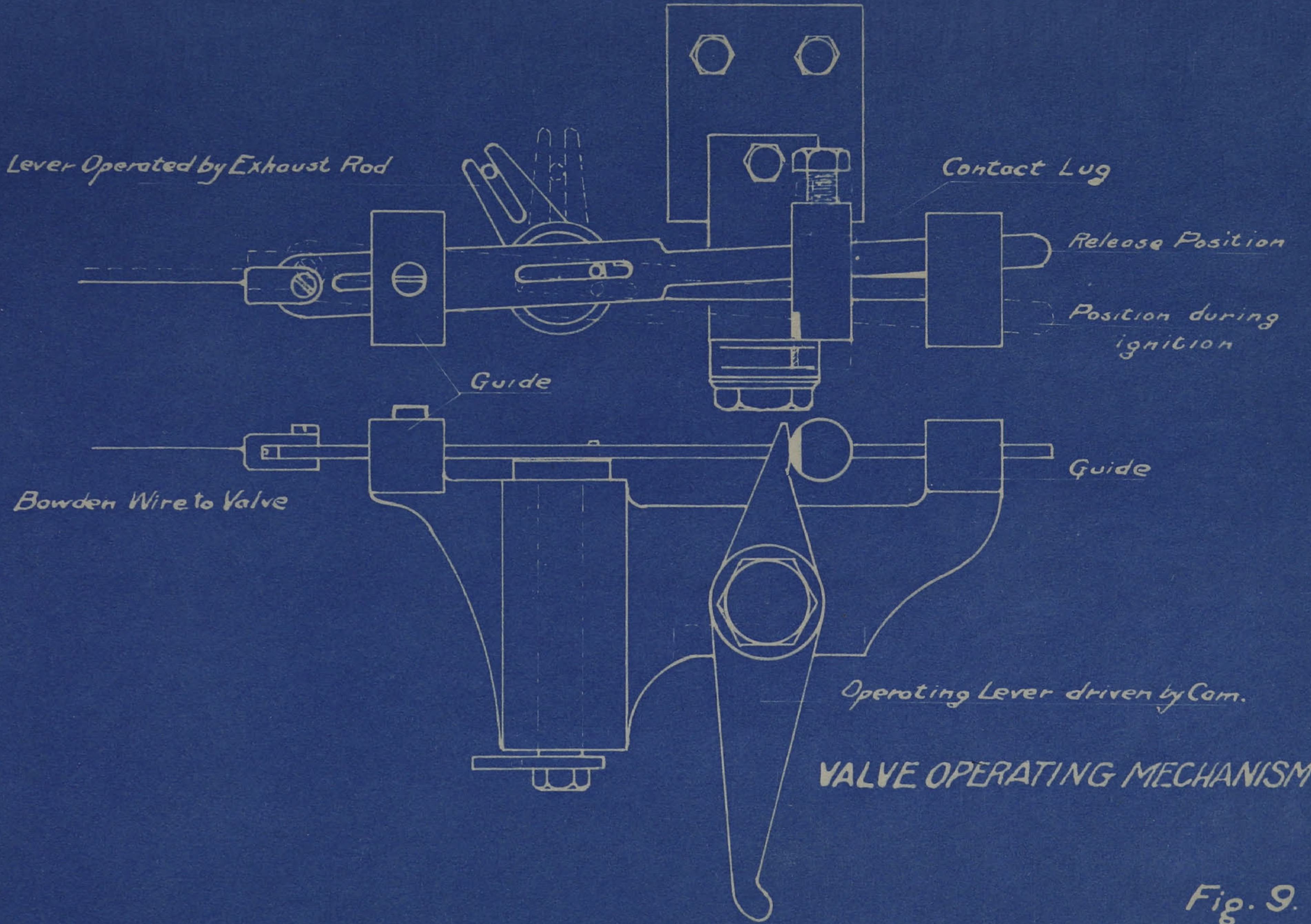
Curve of P-V Exponents.
for expansion line of Average Card
To determine value of n in equation
PV^n = Constant.
average n = 1.09
higher value n = 1.175
lower " n = 1.025
2.2
2.1
2.0
1.9
1.8
1.7
Log P
.44
.50
.60
.70
.80
Log V

Fig. 7.



INJECTION VALVE

Fig. 8



VALVE OPERATING MECHANISM

Fig. 9.

