The effect of water-injection on the performance of internal combustion engines

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Publication Date:
1957

DOI:
https://doi.org/10.26190/unsworks/5265

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Mechanical Eng.

THE EFFECT OF WATER-INJECTION ON THE PERFORMANCE OF INTERNAL COMBUSTION ENGINES

by

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A Higher Degree Thesis submitted to the N.S.W. University of Technology.

June 1957.

SUMMARY

To determine the influence of an anti-detonant injection on the performance of an internal combustion engine two methods were adopted:

1. A mathematical approach by which the theoretical drop of combustion temperature was calculated for an Otto cycle using octane as fuel and water as internal coolant, the water being 25% by weight of the fuel. The method derived holds good for any fuel and coolant.
2. An experimental method in which a series of engine tests were carried out using the Ricardo E-6/S variable compression engine with a manually controlled valve to regulate the coolant flow.

   The main object of this part of the thesis was to determine any improvement in power and economy, when water or water-alcohol at various weight-ratios was used for anti-detonant injections. Observation data, result sheets and performance curves are presented. The graphs are used to analyse test results but may also be used to calculate other values such as percentage gain of power if desired. Economy figures are based on 3s./8^1/2d. per gallon for standard grade petrol and 7s./1d. per gallon for methylated spirit.

From the attached observation and result sheets, variables such as exhaust gas temperature, inlet air temperature and torque values may be plotted.

ACKNOWLEDGMENT.

The author is indebted to Mr. S. Smith for his assistance in installing and calibrating the test equipment and for maintaining the Ricardo E-6/S variable compression engine in good working order during the test period.

Thanks are also due to Mr. G. Reynolds for his help each day in preparing the engine for testing and for his assistance in taking readings during the test runs.

The author is greatly indebted to Mr. J. Grant for drawing the numerous test graphs and to Mr. E.C.Martin, School of Applied Chemistry, who provided data on specific gravity and on heat values of the fuels used.

Finally, thanks are due to Mr. H. Baker from S. Smith and Sons (Aust.) Pty. Ltd., who provided the author with special spark plugs, modified to carry thermocouple wires instead of the centre electrode.

CONTENTS

CHAPTER 1

HISTORY OF ANTI-DETONANT INJECTION.

Water-injection as an anti-detonant for internal combustion engines is a fairly old idea and tests had been carried out before 1900 by Benki in Hungary.

In 1913, Professor B. Hopkinson(1) of the University of Cambridge delivered a paper at a meeting of the Institution of Mechanical Engineers, dealing with his experiences gained when using water as an internal coolant for horizontal gas engines.

In this classical report he pointed out that difficulties with engine liners and pistons, resulting from high temperature differences, could be eliminated by the use of water injection. Furthermore, pre-ignition of the fresh charge could be prevented, because the temperature of the combustion chamber walls and of carbon and tar deposits were kept much below that which is necessary to ignite the incoming charge.

Hopkinson also found that the maximum cycle pressure was reduced by 100 psi, and the explosion became almost inaudible. Indicator cards taken showed that the reduction in maximum pressure was counter-balanced by a slightly raised expansion line, when compared with diagrams of engines without internal coolant.

Professor Hopkinson concluded that, when using water injection, the pressure was better sustained, partly by the formation of steam and partly by the reduction in heat losses, with the result that the indicator cards became fatter and less peaky.

To regulate the water flow into the cylinder, a ball-valve was used and a special spray nozzle was fitted into the cylinder-head to direct the water to those parts of the engine which needed cooling most. So successful was this method that Hopkinson later designed some engines without any external water-jackets.

In the middle 20's a tractor appeared on the market, called "Rumley Oil Pull Tractor", which used kerosene for fuel and water for controlling detonation. At full load, a manually operated needle-valve permitted the entry of just sufficient water to stop detonation, with the result that noiseless running conditions were established. The only drawback to this arrangement was that if resetting of the valve at light load was forgotten, the engine was gradually flooded and trouble from corrosion was experienced.

In 1938 E. L. Barger and J. L. Gale (2) carried out a number of tests, using a "Hercules OOC" and a "McCormick-Deering" engine with water as an internal coolant. Their results may be summed up as follows:

"For low octane fuels of high distillation range, the maximum safe load could not be increased because sufficient water to limit detonation could not be injected without dropping the load. However, for low-octane fuels with a low distillation range, the maximum safe load was increased considerably without appreciable sacrifice of thermal efficiency. Medium-octane fuels (23-30 distillate) responded fairly well to water-injection, showing useful power increase."

At the same time it was found that the use of water-injection increased the thermal efficiency only when operating on such a load that detonation without water-injection had a medium intensity.

On the other hand, M.S. Kuhring (3), who injected water into the air-fuel mixture at the supercharger of a Jaguar MK.IV aircraft engine, found the following results:

"When water was injected at the rate of 83 lb. per 100 lb. of fuel, using a rich mixture, an increase of about 90 B.H.P. was obtained without rise in cylinder temperature and apparent detonation. The specific fuel consumption remained practically constant and quite an appreciable cooling of the air-fuel charge was noted."

At the beginning of World War II, two publications appeared, one in Germany by K. Zinner$_{(4)}$ and the other in U.S.A. by R. Wiebe, J.F. Shultz and J.F. Porter$_{(5)}$. In both papers, a theoretical approach was made to the calculation of engine conditions, when an internal coolant was injected into an engine cylinder.

The American team produced two sets of Mollier diagrams, similar in appearance to the thermodynamic charts, calculated by Hershy, Eberhardt and Hottel. The Mollier diagrams were derived for a chemically correct ethyl-alcohol-air mixture and for a chemically correct octane-air mixture plus water, the latter injected at the rate of 35 lbs. per 100 lbs. of fuel.

During World War II, water and water-alcohol injections were used with great success on supercharged aircraft engines to increase internal cooling at taking-off and at maximum flight speed. Intensive research was carried out by a number of people at various laboratories in the U.S.A. and in Great Britain, and most of the results were published in various N.A.C.A. wartime reports.

The National Research Laboratory of Ottawa, for instance, used a supercharged Armstrong Siddeley Jaguar MK.IV engine. The water for injection was fed from the City main under pressure, passed through a calibrated orifice plate, and then through a copper tube into a distance piece between the supercharger and the carburettor. The water, travelling at high speed, was atomized when leaving the tube while being drawn into the supercharger impeller. Water, in this case, not only suppressed detonation but provided intercooling of the intake air by evaporation of the liquid in the supercharger.

At the same time, R.J. Koenig and G. Heiser (6) studied the effect of water injection on the cooling characteristics of a Pratt and Whitney R 2800 aircraft engine, while T.D. Wear (8) and his team investigated the effect of various internal coolants on knock-limited and temperature-limited power of aircraft engines. A.H. Bell (9) wrote another paper, dealing with the continuous use of internal coolants to suppress knock in aircraft engines, cruising at high power. E.W. Steinitz (10), who also studied the influence of water injection on the performance of aircraft engines pointed out that the use of water as an anti-detonant, with its attendant equipment, complicates the engine and increases the total weight of the aircraft. Danger also exists that the water may freeze unless mixed with an anti-freeze solution.

Further articles, all dealing with the influence of internal coolants on the performance of aircraft engines, were written by the following authors: R.J. Brun and others (11), M.L. Harris and his team (12), R.L. Nelson (14), J.E. Vandeman and O.H. Heinicke (16). Their findings were also published in N.A.C.A. wartime reports between 1943 and 1945.

In 1946, another paper was printed, dealing with the investigations of M.R. Rowe and G.T. Ladd (17). In this article the influence of water and water-alcohol injections on the performance of aircraft engines, with special reference to cylinderhead temperature cooling, was discussed. Data were obtained by using a Wright Cyclone, 9-cylinder engine, operated at low cruise power. Various alcohols were tried in a number of water-alcohol mixtures and the results of the authors' research can be summed up as follows:

"Cylinder cooling, obtained from using water or water-alcohol injection, represents approximately 30% to 40% of the heat of vaporisation of the injected coolant. The major cylinderhead temperature cooling effect is obtained during the "High Temperature and Pressure Peaks" of the combustion cycle, and water provides the greatest degree of cooling at high power output. Water injection can be applied most efficiently at lean fuel-air mixtures, which is important for the economy of operation."

The research work during the period 1942-1946, dealing with the injection of various internal coolants into aircraft engines, established the following facts:

(1) There is no loss of thermal efficiency with low rates of water injection.

(2) Cooling of the engine at high output is not such an important problem.

(3) Supercharged engines will give higher outputs due to cooling of the mixture by the injected water. The higher manifold pressure increases the maximum cylinder pressure and the engine output. Detonation may be fully suppressed by the injected water.

(4) Space and weight requirements for carrying the water in aircraft engines may be excessive unless condensing equipment is used to secure water from exhaust gases.

(5) There is danger of water freezing unless mixed with an anti-freeze solution.

(6) Water provides the greatest degree of cooling at high output.

(7) Water-methanol mixtures provide the greatest saving in engine critical altitude.

(8) Water injection will permit the use of 91 and 87 octane fuels in place of 100 octane for equivalent power up to take-off horsepower.

(9) Water injection can be used most efficiently at lean

fuel-air mixtures.

After World War II, experiences gained from the injection of anti-detonants into aircraft engines, were used on automobile and tractor engines. Work in this field was stimulated by the following:

Firstly, by the absence of fuels which could fully satisfy the octane requirements of modern car engines with compression ratios, expected to go up to 10:1.

Secondly, by the difference in price of premium-standard- and low-grade fuel.

Thirdly, by the possibility of further increase of the compression ratio of engines, already using high-grade fuels.

Fourthly, by the hope of improving the performance and economy of existing and new engines.

At the same time, new coolant injectors, such as the "Vista Meter" were designed, which in many ways proved more efficient than the forerunners of the 1940's. They were simpler in construction, worked more quickly and gave more precise response to changing conditions of load. Furthermore, injection rates could be adjusted to provide the best condition for each operation and these injectors worked equally efficiently at all speeds. Coolant flow was controlled by the manifold vacuum.

Since 1945, quite a number of papers have been published, dealing with internal coolant injection, the most important of these being now briefly discussed.

In 1945, A.T. Colwell, R.E. Cummings and D.E. Anderson(13) undertook extensive investigations on the influence of water and water-alcohol injection into the air-fuel mixture of automobile engines. A "White" 6-cylinder truck engine of 318 cu.in. displacement was used, and tests were carried out with compression ratios 6.5:1 and 7.25:1. Fuels with octane numbers 51.5, 69.8 and 79.5 were used and the carburettor setting changed at various tests. Lean, standard and in some tests rich mixtures were used.

One of the main objects of their research was to determine the relative efficiency of various percentages of both alcohol and water from 0 to 100% in an internal coolant, best suited for power and economy.

