MIXTURE CONTROL TESTING OF A SPARK-IGNITION ENGINE WITH ANTI-DETONANT ADMISSION

by

JOHN GLENN MINGLE, JR.

A THESIS

submitted to

OREGON STATE COLLEGE

in partial fulfillment of
the requirements for the
degree of

MASTER OF SCIENCE

June 1949

APPROVED:

Redacted for Privacy

Professor of Automotive Engineering

In Charge of Major

Redacted for Privacy

Head of Department of Mechanical Engineering

Redacted for Privacy

Chairman of School Graduate Committee

Redacted for Privacy

Dean of Graduate School

E.W. 7/6/49

ACKNOWLEDGMENT

The author is sincerely grateful to W. H. Paul, Professor of Automotive Engineering, for his excellent technical assistance; to S. H. Graf, Head of the Department of Mechanical Engineering, and Standard Oil Company of California for their consideration in awarding the fellowship enabling this graduate work; and to Mrs. Stephanna M. Mingle who aided in the preparation of this manuscript.

TABLE OF CONTENTS

LIST OF FIGURES

MIXTURE CONTROL TESTING OF A SPARK-IGNITION ENGINE WITH ANTI-DETONANT ADMISSION

I. INTRODUCTION

Since the work reported herein concerns the operation of an internal combustion engine, it seems fitting to present a brief review of the combustion process and the variables which tend to influence the performance of the engine.

THE COMBUSTION PROCESS

The combustion process may be described as a process whereby chemical energy of the fuel is released to form kinetic energy in a gas. This kinetic energy is evidenced by a rise in temperature and pressure in the combustion chamber. Thus, performance of an engine depends primarily on the energy of the mixture that is supplied to the engine and on the efficiency with which the engine converts this energy supply into work. (5, p. 461)*

The principal basic items which regulate the combustion process are the temperatures and the pressures throughout the cycle, the time available for the combustion process to occur, and the rate of flame propagation across

---

*Numbers in parentheses refer to bibliography.

the combustion chamber. (4, p. 190)

The limiting factor of the rate of combustion is detonation. Although many investigations have been made, present knowledge on the combustion mechanism leading to detonation is not complete. One of the most recent descriptions of this mechanism has been presented by Edward F. Obert.

> From experimental evidence the following sequence of events appears probable: A flame is initiated by the spark plug and travels at a rapid but orderly rate across the combustion chamber. The release of chemical energy during this process causes the temperature, and therefore the pressure, to increase. The first portion of the mixture to be burned expands and compresses the other portions. This compression increases the temperature of the unburned mixture. Similarly, when the final portion is burned and expands, the previously burned portions are compressed, with a consequent rise of temperature.
>
> While the flame is traveling across the combustion chamber, the portion of mixture being compressed ahead of the flame may reach a spontaneous ignition temperature. If the mixture is held at or above this critical temperature for a finite time an explosive reaction, called detonation, may occur without awaiting the arrival of the flame. Apparently in some cases the flame may pass entirely through the end portion of the charge before detonation occurs. This would indicate that the release of chemical energy is not completed in the flame front.
>
> In the normal operation of an engine, detonation occurs as the piston is proceeding downward on the expansion stroke; hence, when a mixture detonates, the volume of the combustion chamber is smaller than it would be for a burning period that took a greater amount of time. Since the combustion time is shortened, more energy will be released before the piston has traveled far on the expansion stroke. Because of this effect, if for

> no other reason, detonation in the engine should cause higher average combustion temperatures. Note that the momentary pressure unbalance created by detonation will travel through the chamber and compress the initially burned (hottest) portion to the extreme pressure of detonation, thus momentarily raising its temperature still higher. (7, p. 52)

PERFORMANCE VARIABLES

Most of the variables which affect the performance of a four-stroke cycle, spark-ignition, gasoline engine are those variables which affect the combustion process. If one particular engine is considered, those variables involving the physical characteristics of the engine such as piston speed, distribution, and combustion chamber design may be regarded as constants. Also, absolute manifold pressure, per cent of rated load applied to the engine, and engine speed are the principal items which determine the power datum point. Therefore, these items may be classed as independent variables.

The variables that have the most marked effect on the combustion process and its limiting factor, detonation, are air-fuel ratio, time of ignition, and fuel characteristics.

1. Air-Fuel Ratio. Maximum power output is obtained when there is sufficient fuel in the air-fuel mixture to obtain complete combustion with all the oxygen

present in the combustion chamber. This requires an air-fuel ratio of approximately thirteen to one. A lower air-fuel ratio (richer mixture) than this best-power mixture results in a loss of power and efficiency which produces a higher brake specific fuel consumption. A higher air-fuel ratio (leaner mixture) than that of the best-power mixture results in a power loss but yields an increase in efficiency and a lower brake specific fuel consumption up to the air-fuel ratio giving best economy. Best economy is obtained at an air-fuel ratio of approximately 15 to 1. As the air-fuel ratio is increased beyond the point of best economy, both the power and efficiency decrease, thereby, yielding a higher brake specific fuel consumption. Mixtures other than the best-power mixture have slower rates of combustion. Mixtures leaner than the chemically-correct mixtures are most susceptible to detonation due to the slower rate of combustion and a high prevailing temperature.

2. Time of Ignition. For maximum power, the mixture must be ignitied at such a time during the compression stroke that the maximum pressure during the cycle will occur just after the piston has reached the top dead center position. A spark retard from this best-power timing results in a power loss because the peak pressure occurs too late in the cycle and will be reduced in

magnitude. A spark advance from this best-power timing also results in a power loss because the peak pressure will occur too soon in the cycle producing excessive negative work at the end of the compression stroke due to the excessive maximum pressure before the piston has reached the top dead center position. The combination of excessive combustion pressure and the corresponding temperature is most conducive to detonation.

3. Fuel Characteristics. The three characteristics of gasoline which affect the combustion process are the molecular structure, the rate of burning, and the temperature of self-ignition. The molecular structure is related to the knocking tendency of the fuel, and in general, it appears that the more compact molecular structures are associated with lower knocking tendencies. (6, p. 157) The self-ignition temperature determines the degree of temperature and pressure rise in the cylinder without experiencing detonation. The rate of burning is responsible for the rate of flame propagation. The combination of rate-of-flame propagation, self-ignition temperature, and temperature rise, determine whether detonation occurs. Obviously there will be no detonation if the flame progresses through the unburned fraction before the temperature rises to a point of self-ignition.

The one variable that is the key to the performance

of a combustion engine is the compression ratio. It has been pointed out in the preceding paragraphs that the limiting factor of the rate of combustion is detonation. It has also been pointed out that the two most important factors in promoting detonation are the temperatures and pressures existing during the combustion process. If the compression ratio is increased the pressures and temperatures in the cycle will also increase. Thus, increasing the compression ratio lends the process more susceptible to detonation. (6, p. 159)

However, it has long been recognized that the efficiency of spark ignition engines can be improved substantially by raising the compression ratio. This knowledge dates back some 65 years to the time of Sir Dugald Clerk who was the first to deduce both on theoretical and practical grounds that the Otto cycle engine was the most efficient of the engines then in use because of its expansion ratio. Moreover, he also recognized knock or "pre-ignition" as he called it as one of the limiting factors to high compression ratios. (1, p. 1)

SUPPRESSING DETONATION

The problem of eliminating the knock induced by the higher compression ratio has been solved by increasing the antiknock quality of the fuel. However, as pointed out by

Holaday, an increase in the overall antiknock quality from the present level requires an excessive capital investment by the refiners, and is obtained at the expense of decreased production of fuel oil fractions and straight-run Diesel fuel. (3, p. 6)

Thus, in the interest of economy, supply and demand, and conservation of natural resources, the engineer has been compelled to investigate the possibilities of obtaining higher efficiency by utilizing a higher compression ratio with a low octane fuel and suppressing the resulting detonation by some other means. The addition of small quantities of tetraethyl lead to gasoline will effectively increase the antiknock quality, but this process has some limitations. The Surgeon General of the United States has specified that the quantity of tetraethyl lead per gallon shall not exceed three milliliters. Also, the economical relationship between the quantity of tetraethyl lead added and the effective increase in antiknock quality is governed by the lead susceptibility of the gasoline.

Recent developments indicate that much can be accomplished in suppressing detonation by the introduction of certain internal coolants with the charge at the proper time and in the correct quantity. However, this is not a new practice. As early as 1913 it was found that not only was the charge cooled and pre-ignition prevented by water

injection, but in addition, the cylinder walls, head, and pistons were cooled, doing away entirely with the cylinder cooling jackets. (8, p. 2)

Recent investigations have substantiated and elaborated on this theory. The following has been presented by A. T. Colwell and associates.

> The effect of any cooling agent upon detonation is classified by thinking of the process of combustion in stages. Taking for example an average passenger-car engine, stage one is the ignition; the gas pressure at this stage is compression pressure or around 150 psi. The second stage is when the flame has traveled about one-third the distance; the gas pressure in the unburned portion then is about 230 psi. The third stage is when the flame front has moved two-thirds of the way across the combustion chamber; the gas pressure in the unburned one-third being 350 psi. From this point on, what happens depends on the temperature of the last gas to burn. This gas, if in the hot area, would be ready to detonate with very little more flame movement. If this last gas is in a cool area, or has had the benefit of a cooling medium, or its temperature is held down by the work done and the work going on in vaporizing a cooling medium, then, obviously, the flame front completes the travel without self-ignition taking place in this last gas, and thus there is no detonation. (2, p. 358)

In determining the best cooling medium, in view of the above, it might be concluded that the best coolant would be the one having the highest latent heat of vaporization. It is not conclusive however that the latent heat of vaporization is the only factor involved.

Colwell points out:

> Alcohol, when used alone as a motor fuel, has an octane rating of 90 to 100, but tends to

> have a higher value when used as a blending agent. Results indicate that burning characteristics of the charge are altered, reducing the rate of pressure rise, lowering peak values, and fattening the indicator diagram. The bmep thus developed is, therefore, greater because the expansion pressure points on the diagram are in a more advantageous position relative to the crank throw, whereas a higher peak pressure be so placed that the force applied to the crank is less effective in producing torque. Water itself will reduce the degree of pressure rise and peak pressure (primary cause of detonation), but will contribute little toward fattening the diagram. However, if a slow-burning fuel, such as alcohol, is used as an anti-detonant instead of, or with, water, the peak pressure will appear at a greater number of degrees past top center and the diagram will have a flatter top, fattening the diagram on the expansion side. (2, p. 358)

Another phase of the same problem has been presented by Rowe and Ladd.

> Basically, calculations have shown that the cylinder-head temperature cooling obtained using water or water-alcohol injection represents approximately 30 to 40% of the available heat of vaporization of the injected water. However, this heat of vaporization is not considered to be the total resultant effect of water injection. Test result calculations on the compression of gaseous fluids have demonstrated that the work exponential of the basic equation $PV^n = K$ will be reduced as water is injected into the work cycle, and that this reduction may amount to 10-12%. This change indicates that the specific heat during the combustion cycle would have to be taken into account in any truly theoretical analysis of water injection. Too, calculations made at absolute intake charge pressure conditions, for constant speed and increasing horsepower at constant cylinder-head temperatures, show that the slope of specific brake water consumption does not change when going from a pressure condition of complete vaporization to that where the pressure is too high to permit vaporization. This is important and indicates that the major cylinder-head temperature cooling

> effect is obtained during the high temperature and pressure peaks of the combustion cycle, since it does not appear to matter whether the water enters the cycle as either vapor or liquid.
>
> General experience at the Wright Aeronautical Corp. has taught that water and water-methanol mixtures are best suited for detonation inhibitors and internal coolants. The addition of 50% methanol to water by volume is considered to be optimum for water-methanol mixtures. Tests with pure alcohol, either methyl or ethyl, have indicated decreases in detonation limited power outputs as compared to 100-octane fuel. Investigation of water vapor injection into induction air ahead of supercharging has shown that this medium is an effective detonation inhibitor at lean fuel-air ratios only. (9, p. 26)

The quantity of anti-detonant necessary to suppress detonation depends on a number of factors such as the difference between the octane rating of the fuel and the octane requirement of the engine, engine speed, the mixture ratio, and the fuel characteristics. Colwell found that the quantity of alcohol-water required for best border line knock performance ranged from 25.5 per cent by weight at low speed to 16 per cent at high speed, with a maximum quantity admitted at the speed of maximum torque. (2, p. 363) Most of the investigators have found by experience that the quantity of internal coolant should not exceed 20 per cent of the total liquid charge.

Thus, a summation of the opinions of investigators in the field are as follows:

1. Detonation may be suppressed by internal

coolants.

2. Two fluids well suited for this purpose are water and a mixture of 50 per cent water and 50 per cent methyl alcohol by volume.
3. The average optimum quantity of internal coolant is approximately 20 per cent of the total liquid charge.
4. These two coolants have different operating characteristics.
    a. Water alone offers the best rate of cooling.
    b. Water-methanol mixtures permit the greatest power output.
5. Water is most effective at lean mixtures only.

II. TEST PROCEDURE

The object of this investigation was to determine the merits of two selected internal coolants with regard to their relative performance at full-load over a range of air-fuel mixture ratios and engine speeds.

The internal coolants were; (1) water, and (2) a mixture of 50 per cent methyl alcohol and 50 per cent water by volume. These two coolants were selected because in the opinion of major investigators in this field, these two coolants have many advantages over other liquids.

The air-fuel mixture ratio range was from 10 to 17. These particular limits were selected because they represent the limit of useable mixture ratios in a spark-ignition internal-combustion engine.

The engine speeds at which the observations were made were 1200 rpm, 1800 rpm, 2400 rpm, and 3000 rpm. These four speeds were selected because they are representative of the engine's useful full-load power range.

