THE USE OF WATER AS AN ANTI-DETONANT

Thesis for the Degree of M. S.

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THESIS

Gas + oil engines
Title

THE USE OF WATER AS AN ANTI-DETONANT.

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THESIS

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THE AUTHOR.

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Introduction.

Most automobile drivers have noticed that the engine runs smoother on damp rainy days or at night than on hot dry days. It has often been suggested that water be used to give this advantage at all times. Several devices are on the market at the present time which supply some moisture to the fuel charge. Report No. 45, Nat. Adv. Com. for Aeronautics, contains the results of some tests made by V. W. Brinkerhoff on a class B, U. S. Army truck engine with a compression ratio of 3.71 to 1. He found that the injection of water into the intake manifold had little effect on power or efficiency for quanities of water up to 0.4 pounds per pound of gasoline. For amounts of water above 0.4 pounds there was a decided decrease in power.

Most automobile carburetors are adjusted to give a rich, power mixture. Any decrease in the richness of this mixture will result in an increase in economy. Consequently, it is probable that any increase in fuel economy obtained with the various water supplying devices is due to the extra air supplied by them rather than to the effect of the water.

For years the makers of oil engines and kerosine engines and tractors have used water to prevent pre-ignition. The water is supplied by a mixer, an air cleaner using water or by a special carburetor. Considerable increase in the compression ratio of kerosine tractors has been effected by this use of water to prevent pre-ignition, detonation and chemical "cracking" of the fuel.

The use of water as an anti-detonant for automobile engines instead of tetra ethyl lead has been suggested by Marc Chauvierre in an article "L'eau, antidetonant" in La Technique Automobile et Aerienne Vol. 16, No. 129. He gives some very interesting and favorable results obtained from tests run on a single cylinder gas engine.

Object of Investigation.

The purpose of the present investigation is to determine; first, the value of water as an anti-detonant, and second, its effect upon (a) the actual engine and (b) the ideal cycle.

Equip ment and Apparatus Used for Tests.

The experimental tests were run on a Reo Flying Cloud six cylinder engine having a bore of 3¼ inches and a stroke of 5 inches. The normal compression ratio of this engine was 4.91 to 1. Four special heads were obtained giving compression ratios up to about 8 to 1. These cylinder heads are shown in Fig. 1. Number 5 head was never used.

The volume of each combustion space was carefully measured by means of a burette and when necessary metal was removed by machining to make these volumes nearly the same for each cylinder. Table I gives the data for these cylinder heads. They will be designated by the numbers shown in the table and on the figure, Number 1 is the standard head. Tests run with No 4 head were not satisfactory, so some changes were made in the head by grinding.

Table I.

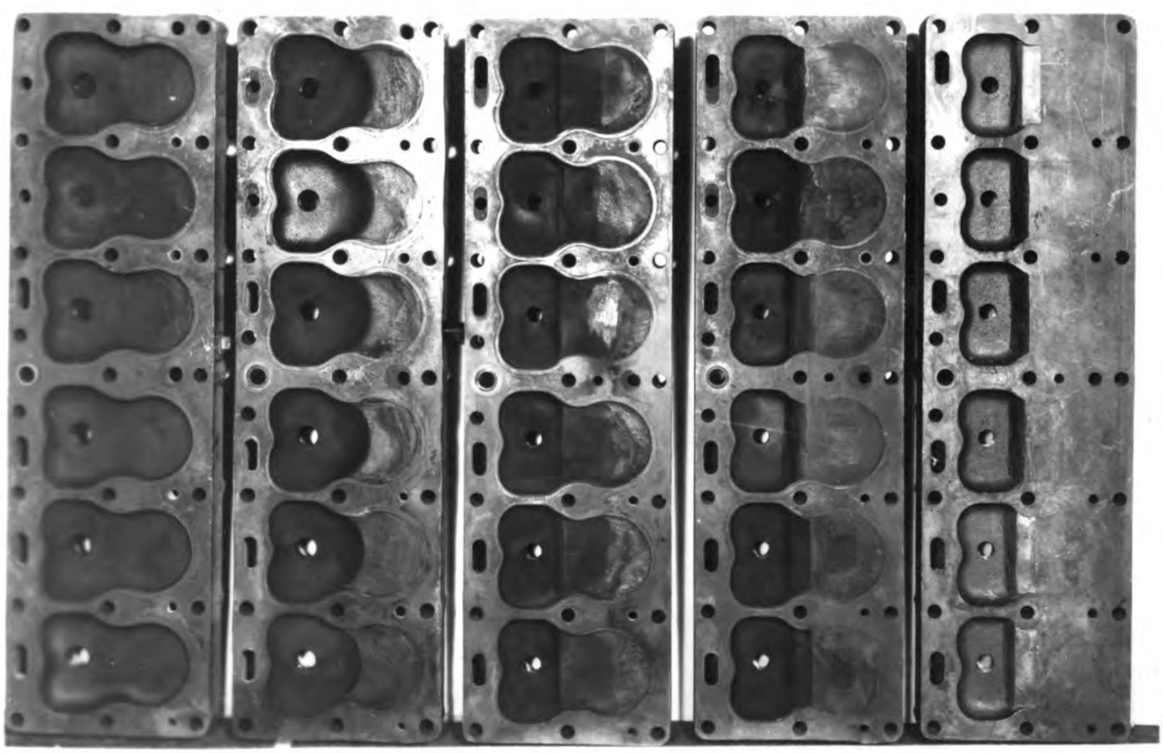
Combustion Chamber Volumes and Observed Compression Pressures.

Displacement of one cylinder = 41.6 cu. in.

Head No.	Cylinder Number. 1	2	3	4	5	6	Aver-age.	Comp. Ratio.
	Volumes in c.c.							
1	173	173	173	176	173	174	174	4.91
2	150	150	150	150	155	152	151	5.51
3	133	132	132	135	127	134	132	6.16
4before	106	110	111	110	109	109	109	7.24
4after	111	112	116	112	109	111	112	7.08
	Compression Pressures in lbs./sq.in. Measured with Okill Compression Gage.							
3	111	111	110	102	115	110	110	
4before	131	130	131	122	135	132	130	
4after	139	139	139	137	140	140	139	

Note: the valves were ground at the same time that the changes were made in No. 4 head.

The engine was connected to a 100 H.P. Sprague electric dynamometer which can be seen in Fig. 2. The dynamometer control panel is shown in Fig. 3. The electric tachometer and speed counter is attached to the control panel. The automatic control for the tachometer and speed counter and the fuel weighing device was not used for these tests because this method of measuring



No. 1. No. 2. No. 3. No. 4. No. 5.

Fig. 1.

Cylinder Heads Used for the Tests.

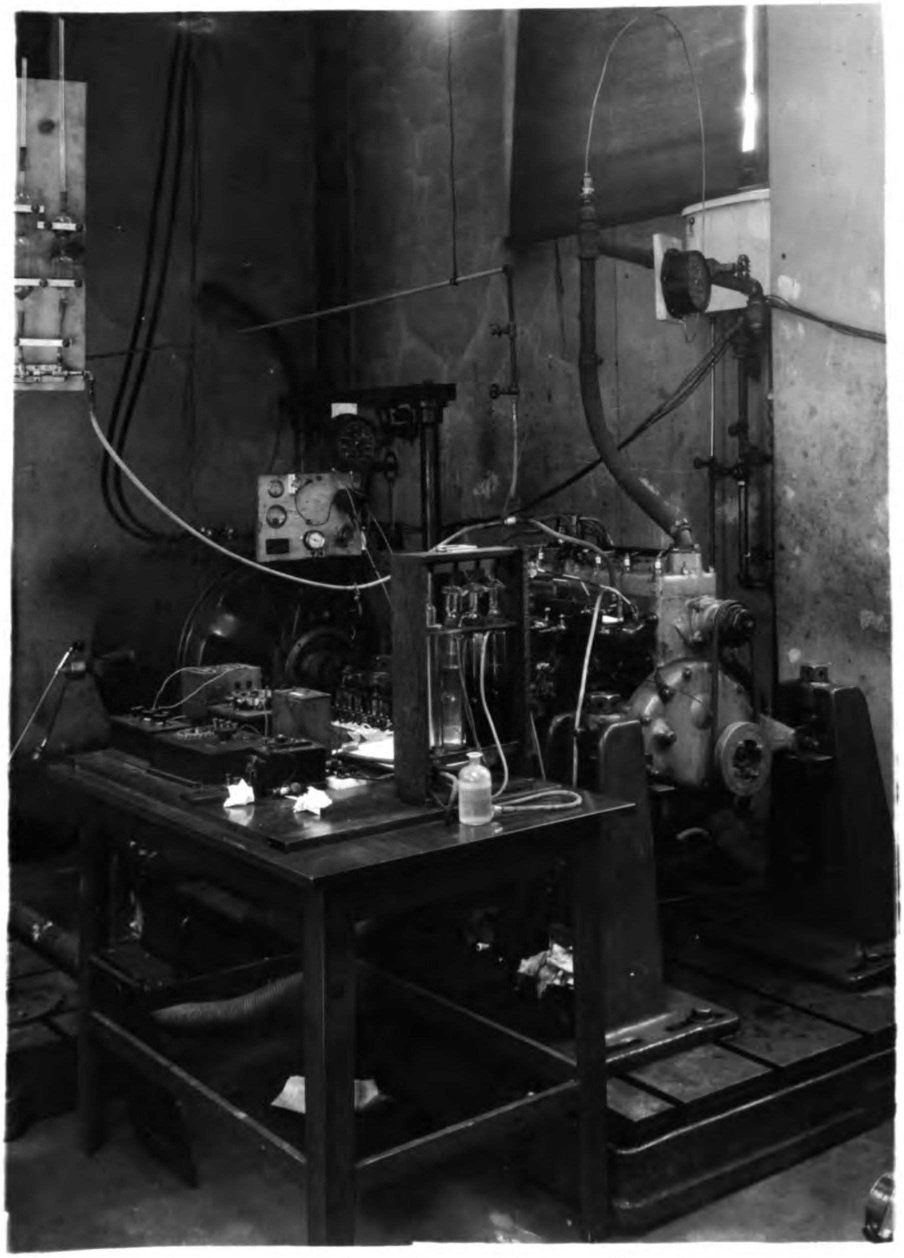


Fig. 2.