The authors found that a water-alcohol mixture (50-50 by volume) was the best average combination, particularly at the price of 74¢ per gallon of alcohol at the time of the investigations. A curve shown in their article represents a typical analysis, comparing the operation cost of a vehicle requiring 79.5 octane petrol (20¢ per gal.) against 69.8-octane petrol (17¢ per gal.) but using (50-50) water alcohol injection. The abscissa indicates the per cent of running time at full load, while the ordinates represent the permissible cost per gallon alcohol for equal fuel cost. From this graph it can be seen that for a truck running 40% of the time at full throttle, the allowable cost per gallon alcohol worked out 74¢.

When comparing the performance curves of the 69.8-octane plus water-alcohol and 79.5-octane petrol, for instance, there is not a great deal of difference and it can be concluded that the anti-detonant is responsible for the increase of about 10 octane numbers at 1200 r.p.m. Water-alcohol consumption at this point is about 28% by weight of the total fuel.

In addition, the tests conducted indicate that the use of water has many advantages in respect of engine wear and engine life. Water injection seems to eliminate carbon formation in the engine, thus reducing the building up of sludge in the oil, spark plug fouling, and improper valve seating. Also ring wear is held to a minimum as a result of the controlled engine conditions.

In July 1946, W.P. Green and C.A. Shreeve Jr.(19) published a paper, in which they presented new data regarding the use of water in very high compression engines to increase part-load efficiencies with low octane fuels.

The authors used a single-cylinder N.A.C.A. Universal test engine and the experiments were carried out at the Mechanical Engineering Laboratory of the University of Maryland, U.S.A.

Green and Shreeve confirmed the results obtained by their predecessors, namely, that injection of water into the engine cylinder decreases the maximum pressure and tends to decrease the slope of the expansion line with little or no change in thermal efficiency.

The authors also found that water injection in high compression engines (with compression ratios higher than 10:1), operating most of the time at part throttle, will improve economy.

In 1948, E.F. Obert(26) published a paper on detonation and internal coolants. In this excellent article, the effect of internal coolants on the combustion process, on detonation and on engine performance were investigated.

For the tests, the author used a C.F.R.-F2 engine, the speed was kept constant at 900 r.p.m. and the throttle fully open. Compression ratios varied between 5.9:1 and 6.45:1. Obert's investigations proved that water, if injected into an unheated manifold of an unsupercharged high-speed engine, has no time to vaporize during the compression stroke. Complete vaporization of the internal coolant is probably not secured until the combustion process is under way.

The author pointed out that the slowing of the combustion rate and the vaporization of the water contained in the end charge, are responsible for the lower temperature of the latter, thus suppressing detonation.

Finally, Obert showed that the injection of water or water-alcohol reduces appreciably the coolant load (the percentage of the total heat transferred to the jacket-water) without any special loss in power output.

About the same time as Obert, J.C. Porter and his team(30) worked on the boosting of engine performance with alcohol-water injection. They used a Crosley 1947 engine, the compression ratio being raised to 9:1. Regular petrol of 73 octane number was used, with the spark advance set for trace knocking at maximum power.

Porter reported that water alone, at a compression ratio of 9:1, did not suppress detonation even with an 80 octane number base fuel since flooding ensued.

Retarding the initial spark setting eleven degrees, compared with the setting for maximum power, was not sufficient to overcome detonation even with an 85 octane number base fuel.

In September 1948, R.I. Potter(34) published his results on the use of anti-detonant injection in a high compression engine. The author investigated the possibility of a low octane number petrol to supply the basic cruising fuel requirements and the injection of an internal coolant to satisfy the high octane needs during peak load conditions.

Potter's tests verified Obert's investigations, namely, that by means of a suitable anti-detonant injection it is possible to obtain a remarkable increase in octane performance of premium fuels, raising these petrols of 85-90 road octane numbers up to 100 or better, during the periods of high octane requirements.

Basic fuels of the aromatic, naphthenic, paraffinic and olefinic type can be extended to essentially 100 octane number by means of an anti-detonant injection, the amount added being varied to suit the requirements of the specific fuel.

Tests were carried out by using the "Modified Border Line" test procedure to obtain the road octane knock rating over the speed range. This method and the "Modified Uniontown" method are based on the ability of the fuel to tolerate spark advance.

In October 1948, the outcome of Potter's research was further discussed in a paper, called "Getting More M.P.G.'s from Octane Numbers"(35). Potter and his team used various alcohol-water mixture, some with a small percentage of tetra-ethyl-lead as anti-detonants. The mixtures were injected with the help of an improved model of the "Vita-Meter." A fleet of 188 tank-trucks was operated with these injectors, using a (45-55)% isopropanol-water solution. The full throttle injection rate averaged 0.225 lb. of coolant mixture per lb. of fuel. The overall use of the internal coolant was found to be 5.5% of the petrol consumed. The anti-detonant added 10 octane numbers on top of a 63 octane number straight-run petrol.

An (85-15)% methanol-water solution, containing 3 cc. of T.E.L. per gallon, was yielding the same results for a 2% overall usage rate. The quoted octane numbers were A.S.T.M. motor octane numbers but the increments were road octane numbers.

It is interesting to note that the above mixture produced a super-charging effect, indicating evaporation in the manifold with a consequence decrease in temperature and increase in the charge density.

About a year later, R.W. Scott, G.S. Tobias and P.L. Haines(39) published a paper, dealing with the antiknock quality requirements of high compression ratio car engines. Sixteen different petrols, varying in composition and octane number, were used in cars with compression ratios varying from 7.5:1 to 12.5:1. The "Borderline Technique" was used to study the road anti-knock quality of the fuels under observation.

The fuels studied covered a range of about 79-102 by the motor method and 90-100 by the research method on all cars, the spark advance with increasing speed was normal in relation to current automotive practice.

The authors concluded that as the compression ratio is increased, the research and motor method octane requirements increase in an orderly manner but the increase becomes smaller as the compression ratio increases. In high compression ratio-engines, the research octane number requirement is limited and fuels of great sensitivity can be tolerated with increasing compression ratios. Results obtained indicate that premium grade petrols with anti-detonant injection could have been used instead of the high octane number fuels.

In 1950, J.C. Porter, M.M. Gilbert, H.A. Lykins and R. Wiebe(41) published the outcome of their extensive research on alcohol-water injection for high compression tractor and automobile engines.

Investigations were carried out to find the alcohol injection requirements for a Minneapolis-Moline tractor engine with a C.R. 8:1, speed 1250 r.p.m., spark advance of 19 degrees, and also for a General Motors high compression engine (HC-125), C.R. 10:1, speed 1000 r.p.m. Various alcohol mixtures were used in conjunction with a reference fuel of 85 octane number and two other petrols of 86/81 and 91/82 octane numbers. The authors mentioned that the engines did not react differently if the alcohol was introduced by injection rather than as a blend.

Tests were carried out at part and full throttle, and the alcohol-water requirements were observed.

In a very instructive table, the knock ratings of various alcohol blends are presented. Octane numbers are shown in Research, Motor, Modified Uniontown and Borderline methods. The figures were obtained from tests on two engines - an Oldsmobile, C.R. 10:1 and a Plymouth C.R. 6.6:1. From the figures in the table the conclusion may be drawn that the maximum octane requirement of an engine is in better agreement with the Research method octane number than with the Motor method, and that the agreement is even better at high compression ratios.

In the same paper, the authors investigated also the influence of tetraethyl lead, added to the anti-knock mixture, on required injection ratios (anti-detonant to petrol). At part-throttle operations it was found that the reduction of this ratio was proportional to the quantity of T.E.L. added. At full throttle, however, the reduction of the ratio was more pronounced for small quantities of T.E.L.

From the test results it can be seen again, that premium grade petrols, in combination with alcohol-water injection, will give a knock-free performance over the full speed range. Higher compression ratios will be possible, which in turn will improve the economy, an important factor for farmers and commercial operators.

In 1953, J.C. Porter and R. Wiebe(42) published the contents of their previous paper(41) in a different form. In the new article, called "Alcohol as an Anti-knock Agent in Automotive Engines", the actual potential performance gain of an engine, derived from the blending of the base stock petrol with alcohol or by the injection of an alcohol-water mixture, is not given on the octane number scale but on the so-called performance number scale. This is more realistic than any octane number scale.

The authors write that "For performance number gains of 5-10 with present days engines and where the base fuel, used with injection, is in the octane range below 85, a (50-50) alcohol-water mixture appears as good as an 85-100 ethyl alcohol mixture and more economical.

For very small octane gains, 100% water may be used. For performance number gains of 10-30, the alcohol content of the mixture should be between 50-100 to prevent any extensive injection requirements.

Methanol is for all practical purposes equivalent to ethyl alcohol in anti-detonant mixtures, but isopropyl alcohol alone is somewhat less effective."

In May 1954, R. Wiebe and J.D. Hummell (43) published the results gained with alcohol-water injection in army trucks and farm tractors. This paper is actually a continuation of two previous publications of Wiebe and others, taking into account the influence of internal coolants on lubricating oils and engine wear, as well as fouling.

To be able to compare results, three 2 1/2-ton trucks were used; one was run with a prototype 80 octane number combat grade fuel, the second with a 68 octane number straight run petrol in combination with alcohol-water injection and the third one was operated for about 2,500 miles with the 80 O.N. combat grade fuel and for the remaining miles with 68 O.N. straight-run petrol plus injection. The trucks were driven at double the rated pay load (10,000 lb.) over roads in various conditions for approximately 8,000 miles. Lubricating oil and filter cartridges were changed every 5,000 miles. During the road tests, the octane requirements were determined with and without alcohol-water injection by means of the Modified Uniontown method at approximately 1,700 miles intervals.

At the same time, the Agricultural Engineering Department of the Ohio Agricultural Experiment Station supervised the operation of 50 tractors on various farms. Each tractor carried a calibrated injector, and specially denatured (50-50) alcohol-water mixture was used as the anti-detonant.

From the tests conducted, the authors draw the following conclusion: "Alcohol-water injection in combination with a petrol of 10 octane numbers lower than the engine octane requirement gave knock-free operation in two army-type trucks. This combination of straight-run petrol with injection resulted in cleaner engines (less sludge, varnish and combustion chamber deposits) and less cylinder wear than when engines were operated with a prototype petrol of 80 O.N.

Since carburettors were all set at the same flow rate, no fuel economy was realised.

The conclusion was also drawn that alcohol-water injection will provide an increase up to about 15 O.N. in the distillate and kerosene range.

Furthermore, with any given fuel which satisfies the octane requirements, the compression ratio of the tractor engines can be increased by about 1.5 numbers when using injection. It was found that the tractor engines were sufficiently strong to stand up to the extra compression ratios for one year without failure. However, to ensure trouble-free operation with an injector, the alcohol-water mixture must contain a corrosion inhibitor to stop clogging of the injector passage. Again, it was observed that straight-run petrol in combination with alcohol-water injection results in cleaner engines and consequently longer engine life.'