In order to determine the merits of the two selected internal coolants, twelve performance tests were performed. Four tests, one for each of the four engine speeds selected, were completed by operating the test engine on the reference fuel without admitting an internal coolant. Four tests were made by operating the test engine on the reference fuel and admitting water as the internal

coolant, and four tests were completed operating the test engine on the reference fuel and admitting an alcohol-water mixture as the anti-detonant. The independent variable during each test was the air-fuel ratio.

In order to complete each test, it was necessary to conduct a series of observations. Each individual observation was made at a different fuel rate while all other variables were held rigidly constant. During each observation proper data were taken in order to determine brake specific fuel consumption, corrected brake horsepower, brake thermal efficiency, and the air-fuel ratio. The results of these determinations were plotted, corrected brake horsepower as the abscissa and brake specific fuel consumption as the ordinate, forming the mixture control curve. Typical mixture control curves may be seen in Figure 1. Note that the maximum power points and the best-economy points are sharply defined. Each point on the curve represents an air-fuel ratio corresponding to the fuel rate at which that particular observation was made.

The procedure for conducting each series of observation at the selected engine speed consisted mainly of making certain that the engine had reached equilibrium, regarding operating temperatures and load, to the end that a constant engine speed was maintained with a variation of plus or minus ten revolutions per minute. In order to

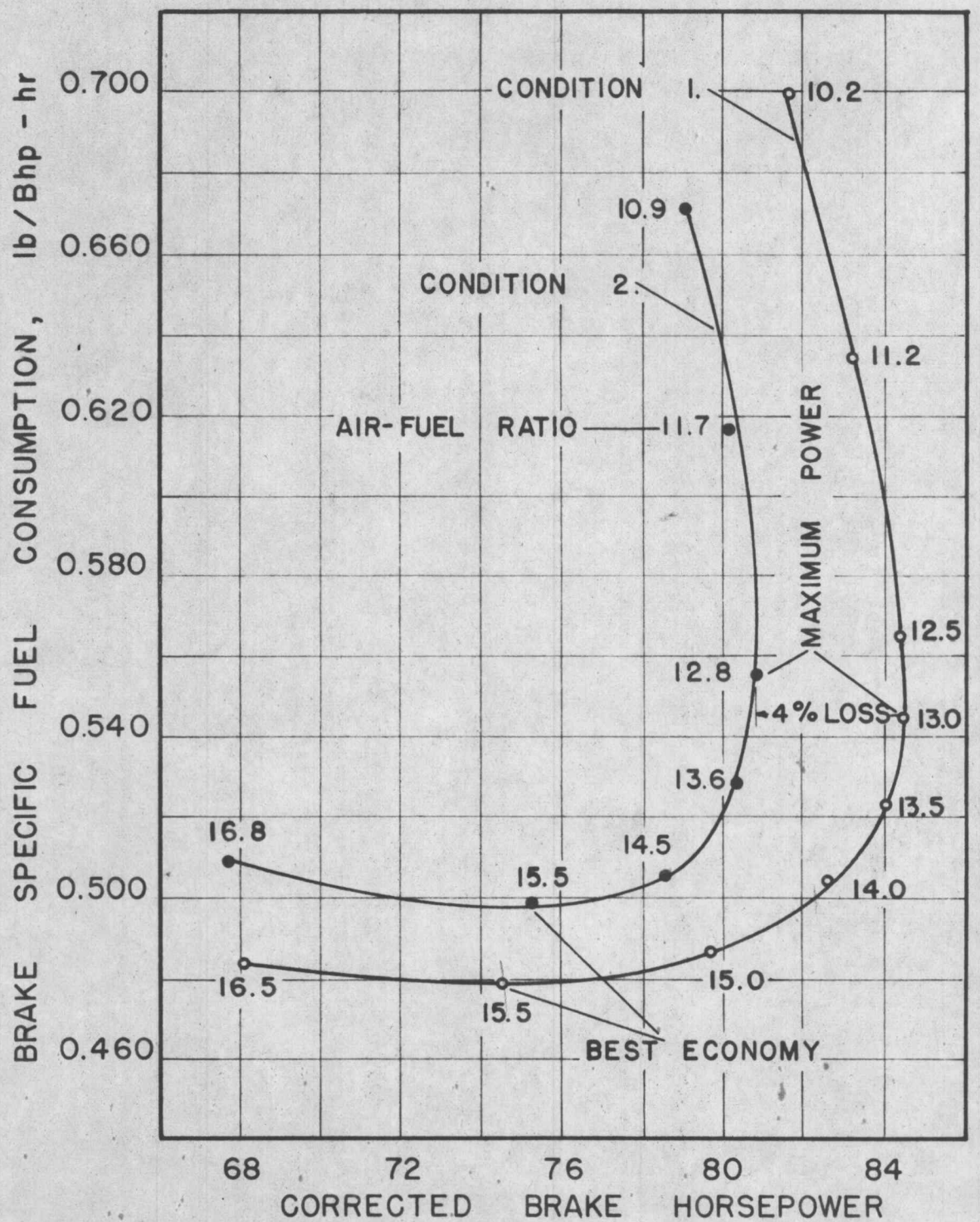

FIGURE 1. TYPICAL MIXTURE CONTROL CURVES.

determine the first point on each curve, the needle valve on the carburetor was adjusted to meter a fuel rate to give an air-fuel ratio of approximately ten to one. After the engine had reached equilibrium and maintained a constant speed, two consecutive observations were made of the following data: engine speed--rpm, net beam load--pounds, spark advance--degrees BTC, fuel rate--seconds for one-half pound of fuel, water temperature in--degrees Fahrenheit, water temperature out--degrees Fahrenheit, oil temperature--degrees Fahrenheit, the pressure drop across the orifice of the air meter--inches of water, internal coolant consumption--seconds for one-tenth pound, wet and dry bulb temperatures--degrees Fahrenheit, barometric pressure--inches of mercury, and ambient temperature at the barometer--degrees Fahrenheit. The two consecutive observations should coincide within one per cent deviation. Before progressing to the next fuel rate, the brake specific fuel consumption, corrected brake horsepower, and the air-fuel ratio were computed and the point plotted on the curve.

The above operation was repeated, each time decreasing the fuel rate to give approximately the next higher air-fuel ratio until the leanest feasible mixture was obtained. It was not considered feasible to operate the engine on an air-fuel ratio greater than seventeen to one

or on any air-fuel ratio that gave a power loss greater than 25 per cent. As the test progressed, it was at times necessary to adjust the fuel rate to give an intermediate air-fuel ratio in order to locate an optimum point on the curve or to bracket a particular point about which there was some doubt.

It should be reiterated that, in order to obtain data on engine performance which were intended to be compared quantitatively, all variables had to be held constant at their selected operating levels with the exception of the fuel rate.

The brake thermal efficiencies for each observation were plotted, thermal efficiency as ordinate and air-fuel ratio as abscissa. The maximum thermal efficiency corresponding to the point of best economy was sharply defined.

III. EQUIPMENT

The principal equipment used in this investigation consisted of the following: a four-stroke cycle, liquid-cooled, gasoline engine; a direct-connected electric dynamometer for applying the load; appropriate equipment for measuring the consumption of air, fuel, and internal coolant; and small instruments necessary for determining engine speed, atmospheric pressure, atmospheric water vapor pressure, and temperatures.

ENGINE

The test engine, Figure 2, had the following general specifications before being slightly modified to accommodate the work reported herein:

Manufacturer	Nash-Kelvinator Corporation
Year	1941
Model	4160
Number of cylinders	6
Bore, inches	3-3/8
Stroke, inches	4-3/8
Displacement, cubic inches	234.8
Compression ratio	6.3
Rated output, brake horsepower	105
Rated speed, revolutions per minute	3400
Number of main bearings	7

This engine is a valve-in-head type, equipped for dual-ignition. The carburetor, an Ensign, has an adjustable main jet for control of the fuel rate. The spark advance is controlled manually. An external oil pump, seen

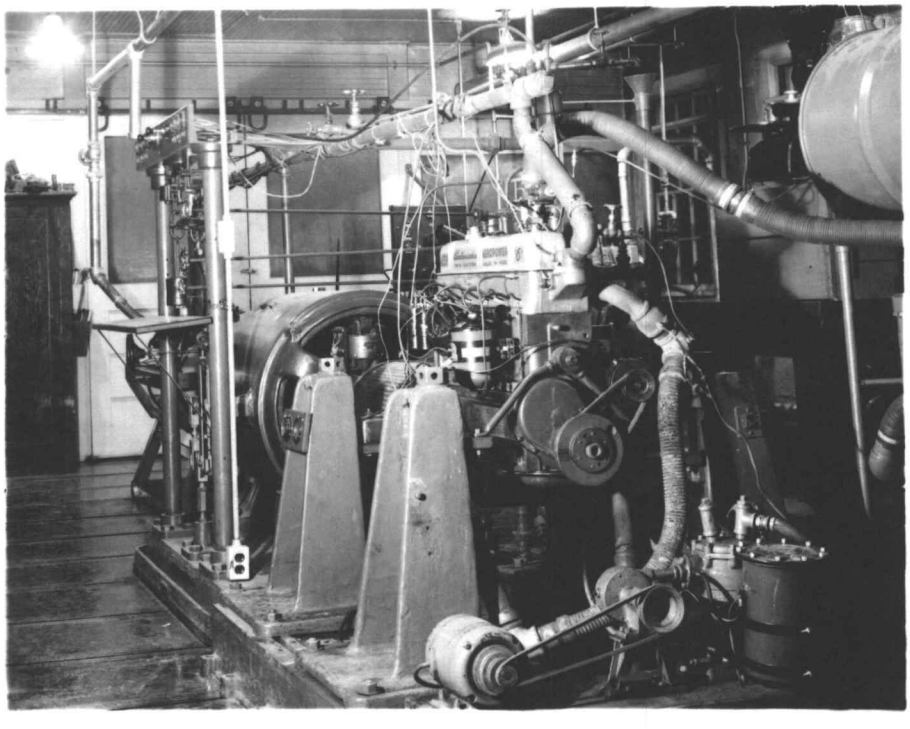

Figure 2. The Test Engine and Dynamometer

in foreground of Figure 2, circulates oil from the crankcase of the engine through a heat exchanger and back to the crankcase. The rate of cooling water flowing over the heat exchanger is controlled manually to maintain the lubricant at a constant temperature. The cooling water for the engine's block and head is circulated through a heat exchanger by the engine-driven water pump. The temperature of the water entering the block is regulated by controlling the rate of water passing over the heat exchanger. The temperature of the water leaving the head is controlled by a gate valve in the outlet line, thereby controlling the rate of circulation.

DYNAMOMETER

The electric dynamometer, Figure 2, a General Electric, one hundred horsepower, direct-current generator, is cradled in ball bearing trunions, and the turning effort exerted by the stator was measured directly in pounds on a beam scale.

The restraining torque was controlled from the control panel, Figure 3, by increasing or decreasing the field resistance of the generator. The electrical energy was dissipated into banks of cast iron resistors. The generator was separately excited by means of a 7.5 kilowatt motor-generator set. This type of unit is one of the most

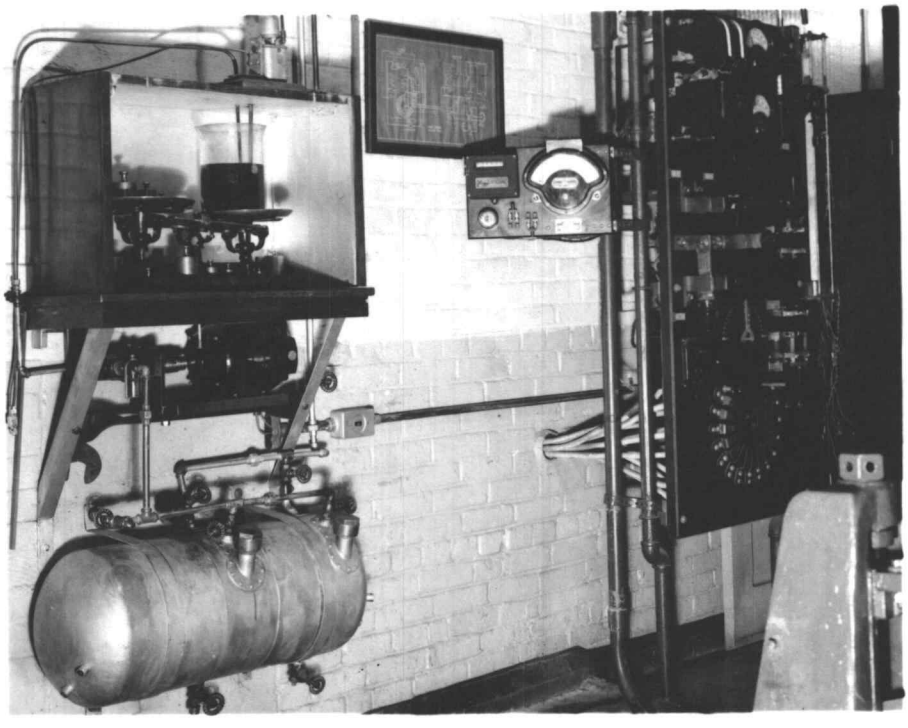

Figure 3. Fuel Weighing System and Load-Control Panel

accurate of all torque-measuring devices. It is estimated that error is less than one per cent.

FUEL-CONSUMPTION MEASURING EQUIPMENT

A pan balance mounted in a cabinet, Figure 3, provided the means of measuring the fuel consumed by the test engine. The fuel, contained in a tank below the cabinet, is transferred to a three-liter beaker on the balance pan by means of a gear pump driven by a direct-current motor. The pump and motor are mounted directly underneath the cabinet. An Autopulse fuel-transfer pump, mounted on top of the cabinet, is used to boost the fuel from the beaker to the engine-driven fuel pump. A stop watch, which could be read to one-tenth second, was employed to determine the time for the engine to consume one-half pound of fuel.

Measurements were always started with the fuel at the same level in the beaker to avoid any error that might be introduced by the effects of bouyancy on the fill and supply tubes immersed in the gasoline. The fuel-measuring system is accurate to within one per cent.