General View of the Test Equipment.

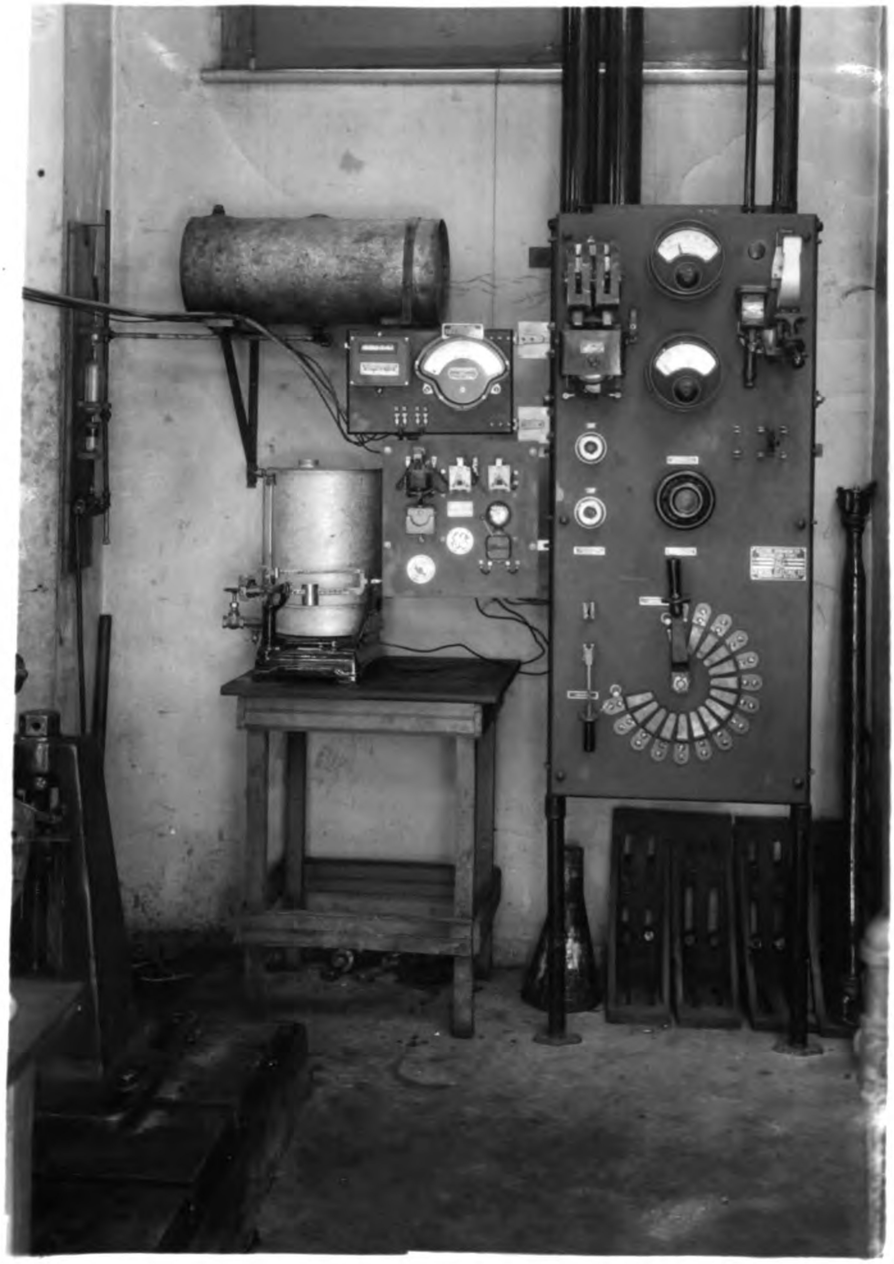


Fig. 3.

View of Dynamometer Control Panel and Fuel Measuring Equip ment.

the fuel required much longer runs than the gasoline measuring pipette shown in the upper left hand corner. The pipette held just 210 c.c. of gasoline and the time required by the engine to consume this amount was found with a stop watch. This size of pipette is such that it gives the gasoline consumption in pounds per hour when a constant, 20, is divided by the time in minutes required for it to empty.

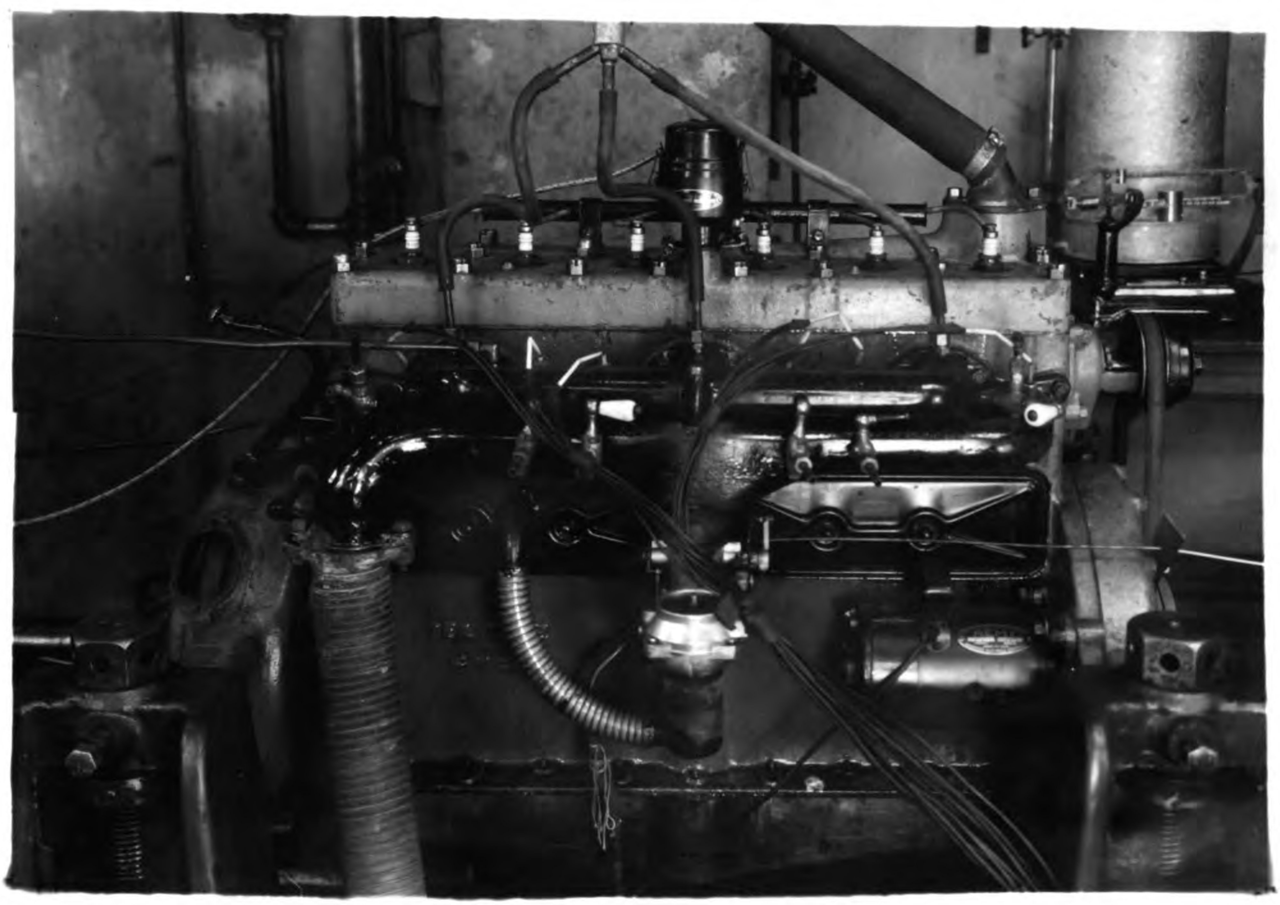


Fig. 4.

General View of the Manifold Arrangement.

The water injected was measured by similar pipettes shown in the upper left of Fig. 2. Three pipettes with wolumes of 56 c.c., 105 c.c., and 210 c.c. respectively were used, depending upon the rate of water injection.