Finally, in 1955 C.F. Kettleborough and E.E. Milkins$_{(44)}$ published the results of their experiments on the effect of water injection into a petrol engine in need of an overhaul. Tests were carried out at the University of Melbourne and a Ricardo E-6 type variable compression engine was used to find what improvements, if any, could be obtained by the use of water injection on an engine with wrong valve timing sequence.

Fuel consumption loops at half-throttle and constant engine speed of 2,000 r.p.m. were plotted for three groups of tests:

The tests were carried out at compression ratios 5.5:1, 6.2:1, 7:1 and 7.55:1. The C.R. 6.2:1 proved to be the highest useful compression ratio for the condition of the engine and the fuel used.

At each compression ratio, the minimum quantity of water was found by setting the control valve to give the water flow just necessary to eliminate knocking at the correct air-fuel ratio. The water-fuel weight ratio decreased therefore with increase of fuel flow.

The authors found that for the compression ratio 5.5:1, using water injection, the specific fuel consumption was increased and the power output decreased compared with no injection. For higher compression ratios, the engine could only be run with water injection as the engine was prone to detonate. At a C.R. 7.55:1, the minimum amount of water for the correct air-fuel ratio was found to be 1.65 lb. per lb. of fuel. This quantity of water eliminated detonation completely but it was found that the water could be cut down to about 1/2 lb. per lb. of fuel if a small intensity of detonation could be tolerated.

Kettleborough and Milkins concluded that, for any given compression ratio, there is a fall in thermal efficiency and power as the quantity of water is increased above the minimum demand to eliminate detonation.

Inspection of the engine after completion of the tests proved that the piston, cylinder head and valve were entirely free of carbon deposits.

In the light of the previous published information, the author of this thesis feels that tests conducted over a range of compression ratios and speeds, using water or water-alcohol injection as anti-detonant, would accomplish the following:

(1) Give fresh data on the best coolant-fuel weight ratio needed at full and part throttle operations, when conducted with spark setting for trace-knocking.

(2) Determine whether worthwhile gains in power and economy could be made by using water or (50-50) water-ethyl alcohol as anti-detonant, when conducting tests with spark settings as in (1).

(3) Find the combined specific fuel cost (petrol plus alcohol) and present the results in graph form such that the combined fuel costs and combined specific fuel costs may be read off directly.

(4) Find the influence of water or water-alcohol injection on the internal condition of the engine after at least 100 hours running time.

(5) Measure the mean-temperature of the gases in the combustion chamber of the engine. These readings should give an indication of the drop of the maximum cycle temperature when using internal coolants at various weight ratios.

CHAPTER 2

Influence of an Anti-detonant Injection on a Detonating Combustion Process.

To understand the influence of an anti-detonant injection on the complex combustion process, it should be remembered that the latter is strongly influenced by the rate of burning, the rate of pressure rise and the temperature of the charge during combustion. While the rise in pressure through the combustion chamber is fairly uniform, the rise of temperature is usually non-uniform.

From knowledge of the combustion process it is generally assumed that the unburned portion of the mixture in the combustion chamber, being further compressed by the approaching flame-front, may reach self-ignition temperature. If the unburned mixture, often referred to as end-gas, is held at or above this critical temperature for a finite time, instantaneous burning of the end-charge may occur before the arrival of the flame. This nearly instantaneous burning of the remaining charge in the combustion chamber is called detonation. D.C. Miller[22], an authority on detonation, describes it as an abnormally high rate of energy release, which causes a definite and measurable pressure differential within the combustion chamber.

Practically, one speaks of detonation when a pressure differential of such an intensity exists that the combustion chamber walls vibrate, thus producing a high-pitched sound called "knocking".

The best way to illustrate detonation in an internal combustion engine is to take photos of pressure-rate/time diagrams with the help of a cathode-ray indicator and to note the degree of sound intensity. From such photos it can easily be seen that detonation does not occur at a critical compression ratio, but is an inherent characteristic of the engine and occurs in varying degrees of intensity, depending on the compression ratio and other engine variables.

In the normal operation of an engine, detonation takes place as the piston starts the working stroke. Since the total combustion time in a detonating engine is shorter than in a non-detonating one, a greater amount of energy will be released before the piston has moved far. For this reason, detonation should produce a higher average combustion temperature.

E.F. Obert(26), who has carried out special tests to establish the influence of water and water-alcohol injection on the combustion process, found that detonation is responsible for an increase in heat losses to the cooling water jacket and a corresponding decrease of the energy removed in the exhaust gases.

Thus, the exhaust gas temperature of a detonating engine should decrease with the increase of knock intensity of an engine.

Obert connects energy losses of the gas in a detonating engine with the following factors:

1) Increased radiation from the gas, brought about by the higher temperature of the engine, probably due to the quicker release of chemical energy.

2) Increased conduction from the gas, as the pressure differential and the gas vibrations lead to a more thorough scrubbing of the combustion chamber walls and piston crown, thus reducing the film resistance. At the same time the higher temperature itself will increase heat losses due to conductivity.

3) Hysteresis losses in the combustion chamber walls and piston crown from forced vibrations.

4) Lesser amount of chemical energy liberated, because of

the different types of reactions, or resulting from the storage of vibrational energy.

To reduce these losses, which may cause a serious reduction in power output of an engine, injection of an internal coolant was introduced by various research workers. Their work was discussed briefly in chapter one.

Scientists agree on the fact that an anti-detonant, if introduced with the charge and in the right quantity, will make knocking disappear and provide the means of satisfying the engine demand for better fuels without increasing the octane number.

It is now generally believed that the slowing down of the combustion process and the cooling of the end-gas by vaporization of the cooling medium are, in principle, responsible for this improvement.

The slowing down of the combustion process, brought about by the addition of the cooling medium to the fresh charge, has the tendency to dilute the available oxygen, thus reducing the flame temperature. At the same time, the higher mean specific heat, derived from the water injected, will also have the tendency to decrease the maximum cycle temperature.

As mentioned previously, a diminution of the flame temperature will reduce the rate of burning and have a benevolent influence on the progress of combustion.

Thus, the cooling of the end-gas, especially if water with its high latent is used, enables the flame front to finish its travel without the end-gas reaching its critical temperature and so self-igniting. The latent heat of some of the cooling media is as follows:

Water	970	B.T.U./lb.
Methyl alcohol	473	' '
Ethyl alcohol	376	' '
Ethyl alcohol and water, (50-50) by volume	675	' '
Petrol	135-150	' '

Tests carried out by R.J. Brun (11), during which water was directly injected into the end-zone of the charge at the instance of knocking, showed that knocking could be completely eliminated.

High-speed photos of quenched and unquenched combustion cycles, which were taken by C.D. Miller (22) revealed a definite relative increase of pressure during the expansion stroke when water is used as an internal coolant. Quenching seems to reduce the temperature of the end-gas immediately, to such an extent that any knock reaction and the coupled energy losses disappear.

Finally, it should be mentioned that investigations by Obert have shown that when a liquid and a charge are compressed together, vaporization of the liquid will cool the gas and water, with its high latent heat, is particularly suited for this purpose. In an unsupercharged engine, water has not sufficient time to vaporize until the combustion process is well under way, but in supercharged engines an increase in output may be obtained, derived from the smaller compression work.

CHAPTER 3

Theoretical Investigation of the Thermodynamic Effect of Adding an Internal Coolant to an Octane-Air Mixture.

In Chapter 2 it was quoted that the addition of an internal coolant to the fresh charge has the tendency to reduce the flame temperature and maximum pressure, with the result that in a knocking engine detonation disappears.

In this chapter, a theoretical approach is adopted to determine the maximum cycle temperature in a petrol engine, when water is used as an internal coolant. The result, if compared with the figure read from the equivalent Hottel chart, will indicate the drop in combustion temperature.

Graphs calculated and plotted by Wiebe, Shultz and Porter(5) for octane as fuel and water as an anti-detonant, (Figs. 5 and 6), may be used to find the maximum cycle temperature and pressure for water-fuel weight ratios in the neighbourhood of 0.30.

For the following calculations, the proportion of 25% by weight of the fuel was used, as this proportion is the same as one of the ratios employed during the experimental work. The residual gases were assumed as 5% by volume, a figure found in engines with compression ratios about 7:1. The inlet pressure "p_1" was assumed as atmospheric.

The general equation for the reaction of a chemically correct octane-air mixture, when using a certain percentage of water as internal coolant, can be written:

$$C_8H_{18} + 12.5\ O_2 + 47N_2 + rH_2O_{liqu.} =$$
$$8CO_2 + 9H_2O + 47N_2 + rH_2O_{vap.} \quad \ldots\ldots\ldots\ldots\ldots \quad (3.1)$$

The number of lb.-mols of liquid water is found as follows:

Assuming a weight ratio of 0.25 and the molecular weights of water and octane as 18 and 114 respectively, then,

$$r \times \frac{18}{114} = 0.25 \text{ or } r = \underline{1.585} \text{ lb.-mols} \quad \ldots\ldots\ldots\ldots\ldots \quad (3.2)$$

Equation (3.1), for 0.95 lb.-mols of fresh charge, when using a chemically correct mixture, takes the form :

$$\frac{0.95}{62.085}\ (C_8H_{18} + 12.5\ O_2 + 47N_2 + 1.585H_2O_{liqu.}) \text{ or}$$

$$0.0153C_8H_{18} + 0.1913\ O_2 + 0.7191N_2 + 0.0243H_2O_{liqu.} =$$
$$0.1224CO_2 + 0.1377H_2O_{vap.} + 0.7191N_2 + 0.0243H_2O_{vap.} \quad (3.3)$$

5% of residual gases, therefore, are comprised of:

$$0.00612CO_2 + 0.006775H_2O_{vap.} + 0.03595N_2 + 0.001215H_2O_{vap.}$$

The total charge of one lb.-mol then consists of :

$$0.0153C_8H_{18} + 0.1913\ O_2 + 0.7550N_2 + 0.0061CO_2 +$$
$$0.0243H_2O_{liqu.} + 0.0080H_2O_{vap.} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.4)$$

The next step is to find the internal energy of the charge before and after compression. For this purpose, a constant volume line of one lb.-mol at N.T.P. has to be drawn, by plotting internal energy versus entropy. The internal energy and the entropy of the constituents of equ. (3.4) can be taken from printed data. L. Lichty's "Thermodynamics" was used in this case. The internal energy and entropy of the charge are found from the sum of the values of the constituents, since they are not absolute values but are measured above the datum of N.T.P.

Internal energy of the constituents with the exception of water, in B.T.U. above 520^o.

Temp. oR	C_8H_{18} 0.0153 mols	O_2 0,1913 mols	N_2 0.7550 mols	CO_2 0.0061 mols	H_2O vap. 0.0080 mols	Intern. energy
520	0	0	0	0	0	0
600	51.75	77	298	3.5	3.9	434.15
700	124	176	676	8	9	993
800	204	278	1055	13	14	1564
900	292	381	1435	18	19	2145
1000	388	486	1820	23.5	24	2741.5
1100	494	595	2210	29	29	3357
1200	606	704	2610	35	35	3990
1300	727	816	3010	41	40	4634
1400	855	730	3430	47	46	5308
1500	995	1050	3840	53.5	52	5990.5

Table No.1

As Lichty's textbook does not give entropy values for C_8H_{18} (n-octane), figures of $\int C_p.\frac{dT}{T}$ were taken from "Selected values of properties of hydrocarbons", circular of the National Bureau of Standards, printed by U.S. Dept. of Commerce.