AIR-CONSUMPTION MEASURING EQUIPMENT

An orifice meter assembly, Figure 4, was employed to measure the weight of air consumed by the engine per

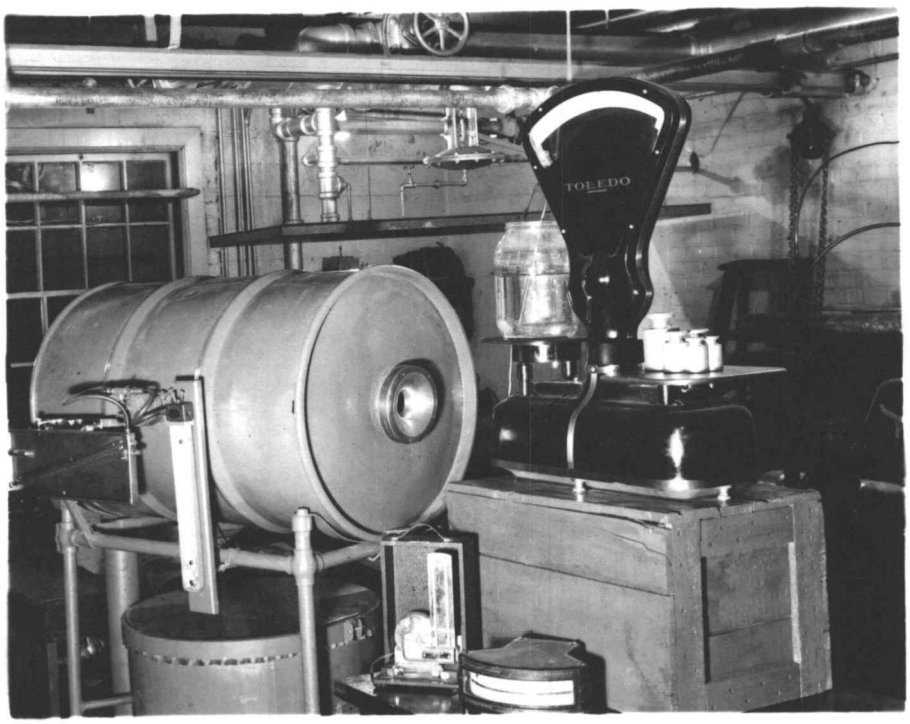

Figure 4. Air Meter and Anti-Detonant Weighing System

unit of time. The orifice is 2.125 inches in diameter and is mounted in the end of a fifty-five gallon steel drum. A three-inch inclined draft gauge, readable to 0.01 inch of water, is utilized to obtain the pressure drop across the orifice. A second drum, the steel ends of which had been removed and covered with a sheet of rubber, is connected between the engine and the orifice drum. The rubber heads help dampen the surge of air and to steady the column of water in the draft gauge. The error involved in computing pounds of air consumed per hour with this equipment is estimated at one per cent.

ANTI-DETONANT-CONSUMPTION MEASURING EQUIPMENT

The internal coolant was admitted into the manifold of the engine at the junction of the manifold and downdraft carburetor flanges. The anti-detonant was contained in a one-gallon glass jar which was placed on one pan of a Toledo pendulum balance, Figure 4. The internal coolant was allowed to flow to the engine by gravity, assisted by the manifold vacuum, and the rate of flow was controlled by a manual-set needle valve. A stop watch was used to determine the time for consumption of one-tenth of a pound of the internal coolant. Since the rate of flow was very low, the error introduced was approximately three per cent.

MISCELLANEOUS

A Hahn and Kolb chrono-tachometer was employed to determine the speed of the engine in revolutions per minute. This instrument was calibrated on a synchronous electric motor and found to be accurate to within three-tenths per cent. A number of speed measurements were obtained during each observation period and an average value was used in computing brake horsepower.

An H and B Instrument Company, Type MP motorized hygrometer was used to determine the atmospheric water vapor pressure. The vapor pressure and the dry bulb temperature were used in determining the factor for correcting the observed brake horsepower to standard conditions. Atmospheric pressure was determined from a standard mercury barometer. Corrections were made for ambient temperature of this instrument.

MODIFICATIONS

Certain modifications were made on the test engine. First, the compression ratio was increased from 6.3 to 7.0. This was done in order to increase the engine's octane-requirement of the fuel. The increase in compression ratio was accomplished through decreasing the combustion recess by removing a layer of metal, 0.087 inch in depth, from the

joining-face of the head.

Second, the distributor and the ignition system were modified. The centrifugal spark advance mechanism was made inoperative by locking the weights in the fully-retarded position. An arm was attached to the distributor housing for manual control of the spark timing. The fly wheel of the engine was calibrated in five-degree intervals, from ten degrees after top dead center to sixty degrees before top dead center, to correspond with the piston travel of numbers one and six cylinders. Time of ignition was observed with a pistol-type "Sun" power timing light, and could be determined to within plus or minus one crankshaft degree. The left bank of ignition wires was removed and the "left" ignition coil disconnected from the distributor. Thus, by firing only one spark plug per cylinder, the spark plug being located at the edge of the cylindrical combustion chamber, the tendency to detonate would be increased (11, p. 100) and the effect of the anti-detonant more easily observed.

Third, an adapter was made to accommodate the admission of internal coolant. This adapter was a one-half inch thick spacer machined to exactly the same cross sectional dimensions as the manifold riser flange. A one-eighth inch copper tube, through which the internal coolant was allowed to flow, was inserted through this

spacer and extended to the center of the riser area. The tube was horizontal and cut off squarely at its termination in the riser.

IV. DETERMINATION OF THE TEST CONSTANTS

As outlined in the introduction to this investigation, there are many variables which affect the performance of an internal combustion engine. In order to observe the change of performance with respect to one of these variables, all other variables must be held constant with the greatest of care. Some of these variables are less important than others and affect the performance to only a minor degree. The operating levels at which they were held constant for this investigation were selected from the SAE Gasoline Engine Test Code and certain manufacturers' specifications. These minor variables and the level at which they were held constant throughout the entire test are as follows:

1. Temperature of the cooling water entering the block--125, plus or minus five, degrees Fahrenheit.
2. Temperature of the cooling water leaving the head--175, plus or minus five, degrees Fahrenheit.
3. Lubricant operating temperature--160, plus or minus five, degrees Fahrenheit.
4. Spark plug electrode gap--0.025, plus or minus 0.002 inch.

5. Distributor breaker dwell angle--34, plus or minus one, degrees.

6. Heat range of spark plugs--Champion J-9.

7. Valve tappet clearance--0.015, plus or minus 0.001, inch.

The major variables, those which affect the engine's performance to a marked degree, and the operating levels at which they were held constant are as follows:

1. Throttle opening--Fully opened.

2. Load. When an engine is driving a dynamometer, the power of the engine is absorbed in generating electricity. Since the restraining force exerted by the generator is applied at some distance from the center of the shaft, the force becomes a torque. The engine was operated at fully-opened throttle, and it was desired to make a test at a definite engine speed. Thus, it was necessary to apply a sufficient amount of torque to keep the engine speed at the desired value. The combination of a fully-opened throttle and the maximum torque necessary to keep the engine operating at a desired speed is known as a full-load condition. This full-load condition was maintained throughout any single complete test.

3. Fuel. The primary fuel used for all the variable mixture tests was a regular grade of commercial gasoline. This fuel was rated on a CFR engine by the ASTM

Motor method at a 75.1 octane number by using primary reference fuels. The specific gravity of the regular gasoline was determined to be 59.0 degrees API. Also, a premium grade of commercial gasoline was used when making some preliminary spark advance observations. The premium fuel was rated by the above method at an 84.2 octane number by using primary reference fuels and double checked by using secondary reference fuels. The specific gravity premium fuel was determined to be 59.8 degrees API.

4. Spark timing. It had previously been determined that when operating the engine with a 6.3 compression ratio, best-power spark advance could be obtained at any engine speed using a regular grade of fuel operating under fully loaded conditions with a thirteen to one air-fuel ratio. Having increased the compression ratio, it was expected that the octane requirement would be higher and that this same best-power spark advance could not be obtained without severe detonation taking place when using the same grade of fuel. Thus, it was necessary to make a series of preliminary full-load observations to determine the maximum degree of spark advance to which the engine could be subjected, at each of the four engine speeds used in the test, without encountering detonation. Several spark-advance settings were made at each speed under full-load conditions, with the spark timing being advanced in five or

ten-degree intervals until the point of audible knock was reached. The corrected brake horsepower developed for each spark-advance setting was obtained. The air-fuel ratio was thirteen to one and all other variables were held constant as indicated above. The results of these preliminary tests are shown in Figure 5. The maximum spark-advance setting without encountering audible knock with regular gasoline, at each of the engine speeds used in the tests were determined to be as follows:

1200 rpm--10 degrees BTC

1800 rpm--15 degrees BTC

2400 rpm--25 degrees BTC

3000 rpm--30 degrees BTC

Also, superimposed on these curves are the results of a similar test conducted with premium gasoline. As the spark advance is increased up to the point of detonation for the regular gasoline, it is noted that this lower octane fuel develops more power than does the higher octane premium fuel. It is also noted that the slopes of all the curves of the premium gasoline are less than the slopes of the curves for regular gasoline. This may be explained by two factors. First, the gain in power for the lower-octane fuel may be laid to the shorter combustion time of this fuel due to incipient detonation. (1, p. 55) Second, it is evident that these two fuels have different

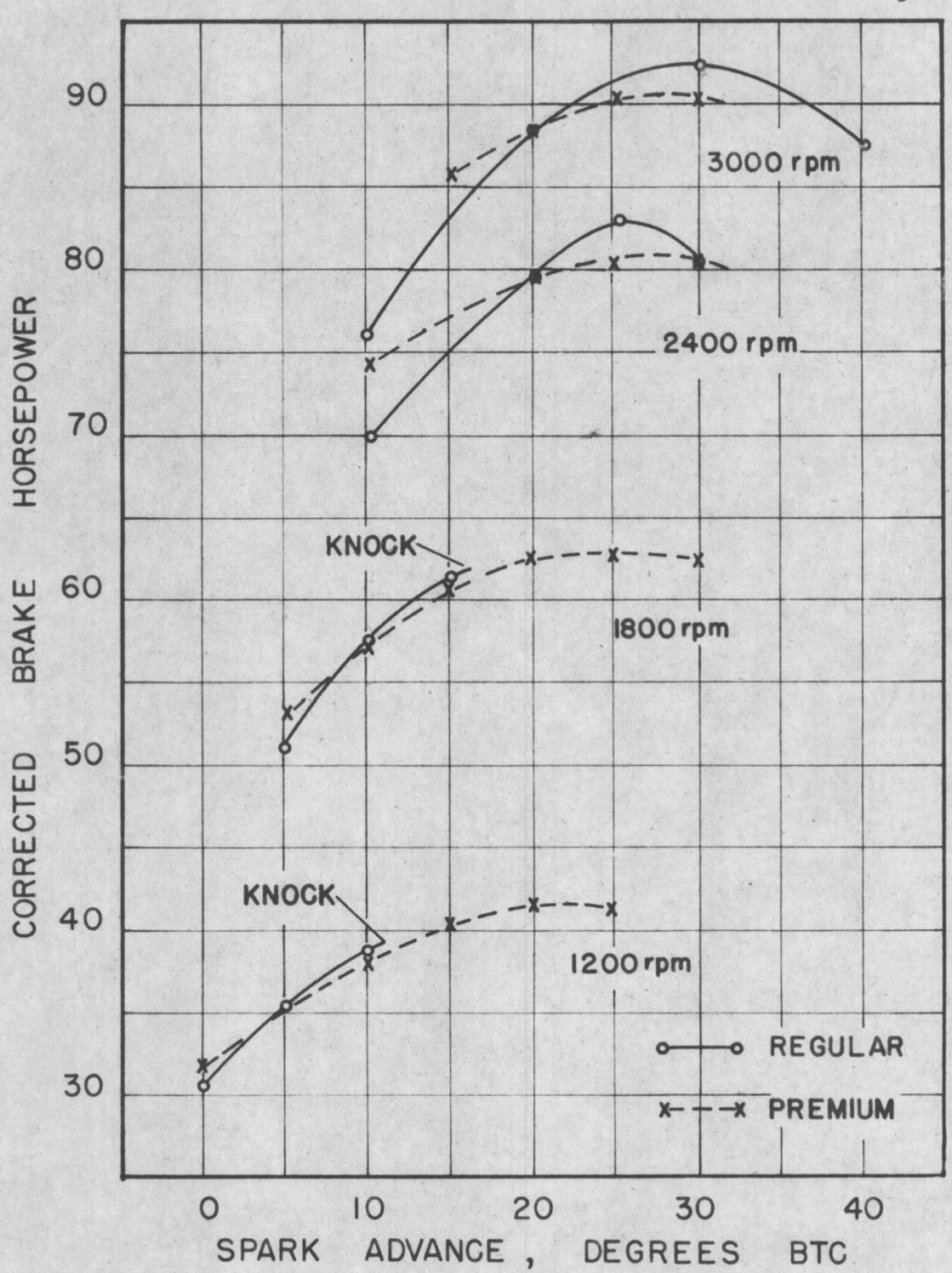

FIGURE 5. MAXIMUM-POWER SPARK ADVANCE WITH TWO GRADES OF FUEL, COMPRESSION RATIO 7.0, FULL LOAD, AND AIR-FUEL RATIO 13.0.

chemical structures. Since these two fuels did not give the same power for the same spark advance, the comparison of these two fuels by the mixture control method of testing was beyond the scope of the work reported.

However, Figure 5, indicates that a fuel with an octane rating of approximately 85 satisfies the requirements of the engine and that a 75 octane fuel does not satisfy the full-load requirement of the engine. This is evidenced by the fact that a knock-free best-power spark advance was obtained with the premium fuel, whereas the regular fuel detonated before a best-power spark advance could be obtained.