The method of injecting the water into the intake manifold is shown in Fig. 4. Standard copper tubing fittings were used and orfice plates were inserted in them. Six sets of water injection nozzles with various size orfices were used to obtain the desired amounts of water. Five of these sets are shown on the front edge of the table in Fig. 5. This method of injecting the water was used to obtain an even distribution of the water to the various cylinders. Each nozzle supplying water to two cylinders.

As shown in Fig. 4 the exhaust manifold was fitted with a gas sampling cock and a thermocouple for each cylinder. The thermocouples were inserted into the exhaust manifold through packing glands made of copper tubing fittings. The thermocouples were made of No. 22 Chromel and Alumel wire inserted in two hole porcelain insulators. The voltage of the thermocouples was measred by a laboratory type potentiometer shown in Fig. 5. A switch board was made of six standard double pole single throw switches. The thermocouples were calibrated in an electric furnace with a standardized Platinum Iridium thermocouple.

The exhaust gas analysis was obtained by a bubbling type Orsat apparatus shown in Figs. 2 and 5. No results of any particular value were obtained.

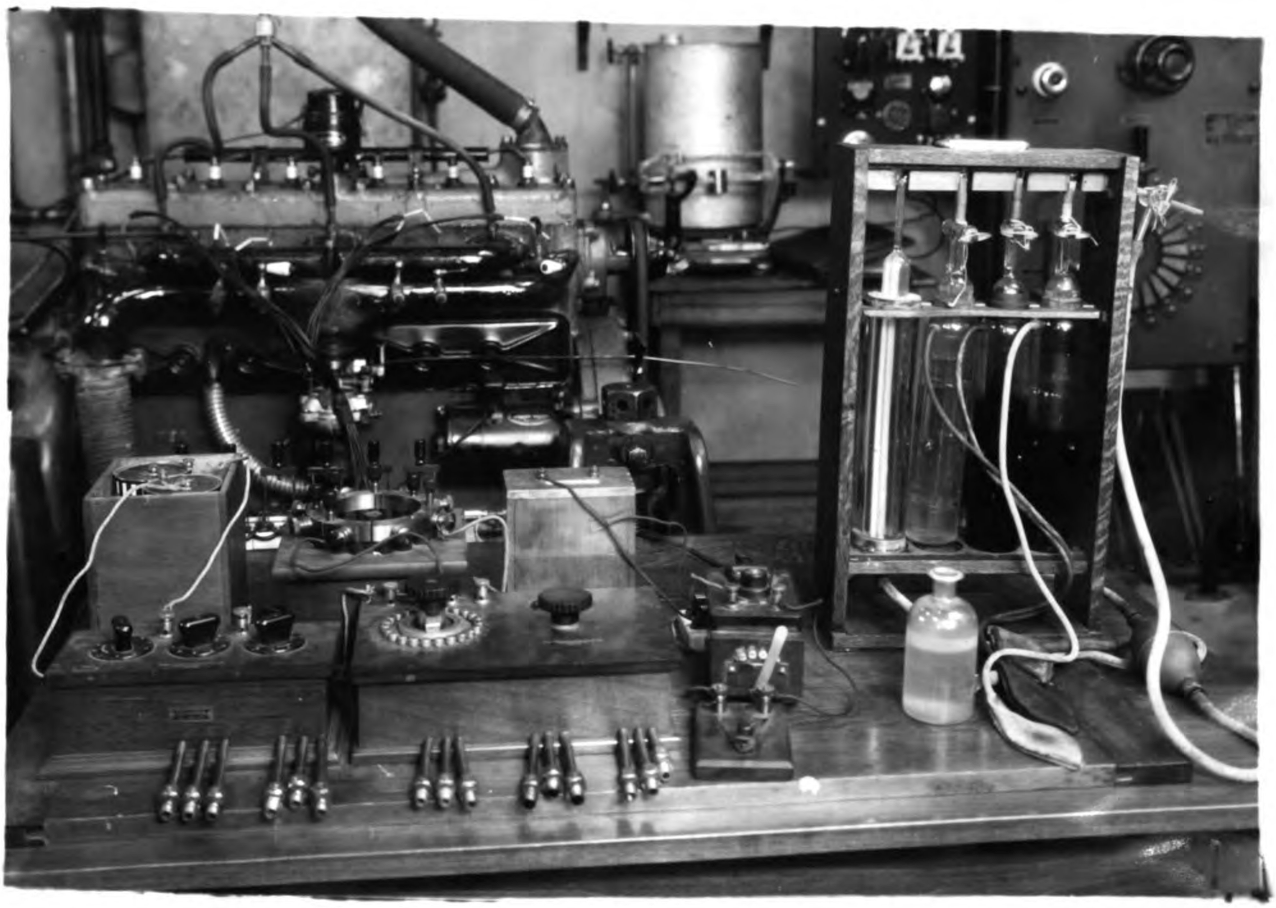


Fig. 5.

Potentiometer and Orsat Apparatus.

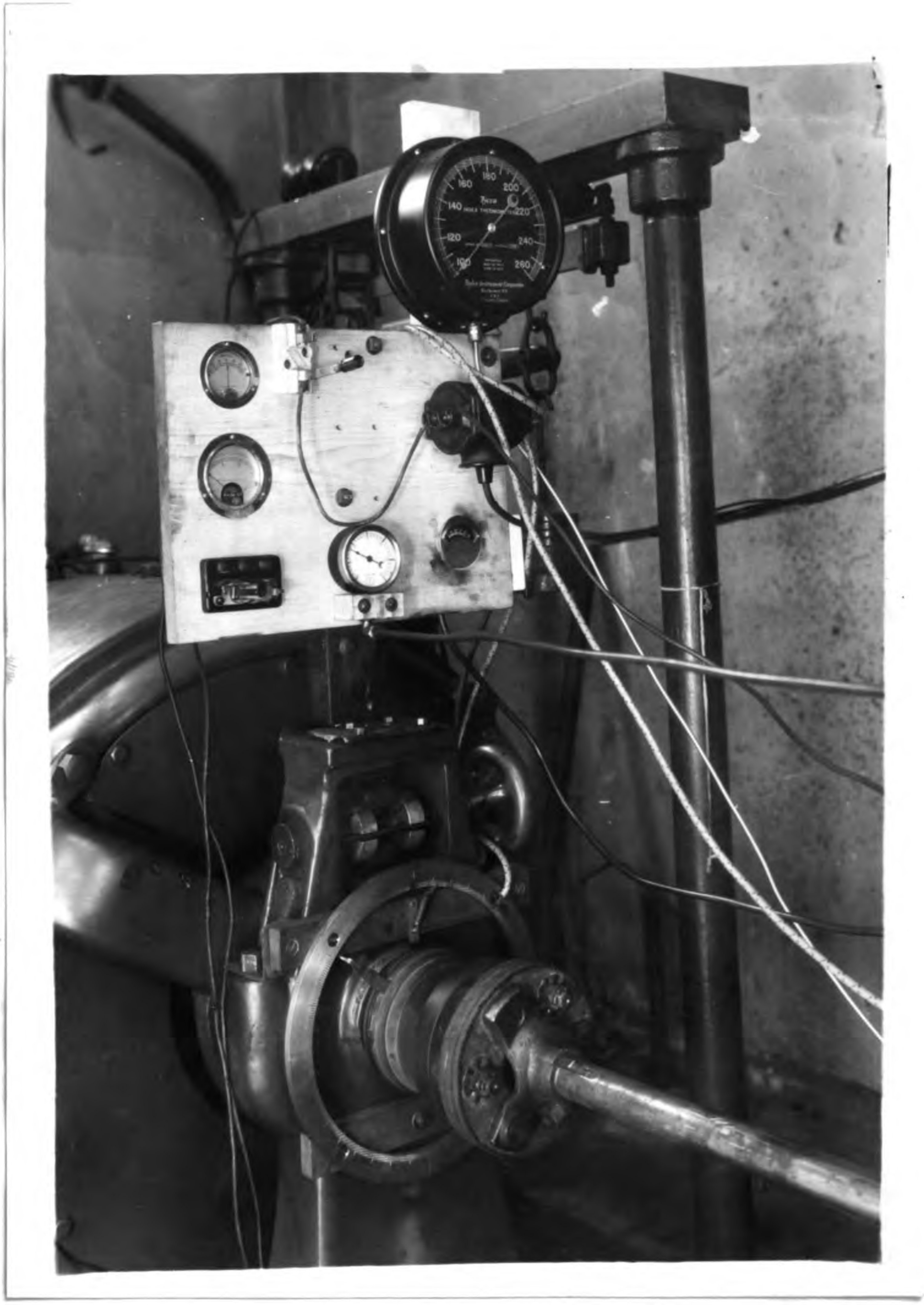


Fig. 6.

View of Device Used for Indicating Spark Advance.

The cooling water was regulated by the thermostat which is standard equipment for this engine and the temperature was given by an indicating thermometer. The oil pan was water cooled and the temperature of the oil shown by another indicating thermometer.

The device for obtaining the spark advance is shown in Fig. 6. It consists of a ring graduated in degrees and a slip ring and pointer. These are insulated from the foundation and are connected in series with the high tension lead from the spark coil to the distributer. The pointer is set to zero with the engine at dead center Readings are obtained by noting the point at which the spark jumps from the revolving pointer to the graduated ring. A switch is provided for short circuiting the device when it is not in use. The device was designed by the author and was constructed in the department shops. A similar device has been used at Purdue University.