To convert the values of $\int C_p.\frac{dT}{T}$ into $\int C_v.\frac{dT}{T}$, the following relation can be used:

$$\int_{T_1}^{T_2} C_v.\frac{dT}{T} = \int_{T_1}^{T_2} C_p.\frac{dT}{T} - R.\ln\frac{T_2}{T_1} \quad \text{......................} \quad (3.5)$$

The entropy values in the publication of the National Bureau of Standards are absolute values and the value at 520°R has to be subtracted to give values above this datum.

Entropy of n-octane in B.T.U./°R/lb.-mol

Temp. °R	$\int C_p.\frac{dT}{T}$ abs.value	$\int C_p.\frac{dT}{T}$ corr.val.	$1.98.\ln\frac{T}{520}$	$\int C_v.\frac{dT}{T}$
520	109.5	0	0	0
600	117	7.7	0.28	6.72
700	125	15.5	0.59	14.91
800	133	23.5	0.86	22.64
900	140.5	31	1.09	29.91
1000	148	38.5	1.30	37.20
1100	155	45.5	1.49	44.01
1200	162	52.5	1.66	50.84
1300	169	59.5	1.82	57.68
1400	176	66.5	1.97	64.53
1500	182	72.5	2.10	70.40

Table No.2

$\int C_v \cdot \frac{dT}{T}$ values of the constituents, with the exception of water, in B.T.U./°R above 520°R.

Temper. °R	C_8H_{18} 0.0153 mols	O_2 0.1913 mols	N_2 0.7550 mols	CO_2 0.0061 mols	$H_2O_{vap.}$ 0.0080 mols	$\int C_v \cdot \frac{dT}{T}$
520	0	0	0	0	0	0
600	0.103	0.144	0.545	0.0062	0.0075	0.8057
700	0.228	0.300	1.125	0.0133	0.0147	1.6817
800	0.347	0.432	1.622	0.0198	0.0210	2.4418
900	0.458	0.552	2.08	0.0259	0.0270	3.1429
1000	0.570	0.662	2.48	0.0317	0.0326	3.7763
1100	0.674	0.764	2.85	0.0370	0.0378	4.3628
1200	0.780	0.860	3.21	0.0422	0.0426	4.9348
1300	0.883	0.952	3.53	0.0468	0.0468	5.4586
1400	0.986	1.040	3.84	0.0514	0.0510	5.9684
1500	1.078	1.118	4.125	0.0560	0.0550	6.4320

Table No.3

Calculation of internal of energy of 0.0243 lb.-mol of wet steam.

0.0243 lb.-mol water = 0.437 lb.

From Callendar's tables for saturated steam, specific volumes and internal energies may be found as follows:

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Temperature	Specif. Volume $\frac{ft.^3}{lb.}$	Content of dry steam in 358 $ft.^3$ lb.	Intern. Energy B.T.U./lb	Int.energy corrected for 520^{o}R as datum B.T.U./lb.	(3)x(5)	Inter. energy of water B.T.U.	Total inter. energy B.T.U.
60 520	1230	0.291	1025.6	0	0	0	0
68 528	922.19	0.388	1028.5	2.9	1.13	0.049x8 =0.392	1.522
71.5 531.5	819	0.437	1029.8	4.2	1.84	0	1.840

Above 531.5^{o}R steam becomes superheated.

Table No.4

Calculation of internal energy of steam in the superheated region.

According to table 4, the water injected becomes superheated at 531.5 ^{o}R, the corresponding specific volume being 819 ft^3/lb. To find the partial pressures, Callendar's general equation may be used:

$$V_s = \frac{R.T.}{P} - C + b \; ft^3./lb. \quad \ldots\ldots\ldots\ldots\ldots \; (3.6)$$

In this equation: $R = 154.17$ ft.-lb./lb. ^{o}K.

P is the pressure in lb./ft^2. absolute and

T is the temperature in degrees K.

$C = \frac{157.52 \times 10^6}{T^{3.33}}$ and

b is a constant = 0.01604

"C" can be calculated or taken from tables.

For K = 303, C = 0.842

and for K = 800, C = 0.0335

Owing to the small values of C and b they may be neglected.

From simplified equation (3.6) "P" may now be calculated as:

$$P = \frac{R.T}{V_s} = \frac{154.17 \times T}{819} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \; (3.7)$$

If "P" should be expressed in lb./in^2. and "T" in degrees R equation (3.7) may be written as:

$$P = \frac{154.17 \times T}{144 \times 1.8 x V_s} \; lb./in^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \; (3.8)$$

$$\text{For } V_s = 819; \quad P = \frac{154.17 \times T}{144 \times 1.8 x 819} = \frac{T}{1377} \; lb./in^2. \; \ldots(3.9)$$

Using equation (3.9), the following values of partial pressures have been calculated:

Partial Pressures

Temperature "T" ^oR	Part. pressure "p" lb./in^2.
531.5	0.385
550	0.400
600	0.435
700	0.508
800	0.581
900	0.654
1000	0.726
1100	0.800
1200	0.872
1300	0.945
1400	1.018
1500	1.090

Table No.5

The internal energy of steam may now be found, using the Callendar's formula:

$$E = 0.36709T - 3.33\frac{c.p}{1400} + 464 \text{ C.H.U./lb.} \quad \ldots\ldots\ldots \quad (3.10)$$

The second member of equ.(3.10) can, for the purpose of the following calculations, be neglected, and if "T" is expressed in degrees "R" instead of degrees "K", formula (3.10) may be written as:

$$E = 1.8(0.36709\frac{T}{1.8} + 464) = 0.36709T + 835 \text{ B.T.U./lb.} \quad ..(3.11)$$

Internal energy of 0.437 lb. of H_2O

above 520°R

(1)	(2)	(3)	(4)	(5)	(6)
Temp. "T" °R	0.36709T	(2)+835	0.437x(3)	Intern. energy above 531.5°R B.T.U.	Intern. energy above 520°R B.T.U.
531.5	195	1030	450	0	1.84
550	202	1037	453	3	4.84
600	220	1055	460	10	11.84
700	257	1092	476	26	27.84
800	293	1128	493	43	44.84
900	330	1165	510	60	61.84
1000	367	1202	525	75	76.84
1100	403	1238	540	90	91.84
1200	440	1275	557	107	108.84
1300	478	1313	571	121	122.84
1400	514	1349	590	140	141.84
1500	550	1385	605	155	156.84

Table No.6

Entropy of 0.437 lb. wet steam

(1)	(2)	(3)	(4)	(5)	(6)
Temper.	Content of dry steam in 0.437 lb. lb.	Content of water in 0.437 lb.	Entropy of steam B.T.U./^{o}R	Entropy of water B.T.U./^{o}R	Total Entropy (4)+(5) B.T.U./^{o}R
520	0.291	0.146	0.610	0.008	0.618
528	0.388	0.049	0.800	0.0035	0.804
531.5	0.437	0	0.900	0.0	0.900

Table No.7

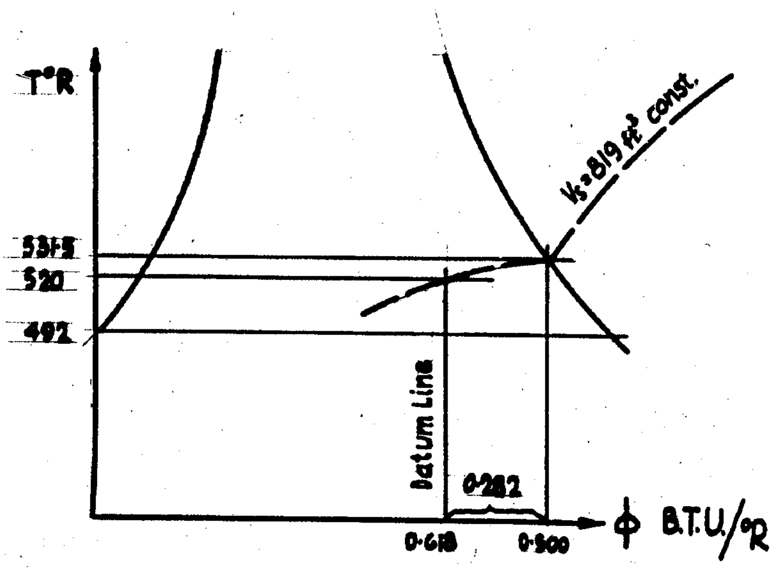


FIG. No 1

"T - Ø" diagram of 0.437 lb. of wet steam.

The entropy of steam in any dry state can be found from the equation:

$$\phi = 1.09876 \log_{10}T - 0.25356 \log_{10}P - 0.002381\frac{c.P}{T} - 0.21964 \quad \ldots\ldots\ldots \quad (3.12)$$

In the above equation, the third member may again be neglected and "P" expressed in terms of "T", using equation (3.7).

Equation (3.12) then takes the form:

$$\phi = 1.09876 \log_{10}T - 0.25356 \log_{10}(0.189T) - 0.21964 =$$

$$0.8452 \log_{10}T - 0.3618 \text{ B.T.U./}^{\circ}\text{K/lb.} \quad \ldots\ldots\ldots\ldots \quad (3.13)$$

In this equation, "T" represents degrees "K". If degrees "R" are to be used, formula (3.13) can be changed to:

$$\phi = 0.8452 \log_{10}\frac{T}{1.8} - 0.3618 \text{ B.T.U./}^{\circ}\text{R/lb.} \quad \ldots\ldots\ldots \quad (3.14)$$

For the following calculations, the difference of entropy at "T" and 531.5°R has to be found and equation (3.14) takes its final form:

$$\Delta\phi = 0.8452 \log_{10}\frac{T}{531.5} \text{ B.T.U./}^{\circ}\text{R/lb.} \quad \ldots\ldots\ldots\ldots\ldots \quad (3.15)$$

Using the above formula, the entropy has been calculated and tabulated.

Entropy of 0.437 lb. of superheated steam above 520°R

in B.T.U./°R.