ANTI-DETONANTS

The two anti-detonant fluids selected were: (1) water, and (2) a mixture of 50 per cent water and 50 per cent methyl alcohol by volume. In order to make certain that the quantity of anti-detonant admitted was constant per cycle regardless of engine speed it was decided that 20 per cent of the fuel rate at an air-fuel ratio of thirteen to one would be the basis for determining the internal-coolant rate. In order to determine the rate at which the internal coolant was to be admitted at each engine speed selected for the investigation, a preliminary full-load test was run at each of the four engine speeds

to determine the number of pounds of fuel per hour consumed at each of these speeds. The internal coolant rate at each engine speed would be 20 per cent of a fuel rate so determined for each speed. These rates are tabulated following:

Engine Speed rpm	Fuel Rate lb/hr	Coolant Rate lb/hr	Per Cent Coolant of Fuel + Coolant
1200	22.6	4.5	16.5
1800	33.5	6.7	16.6
2400	43.2	8.6	16.6
3000	51.7	10.3	16.6

The quantity of internal coolant admitted per cylinder per cycle is 0.00002 pound.

During the mixture control tests, it was desirable to measure the consumption of the anti-detonant during the same time in which the fuel consumption measurement was being made. Fuel consumption measurements indicated that one-half of a pound of fuel was consumed in a certain number of seconds. Therefore, if in measuring the consumption of the anti-detonant one-tenth of a pound of internal coolant--20 per cent of one-half pound--were used as the standard of measurement, the elapsed time for the test quantities of the internal coolant and the fuel to be consumed would nearly coincide.

The metering of the anti-detonant proved rather difficult because of the small quantity to be metered and the construction of the metering apparatus. It was experimentally determined that three per cent error was the

limit of accuracy of this metering system. Thus, allowable time limits for the consumption of the one-tenth of a pound of internal coolant were computed. They are as follows:

Engine Speed rpm	Coolant Rate lb/hr	Allowable Time sec ± 3%	
1200	4.5	80.0	2.4
1800	6.7	53.6	1.6
2400	8.6	41.7	1.3
3000	10.3	35.0	1.0

V. METHODS OF CALCULATIONS

As stated in the preceding section, it was necessary to compute brake specific fuel consumption, corrected brake horsepower, the air-fuel ratio, and brake thermal efficiency. The methods used in computing these and other items used in determining the performance data are as follows:

Observed Brake Horsepower

$$\text{(obs) Bhp} = \frac{2\pi\ PRN}{33000} = \frac{PRN}{5252} = \frac{PN}{3000}$$

where P = Net beam load on the dynamometer, pounds

R = Lever arm on the dynamometer = 1.75 feet

N = Engine speed, rpm

Fuel Consumption Rate

$$W_f,\ \text{lb/hr} = \frac{1/2\ \text{lb} \times 3600\ \text{sec/hr}}{t\ \text{sec}} = \frac{1800}{t}$$

where t = Time in seconds to consume the 1/2 pound of fuel

Brake Specific Fuel Consumption

$$\text{BSFC},\ \frac{\text{lb}}{\text{Bhp - hr}} = \frac{\text{Fuel Rate, lb/hr}}{\text{(obs) Bhp}}$$

Corrected Brake Horsepower

$$\text{(corr) Bhp} = f \times \text{(obs) Bhp}$$

$$f = \frac{29.92}{B - E}\ (T/520)^{1/2}$$

where B = True barometer corrected for temperature, in Hg

E = Atmospheric water vapor pressure, in Hg

T = Absolute intake air temperature at carburetor inlet degrees Rankine

Air Consumption Rate

$$W_a, \text{ lb/hr} = 414\ CD^2\ (P \times B/T)^{1/2}$$

where C = Orifice coefficient, 0.989 to 0.994 (according to P)

D = Orifice diameter, 2.125 inches

B = True barometer, inches Hg

T = Absolute temperature, degrees Rankine

P = Pressure drop across the orifice, inches of water

A chart, Figure 6, was prepared whereby the air consumption rate could be determined quickly and directly. The chart provides for interpolation for barometric pressure but was computed for the one temperature of 80 degrees Fahrenheit which is an average of the 20 degree temperature variation experienced in the laboratory. The effect of a small temperature change on the air flow is negligible since W_a is a function of the reciprocal of the square root of T.

Air Fuel Ratio

$$A/F = \frac{W_a}{W_f + W_{al}}$$

where W_a = Air-consumption rate, pounds per hour

W_f = Fuel-consumption rate, pounds per hour

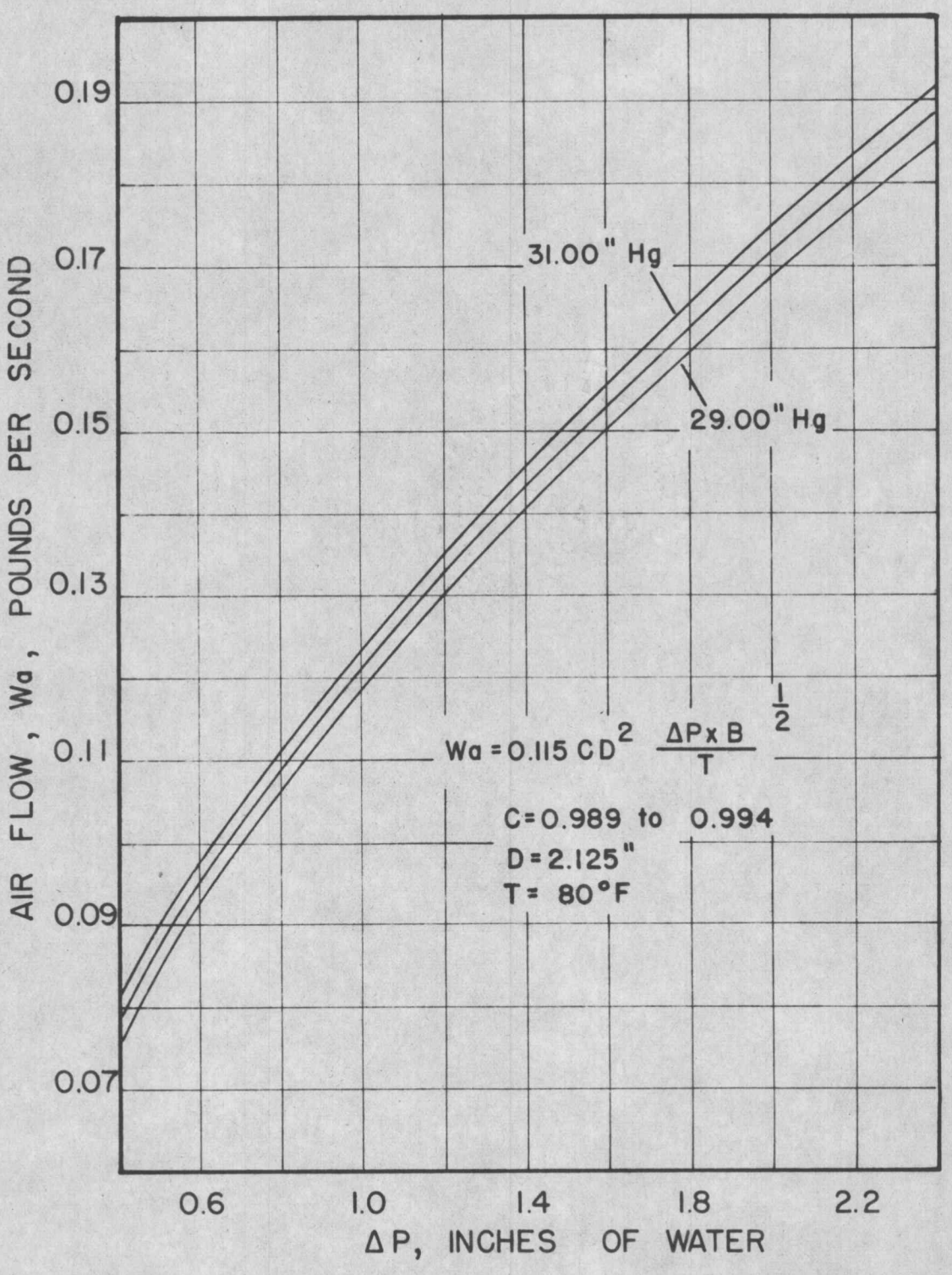

FIGURE 6. CHART USED TO DETERMINE THE AIR-CONSUMPTION RATE.

W_{al} = Alcohol-consumption rate, pounds per hour

Anti-Detonant Rate

$$W_{ad}, \text{ lb/hr} = \frac{1/10 \text{ lb} \times 3600 \text{ sec/hr}}{t \text{ sec}} = \frac{360}{t}$$

where t = Time in seconds to consume 1/10 of a pound of internal coolant

W_{ad} = Anti-detonant consumption rate, pounds per hour

Alcohol Rate

$$W_{al}, \text{ lb/hr} = W_{ad} \times \frac{6.63}{6.63 + 8.33} = 0.443\ W_{ad}$$

where W_{al} = Weight of alcohol consumed, pounds per hour

W_{ad} = Total anti-detonant (alcohol-water) consumption rate, pounds per hour

6.63 = Pounds per gallon of methyl alcohol

8.33 = Pounds per gallon of water

Brake Thermal Efficiency

$$\text{BTE}, \% = \frac{2545}{\text{BSFC} \times \text{HHV}_g}$$

where HHV_g = 18320 + 40 (API - 10)

BSFC = Brake specific fuel consumption pounds per brake horsepower hour

HHV_g = Gasoline higher heating value, Btu/lb

API = Fuel specific gravity on the American Petroleum Institute scale, degrees

When computing Brake Thermal Efficiency with alcohol-water as the anti-detonant,

$$\text{BTE, } \% = \frac{2545 \times (\text{obs}) \text{ Bhp}}{(W_f \times HHV_g) + (W_{al} \times HHV_{al})}$$

where HHV_{al} = Methyl alcohol higher heating value, 12000 Btu/lb

VI. RESULTS

The three mixture control curves and the corresponding thermal efficiency curves for each engine speed selected for this investigation are shown in Figures 7 through 14. The performances when admitting the two internal coolants were compared with the performance without admitting an internal coolant at the points of maximum power and the points of best economy. These comparisons, when taken collectively over the speed range, established certain trends.

MAXIMUM POWER

Consider first the points of maximum power on the mixture control curves, seen in Figures 7, 8, 9, and 10. Note that there is a power loss with the introduction of either internal coolant. At 1200 rpm, either coolant produces a 4.5 per cent loss. At 1800 rpm, either coolant produces a 4.0 per cent loss. At 2400 rpm, water produces a 3.6 per cent loss and alcohol-water a 3.2 per cent loss. At 3000 rpm, water produces a 1.1 per cent loss and alcohol-water a 1.6 per cent loss. Thus, a trend is established at the point of maximum power--the loss of power, occurring when admitting an internal coolant, decreases with increasing engine speed.

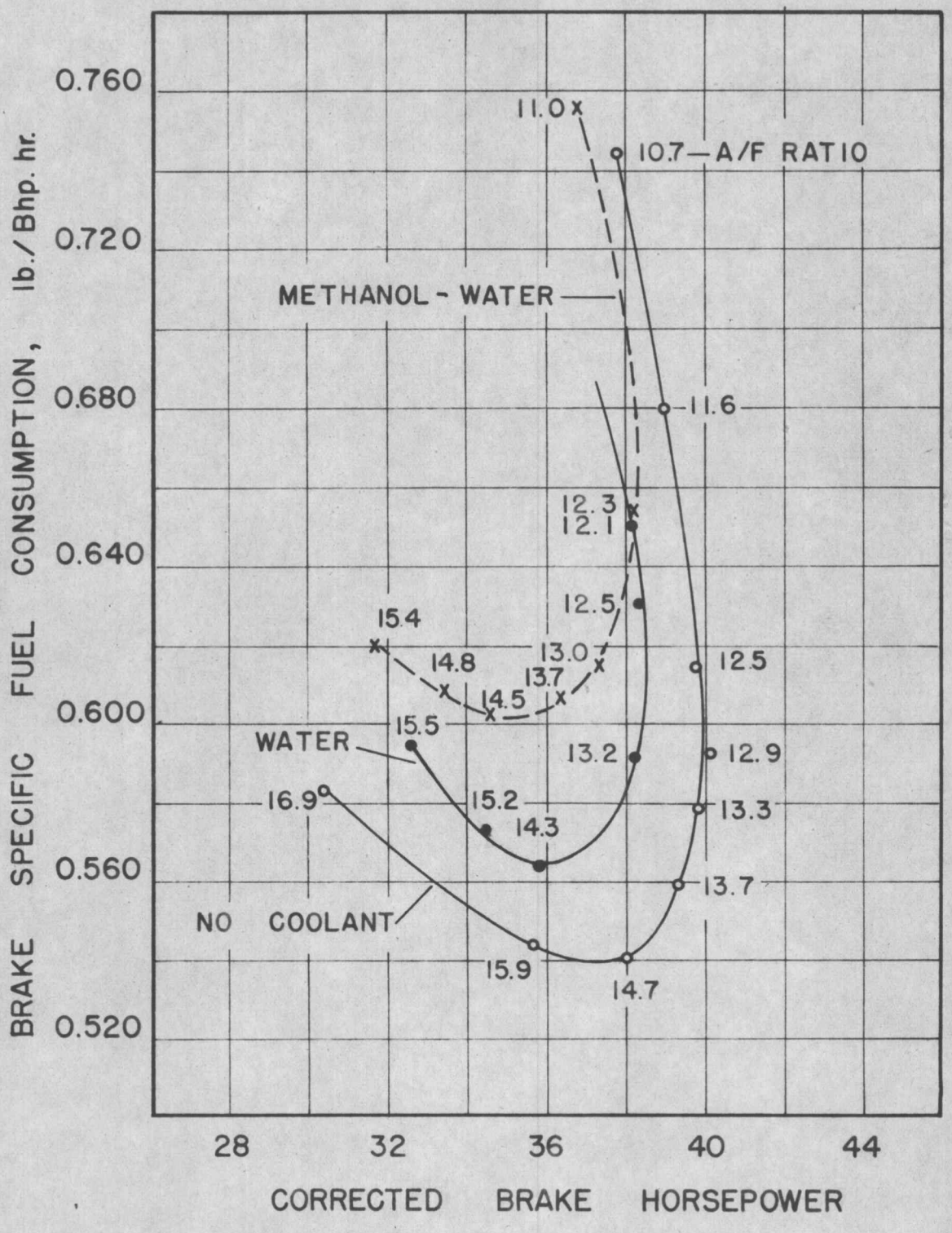

FIGURE 7. MIXTURE CONTROL CURVES AT 1200rpm, FULL LOAD, SPARK ADVANCE 10° BTC, REGULAR FUEL, COOLANT RATE 4.5 lb/hr.