Tests to Determine the Value of Water As an Anti-Detonant.

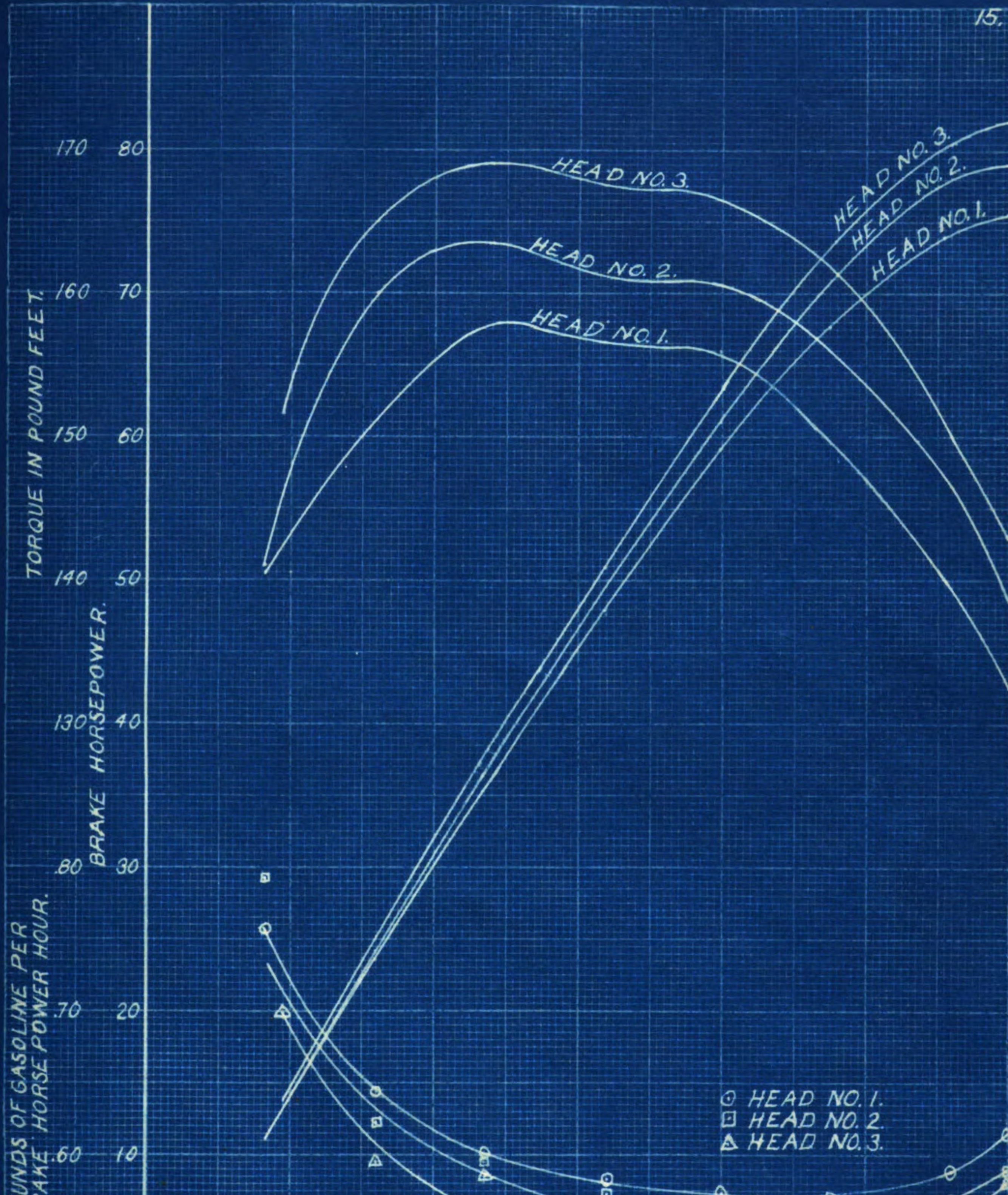
To determine to what extent the advantages of high compression could be realized with water as the anti-detonant a series of tests were run for maximum power conditions. The engine was run with wide open throttle at speeds starting at about 400 R.P.M. and going up by increments of 400 to 2800 R.P.M. and then at 3000 R.P.M. The engine gave the most power at the latter speed and this speed was not exceeded for fear of injuring the engine as the stresses are considerably increased by the increase in compression. During the tests no water was supplied to the engine when using the standard head and just sufficient water to prevent pre-ignition and detonation when using the higher compression heads. To secure the maximum power it was necessary to advance the spark to a point where there was a considerable "spark knock" in all cases. This was probably due to the slight differences in compression pressures in the different cylinders. The variation in these pressures can be seen from an inspection of Table I. The compression pressures were measured by an Okill guage.

The results of these tests for heads Nos. 1, 2, and 3 are shown in Table II and graphically in Fig. 7.

Table II.

Maximum Brake Horse Power Tests.

R.P.M.	B.H.P. corrected	Torque Lb.ft.	Lbs. Gas./B.H.P.hr.	Spark Deg. Advance.	Water/Gas. Lbs./lb.
Cylinder Head Number 1.					
414	11.1	140.5	.756	12	0.0
795	23.5	151.2	.644	16	0.0
1230	37.0	158.0	.600	20	0.0
1595	47.6	156.5	.583	26	0.0
1990	59.2	156.0	.573	30	0.0
2375	67.6	149.5	.569	34	0.0
2775	74.0	140.0	.587	35	0.0
2995	75.6	132.5	.615	35	0.0
Cylinder Head Number 2.					
413	11.1	141.0	.793	15	0.50
795	24.2	160.0	.622	15	0.30
1185	36.9	163.5	.596	20	0.22
1590	47.8	158.0	.513	27	0.18
1995	61.0	160.5	.545	28	0.16
2400	71.4	156.0	.549	32	0.14
2800	78.2	146.5	.568	34	0.13
2998	79.0	138.5	.587	34	0.14
Cylinder Head Number 3.					
474	13.7	151.5	.699	10	0.0
800	25.2	165.0	.595	10	0.0
1190	38.3	169.0	.584	13	0.0
1596	50.9	167.5	.531	20	0.26
2006	63.4	166.5	.517	21	0.21
2390	73.6	161.5	.521	24	0.19
2818	80.5	150.2	.552	28	0.34
3007	82.0	143.0	.580	26	0.32



TORQUE IN POUND FEET.
BRAKE HORSEPOWER.
POUNDS OF GASOLINE PER BRAKE HORSE POWER HOUR.
170
160
150
140
130
.80
.70
.60
.50
80
70
60
50
40
30
20
10
0
HEAD NO. 3.
HEAD NO. 2.
HEAD NO. 1.
HEAD NO. 3.
HEAD NO. 2.
HEAD NO. 1.
⊙ HEAD NO. 1.
⊡ HEAD NO. 2.
△ HEAD NO. 3.
0
500
1000
1500
2000
2500
3000
REVOLUTIONS PER MINUTE.

FIG. 7.

The horse power has been corrected to standard conditions of intake temperature and barometric pressure. The power obtained for head No. 4 was considerably lower than for the other three due to leakage at the higher pressures and other mechanical difficulties. The results obtained with the first three heads were exactly what might be expected for the increase in compression ratio used. The peculiar droop in the torque curve is a charactaristic of this engine and is not due to the water injection.

To determine the effect of the increased compression ratio on the fuel consumption under driving conditions a series of tests were made with the engine developing just enough power to drive the automobile on a smooth level pavement. The results of these tests are given in Table III and graphically in Fig. 8. There is a considerable increase in the miles per gallon of gasoline with the higher compression ratios.

All computations of test results were made according to the standards of the Society of Automotive Engineers.

Table III.

Mileage Test Data. Heads 1, 2, & 3.