(1)	(2)	(3)	(4)	(5)	(6)
Temp. "T" °R	$\frac{T}{531.2}$	$\log_{10}$.(2)	0.8452x(3)	0.437x(4)	0.282+(5)
531.5	1	0.0	0.0	0.0	0.282
550	1.035	0.0149	0.0126	0.0055	0.288
600	1.130	0.0531	0.045	0.0197	0.302
700	1.320	0.1206	0.102	0.0445	0.327
800	1.505	0.1775	0.150	0.0655	0.348
900	1.690	0.2279	0.193	0.084	0.366
1000	1.880	0.2742	0.232	0.101	0.383
1100	2.07	0.3160	0.267	0.116	0.398
1200	2.26	0.3541	0.298	0.130	0.412
1300	2.45	0.3892	0.328	0.143	0.425
1400	2.64	0.4216	0.356	0.156	0.438
1500	2.83	0.4518	0.382	0.166	0.448

Table No.8

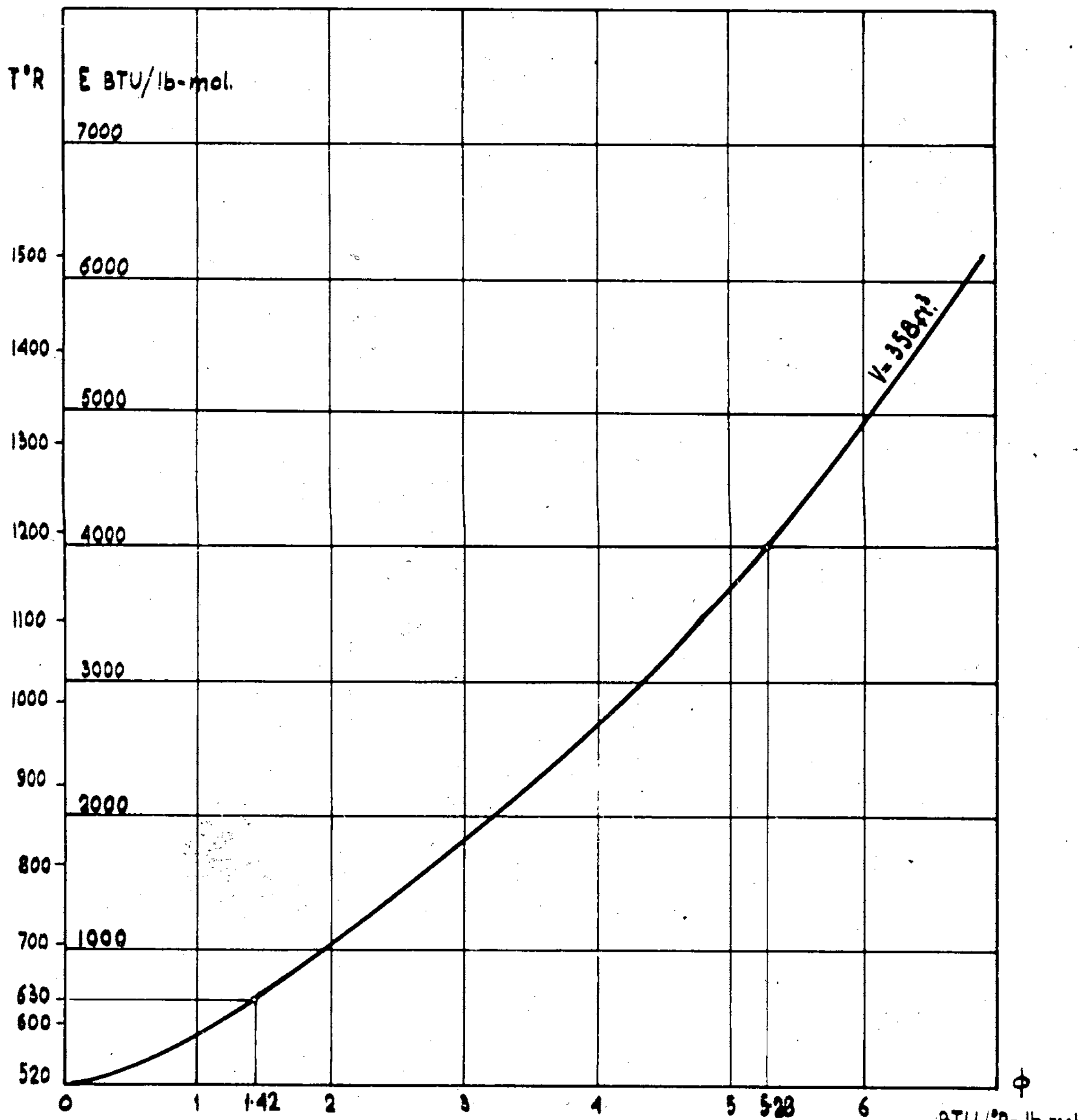
The internal energy and entropy of all constituents can now be found, taking the appropriate values from tables No.1, 2, 3, 4, 6, 7, 8.

Internal Energy and Entropy of all Constituents in B.T.U. above 520°R

Temperature °R	Intern.energy B.T.U./lb.-mol	Entropy B.T.U./°R/lb.-mol
520	0.0	0.0
531.5	76.84	0.350
550	154.84	0.550
600	445.99	1.1077
700	1020.84	2.0087
800	1608.84	2.7898
900	2206.84	3.5089
1000	2818.34	4.1593
1100	3448.84	4.7608
1200	4098.84	5.3468
1300	4756.84	5.8836
1400	5449.84	6.4064
1500	6147.34	6.8800

Table No.9

Using table No.9, the constant volume line can now be plotted (Fig. 2) and used to find the internal energy at the beginning and end of the compression stroke for one lb.-mol of charge.



INTERNAL-ENERGY VERSUS ENTROPY FOR 1 lb-mol. INITAL CYLINDER CONTENT.

FIG. No 2

If the temperature of the charge "T_1" at the beginning of compression be assumed as $630^{o}R$, a figure common for cycles with water-injection, then the corresponding internal energy and entropy read off from Fig.2 are:

$E_1 = 630$ B.T.U., and $\phi_1 = 1.420$ B.T.U./^{o}R.

These figures, as mentioned before, are not absolute values, but are based on $520^{o}R$ as zero value.

For the chosen compression ratio of 7:1, ϕ_2 can now be calculated as follows:

$$\phi_2 = \phi_1 + R.\log_e 7 = 1.420 + 1.987.\log_e 7$$
$$= 1.420 + 3.86 = \underline{5.28 \text{ B.T.U.}/^{o}R} \quad .. (3.16)$$

From the constant volume line, the internal energy "E_2" and compression end-temperature "T_2" can now be found.

$E_2 = \underline{4,000 \text{ B.T.U.}}$ and $T_2 = \underline{1,188^{o}R}$

To find the maximum cycle temperature "T_3", which is the aim of this investigation, the heat released during reaction and the internal energy have to be plotted against temperature for the products of combustion. The intersection of the two curves will provide the solution.

Total heat released can be calculated from the heat of reaction of n-octane, less the heat of dissociation of the dissociation products. For this purpose dissociation quantities have to be found at various temperatures.

A tabular method, developed by the author(45), will be used to calculate the dissociation quantities.

To find the dissociation quantities at a certain temperature, the following equations have to be solved:

$$\frac{(CO)^2 \times (O_2)}{(CO_2)^2} = K_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \tag{3.17}$$

$$\frac{(CO) \times (H_2O)}{(CO_2 \times (H_2)} = K_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \tag{3.18}$$

$$\frac{(OH)^2 \times (H_2)}{(H_2O)^2} = K_3 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \tag{3.19}$$

$$\frac{(H)^2}{(H_2)} = K_4 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \tag{3.20}$$

$$\frac{(O)^2}{(O_2)} = K_5 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \tag{3.21}$$

$$\frac{(N)^2}{(N_2)} = K_6 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \tag{3.22}$$

$$\frac{(NO)^2}{(O_2)(N_2)} = K_7 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \tag{3.23}$$

The quantities in brackets are the partial pressures of the constituents, and K_1 to K_7 the equilibrium constants for an assumed temperature, T.

If combustion takes place at constant volume, as shown in Fig. 3, the maximum pressure "p" for an absolute temperature "T" may be found as follows:

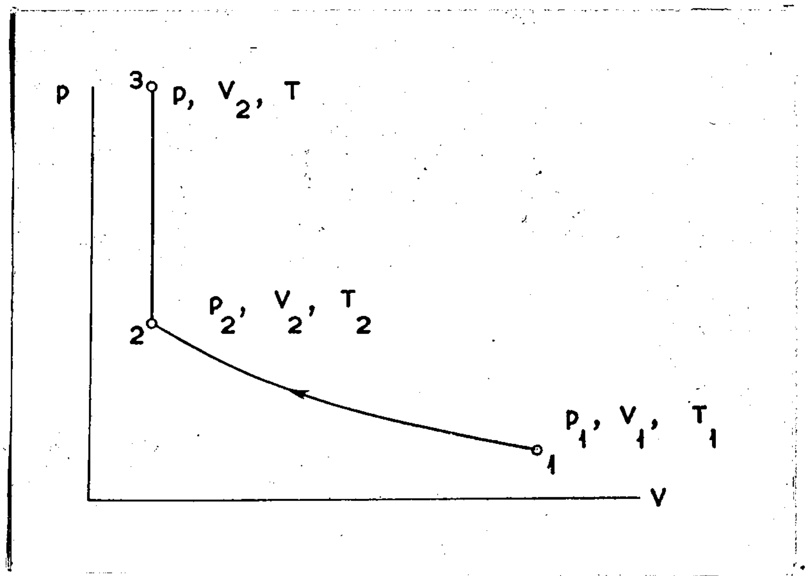


Fig. 3 - Combustion at constant volume.

$$144\ p.V_2 = m_p.G.T. \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.24)$$

$$144\ p_1V_1 = m_m.G.T_1$$

$$\text{therefore: } p = p_1.\frac{V_1}{V_2}.\frac{T}{T_1}.\frac{m_p}{m_m} = p_1.r.\frac{T}{T_1}.\propto \quad \ldots\ldots\ldots \quad (3.25)$$

In the above equations "p" and "T" represent the pressure and temperature of the products of combustion (without regard to the true final value p_3 and T_3), "r" the compression ratio and "$\propto$" the molecular ratio. This is the ratio of the number of molecules of the products of combustion, m_p, to the number of molecules of the mixture before combustion, m_m. If "m_p" is replaced by the number of mols of a constituent, then "p" in equation (3.25) becomes the partial pressure and its value can be inserted in the various equilibrium equations. In cycle calculations for petrol engines it is usual to work with a mixture "m_m" of one lb.-mol.

Equation (3.17) can, therefore, be written:

$$\frac{\left(p_1 . r . \frac{T}{T_1} . \frac{m_{CO}}{m_m}\right)^2 \times \left(p_1 . r . \frac{T}{T_1} . \frac{m_{O_2}}{m_m}\right)}{\left(p_1 . r . \frac{T}{T_1} . \frac{m_{CO_2}}{m_m}\right)^2} = K_1$$

or $$\frac{(m_{CO})^2}{(m_{CO_2})^2} . p_1 . r . \frac{T}{T_1} . \frac{(m_{O_2})}{m_m} = K_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.26)$$

In above equation m_{CO}, m_{CO_2}, m_{O_2}, are the number of lb.-mols of CO, CO_2 and O_2 in the products of combustion at the assumed temperature "T", and "p_1" the inlet pressure in atmospheres. Let x, y, z, 2p, 2s and t be the number of lb.-mols of CO_2, H_2O, OH, H, O, N and NO in the products of combustion and a, b, c, and d represent the number of lb.-mols CO_2, O_2, H_2 and N_2 if no dissociation were to occur, then the products of the combustion gas at point 3 in figure 3 will be:

$$xCO_2 + yH_2O + zOH + 2pH + 2qO + 2sN + tNO + (a-x)CO +$$

$$(b - \frac{x+y+z+t+2q}{2})O_2 + (c - (y+\frac{z}{2}+p))H_2 + (d-\frac{t+2s}{2})N_2 \quad \ldots (3.27)$$

and "m_p" then becomes $(a+b+c+d-\frac{x}{2}-\frac{y}{2}+p+q+s)$ lb.-mols.