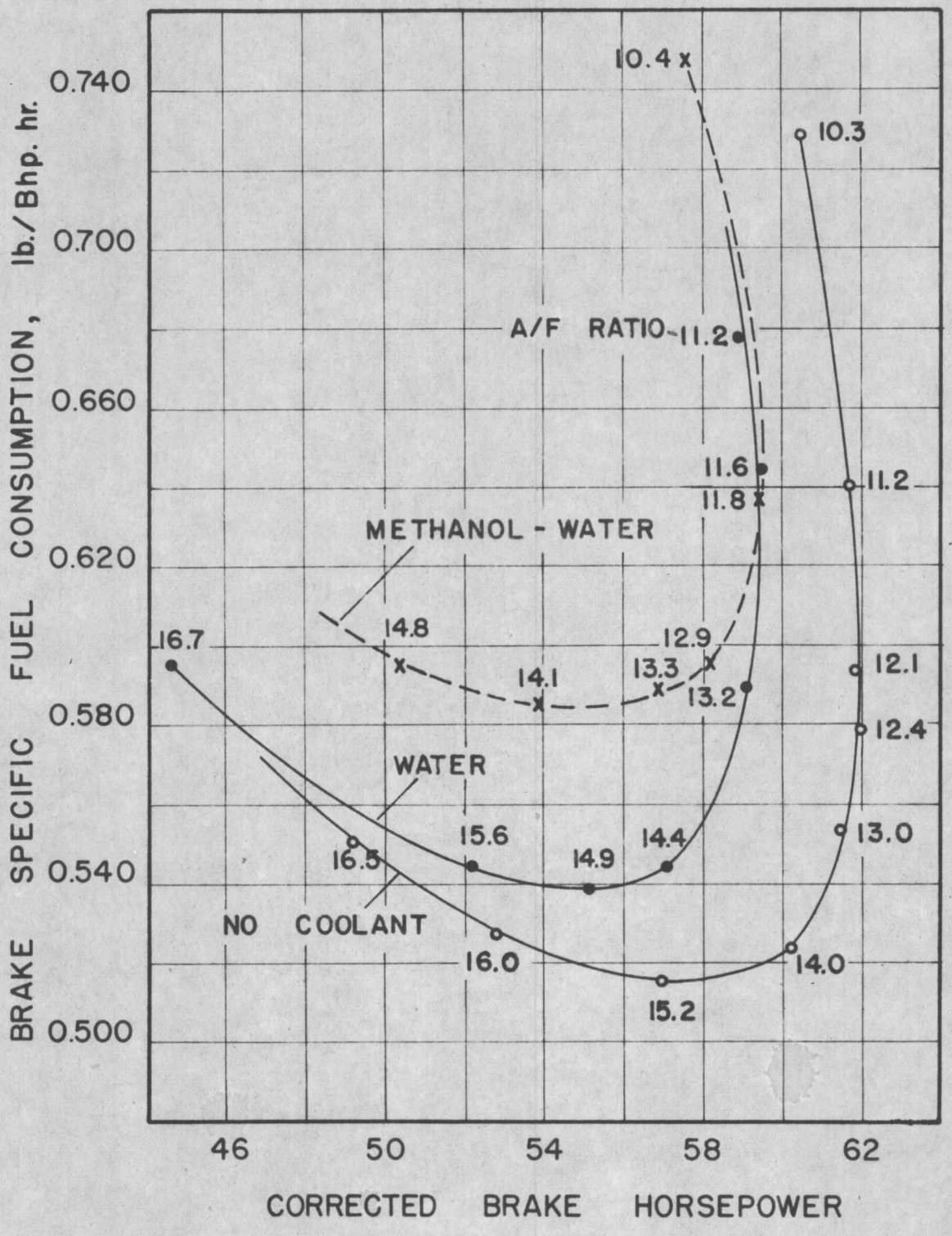

FIGURE 8. MIXTURE CONTROL CURVES AT 1800rpm, FULL LOAD, SPARK ADVANCE 15°BTC, REGULAR FUEL, COOLANT RATE 6.7 lb/hr.

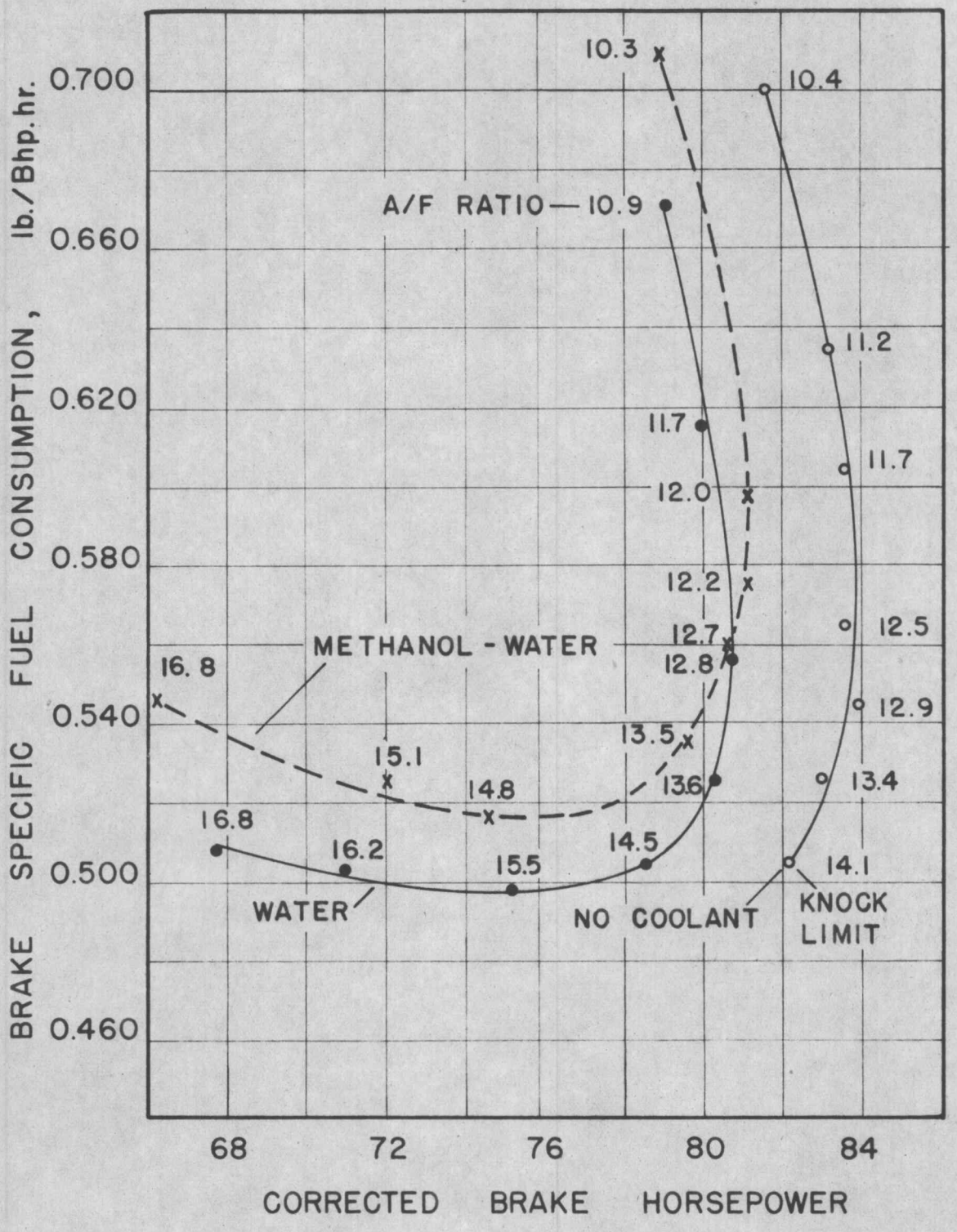

FIGURE 9. MIXTURE CONTROL CURVES AT 2400rpm, FULL LOAD, SPARK ADVANCE 25°BTC, REGULAR FUEL, COOLANT RATE 8.6 lb/hr.

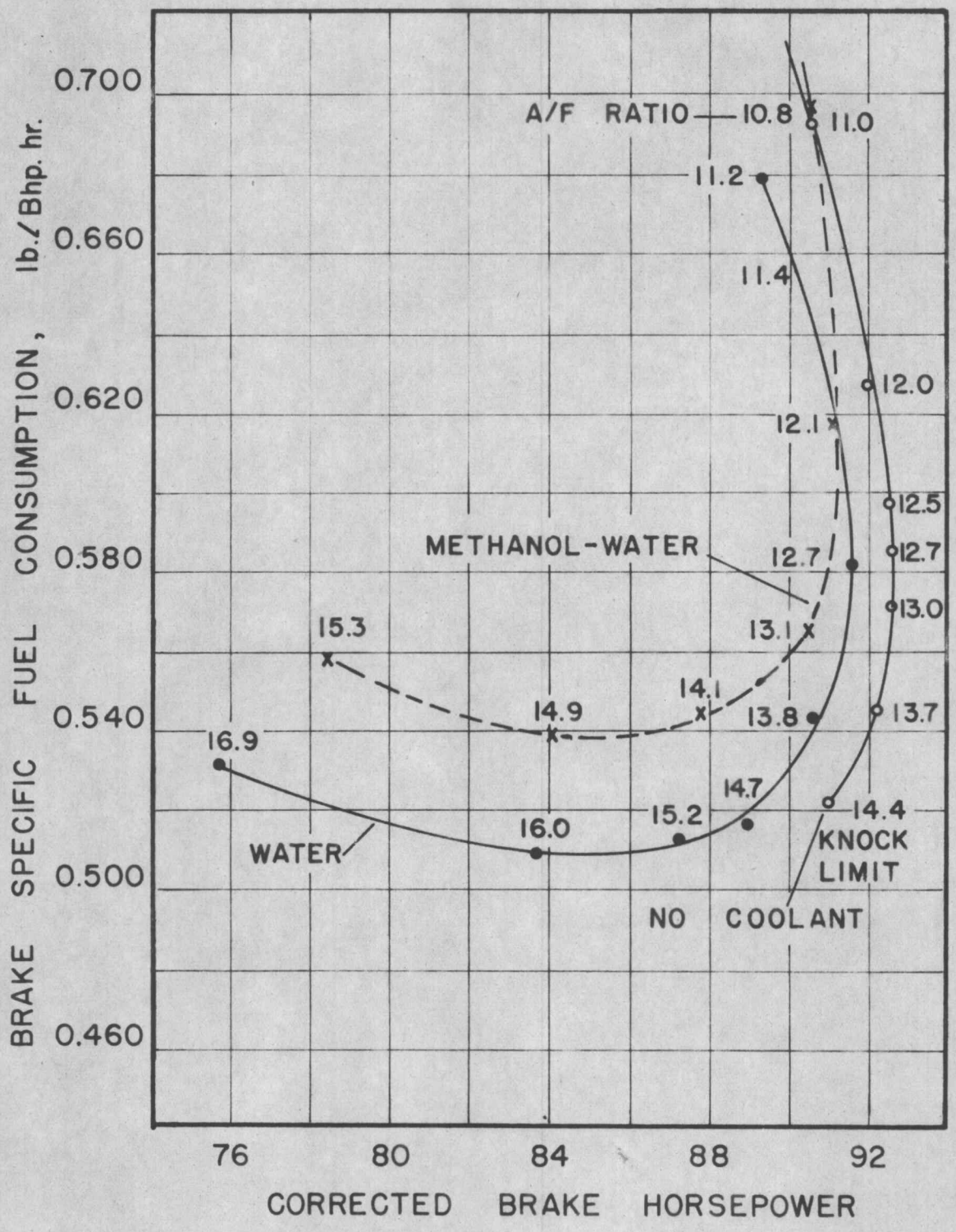

FIGURE 10. MIXTURE CONTROL CURVES AT 3000 rpm, FULL LOAD, SPARK ADVANCE 30°BTC, REGULAR FUEL, COOLANT RATE 10.3 lb/hr.

Also, note that at maximum power there is an increase in specific fuel consumption and a decrease in the air-fuel ratio occurring with the introduction of water and a further increase in specific fuel consumption and a decrease in air-fuel ratio occurring with the introduction of alcohol-water. The curves, at the points of maximum power are rather steep and quantitative analysis of specific fuel consumption and corresponding air-fuel ratio is only approximate. However, a trend is established. At the point of maximum power, the increase in specific fuel consumption and the corresponding increase in air-fuel ratio, occurring when admitting an internal coolant, are a minimum at low and high speeds and are a maximum at an intermediate speed corresponding to the speed of maximum torque.

BEST ECONOMY

Consider now the points of best economy on the mixture control curve seen in Figures 7, 8, 9, and 10. Note that at 2400 and 3000 rpm, when operating without admitting an internal coolant, a knock limited performance point was reached at an air-fuel ratio of 14.1 and 14.4 respectively. At this point detonation was too severe to permit any further leaning of the mixture. Also note that satisfactory knock-free operation was obtained when admitting internal coolants at air-fuel ratios higher than

14.1 and 14.4 respectively. Thus, the best-economy points were obtainable at 2400 and 3000 rpm only when admitting an internal coolant.