Miles / hour.	R.P.M. OF Engine	B.H.P.	Spark Setting Degrees Advance. Head # 1.	Head # 2.	Head # 3.	Miles / Gallon of Gasoline. Head # 1.	Head # 2.	Head # 3.
5	448	.7	18	13	8	13.2	13.6	13.8
10	496	1.5	18	15	14	18.5	19.4	20.5
15	745	2.4	18	15	16	19.8	20.8	21.9
20	994	3.7	20	18	20	20.1	20.9	22.0
25	1240	5.3	24	23	22	19.2	20.5	21.0
30	1490	7.4	27	26	26	18.4	19.3	19.8
40	1985	13.4	34	32	23	17.4	17.2	17.7
50	2480	22.4	28	29	22	14.5	15.4	15.5
60	2980	35.1	35	37	24	11.7	12.3	12.5
65	2230	43.2	35	37	21	11.0	10.8	10.8

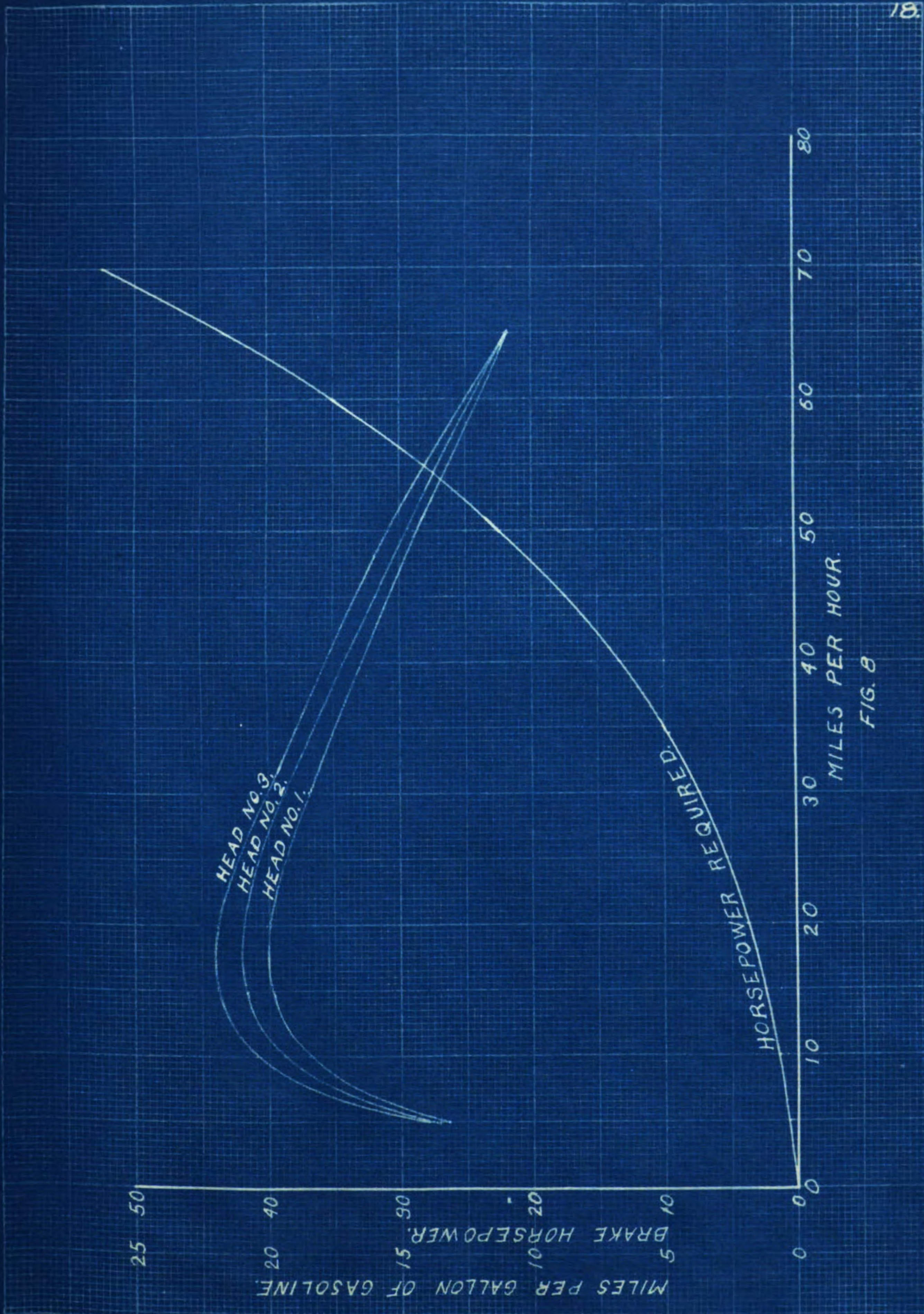

HEAD NO. 3.
HEAD NO. 2.
HEAD NO. 1.
HORSEPOWER REQUIRED.
MILES PER HOUR.
0
10
20
30
40
50
60
70
80
BRAKE HORSEPOWER.
0
10
20
30
40
50
MILES PER GALLON OF GASOLINE.
0
5
10
15
20
25

FIG. 8

Tests to Determine the Effect of Water Upon the Operating Characteristics of the Actual Engine.

To determine the effect of the water in the engine a series of tests were run with wide open throttle and with varying amounts of water. The results of these tests for 1000 R.P.M **are** shown in Table IV and some of the values obtained with a speed of 1000 R.P.M. are shown graphically in Fig. 9. The results obtained with the other heads and speeds were similar and it was thought unnecessary to tabulate them.

Orsat analysis of the exhaust gas was taken but the injection of the water caused variations in the distrubution of the fuel to the cylinders to such an extent that the effect of the water on combustion could not be determined. It is doubtfull if the effect would be sufficient to be noticable as the Orsat apparatus showed only a trace of Carbon Monoxide when there was any excess of Oxygen present.

Considerable difficulty was experienced with the ignition at the higher compressions. Finally a new spark coil was used and this helped to some extent. For successfull operation it would be necessary to secure an ignition system designed for the compressions used.

Table IV.

Effect of Water in the Actual Engine.

Cylinder Head Nos.1,2,3. Full Throttle.

Lbs. Water /lb. of Gasoline	B.H.P.	Lbs. Gasoline /B.H.P. /Hour.	B.M.E.P.	Thermal Eff. %.	Spark Setting Degrees Advance	Exhaust Temperature. °F.
		1000 R.P.M.	Head No. 1.			
0.0	27.8	.584	88.4	23.1	20	950
0.102	27.8	.598	88.4	22.6	20	955
0.258	27.6	.614	87.8	22.0	20	955
0.237	27.6	.724	87.8	18.7	20	955
0.303	27.6	.670	87.8	20.2	20	955
0.381	27.5	.672	87.5	20.1	20	955
0.427	27.5	.679	87.5	19.9	20	945
0.721	27.3	.677	86.8	20.0	23	890
0.994	27.0	.692	85.8	19.5	25	870
1.172	27.0	.697	85.8	19.4	27	830
1.690	26.5	.705	84.2	19.2	30	825
		1000 R.P.M.	Head No. 2.			
0.0	30.2	.612	96.0	22.1	12	940
0.369	30.2	.612	96.0	21.9	13	935
0.695	30.0	.623	95.4	21.7	15	900
0.845	29.6	.613	94.1	22.0	16	895
1.130	29.3	.638	93.1	21.2	18	860
1.850	27.6	.677	87.8	20.0	20	805
		1000 R.P.M.	Head No. 3.			
0.0	30.5	.613	97.0	22.0	12	860
0.240	30.3	.656	96.4	20.6	13	830
0.283	30.0	.656	85.4	20.6	13	820
0.598	29.8	.661	94.8	20.4	12	805
1335	29.4	.666	93.5	20.3	20	720