The equilibrium equations (3.26) can now be written as:

$$(\frac{a-x}{x})^2 \times (b - \frac{x+y+z+t+2q}{2}) \times (p_1 \times r \times \frac{T}{T_1} \times \frac{1}{m_m}) = K_1 \quad .. (3.28)$$

The product $(p_1 \times r \times \frac{T}{T_1} \times \frac{1}{m_m})$ is a constant, as the values

p_1, r, T and m_m are given, and should be called "C_1".

The equilibrium equations can then be written in their final form:

$$(\frac{a-x}{x})^2 . (b - \frac{x+y+z+t+2q}{2}) = \frac{K_1}{C_1} = A \quad \ldots\ldots\ldots\ldots (3.29)$$

$$(\frac{a-x}{x}) . \frac{y}{c - (y+\frac{z}{2}+p)} = K_2 \quad \ldots\ldots\ldots\ldots (3.30)$$

$$\frac{z^2}{y^2} . (c - (y+\frac{z}{2}+p)) = \frac{K_3}{C_1} = B \quad \ldots\ldots\ldots\ldots (3.31)$$

$$\frac{4p^2}{c - (y+\frac{z}{2}+p)} = \frac{K_4}{C_1} = C \quad \ldots\ldots\ldots\ldots (3.32)$$

$$\frac{4q^2}{b - \frac{x+y+z+t+2q}{2}} = \frac{K_5}{C_1} = D \quad \ldots\ldots\ldots\ldots (3.33)$$

$$\frac{4s^2}{d-\frac{t}{2}-s} = \frac{K_6}{C_1} = E \quad \ldots\ldots\ldots\ldots (3.34)$$

$$\frac{t^2}{\left(b - \frac{x+y+z+t+2q}{2}\right).\left(d-\frac{t}{2}-s\right)} = K_7 \quad \text{.................} \quad (3.35)$$

To solve equations 3.29 - 3.35 by the author's method, it is necessary first to estimate the number of lb.-mols of CO_2 in the products of combustion represented as "x" in the above formulae. Assume for the first calculation the number of lb.-mols of CO_2 and insert this value, called "x_1", in the seven equilibrium equations. They may now be written as:

$$\left(b - \frac{x+y+z+t+2q}{2}\right) = A.\left(\frac{x_1}{a-x_1}\right)^2 = F \quad \text{...............} \quad (3.36)$$

$$\frac{y}{c - (y+\frac{z}{2}+p)} = K_2.\left(\frac{x_1}{a-x_1}\right) = G \quad \text{.............} \quad (3.37)$$

$$\text{therefore:} \quad c - (y+\frac{z}{2}+p) = \frac{y}{G} \quad \text{..................} \quad (3.38)$$

$$\text{but} \quad c - (y+\frac{z}{2}+p) = B.\frac{y^2}{z^2} \text{ (equ. 3.31)} \quad \text{...........} \quad (3.39)$$

$$\text{and} \quad \frac{y}{G} = B.\frac{y^2}{z^2} \qquad \therefore\ y = \frac{z^2}{B.G} \quad \text{................} \quad (3.40)$$

$$4p^2 = C.(c - (y+\frac{z}{2}+p)) = C.\frac{y}{G} = C.\frac{z^2}{B.G^2} \quad \text{...........} \quad (3.41)$$

$$2p = \frac{z}{G}\sqrt{\frac{C}{B}} \quad \text{..............................} \quad (3.42)$$

Equation (3.38) can now be expressed in terms of "z" as:

$$c - \frac{z^2}{B.G} - \frac{z}{2} - \frac{z}{2G}\sqrt{\frac{C}{B}} = \frac{y}{G} = \frac{z^2}{B.G^2}$$

$$\text{or } z^2\left(\frac{1}{B.G^2} + \frac{1}{B.G}\right) + \frac{z}{2}\left(1 + \frac{1}{G}\sqrt{\frac{C}{B}}\right) - c = 0$$

$$z^2\left(\frac{1+G}{B.G^2}\right) + \frac{z}{2}\left(1 + \frac{1}{G}\sqrt{\frac{C}{B}}\right) - c = 0$$

$$z^2 + \frac{z}{2}\left(\frac{B.G}{1+G}\right).\left(G + \sqrt{\frac{C}{B}}\right) - c.G.\left(\frac{B.G}{1+G}\right) = 0$$

$$\text{if } \frac{B.G}{1+G} = H \qquad (3.43)$$

$$\text{then: } z^2 + \frac{z}{2}.H.\left(G + \sqrt{\frac{C}{B}}\right) - c.G.H = 0 \qquad (3.44)$$

$$\text{and } z_{1,2} = \frac{-H.\left(G + \sqrt{\frac{C}{B}}\right)}{4} \pm \sqrt{\frac{H^2\left(G + \sqrt{\frac{C}{B}}\right)^2}{16} + c.G.H} \qquad (3.45)$$

But $4q^2 = D.F$, see equation (3.33) and (3.36),

$$\text{therefore } 2q = \sqrt{D.F} \qquad (3.46)$$

Using the positive root-value only, "y" and "p" can now be found by inserting "z" in equations (3.40) and (3.42).

From equation (3.36) "t" can then be found:

$$b - \frac{x_1+y+z+t+2q}{2} = F \qquad (3.36)$$

$$\text{and } t = 2b-2F-x_1-y-z-2q \qquad (3.47)$$

After finding the value of "t", "s" can be calculated, using equation (3.34)

$$4s^2 = E.d - \frac{E}{2}.t - E.s = 0$$

$$s_{1,2} = \qquad (3.48)$$

and finally, $K_7 = \dfrac{t^2}{F.(d-\frac{t}{2}-s)}$ .... (3.49)

In most cases, the value of "s" may be neglected being too small to show any real heat loss due to dissociation of "N_2". Finally, by inserting the values "t" and "s" into equation (3.49), the value of K_7 may be found. If it does not correspond with the original value of the selected temperature "T", a new value "x_2" has to be selected and the calculations repeated.
Using a tabular method, one quickly approaches the value of "K_7" and the corresponding values x,y,z,t,2q and s should satisfy the seven equilibrium equations.

Using equation (3.4), the combustion equation for the initial burning can now be written as:

$0.0153C_8H_{18} + 0.1913\ O_2 + 0.7550N_2 + 0.0243H_2O_{liqu.} +$
$0.0061CO_2 + 0.0080H_2O_{vap.} = 0.1224CO + 0.1377H_2 + 0.1301\ O_2 +$
$0.7550N_2 + 0.0243H_2O_{vap.} + 0.0061CO_2 + 0.0080H_2O_{vap.} =$
$0.1224CO + 0.1377H_2 + 0.1301\ O_2 + 0.7550N_2 +$
$0.0061CO_2 + 0.0323H_2O_{vap.}$ .... (3.50)

If, during the subsequent combustion, a total number of "x" lb.-mols of CO_2 (including the number of lb.-mols of CO_2 of the residual gases) are to be formed, then only (x - 0.0061)lb.-mols of CO will have to burn to CO_2 and, therefore, (0.1285 - x)lb.-mols of CO will remain.

Similarly, for "y" lb.-mols of $H_2O_{vap.}$, which includes the steam remaining from the previous cycle, (y - 0.0323)lb.-mols of H_2 will have to burn to $H_2O_{vap.}$, leaving (0.1700 - y)lb.-mols H_2 free. This number of lb.-mols will have to be reduced further by $(\frac{z}{2} + p)$lb.-mols according to equation (3.27).

Finally, the stock of O_2 will be diminished by the amount of oxygen necessary to burn the calculated amount of CO and H_2.

Therefore, $(0.1301 - \frac{x-0.0061}{2} - \frac{y-0.0323}{2})$lb.-mols O_2 less $(\frac{z+2q+t}{2})$lb.-mols from equation (3.27) will remain in the products of combustion.

Equation (3.27) finally takes the form:

$$xCO_2 + yH_2O + zOH + 2pH + 2qO + 2sN + tNO +$$
$$(0.1285 - x)CO + (0.1493 - \frac{x+y+z+2q+t}{2})O_2 +$$
$$(0.1700 - (y+\frac{z}{2} +p))H_2 + (0.7550 - \frac{2s+t}{2})N_2 \text{ ....} \quad (3.51)$$

Therefore, a= 0.1285, b= 0.1493, c= 0.1700 and d= 0.7550.

<u>Heat released during reaction.</u>

To find the curve of heat released, plotted against temperature, a number of temperatures in the neighbourhood of the maximum cycle temperature have to be selected and the corresponding dissociation quantities found by using the tabulation method.

To solve the problem under consideration, temperatures of 4,900°R and 5,000°R were used in the following calculation.

Values of "K"

T	4,900°R	5,000°R
K_1	1.259×10^{-2}	1.995×10^{-2}
K_2	6.457	6.546
K_3	1.072×10^{-3}	1.738×10^{-3}
K_4	3.981×10^{-3}	5.888×10^{-3}
K_5	1.738×10^{-3}	2.692×10^{-3}
K_6	1.380×10^{-7}	2.089×10^{-7}
K_7	7.413×10^{-3}	8.710×10^{-3}

Table No.10

The next step is to find "C_1" and the constants "A - E" for the selected temperatures, (Equ. 3.29 - 3.34).

$$C_1 = p_1 \cdot r \cdot \frac{T}{T_1} \cdot \frac{1}{m_m} = 1.7 \cdot \frac{4900}{630} \cdot \frac{1}{1} = \underline{54.49}$$

T	4,900°R	5,000°R
C_1	54.49	55.55
A	2.31×10^{-4}	3.59×10^{-4}
B	1.97×10^{-5}	3.125×10^{-5}
C	7.32×10^{-5}	1.06×10^{-4}
D	3.195×10^{-5}	4.84×10^{-5}
E	2.535×10^{-9}	3.76×10^{-9}

Table No.11

Values of "x_1" have now to be assumed and the corresponding value of "K_7" can be found, as indicated in table No. 12.

Table used in finding values of K_7.