At 1200 rpm, there is a 5.4 per cent power loss with the introduction of either coolant. At 1800 rpm, a 3.5 per cent power loss results with the introduction of either coolant. At 2400 and 3000 rpm, while no power loss comparison can be made, it is noted that the points of best economy occur at the same power when admitting either coolant. However, a trend is established in that at the point of best economy at any engine speed, the power loss with the introduction of alcohol-water is equal to the power loss with the introduction of water.

It is noted that the air-fuel ratios occurring at best economy are higher with increasing engine speed, and that the difference between the air-fuel ratio when water is admitted and the air-fuel ratio when alcohol-water is admitted, is 0.2 air-fuel ratio at 1200 rpm and increases to 1.0 air-fuel ratio at 3000 rpm. It is also noted that the difference between the specific fuel consumption for best economy, when water is admitted and the specific fuel consumption when alcohol-water is admitted, is 0.04 lb/bhp hr at both 1200 and 1800 rpm but becomes 0.02 at 2400 and 0.03 at 3000 rpm.

THERMAL EFFICIENCY

The curves of thermal efficiency versus air-fuel ratio for each engine speed are shown in Figures 11, 12, 13, and 14. At 1200 and 1800 rpm, it is noted that the thermal efficiency obtained when no coolant is admitted exceeds the thermal efficiency obtained when either coolant is admitted at all air-fuel ratios. Further, it is noted that the efficiency obtained when admitting water, exceeds the efficiency obtained with alcohol-water at an air-fuel ratio of 13.0 at 1200 rpm and 13.5 at 1800 rpm. At 2400 rpm, the thermal efficiency when alcohol-water is admitted, exceeds the efficiency obtained when no coolant is admitted or when water is admitted, up to the air-fuel ratio of 15.2. Above the air-fuel ratio of 15.2 the efficiency obtained with the admission of water exceeds the efficiency obtained with alcohol-water. Note that the efficiency obtained when no coolant is admitted is below that obtained when admitting water. However, the knock-limited performance point at 14.1 air-fuel ratio when no coolant is admitted prevents any further comparison. At 3000 rpm, the efficiency obtained when admitting alcohol-water exceeds that obtained when admitting either water or when no coolant is admitted, up to an air-fuel ratio of 14.9. Above the air-fuel ratio of 14.9 the efficiency obtained when water is admitted is the highest. Note that the efficiency obtained when water

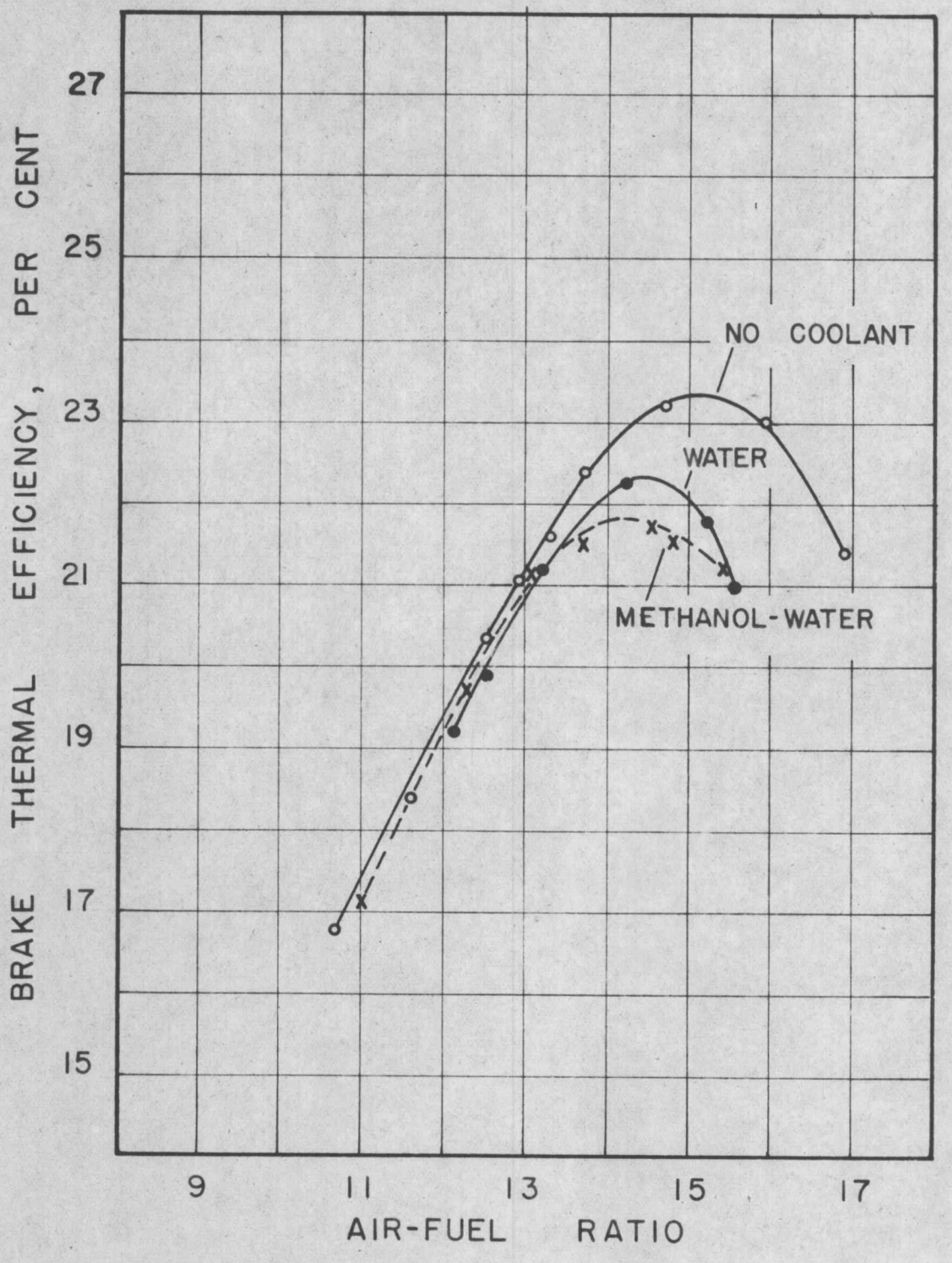

FIGURE 11. THERMAL EFFICIENCY AT 1200 rpm, FULL LOAD, SPARK ADVANCE 10°BTC, REGULAR FUEL, COOLANT RATE 4.5 lb/hr.

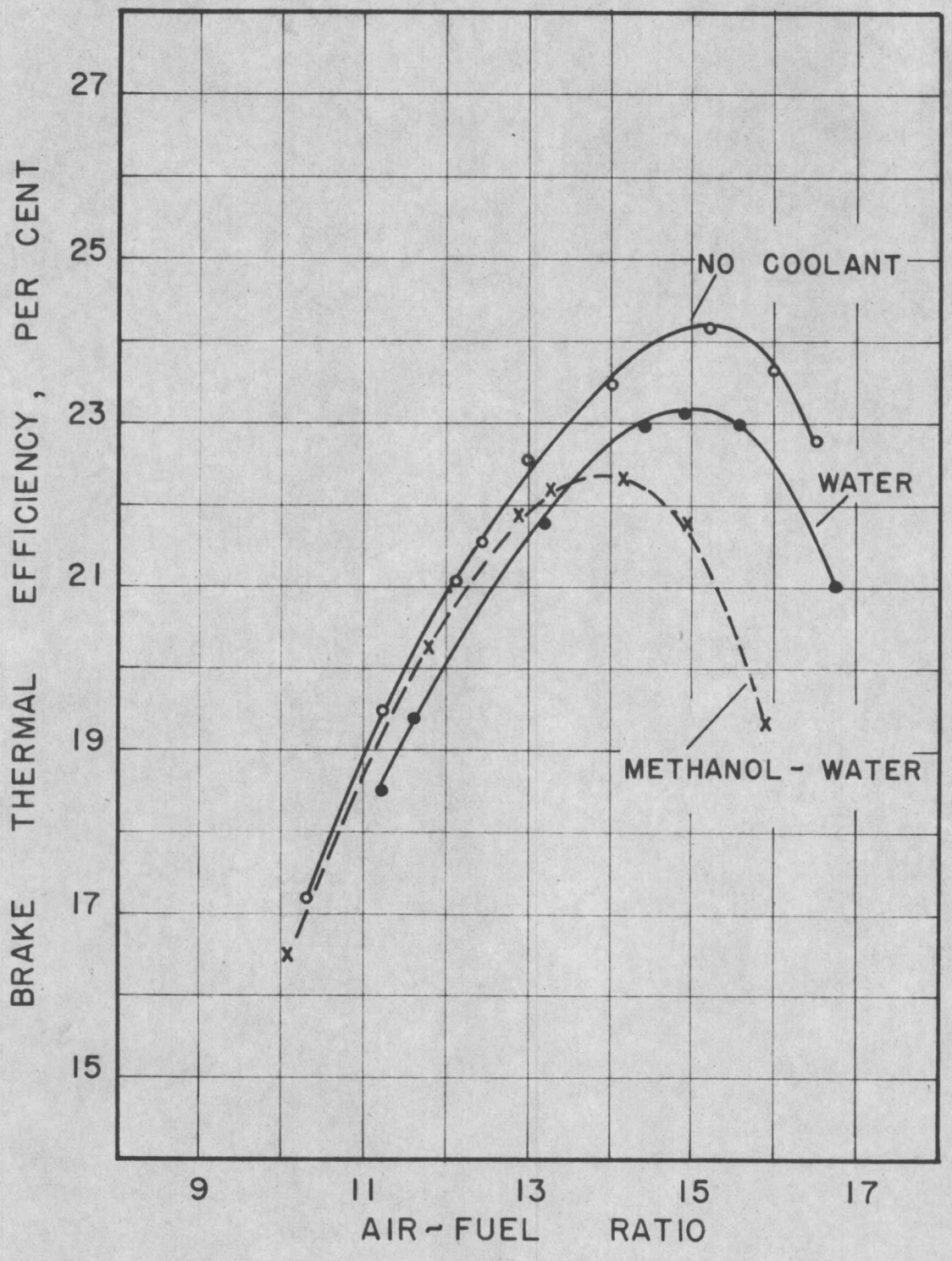

FIGURE 12. THERMAL EFFICIENCY AT 1800rpm, FULL LOAD, SPARK ADVANCE 15° BTC, REGULAR FUEL, COOLANT RATE 6.7 lb/hr.

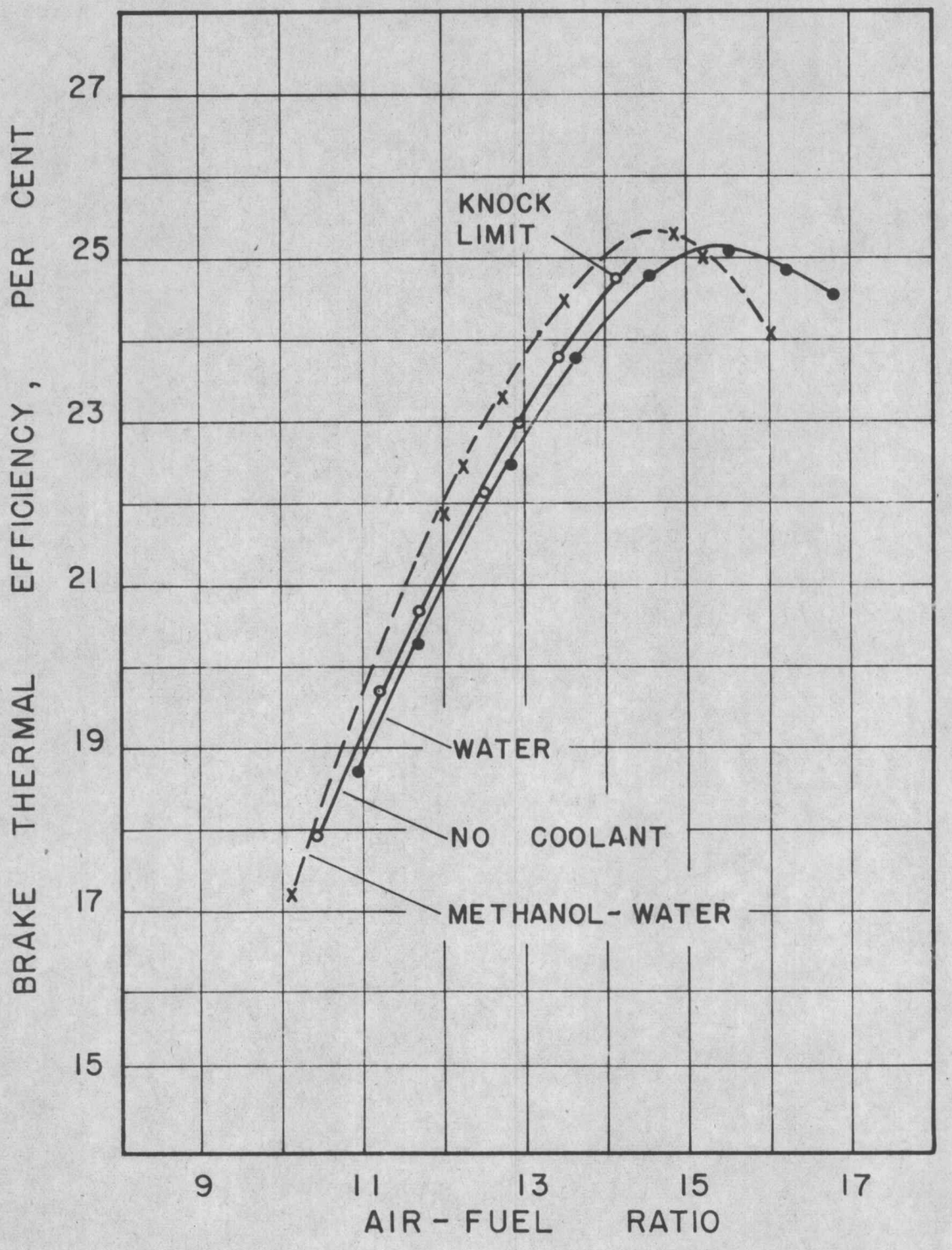

FIGURE 13. THERMAL EFFICIENCY AT 2400 rpm, FULL LOAD, SPARK ADVANCE 25°BTC, REGULAR FUEL, COOLANT RATE 8.6 lb/hr.