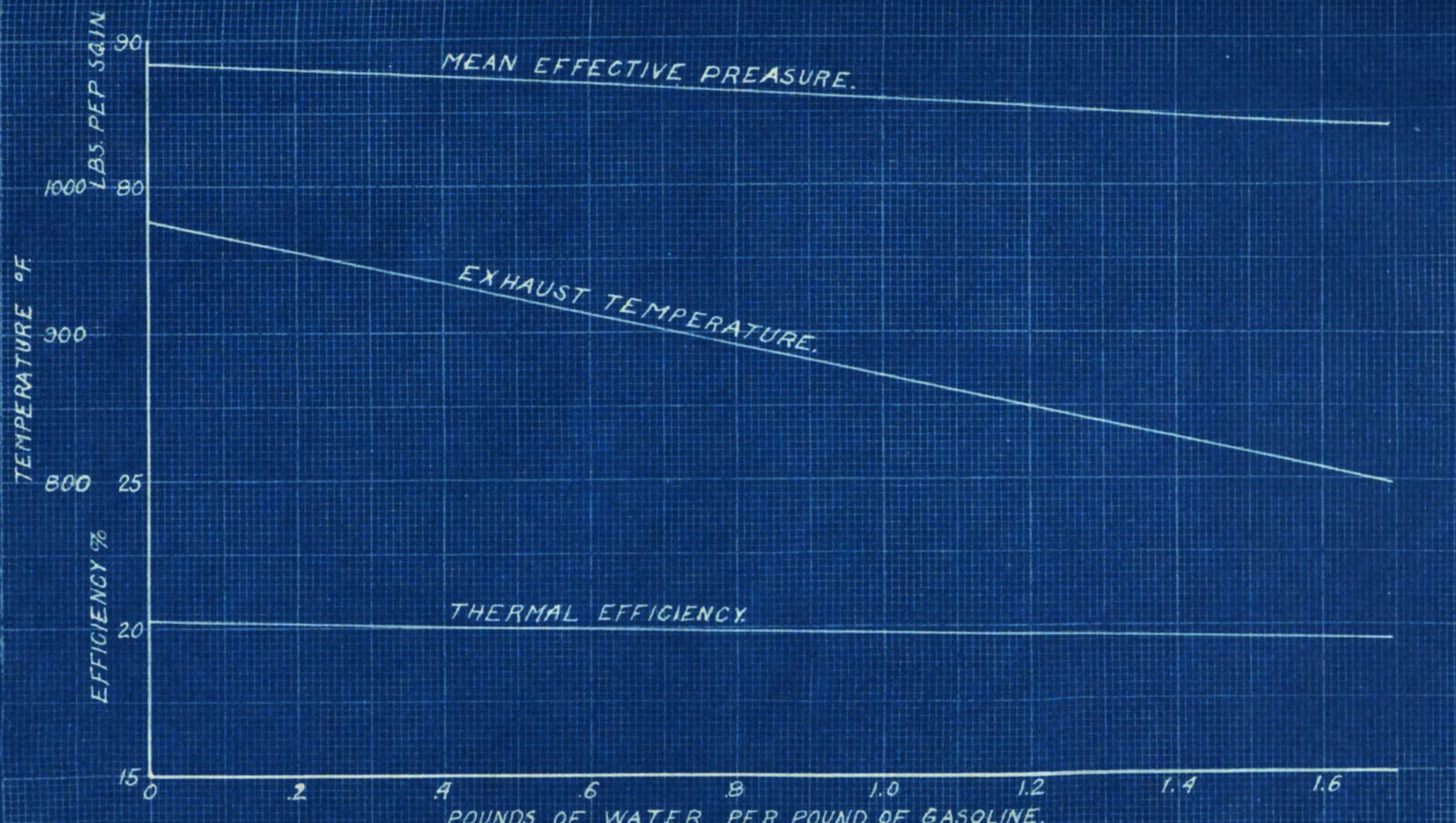

MEAN EFFECTIVE PREASURE.
EXHAUST TEMPERATURE.
THERMAL EFFICIENCY.
TEMPERATURE °F.
LBS. PEP SQ.IN.
EFFICIENCY %
90
80
25
20
15
1000
900
800
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
POUNDS OF WATER PER POUND OF GASOLINE.

FIG. 9.

Calculations to Determine the Effect of Water upon the Ideal Otto Cycle.

The method of analysis for the Otto cycle developed by Goodenough and Baker in "A Thermodynamic Analysis of Internal-Combustion Engine Cycles", Bulletin No. 160, University of Illinois, 1927, was applied to the problem of water injection. The assumption was made that the water was in the vapor state when it entered the cylinder. This was not true for the method of injection used in the tests, however it simplifies the computation of the initial conditions. This assumption entirely neglects the effect of the latent heat of the water which would only cause a lowering of the initial temperature and a corresponding decrease of temperature through out the cycle. For accurate results this effect should not be neglected but for determining the other effects of the water on the cycle it is permissible.

The results of the calculations made in applying the above analysis are given in Table V and graphically in Fig. 10. A sample of the computation and a short discussion of the theory, especially the adaptation to this problem, are given in the Appendix. Subscript (1) refers to the initial conditions at the beginning of compression, (2) to the end of compression and the

Table V.

Calculated Data.

Mols Water/Mol Gas.	0	1	3	6	10
Lbs. Water/Lb. Gas.	0	.158	.473	.947	1.579
M.E.P. Lbs./sq.in.	180.67	178.20	173.33	166.43	157.73
Efficiency %	36.17	36.24	36.37	36.55	36.73
Work B.t.u./Mol Gas	775166	776567	779339	783245	787093
T_1	621	620	619	618	617
p_1	14.7	14.7	14.7	14.7	14.7
T_2	1055	1053	1050	1047	1044
p_2	124.87	124.83	124.68	124.52	124.37
T_3	5007	4956	4860	4724	4558
p_3	632.57	626.41	613.43	594.82	572.12
x_3	.8239	.8358	.8572	.8832	.9114
y_3	.9731	.9749	.9780	.9818	.9858
T_4	3683	3634	3543	3416	3262
p_4	92.10	90.93	88.61	85.35	81.39
x_4	.9792	.9820	.9863	.9910	.9949
y_4	.9957	.9961	.9969	.9979	.9987
Mols unburned CO 3	1.480	1.380	1.201	.983	.748
" " CO 4	.175	.151	.115	.076	.043
" " H_2 3	.254	.264	.278	.288	.285
" " H_2 4	.041	.041	.039	.033	.026

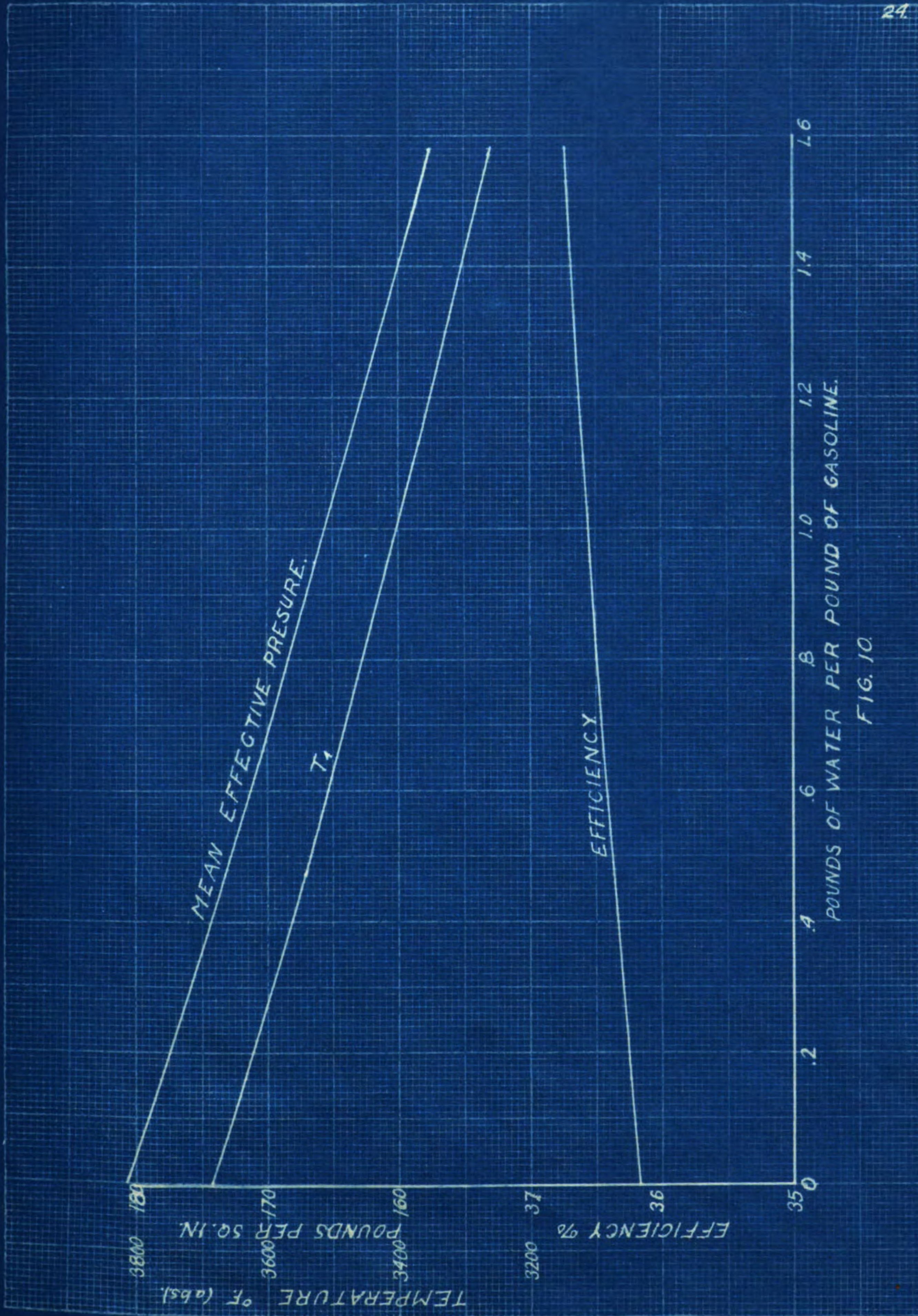

MEAN EFFECTIVE PRESURE.
T_1
EFFICIENCY
POUNDS OF WATER PER POUND OF GASOLINE.
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
POUNDS PER SQ. IN.
180
170
160
EFFICIENCY %
37
36
35
TEMPERATURE °F (abs.)
3800
3600
3400
3200

FIG. 10.

beginning of combustion, (3) to the end of constant volume combustion, and (4) to the end of expansion. The symbols (x) and (y) refer to the percent of carbon monoxide and hydrogen burned to carbon dioxide and water.

The results for point (3) at the end of constant volume combustion are obtained by the use of the equation for chemical equilibrium. From the values of x_3 and y_3 the amounts of unburned carbon monoxide and hydrogen were calculated. The injection of water decreases the amount of the carbon monoxide slightly due to the lower temperature. The increased concentration of the water causes a slight increase in the unburned hydrogen in spite of the lower temperature. However, this is a very slight slowing up of the combustion of the hydregen and can not be interpeted as a dissocation of the injected water. This effect is very small at the temperature obtained with the ideal cycle and would be so small as to be practically negligible at the much lower temperature found in the actual engine.

Conclusions.

The results of these tests and computations are not necessarily conclusive but they seem to indicate that water is a very good anti-detonant and that its effect is not as complicated as previously believed. The effect seems to be that of lowering the temperatures through out the cycle due to its high latent heat and high specific heat and a dilution of the fuel charge. The lower temperature and the dilution prevent pre-ignition, detonation, and chemical "cracking" and cause a slight increase in the completness of combustion. The dilution probably accounts for the decrease in power that we found with increasing amounts of water. The freedom from carbon deposits usually experienced is due to the prevention of chemical "cracking". It is doubtfull if the water injection would have much effect on carbon deposits previously formed. V. W. Brinkerhoff, in the report previously mentioned, gives the results of some tests run for this purpose, however much larger amounts of water were supplied to remove the carbon than were used for anti-detonation purposes. The power developed was greatly reduced and even then the removal of carbon was very slow.