	4,900°R		5,000°R	
	1st Trial	2nd Trial	1st Trial	2nd Trial
x_1	0.11	0.1085	0.1050	0.1055
$a-x_1$	0.185	0.020	0.0235	0.0230
$\frac{x_1}{a-x_1}$	5.95	5.425	4.64	4.58
$(\frac{x_1}{a-x_1})^2$	35.4	29.5	19.9	21
$F=A(\frac{x_1}{a-x_1})^2$	$\frac{7.95}{10^5}$	$\frac{6.82}{10^5}$	$\frac{7.15}{10^5}$	$\frac{7.55}{10^5}$
$G=K_2(\frac{x_1}{a-x_1})$	38.4	35	29.2	30
$H=\frac{B.G}{1+G}$	$\frac{1.92}{10^5}$	$\frac{1.915}{10^5}$	$\frac{3.025}{10^5}$	$\frac{3.02}{10^5}$
z as per equ.(3.45)	0.011	0.0105	0.01205	0.01218
$y=\frac{z^2}{B.G}$	0.16	0.1595	0.1575	0.158
$p=\frac{z}{2.G}\sqrt{\frac{C}{B}}$	Not cal.	$\frac{5.77}{10^4}$	Not cal.	$\frac{7.47}{10^4}$
$2q=\sqrt{F.D.}$	$\frac{5.04}{10^4}$	$\frac{4.67}{10^4}$	$\frac{5.88}{10^4}$	$\frac{6.05}{10^4}$
$t=2b-2F-x-y-z-2q$	0.0012	0.006	0.0092	0.0072
s= as per equ.(3.48)	Not cal.	$\frac{2.1}{10^5}$	Not cal.	$\frac{2.6}{10^5}$
$K_7=\frac{t^2}{F\times(d-\frac{t}{2}-s)}$	$\frac{2.43}{10^4}$	$\frac{7.02}{10^3}$ Value acc.	$\frac{1.59}{10^2}$	$\frac{9.15}{10^3}$ Value acc.

Table No. 12

Note: The correct values of "K_7" at 4,900 and 5,000°R are shown in Table No. 10.

After calculating the correct dissociation quantities, the internal energy and the heat of reaction have been found and are tabulated in Tables 13 and 14.

Internal energy of products of combustion.

T	4,900°R			5,000°R		
	Int.ener. B.T.U./ lb-mol	Number of mols	Int.ener. B.T.U.	Int.ener. B.T.U./ lb-mol	Number of mols	Int.ener. B.T.U.
CO_2	50,069	0.1085	5,435	51,365	0.1055	5,420
H_2O	38,791	0.1595	6,200	39,885	0.1580	6,300
OH	25,643	0.0105	269	26,319	0.01218	320
H	13,048	0.00058	7	13,346	0.00075	10
O	"	0.00047	6	"	0.00061	9
N	"	0.00004	-	"	0.00005	-
No	27,874	0.006	167	28,570	0.0072	205
CO	27,226	0.020	544	27,912	0.023	642
O_2	28,874	0.0068	196	29,616	0.0076	225
H_2	25,418	0.0049	125	25,819	0.0055	142
N_2	26,905	0.752	20,200	27,589	0.7514	20,700

$E_3 = \underline{\underline{33,149}}$ $\qquad$ $E_3 = \underline{\underline{33,973}}$

Table No. 13

Heat of reaction

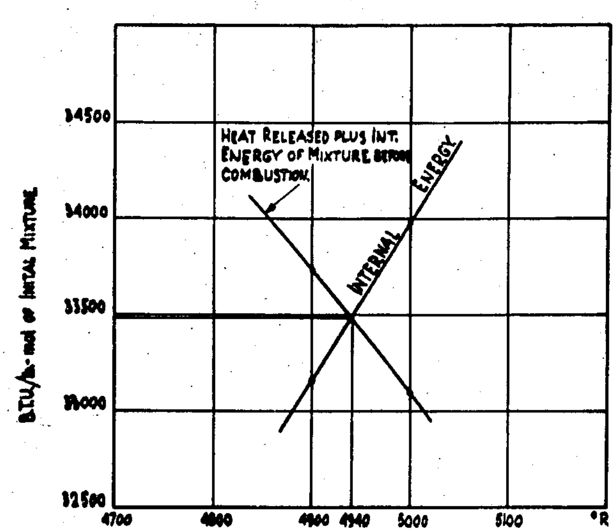
			T = 4,900°R			T = 5,000°R		
No.	Reactions	Chem.Ener. B.T.U./lb.mol	Mols	Pos. values	Neg. values	Mols	Pos. values	Neg. values
1	$C_8H_{18}+12.5O_2=8CO_2+9H_2O$	2,201,618	0.0153	33,700		0.0153	33,700	
2	$CO_2 \rightarrow CO+\frac{1}{2}O_2$	121,181	0.02		2,424	0.023		2780
3	$H_2 \rightarrow OH+\frac{1}{2}H_2$	115,500	0.0105		1,210	0.01218		1405
4	$\frac{1}{2}H_2 \rightarrow H$	92,882	0.00058		54	0.00075		70
5	$\frac{1}{2}O_2 \rightarrow O$	105,833	0.00047		50	0.00061		64
6	$\frac{1}{2}N_2 \rightarrow N$	157,500	0.00004		6	0.00005		8
7	$NO \leftarrow \frac{1}{2}N_2+\frac{1}{2}O_2$	38,746	0.006		230	0.0072		278
				33,700	3,974		33,700	4,605
				- 3,974			- 4,605	
				H_3=29,726			H_3=29,095	

Table No.14

On page 27 the internal energy "E_2" was found to be 4,000 B.T.U. If this value is added to the values of "H_3" at 4,900°R and 5,000°R, two points of the graph for "Heat released" plus internal energy "E_2" are established, and a straight line can be drawn through them.

A more accurate way, however, to find the true shape of the graph is to calculate additional values.

As the internal energy is practically a linear function of the temperature for small temperature differences, two points are sufficient for plotting it, (Fig. 4).



INTERNAL ENERGY AND HEAT RELEASED DURING REACTION VERSUS TEMPERATURE FOR THE PRODUCT OF COMBUSTION.
FIG. No 4

In the following table, the maximum combustion temperatures "T_3" are tabulated. They have been found for the cycle under consideration, when using (i) no internal coolant and (ii) water as internal coolant, with weight ratios of 0.35 and 0.25 respectively.

Maximum combustion temperatures

Weight ratio of water	Max. combustion temperature $^{\circ}R$	Authority used:
0	5040	Hottel charts
0.25	4940	Tabulation method (Weiss)
0.35	4900	Wiebe's diagrams

Table No.15

The above table indicates that as the weight ratio of water increases from 0 to 0.35, the maximum cycle temperature falls by 140°F.

This drop of temperature, in addition to the facts explained previously, seems to be sufficient to bring the temperature of the end-gas just below the critical value, with the result that detonation disappears.

In conclusion, it can be said that the tabulation method can be used for any fuel with or without internal coolants, and will be of value in solving thermodynamic problems when the relevant charts are not available.

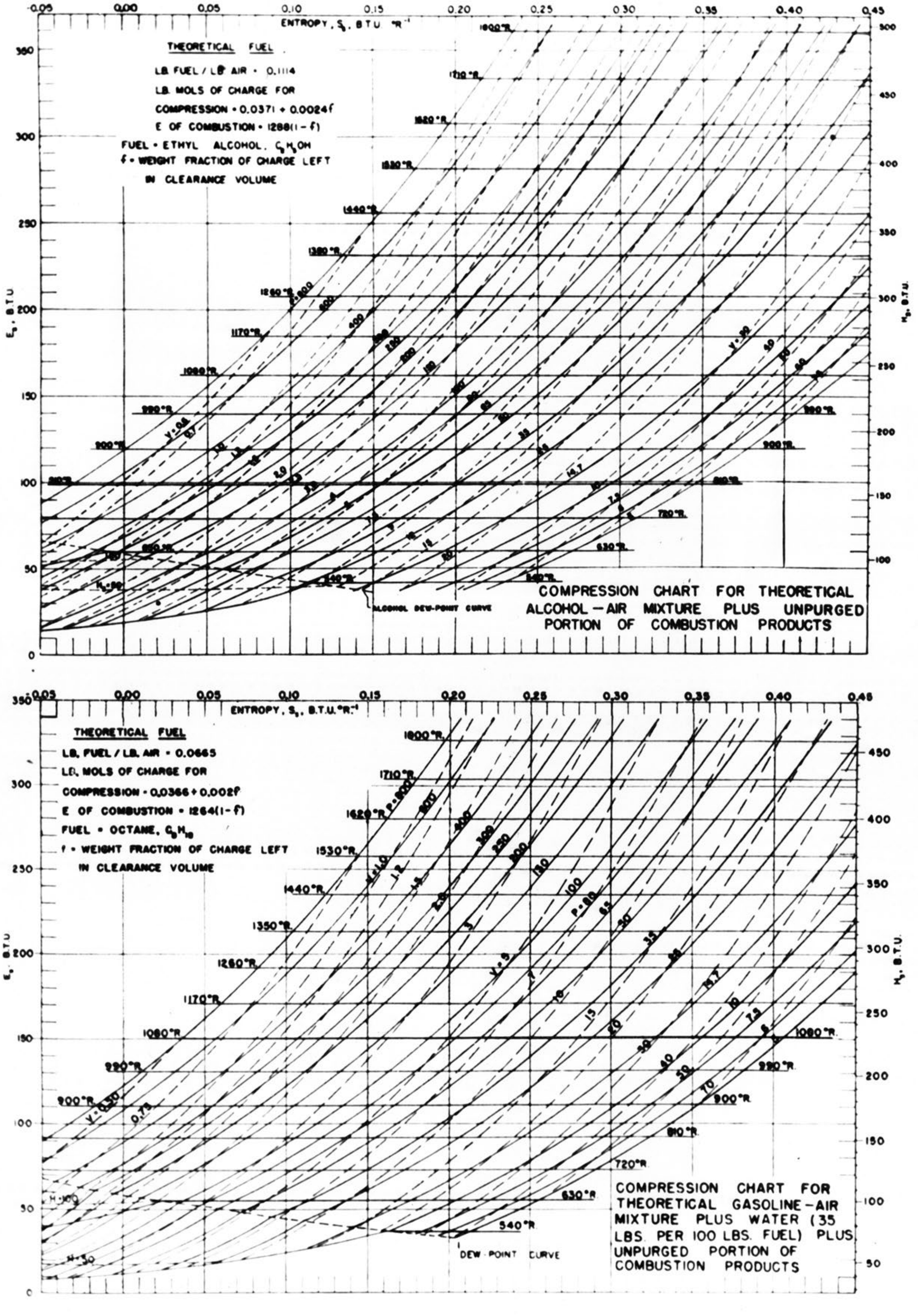
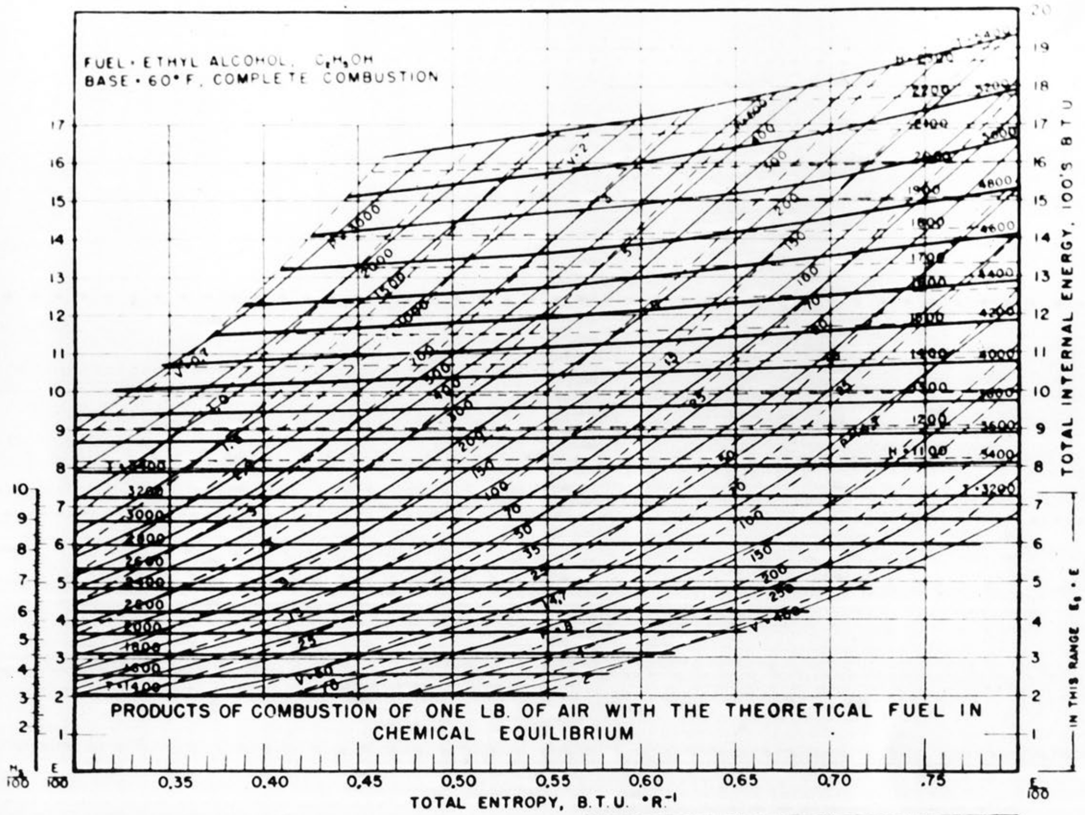