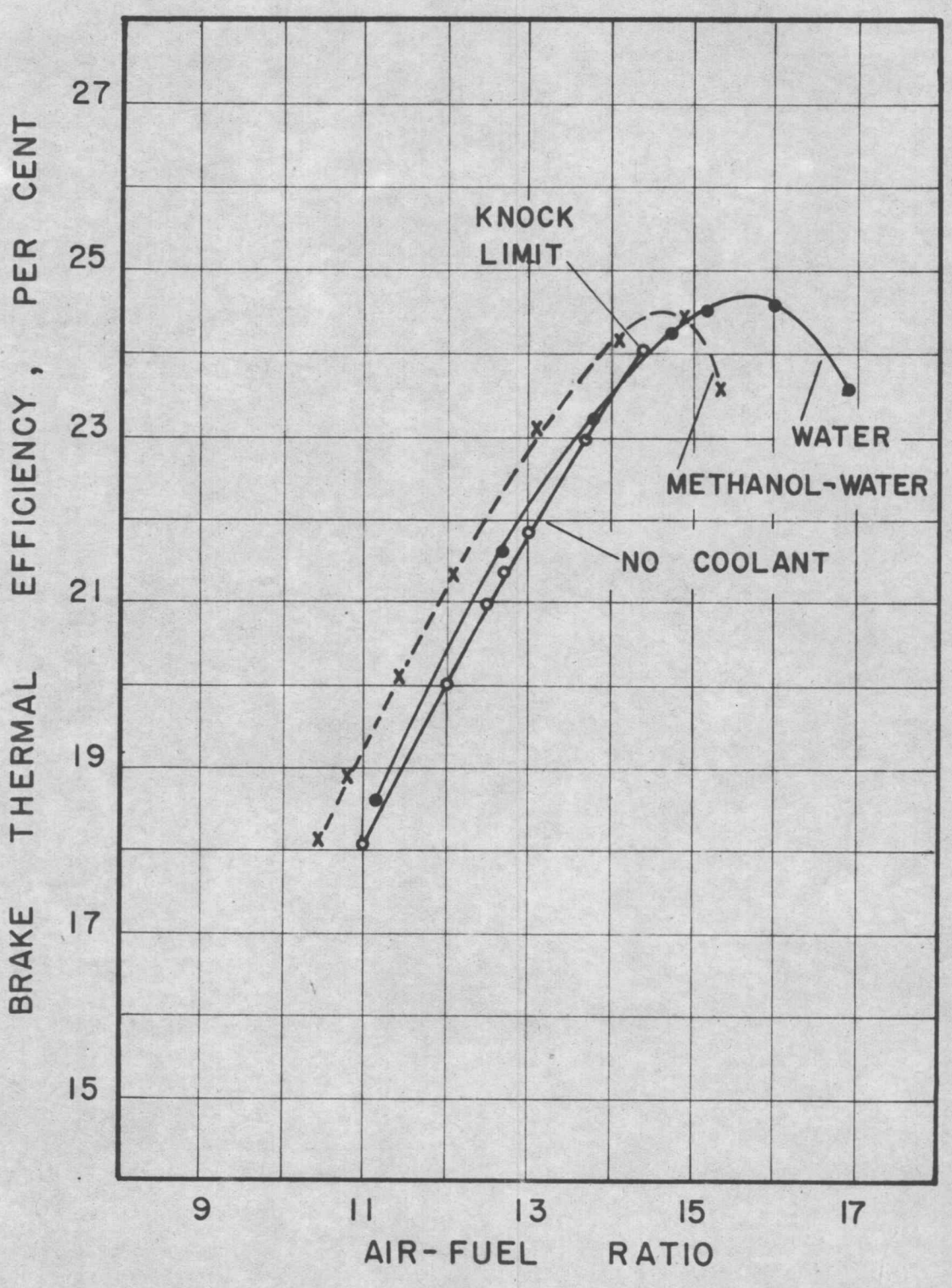

FIGURE 14. THERMAL EFFICIENCY AT 3000 rpm, FULL LOAD, SPARK ADVANCE 30° BTC, REGULAR FUEL, COOLANT RATE 10.3 lb/hr.

is admitted exceeds the efficiency when no coolant is admitted between the range of 11.2 and 14.0 air-fuel ratio. Here again when no coolant is admitted a knock-limited performance point prevents any further comparison.

Therefore, a general trend is established whereby the efficiency obtained when admitting alcohol-water exceeds the efficiency when admitting water in the rich mixture range, and the efficiency obtained when admitting water exceeds that obtained when admitting alcohol-water in the lean mixture range. The air-fuel ratio at which the efficiency with water exceeds that obtained with alcohol-water, increases with increasing engine speed. Also, the efficiency when no coolant is admitted exceeds the efficiency when either coolant is admitted at low speed but is exceeded by the efficiency obtained when either coolant is admitted at high speeds.

SUMMARY OF RESULTS

1. At the point of maximum power, the power loss which occurs when admitting an internal coolant, decreases with increasing engine speed.

2. At the point of maximum power, the increase in specific fuel consumption and the corresponding air-fuel ratios which occur when admitting an internal coolant, are a minimum at low and high speeds and are a maximum at an

intermediate speed corresponding to the speed of maximum torque.

3. At any engine speed, the power loss occurring at the point of best economy with the introduction of alcohol-water is equal to the power loss when admitting water.

4. At the points of maximum economy, the difference between the air-fuel ratio occurring when water is admitted and the air-fuel ratio occurring when alcohol-water is admitted, increases with increasing engine speed.

5. The thermal efficiency obtained when water is admitted exceeds that obtained when alcohol-water is admitted at a certain air-fuel ratio which increases with increasing engine speed.

6. The thermal efficiency obtained when no internal coolant is admitted, exceeds the efficiency when either coolant is admitted at low speed, and is exceeded by the efficiency obtained when either coolant is admitted at high speed.

VII. DISCUSSION

The following discussion is offered as a logical explanation of the results set forth on the preceding pages.

POWER LOSS FROM WATER ADMISSION

The power loss occurring at maximum power when admitting water as an internal coolant may be accounted for by the heat required to vaporize and superheat the water.

The latent heat of vaporization of water was taken as the change of enthalpy from saturated liquid to saturated vapor at the brake mean effective pressure for each engine speed.

At 1200 rpm,

1.8 hp loss x 2545 Btu/bhp hr = 4580 Btu/hr

4.5 lb water/hr x 887 Btu/lb = 4000 Btu/hr

$\frac{4580}{4000}$ x 100 = 115 per cent vaporized

At 1800 rpm,

2.5 hp loss x 2545 Btu/bhp hr = 6360 Btu/hr

6.7 lb water/hr x 884 Btu/lb = 5920 Btu/hr

$\frac{6360}{5920}$ x 100 = 108 per cent vaporized

At 2400 rpm,

3.0 hp loss x 2545 Btu/bhp hr = 7635 Btu/hr

8.6 lb water/hr x 883 Btu/lb = 7590 Btu/hr

$$\frac{7635}{7590} \times 100 = 101 \text{ per cent vaporized}$$

At 3000 rpm,

$$1.0 \text{ hp loss} \times 2545 \text{ Btu/bhp hr} = 2545 \text{ Btu/hr}$$

$$10.3 \text{ lb water/hr} \times 904 \text{ Btu/lb} = 9300 \text{ Btu/hr}$$

$$\frac{2545}{9300} \times 100 = 27 \text{ per cent vaporized}$$

Since a constant quantity of water (2×10^{-5} pound) was admitted per cycle, it is observed that the per cent of water vaporized is not a linear function of the time available. The time available includes the time required to complete the compression and expansion strokes. This may be clearly seen in the following table.

Engine Speed rpm	Time Per Cycle second		Per Cent Vaporized per cent	
	Total	Dif	Total	Dif
1200	0.050		115	
		0.017		7
1800	0.033		108	
		0.008		7
2400	0.025		101	
		0.005		74
3000	0.020		27	

From the table it is evident that there is a problem involving the time available to vaporize the water and the prevailing rate of vaporization. The rate of vaporization of a liquid is a function of the rate of heat transfer and depends on the existing surface to volume ratio. Thus, it may be said that the degree to which vaporization takes

place depends on two factors: (1) engine speed--representing the time available, and (2) the degree of atomization--representing the relative amount of exposed surface available for heat transfer.

This is somewhat verified by a statement of Edward F. Obert:

> When a liquid and gas are compressed, vaporization of the liquid will cool the gas and less work will be required for the compression. Water, with its high latent heat, is particularly suited for this purpose, but it must be finely atomized and sufficient time allowed for its vaporization. .....Evidently, in normal operation of an unsupercharged engine the water does not vaporize until after the combustion process is well under way. (1, p. 57)

No quantitative data regarding the degree of atomization were taken during the course of this investigation and no provisions were made to control atomization. Due to the construction of the flow mechanism (refer to page 25 of this text), the atomization was dependent upon turbulence in the manifold and combustion chamber.

POWER LOSS FROM ALCOHOL-WATER ADMISSION

The above discussion concerning heat loss due to vaporization of the water does not apply to the methanol-water mixture. The latent heat of vaporization of the methanol-water mixture is only 70 per cent of the latent heat of vaporization of water, indicating that more than

the evaporation heat loss must be accounted for. As will be outlined in the following paragraphs, it is believed that not all of this additional heat loss represents the work required to superheat water in the methanol-water mixture.

As shown by Colwell (refer to page 8 of this text) and by the thermal efficiency plots (Figures 11, 12, 13, and 14), the chemical energy of the alcohol adds to the energy of combustion. Since alcohol is a slower burning fuel than gasoline, it may be concluded that the rate of combustion of the total air-fuel mixture was retarded, and the spark timing would have to be advanced in order to make use of this additional energy. As shown by Seshadri (10, p. 41), a greater spark advance was necessary to obtain maximum power when admitting a constant quantity of a methanol-water solution as compared with the spark advance necessary to obtain maximum power when admitting the same quantity of water. However, the spark advance was maintained at a constant setting and a decrease in power was observed when alcohol was admitted as an internal coolant. Thus, the additional power loss may be considered as energy rejected to the exhaust gases as a result of the slower rate of burning. This may account for the additional heat loss mentioned above.

The overall conclusions considering this discussion

on the heat loss processes of the two coolants follow:

1. Atomization of the coolant is an important factor in the effectiveness of an internal coolant.
2. When admitting water, the power loss may be accounted for by the latent heat of vaporization plus some superheat.
3. When admitting alcohol-water, the power loss may include energy rejected to the exhaust gases as a result of the slower rate of burning.

THERMAL EFFICIENCY

As shown by the thermal efficiency curves and as pointed out in the results reported herein, alcohol-water produces a high thermal efficiency in the rich mixture ranges, and water produces a high thermal efficiency in the lean mixture range.

High thermal efficiency obtained with alcohol-water admission at rich mixtures may be attributed to the finite quantity of chemical energy added to the combustion process. It is believed that still higher thermal efficiency may be obtained by making better use of the potential energy of the alcohol through adjustment of spark timing to compensate for the slower combustion rate.

The drop in thermal efficiency obtained with

alcohol-water admission at lean mixtures may be attributed to the additional reduction in the rate of combustion from the leaner mixture. The summation of this reduction, the reduction attributed to lower temperatures resulting from the vaporization process, and the reduction caused by the slower burning alcohol, causes the total rate of combustion, when alcohol-water is admitted, to be reduced to a point where the process is not wholly completed by the time the exhaust valve opens; thereby, losing a greater amount of energy to the exhaust gases.

It seems to follow that high thermal efficiencies are obtained with water admission at lean mixtures because of the difference in the rate of combustion.

OPTIMUM PERFORMANCE POINTS

The decrease in air-fuel ratio and the increase in brake specific fuel consumption for corresponding points of maximum power and best economy with the introduction of either internal coolant are directly caused by the loss of power at these points.

Brake specific fuel consumption is directly proportional to the pounds of fuel supplied and inversely proportional to the brake horsepower produced with this quantity of fuel. Therefore, since less power was developed for any one air-fuel ratio, the specific fuel

consumption was correspondingly higher. This had the effect of moving the mixture control curve up and to the left with respect to the co-ordinate axes. Thus, the optimum performance points when admitting an internal coolant were above and to the left of the datum curve as a result of the power loss described in the preceding section.

CONCLUSIONS

From the results of the performance tests and the logical justification for these results, it is evident that the performance of an engine has certain definite characteristics when an internal coolant is admitted with the charge as compared to the performance without an internal coolant, and that these characteristics may be attributed to a power loss resulting from the absorption of energy by the vaporization of water and from a reduction in the rate of burning of the mixture by the addition of alcohol.

Thus, it follows that it is not possible to obtain better performance from an unsupercharged engine with anti-detonant admission <u>unless</u> the loss of power resulting from detonation when no anti-detonant is admitted exceeds the inherent loss of power resulting from the anti-detonant admission.

VIII. DATA

DATA

TEST SERIES NUMBER I

Engine speed-1200 rpm, Spark advance-10 deg btc, Full load, Fuel-75 octane, Internal coolant-none.