It would be of considerable interest to run a series of tests on an engine designed for high pressures. Preferably a single cylinder engine as that would eliminate the distribution difficulties. A high speed variable compression engine of the type used for testing fuels would be very good for this purpose. A study of cards taken with a high speed indicator would also be of value.

Appendix.

The only deviation from the analysis of Goodenough and Baker in "A Thermodynamic Analysis of Internal-Combustion Engine Cycles", Bulletin No. 160, University of Illinois, 1927, is in the computation of initial conditions. Goodenough and Baker assume the temperature of the residual gas to be the same as the temperature at the end of expansion. For this investigation the temperature of the residual gas has been found as suggested by Professor G. B. Upton in his discussion of the paper "Efficiencies of Otto and Diesel Engines" by Ellenwood, Evans, and Chwang presented at the 1927 meeting of the Oil and Gas Power Division of the American Society of Mechanical Engineers.

As far as the residual gas is concerned the action after the opening of the exhaust valve is a continuation of the adiabatic expansion. Of course there is no usefull work done but the residual gas in expanding is doing work in pushing the rest of the gas out of the cylinder. Or the entire products of combustion can be considered as expanding adiabaticly through an orfice which in this case is the exhaust valve. For this reason the temperature of the residual gas is much lower than the temperature at the opening of the exhaust valve.

The equation pv^k = constant was applied for determining this temperature as it is much simpler than the more accurate analysis and the error incured would have a negligilbe effect on the final results. The exponent k was taken as the ratio of the mean specific heats at constant pressure and constant volume for the particular temperature range involved.

The following sample of the computation shows the method of handling the injection water. As before stated the injected water is assumed to be in a vapor state when taken into the engine.

For a complete discussion of this theory and method of analysis consult the bulletin mentioned above.

Sample Computation for Water Injection.

Otto Cycle. Expansion to Suction Volume.

1. Compression Ratio $= \frac{V}{V_c} = 5.$ Heat Losses : None.

   Fuel : C_8H_{18} (Gasoline) with 100% theoretical air.

2. Suction Pressure = Exhaust Pressure = 14.7 lb./sq.in.

$$C_8H_{18} + 12.5\ O_2 + 47.25\ N_2 + n\ H_2O =$$
$$8\ CO_2 + (9 + n)\ H_2O + 47.25\ N_2.$$

Fuel Mixture, Mols.			Products Mixture, Mols.		
3.	C_8H_{18}	$= 1.00$	8.	CO_2	$= 8.00$
4.	O_2	$= 12.50$	9.	H_2O	$= 9.00 + n$
5.	H_2O	$= n$	10.	N_2	$= \underline{47.25}$
6.	N_2	$= \underline{47.25}$	11.	n_2	$= 64.25 + n$
7.	n_1	$= 60.75 + n$			

Molar Specific Heat Equations:

12. $\gamma_{p(C_8H_{18})} = 38.327 + 38.00 \times 10^{-3}\ T.$

13. $\gamma_{p(O_2,\ N_2,\ CO)} = 6.93 + 0.0 \times 10^{-3}\ T + 0.12 \times 10^{-6} T^2.$

14. $\gamma_{p(CO_2)} = 7.15 + 3.9 \times 10^{-3}\ T - 0.60 \times 10^{-6}\ T^2.$

15. $\gamma_{p(H_2O)} = 8.33 - 0.276 \times 10^{-3}\ T + .423 \times 10^{-6}\ T^2.$

16. $(3) \times (12) = 38.327 + 38.00 \times 10^{-3}\ T.$

17. $((4) + (6)) \times (13) = 414.0675 + 7.17 \times 10^{-6}\ T^2.$

18. $(5) \times (15) = n(8.33 - 0.276 \times 10^{-3}\ T\quad .423 \times 10^{-6} T^2).$

19. $\gamma_{p(\text{Fuel Mixture})} = \frac{(16) + (17) + (18)}{(7)}.$

20. $(8) \times (14) = 57.2 + 31.2 \times 10^{-3}\ T - 4.8 \times 10^{-6}\ T^2.$

21. $(9) \times (15) = (9.00 + n)(8.33 - 0.276 \times 10^{-3}\ T$

$+ .423 \times 10^{-6}\ T^2).$

22. $(10) \times (13) = 327.4425 + 5.670 \times 10^{-6}\ T^2.$

23. $\gamma_{p(\text{Products Mixture})} = \dfrac{(20) + (21) + (22)}{(11)}.$

Let $n = 10$ then:

$\gamma_{p(19)} = 7.5717 + .4981 \times 10^{-3}\ T + .1611 \times 10^{-6}\ T^2.$

$\gamma_{p(23)} = 7.3120 + .3496 \times 10^{-3}\ T + .1200 \times 10^{-6}\ T^2.$

Temperature of Fuel Mixture = 520°

Assume: $T_1 = 600°$ $T_4 = 3270°$ $p_4 = 82$ lb./sq.in.

Let $k = 1.28$ then:

$T_5 = \left\{\dfrac{14.7}{82}\right\}^{\frac{.28}{1.28}} \times 3270 = 2250°$

Mean Temperature between 4 and 5 = 2760°

γ_{pp} (for 2760°) = 9.1910

γ_{vp} (for 2760°) = 7.2055

$k = \dfrac{9.1910}{7.2055} = 1.276$

$T = 3270 \times \left\{\dfrac{14.7}{82}\right\}^{\frac{.276}{1.276}} = 2250°$

24. γ_{pm} = (19) for 560° = 7.9011

25. γ_{pp} = (23) for 1425° = 8.0539

26. $\beta = \dfrac{(25)}{(24)} = 1.0193$

Computation of Initial Conditions.

		600	610	620
	Let T_1 =	600	610	620
27.	$\frac{rT_5}{T_1}$	18.75	18.44	18.15
28.	(27) - 1	17.75	17.44	17.15
29.	$T_5 - T_1$	1650	1640	1630
30.	$T_1 - T_a$	80	90	100
31.	$\frac{(29) \times (26)}{(30)}$	21.02	18.57	16.61

By Graphical Interpolation:

$T_1 = 617°$ $\qquad \frac{n_1}{n_c} = 17.24$

$n_c = \frac{70.75}{17.24} = 4.1038$

Composition of Residual Gas, Mols.

$CO_2 = 8 \times \frac{4.1038}{74.25} = .442$

$H_2O = 19 \times \frac{4.1038}{74.25} = 1.050$

$N = 47.25 \times \frac{4.1038}{74.25} = \underline{2.612}$

4.104

Charge at End of Suction, Mols.

32. C_8H_{18} = 1.000

33. CO_2 = .442

34. H_2O = 11.050

35. N_2 = 49.862

36. O_2 = $\underline{12.500}$

37. n_1 = 74.854

The Adiabatic Compression

Let T_2 =	1030	1040	1050	T_1 = 617
$\emptyset(N_2,O_2)$	34.3690	34.4180	34.4666	31.7940
$\emptyset(CO_2)$	39.5300	39.6127	39.6949	35.4766
$\emptyset(H_2O)$	43.9576	44.0205	44.0829	40.6761
38. $\emptyset(C_8H_{18})$	291.2575	291.9887	292.7164	256.9395
39. ((35) + (36)) $X\emptyset(N_2,O_2)$	2143.3196	2146.3753	2149.4061	1982.7374
40. (33) x $\emptyset(CO_2)$	17.4723	17.5088	17.5451	15.6807
41. (34) x $\emptyset(H_2O)$	485.7315	486.4263	487.1160	449.4709
$n\emptyset$= (38) + (39) + (40) + (41)	2937.7809	2942.2993	2946.7836	2704.8285
$n\emptyset(T_2)-(n\emptyset(T_1))$	232.9524	237.4708	241.9551	---------

4.571 x (37) x $\log_{10}5$ = 239.1579

By Graphical Interpolation:

42. $T_2 = 1044°$

43. $p_2 = \dfrac{(1) \times (2) \times (42)}{T_1}$ = 124.37 lb./sq.in.

44. = 8.4603 Atmospheres.

45. $V_2 = \dfrac{n_1RT_2}{p_2} = \dfrac{(37) \times 1544 \times (42)}{(43) \times 144}$ = 6,737 cu. ft.

The Combustion Process.

Equilibrium Mixture of Gas, Mols.

CO_2 = 8.442x

CO = 8.442(1 - x)

H_2O = 20.050y

H_2 = 20.050(1 - y)

N_2 = 49.754

O_2 = 14.246 - 4.221x - 10.025y

n_e = 92.600 - 4.221x - 10.025y

47. n_1 = 8.442. 48. n_2 = 20.050. 49. n" = 49.862

50. n_s' = 14.246 51. $n_1+n_s'+n''$=72.55. 52. n_p=78.354.

For T_2 = 1044° $H_{v(CO)}$ = 121,450. $H_{v(H_2)}$ = 103,345.

H_m = 2,151,346. n_1H_{CO} =1,025,281. $n_2H_{H_2}$=2,072,067.