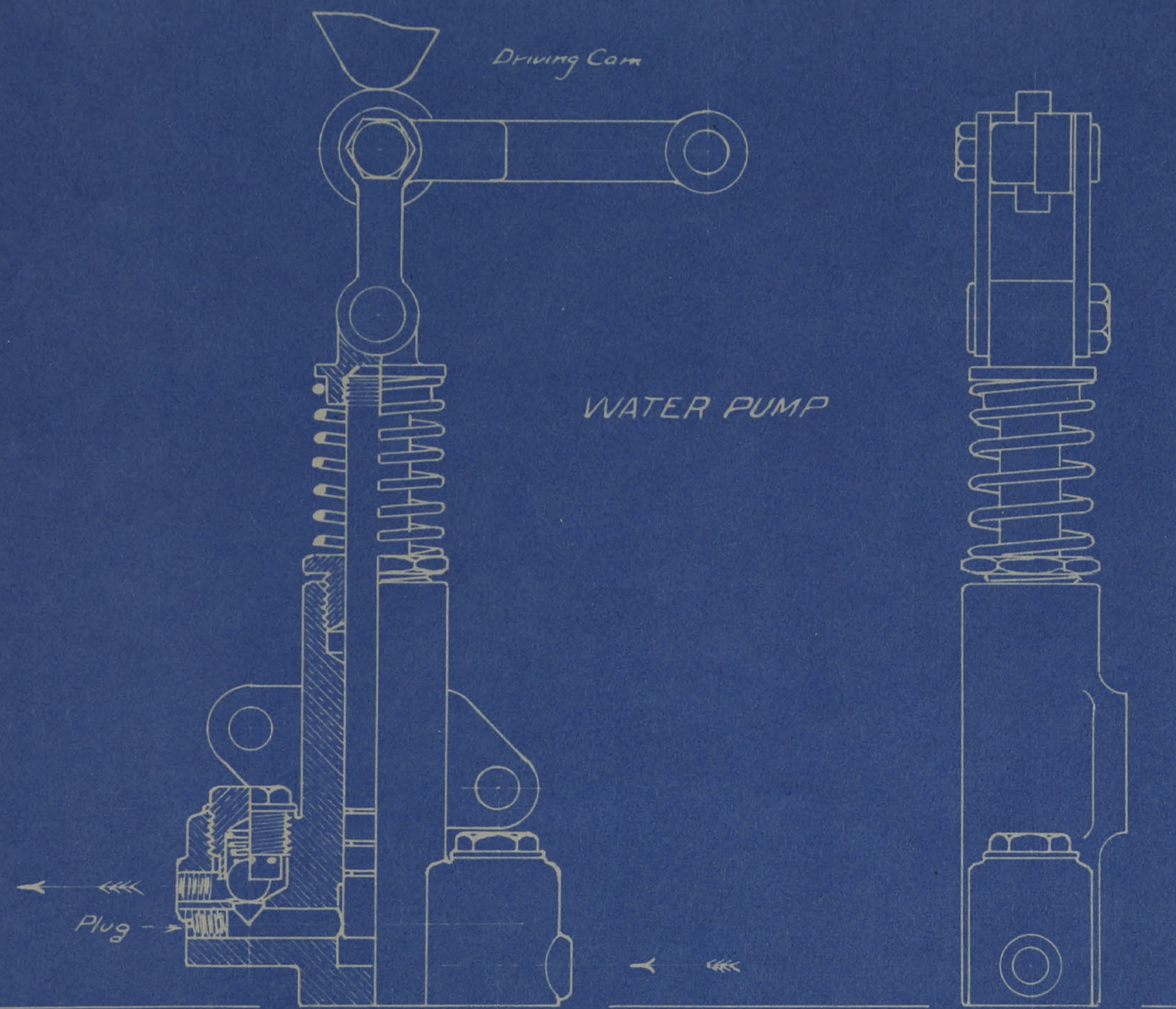
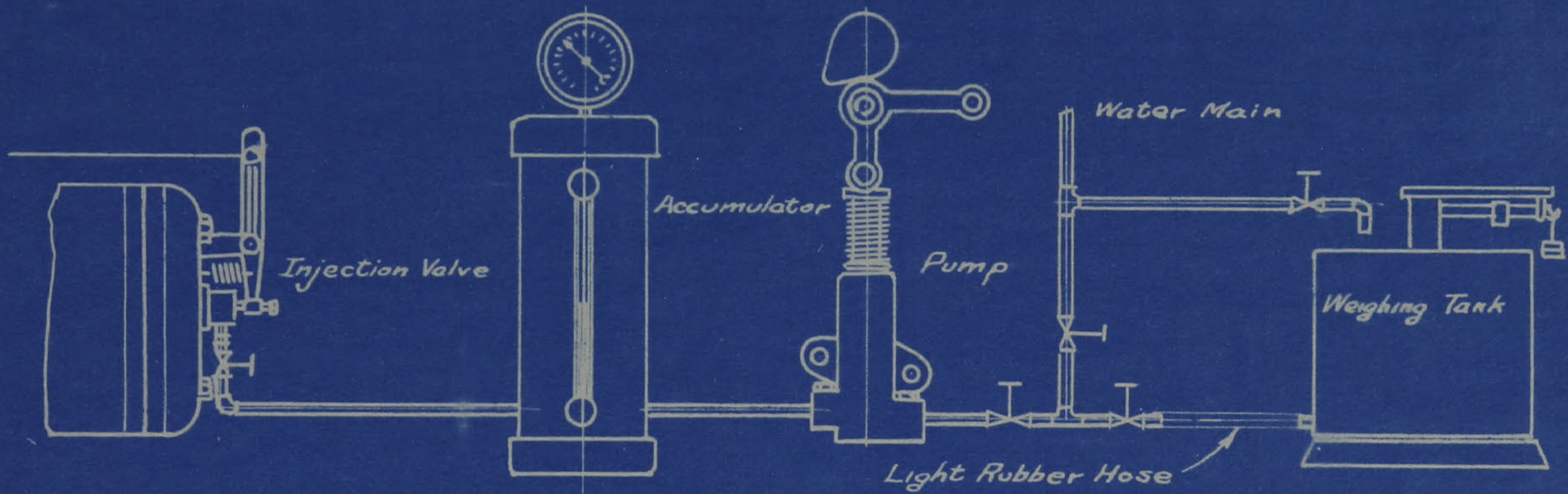