Test No.	Rpm	Beam Load	Fuel Rate	Air Rate	A/F Ratio	Obs Bhp	Bsfc	Cor Bhp	Bte
1	1190	91.7	27.1	291	10.7	36.4	0.745	37.7	16.8
2	1200	94.0	25.6	295	11.6	37.6	0.680	39.0	18.4
3	1200	96.0	23.6	295	12.5	38.4	0.615	39.8	20.4
4	1208	96.1	23.0	295	12.9	38.7	0.593	40.1	21.1
5	1200	96.1	22.3	295	13.3	38.5	0.579	39.8	21.6
6	1200	94.3	21.0	289	13.7	37.7	0.559	39.3	22.4
7	1205	89.4	19.5	286	14.7	36.0	0.540	38.0	23.2
8	1210	85.5	18.8	297	15.9	34.5	0.544	35.8	23.0
9.	1190	73.6	17.1	289	16.9	29.2	0.586	30.4	21.4

TEST SERIES NUMBER II

Engine speed-1800 rpm, Spark advance-15 deg btc, Full load, Fuel-75 octane, Internal coolant-none.

Test No.	Rpm	Beam Load	Fuel Rate	Air Rate	A/F Ratio	Obs Bhp	Bsfc	Cor Bhp	Bte
1	1795	95.8	41.2	424	10.3	57.3	0.729	60.4	17.1
2	1800	97.8	37.6	423	11.2	58.6	0.641	61.7	19.5
3	1795	98.2	34.9	423	12.1	58.7	0.594	61.8	21.1
4	1800	98.1	34.0	423	12.4	58.9	0.579	62.0	21.6
5	1800	97.3	32.4	423	13.0	58.4	0.554	61.5	22.6
6	1810	94.6	30.5	423	14.0	57.1	0.534	60.2	23.5
7	1800	90.5	28.0	423	15.2	54.3	0.516	56.9	24.2
8	1810	83.5	26.6	426	16.0	50.4	0.528	52.8	23.7
9	1800	78.4	25.9	428	16.5	47.0	0.550	49.0	22.8

TEST SERIES NUMBER III

Engine speed-2400 rpm, Spark advance-25 deg btc, Full load,

Fuel-75 octane, Internal coolant-none.

Test No.	Rpm	Beam Load	Fuel Rate	Air Rate	A/F Ratio	Obs Bhp	Bsfc	Cor Bhp	Bte
1	2395	97.4	54.5	566	10.4	77.6	0.701	81.5	17.9
2	2398	99.0	50.3	564	11.2	79.1	0.635	83.1	19.7
3	2400	99.5	48.1	564	11.7	79.6	0.605	83.6	20.7
4	2398	99.6	45.0	561	12.5	79.6	0.565	83.6	22.1
5	2400	100.0	43.6	561	12.9	80.0	0.545	84.0	23.0
6	2400	98.8	41.6	558	13.4	79.0	0.527	83.0	23.8
7	2400	97.8	39.7	558	14.1	78.3	0.506	82.2	24.8
8	Detonation too severe for further operation.								

TEST SERIES NUMBER IV

Engine speed-3000 rpm, Spark advance-30 deg btc, Full load,

Fuel-75 octane, Internal coolant-none.

Test No.	Rpm	Beam Load	Fuel Rate	Air Rate	A/F Ratio	Obs Bhp	Bsfc	Cor Bhp	Bte
1	3007	86.9	60.1	664	11.0	87.0	0.692	90.6	18.1
2	2995	88.3	55.4	661	12.0	88.2	0.627	92.0	20.0
3	3010	88.5	53.0	661	12.5	88.8	0.597	92.5	21.0
4	3005	88.8	52.0	660	12.7	88.9	0.585	92.6	21.4
5	3005	88.8	50.7	660	13.0	88.9	0.571	92.6	21.9
6	3010	88.2	48.1	660	13.7	88.5	0.544	92.2	23.0
7	3004	87.3	45.6	656	14.4	87.4	0.522	91.0	24.0
8	Detonation too Severe for further operation.								

TEST SERIES NUMBER V

Engine speed-1200 rpm, Spark advance-10 deg btc, Full load,

Fuel-75 octane, Internal coolant-water.

Test No.	Rpm	Beam Load	Fuel Rate	Air Rate	A/F Ratio	Obs Bhp	Bsfc	Cor Bhp	A.D. Rate	Bte
1	1210	89.7	23.6	286	12.1	36.2	0.652	38.2	4.58	19.2
2	1196	92.1	23.2	289	12.5	36.7	0.631	38.3	4.57	19.9
3	1200	91.7	21.6	285	13.2	36.6	0.592	38.2	4.57	21.2
4	1210	89.5	20.1	286	14.2	35.7	0.563	37.7	4.50	22.3
5	1200	82.4	19.4	285	15.2	33.0	0.575	34.3	4.58	21.8
6	1205	79.2	19.0	285	15.5	31.9	0.596	33.1	4.58	21.0

TEST SERIES NUMBER VI

Engine speed-1800 rpm, Spark advance-15 deg btc, Full load,

Fuel-75 octane, Internal coolant-water.

Test No.	Rpm	Beam Load	Fuel Rate	Air Rate	A/F Ratio	Obs Bhp	Bsfc	Cor Bhp	A.D. Rate	Bte
1	1794	93.9	38.1	426	11.2	56.2	0.678	58.9	6.76	18.5
2	1795	94.8	36.7	424	11.6	56.7	0.645	59.4	6.75	19.4
3	1807	93.5	32.4	428	13.2	56.3	0.575	59.0	6.66	21.8
4	1799	90.9	29.7	426	14.4	54.5	0.545	57.1	6.59	23.0
5	1790	88.2	28.4	422	14.9	52.6	0.539	55.1	6.68	23.2
6	1790	83.6	27.2	422	15.6	49.9	0.544	52.2	6.92	23.0
7	1795	71.7	25.4	424	16.7	42.6	0.595	44.6	6.88	21.0

TEST SERIES NUMBER VII

Engine speed-2400 rpm, Spark advance-25 deg btc, Full load,

Fuel-75 octane, Internal coolant-water.

Test No.	Rpm	Beam Load	Fuel Rate	Air Rate	A/F Ratio	Obs Bhp	Bsfc	Cor Bhp	A.D. Rate	Bte
1	2400	95.1	51.0	558	10.9	76.0	0.671	79.0	8.54	18.7
2	2396	96.3	47.4	555	11.7	76.9	0.616	79.9	8.60	20.3
3	2400	97.0	43.2	554	12.8	77.6	0.556	80.7	8.54	22.5
4	2406	96.2	40.8	555	13.6	77.2	0.528	80.3	8.64	23.8
5	2396	94.6	38.2	555	14.5	75.6	0.505	78.6	8.74	24.8
6	2405	88.5	35.4	548	15.5	71.0	0.499	75.2	8.54	25.1
7	2400	83.7	33.8	546	16.2	67.0	0.504	70.9	8.41	24.9
8	2400	80.0	32.6	549	16.8	64.0	0.509	67.7	8.61	24.6

TEST SERIES NUMBER VIII

Engine speed-3000 rpm, Spark advance-30 deg btc, Full load,

Fuel-75 octane, Internal coolant-water.

Test No.	Rpm	Beam Load	Fuel Rate	Air Rate	A/F Ratio	Obs Bhp	Bsfc	Cor Bhp	A.D. Rate	Bte
1	3000	84.3	57.1	638	11.2	84.3	0.678	89.3	10.41	18.5
2	3000	86.5	50.3	637	12.7	86.5	0.581	91.6	10.00	21.6
3	3000	85.5	46.4	637	13.8	85.5	0.543	90.6	10.35	23.1
4	3000	84.0	43.4	637	14.7	84.0	0.516	89.0	10.50	24.3
5	3010	82.0	42.2	641	15.2	82.4	0.512	87.3	10.22	24.5
6	3000	79.0	40.1	640	16.0	79.0	0.509	83.6	10.22	24.6
7	3000	71.5	38.0	641	16.9	71.5	0.531	75.7	10.42	23.6

TEST SERIES NUMBER IX

Engine speed-1200 rpm, Spark advance-10 deg btc, Full load, Fuel-75 octane, Internal coolant - methyl alcohol and water (50-50 by volume).

Test No.	Obs Bhp	Air Rate	Gaso Rate	A.D. Rate	Alco Rate	Total Fuel	Bsfc	Cor Bhp	A/F Ratio	Bte
1	35.8	299	25.0	4.63	2.05	27.1	0.756	36.8	11.0	17.1
2	37.1	299	22.3	4.56	2.02	24.3	0.655	38.2	12.3	19.8
3	36.5	291	20.4	4.53	2.01	22.4	0.614	37.5	13.0	21.2
4	35.4	295	19.5	4.50	2.00	21.5	0.607	36.4	13.7	21.5
5	33.7	295	18.3	4.55	2.02	20.3	0.603	34.6	14.5	21.7
6	32.6	295	17.8	4.63	2.05	19.9	0.610	33.5	14.8	21.5
7	30.8	295	17.1	4.51	2.00	19.1	0.620	31.7	15.4	21.2

TEST SERIES NUMBER X

Engine speed-1800 rpm, Spark advance-15 deg btc, Full load, Fuel-75 octane, Internal coolant - methyl alcohol and water (50-50 by volume).

Test No.	Obs Bhp	Air Rate	Gaso Rate	A.D. Rate	Alco Rate	Total Fuel	Bsfc	Cor Bhp	A/F Ratio	Bte
1	53.6	427	39.0	6.92	3.06	42.1	0.785	56.1	10.1	16.5
2	56.8	426	33.3	6.59	2.92	36.2	0.637	59.4	11.8	20.3
3	55.5	425	30.0	6.76	3.00	33.0	0.595	58.1	12.9	21.9
4	54.4	424	28.9	6.92	3.06	32.0	0.590	56.9	13.3	22.2
5	51.5	425	27.2	6.82	3.02	30.2	0.586	53.9	14.1	22.3
6	48.1	424	25.8	6.58	2.92	28.7	0.595	50.4	14.8	21.8
7	39.6	425	23.6	6.91	3.06	26.7	0.674	41.5	15.9	19.4

TEST SERIES NUMBER XI

Engine speed-2400 rpm, Spark advance-25 deg btc, Full load, Fuel-75 octane, Internal coolant - methyl alcohol and water (50-50 by volume).

Test No.	Obs Bhp	Air Rate	Gaso Rate	A.D. Rate	Alco Rate	Total Fuel	Bsfc	Cor Bhp	A/F Ratio	Bte
1	73.8	554	51.9	8.57	3.80	55.7	0.755	77.2	10.1	17.2
2	77.5	554	42.6	8.44	3.74	46.3	0.598	81.1	12.0	21.9
3	77.5	546	40.8	8.85	3.92	44.7	0.576	81.1	12.2	22.5
4	77.2	548	39.2	8.78	3.89	43.1	0.559	80.8	12.7	23.3
5	76.1	548	36.8	8.57	3.80	40.6	0.534	79.7	13.5	24.4
6	72.6	558	33.8	8.56	3.80	37.6	0.517	74.6	14.8	25.3
7	68.8	546	32.2	8.68	3.85	36.1	0.525	72.0	15.1	25.0
8	63.3	554	30.8	8.31	3.68	34.5	0.545	66.2	16.0	24.0

TEST SERIES NUMBER XII

Engine speed-3000 rpm, Spark advance-30 deg btc, Full load, Fuel-75 octane, Internal coolant - methyl alcohol and water (50-50 by volume).

Test No.	Obs Bhp	Air Rate	Gaso Rate	A.D. Rate	Alco Rate	Total Fuel	Bsfc	Cor Bhp	A/F Ratio	Bte
1	86.5	643	57.0	10.1	4.84	61.8	0.715	90.2	10.4	18.1
2	86.9	645	54.9	11.0	4.88	59.8	0.689	90.6	10.8	18.9
3	87.3	645	51.5	11.0	4.88	56.4	0.645	91.0	11.4	20.1
4	87.4	645	48.6	10.8	4.79	53.4	0.611	91.1	12.1	21.3
5	86.7	641	44.2	10.8	4.79	49.0	0.565	90.5	13.1	23.1
6	84.2	645	40.9	11.1	4.91	45.8	0.545	87.8	14.1	24.1
7	80.5	645	38.5	11.0	4.88	43.4	0.539	84.0	14.9	24.4
8	75.1	643	37.1	11.0	4.88	42.0	0.559	78.4	15.3	23.6

BIBLIOGRAPHY

1. Campbell, J. M., Caris, D. F., and Withrow, L. L. Increasing the thermal efficiencies of internal combustion engines. Paper presented at the SAE national west coast meeting, San Francisco, Calif. Aug. 16-20, 1948.

2. Colwell, A. T., Cummings, R. E., and Anderson, D. E. Alcohol-water injection. SAE transactions. 53:358-371, 1945.

3. Holaday, W. M. The efficient production and utilization of motor gasoline. Paper presented at the SAE annual meeting, Detroit, Mich. Jan. 12-16, 1948.

4. Jennings, Burgess H. and Obert, Edward F. Internal combustion engines; analysis and practice. Scranton, Pa., International textbook co., 1944, 471 p.

5. Lichty, Lester C. Internal-combustion engines. New York, McGraw-Hill book co., 1939, 603 p.

6. Maleev, V. L. Internal-combustion engines; theory and design. New York, McGraw-Hill book co., 1945, 636 p.

7. Obert, Edward F. Detonation and internal coolants. SAE quarterly transactions. 2:52-59, 1948.

8. Potter, R. I. Use of anti-detonant injection in a high compression ratio engine. Paper presented at the SAE summer meeting, French Lick, Ind. June 6-11, 1948.

9. Rowe, M. R. and Ladd, G. T. Water injection for aircraft engines. SAE transactions. 54:26-37, 1946.

10. Seshadri, E. V. Performance of a spark-ignition engine with admission of some internal knock suppressing coolants. A thesis submitted to Oregon State College, June, 1949.

11. Taylor, C. Fayette and Taylor, Edward S. The internal combustion engine. Scranton, Pa., International textbook co., 1938. 322 p.