$n_1H_{CO} + n_2H_{H_2} - H_m$ = 946,002.

		4500	4550	4600
	T_e	4500	4550	4600
53.	Δu_D (1044 to T)	20,688	21,058	21,432
54.	Δu_{H_2}	20,263	20,614	20,967
55.	Δu_{CO_2}	38,033	38,638	39,245
56.	Δu_{H_2O}	31,972	32,660	33,358
57.	(51) x (53)	1,500,914	1,527,758	1,554,892
58.	½(48) x (53)	207,397	211,106	214,856
59.	3/2 (47) x (53)	261,972	266,657	271,393
60.	(48) x (54)	406,273	413,311	420,388
61.	(47) x (55)	321,075	326,182	331,306
62.	(48) x (56)	641,039	654,833	668,828

63.	(59) - (61) + $n_1 H_{CO}$	966,178	965,756	965,368
64.	(57)+(60)+946,002	2,853,189	2,887,071	2,921,282
65.	$n_2 H_{H_2}$+ (60)-(62)+(58)	2,044,698	2,041,651	2,038,483
	$a = \frac{(63)}{(65)}$	.4725	.4730	.4736
	$b = \frac{(64)}{(65)}$	1.3954	1.4141	1.4331
	$c = K_p$(water gas)	6.6710	6.7744	6.8781
	c(b-1)	2.6377	2.8053	2.9789
66.	c(b-1)-(a+b)	.7698	.9182	1.0722
67.	$(66)^2$	.5926	.8431	1.1496
68.	2a(c-1)	5.3591	5.4626	5.5677
69.	2b x 2a(c-1)	14.9562	15.4493	15.9581
70.	(67) + (69)	15.5488	16.2924	17.1077
71.	$\sqrt{(70)}$	3.9432	4.0364	4.1361
72.	(66) + (71)	4.7130	4.9546	5.2083
73.	$x = \frac{(72)}{(68)}$	.8794	.9070	.9354
	ax	.4155	.4290	.4430
74.	b-ax = y =	.9799	.9851	.9901
	n_s'	14.246	14.246	14.246
	$\frac{1}{2}n_1 x$	3.7119	3.8284	3.9483
	$\frac{1}{2}n_2 y$	9.8235	9.8756	9.9258
	$n_e' = n_s' - \frac{1}{2}n_1 x - \frac{1}{2}n_2 y$	.7106	.5420	.3719
	$\frac{1}{2}\log n_e'$	$\bar{1}.92582$	$\bar{1}.86700$	$\bar{1}.78522$
	log x	$\bar{1}.94419$	$\bar{1}.95761$	$\bar{1}.97100$
	log (1-x)	$\bar{1}.08135$	$\bar{2}.96848$	$\bar{2}.81023$
	$\frac{1}{2}\log \frac{(37) \times (42)}{(44)}$	1.98277	1.98277	1.98277

75. $R(x, y) = \log x$

$-\log(1-x) - \log n_e'$			
$+\frac{1}{2}\log \frac{(37) \times (42)}{(44)}$	2.91979	3.10490	3.35832
$\frac{1}{2}\log T_3$	1.82661	1.82901	1.83130
$\log K_p(CO)$	1.38659	1.32307	1.26092

76. $L(T) = \frac{1}{2}\log T_3 +$

$\log K_p(CO)$	3.21320	3.15208	3.09230

By Graphical Interpolation:

77. $T_3 = 4558°$

$x_3 = .9114$

$y_3 = .9858$

$n_e = 92.600 - 4221x - 10.025y = 78.8704.$

78. $p_3 = \frac{n_e T_3 p_2}{n_1 T_2} = 572.1227$ lb./sq.in. $= 38.9199$ Atm.

The Expansion Process.

Let T_4 be 3270° then K_p(water gas) = 4.0083

Let x_4 =	.994	.996	.998
cx	3.9843	3.9923	4.0003
$cx + (1-x)$	3.9903	3.9963	4.0023
$y_4 = \dfrac{cx}{cx + (1-x)}$	.9985	.9990	.9995
n_s'	14.246	14.246	14.246
$\frac{1}{2}n_1 x$	4.1957	4.2041	4.2126
$\frac{1}{2}n_2 y$	10.0100	10.0150	10.0200
$n_e' = n_s' - \frac{1}{2}n_1 x - \frac{1}{2}n_2 y$	.0403	.0269	.0134
$\frac{1}{2}\log n_e'$	$\bar{1}.30266$	$\bar{1}.21408$	$\bar{1}.06355$
$\log x$	$\bar{1}.99739$	$\bar{1}.99826$	$\bar{1}.99913$
$\log (1-x)$	$\bar{3}.77815$	$\bar{3}.60206$	$\bar{3}.30103$
$\frac{1}{2}\log \dfrac{n_e T_3 r}{p_3}$	2.33224	2.33224	2.33224
79. $R(x,y) = \log x - \log (1-x) - \frac{1}{2}\log n_e' + \frac{1}{2}\log \dfrac{n_e T_3 r}{p_3}$	5.24082	5.51356	5.96679

Let T_4 =	3260	3270	3280
$\frac{1}{2}\log T_4$	1.75661	1.75728	1.75794
$\log K_p(CO)$	3.59223	3.56765	3.54322
80. $L(T) = \frac{1}{2}\log T_4 + \log K_p(CO)$	5.34884	5.32493	5.30116

T_4	3260	3270	3280
By Interpolation x	.9949	.9947	.9945
$c = K_p$(water gas)	3.9848	4.0083	4.0318
cx	3.9645	3.9871	4.0096
$cx + (1-x)$	3.9696	3.9924	4.0151
$y = \dfrac{cx}{cx + (1-x)}$	.9987	.9987	.9986

	T_4	3260	3270	3280	T_3 =4558
	$H_p(CO)$	119,789	119,776	119,763	118,924
81.	$\dfrac{n_1(1-x)H_p}{T}$	1.5820	1.6309	1.6953	19.5153
	$H_p(H_2)$	107,337	107,341	107,345	106,327
82.	$\dfrac{n_2(1-y)H_p}{T}$	.8582	.8556	.9186	6.6416
	$\phi_{Diat.}$	40.6406	40.6597	40.6787	42.9067
	ϕ_{CO_2}	51.3103	51.3457	51.3811	55.2778
	ϕ_{H_2O}	52.6761	52.7066	52.7370	56.5905
83.	ϕ_D x (49)	2026.4216	2027.3740	2028.3213	2139.4139
84.	ϕ_{CO_2}x(47)	433.1616	433.4604	433.7592	466.6552
85.	ϕ_{H_2O}x(48)	1056.1558	1056.7673	1057.3769	1134.6395
	log x	$\bar{1}.99778$	$\bar{1}.99769$	$\bar{1}.99760$	$\bar{1}.95971$
86.	$n_1$log x	-.01874	-.01950	-.02026	-.34013
	log y	$\bar{1}.99944$	$\bar{1}.99944$	$\bar{1}.99939$	$\bar{1}.99379$
87.	$n_2$log y	-.01123	-.01123	-.01223	-.12451
	log r	.69897	.69897	.69897	0.0
88.	$n_p$log r	54.76710	54.76710	54.76710	--------
89.	(86)+(87)				
	-(88)	-54.79707	-54.79783	-54.79959	-.46464

	T_4	3260	3270	3280	T_3 = 4558
90.	4.571(89)	-250.4774	-250.4809	-250.4889	-2.1239
	S = (81) + (82) + (83) + (84) + (85) - (90)	3768.6566	3770.5771	3772.5602	3768.9894
91.	$S_4 - S_3$	-.3328	1.5877	3.5708	---------

By Graphical Interpolation:

$T_4 = 3262$

$x_4 = .9949$

$y_4 = .9987$

$n_4 = 92.600 - 4.221x - 10.025y = 78.3885.$

$p_4 = \dfrac{n_4 T_4(2)}{(37)T_1} = 81.39$ lb./sq.in.

$V_4 = \dfrac{n_4 \times 1544 \times T_4}{144 \times p_4} = 33{,}685$ cu. ft.

Work Done During the Cycle.

$T = 617°$ $T = 3262°$

92. $H_{v(C_8H_{18})}$ at $T_1 = 2,147,414$

93. $n_1(1-x)\ H_{v(CO)}$ at $T_4 = 5,018$

94. $n_2(1-y)H_{v(H_2)}$ at $T_4 = 2,713$

95. $n_1\Delta u_{CO_2} = 226,094$

96. $n_2\Delta u_{H_2O} = 405,591$

97 $n''\Delta u_D = 720,905$

Work = (92) - (93) - (94) - (95) - (96) - (97) =

Work = 787,093 B.t.u. per Mol of Gasoline.

Efficiency of the Cycle.

$H_{p(C_8H_{18})}$ at $520° = 2,143,000$

$$\text{Eff.} = \frac{787,093 \times 100}{2,143,000} = 36.7286\ \%.$$

Mean Effective Pressure.

$V_4 - V_3 = 26,948$ cu. ft. Displacement.

$$\text{M.E.P.} = \frac{\text{Work} \times J}{(V_4-V_3)144} = \frac{787.093 \times 777.64}{26,948 \times 144} = 157.73 \text{ lb/sq.in.}$$

-----Finis-----

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