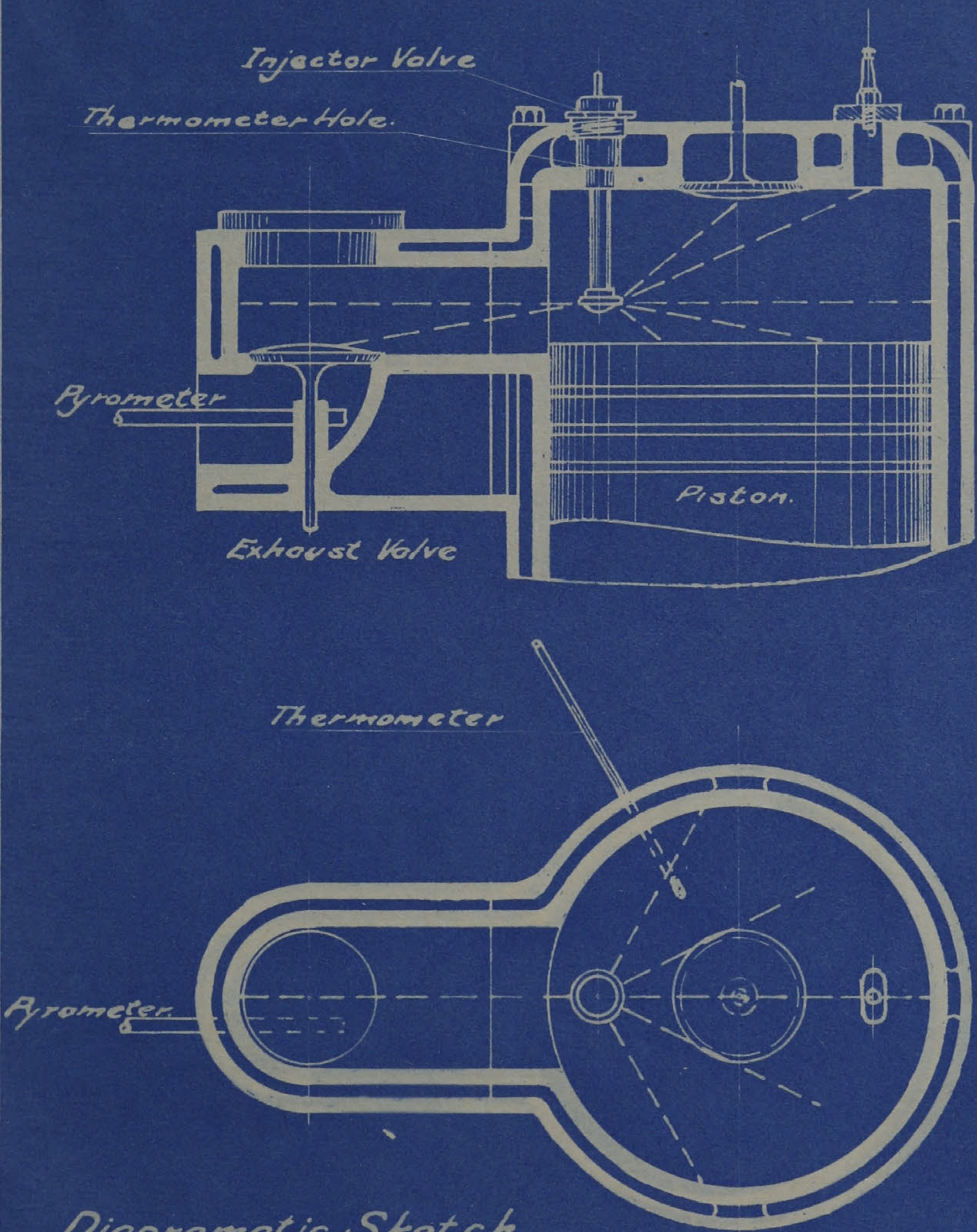
Fig. 10



LAYOUT OF COOLING SYSTEM

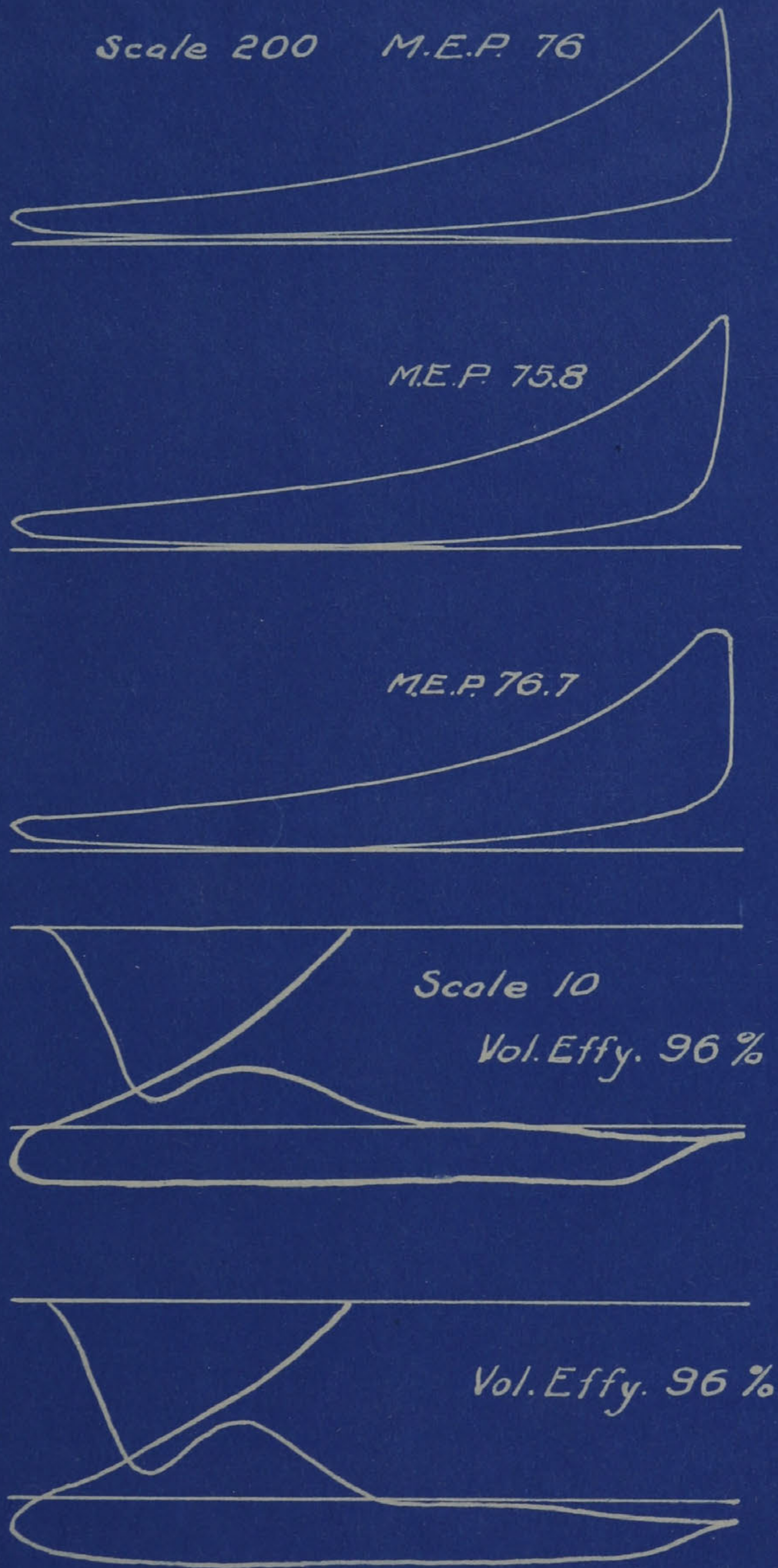
NOTE :- Provision is made in water supply to pump for pressure feed or feed from a weighing tank. The ACCUMULATOR is fitted with a water glass and a pressure gauge.

Fig. 11.



Diagramatic Sketch
Showing POSITION of JET in CYLINDER
and DIRECTION of STREAMS.
Also position of thermometer in head and
pyrometer under exhaust valve.

Fig. 12.



AVERAGE INDICATOR CARDS Fig 13.