ENTROPY, S_s, B.T.U. °R.$^{-1}$
THEORETICAL FUEL
LB. FUEL / LB. AIR = 0.1114
LB. MOLS OF CHARGE FOR
COMPRESSION = 0.0371 + 0.0024f
E OF COMBUSTION = 1288(1 − f)
FUEL = ETHYL ALCOHOL, C_2H_5OH
f = WEIGHT FRACTION OF CHARGE LEFT
IN CLEARANCE VOLUME
E_s, B.T.U.
H_s, B.T.U.
ALCOHOL DEW-POINT CURVE
COMPRESSION CHART FOR THEORETICAL
ALCOHOL – AIR MIXTURE PLUS UNPURGED
PORTION OF COMBUSTION PRODUCTS
ENTROPY, S_s, B.T.U. °R.$^{-1}$
THEORETICAL FUEL
LB. FUEL / LB. AIR = 0.0665
LB. MOLS OF CHARGE FOR
COMPRESSION = 0.0366 + 0.002f
E OF COMBUSTION = 1264(1 − f)
FUEL = OCTANE, C_8H_{18}
f = WEIGHT FRACTION OF CHARGE LEFT
IN CLEARANCE VOLUME
E_s, B.T.U.
H_s, B.T.U.
DEW POINT CURVE
COMPRESSION CHART FOR
THEORETICAL GASOLINE-AIR
MIXTURE PLUS WATER (35
LBS. PER 100 LBS. FUEL) PLUS
UNPURGED PORTION OF
COMBUSTION PRODUCTS

Fig. 5



T - TEMPERATURE, °R.
P - PRESSURE, LBS/IN^2 (Solid Lines)
V - VOLUME, CU. FT. (Dashed Lines)
E - TOTAL INTERNAL ENERGY, B.T.U.
E_s - INTERNAL ENERGY, EXCLUSIVE OF INT. ENERGY OF COMBUSTION
H - TOTAL ENTHALPY, E + PV(144/778), B.T.U.
H_s - E_s + PV(144/778), B.T.U.

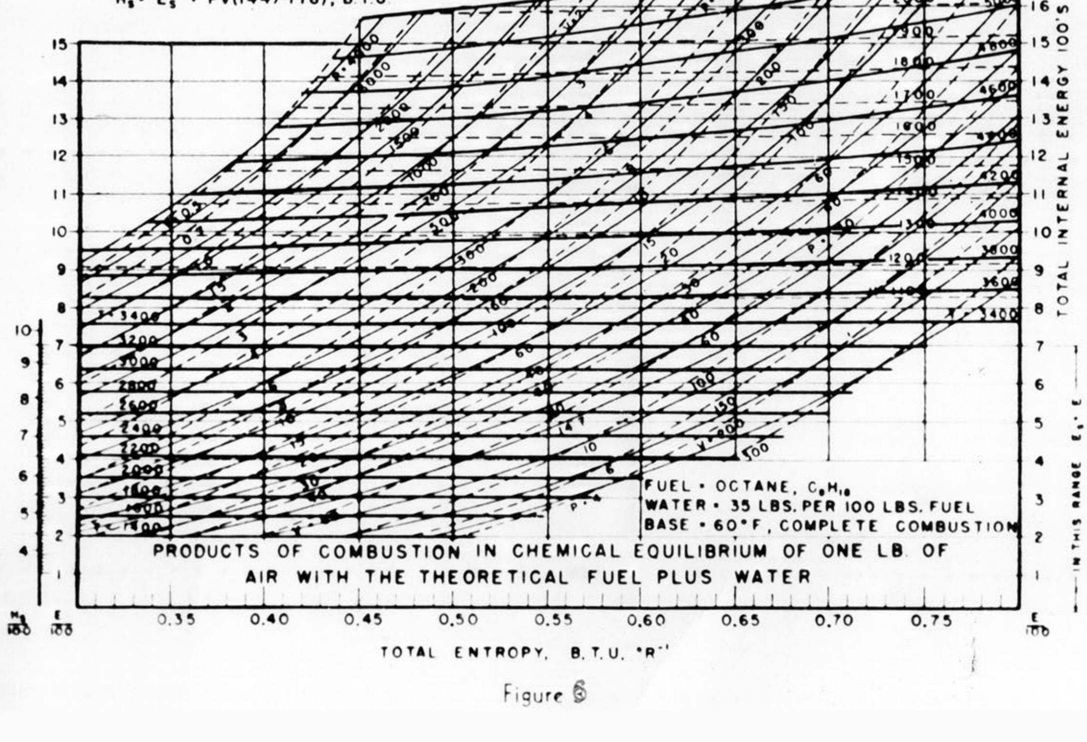


Figure 6

CHAPTER 4

Laboratory Tests

4 - 1. Aim of investigation

The experimental part of the thesis was carried out with the aim of investigating the effect of internal coolants on the performance and economy of an internal combustion engine.

At the same time, an investigation was to be made of variables such as spark advance, inlet air and exhaust gas temperatures. Furthermore, the interior condition of the engine was to be determined before and after the test period by removing the cylinder head and making photos of the liner and piston crown. Variations in carbon and tar deposits could thus easily be detected.

Laboratory tests were to be performed with the help of a Ricardo E-6/S variable compression engine and a manually controlled valve, which, by its design, should have enabled the author to make considerable variations in the amount of coolant injected.

It had also been planned to replace the control valve by an automatic injection equipment, marketed for cars and to compare the results. Unfortunately, it was found that an automatic device would not work correctly with a single-cylinder engine, such as the Ricardo. Therefore, this part of the test had to be abandoned.

4 - 2. Test equipment and instrumentation

The equipment used consisted of the Ricardo E-6/S variable compression engine, complete with electrical brake and the necessary instruments for measuring fuel and air consumption. A new panel was added adjacent to the engine, carrying the water metering equipment, the manually operated coolant valve, a coolant tank, exhause manifold vacuum gauge and a pyrometer for reading exhaust gas temperatures.

Description of engine

The engine is a single cylinder, poppet-valve, four-stroke, water-cooled, vertical engine, the specification of which is given in Table 16. The engine is very rigidly constructed and is able to withstand firing pressures up to 1,8000 lb./in^2 without any damage to its parts.

The cast-iron cylinder is provided with a dry liner made from high-phosphorous cast iron, the latter being hardened and ground. The outer surface of the cylinder jacket is threaded and is in contact with the internal thread of a worm-wheel, the rotation of which, by means of a worm, raises or lowers the cylinder relative to the centre-line of the crankshaft, (Fig. 7).

The compression ratio may be changed while the engine is running. The position of the cylinder is determined by means of a micrometer, (Fig. 8). For a particular compression ratio the micrometer setting may be obtained from a calibration curve provided by the makers of the engine.

The combustion chamber, when using the E-6S as a petrol engine, is cylindrical in shape, and is bounded by the flat surfaces of piston and cylinder head. Such a combustion chamber is very compact and retains the cylindrical form at the various compression ratios. The spark plug, inclined at 45 degrees, is situated at the side of the combustion chamber between the two valves. Provision is made to mount the spark plug at the opposite side of the combustion chamber if desired. This alternate hole for the spark plug is used to house the pressure pick-up for the "Farnboro" indicator or an adapter, which is made to accommodate the capacity type pressure cell used for the Southern Instruments electronic indicator.

The magneto is driven from the camshaft by a vernier coupling. The timing of the ignition is controlled by a lever on the magneto the position of which indicates the spark advance, in crankshaft degrees, on a calibrated segment.

The carburettor has a taper needle valve, which is used to regulate the quantity of fuel passing through the main jet and thus controls the mixture strength within wide limits. A vertical plate, divided into 72 divisions, has been attached to the carburettor housing and a small arm, fastened to the needle valve spindle, rides on this plate. This device makes it possible to reproduce needle valve settings accurately and to supply an arbitrary number to each position, (Fig. 9).

Engine Specification

Ricardo E - 6/S variable compression engine, Serial No. 31/50	
Number of cylinders	1
Bore	3 in.
Stroke	$4\frac{3}{8}$ in.
Swept volume	506 cc.
Speed range	1000 - 3000 r.p.m.
Compression ratio	4.5 to 22.1:1
Tappet clearance - inlet	0.006 in.
" " - exhaust ..	0.008 in.
Inlet valve opens	8 degrees b.t.d.c.
" " closes	36 degrees a.b.d.c.
Exhaust valve opens	43 degrees b.b.d.c.
" " closes	6 degrees a.t.d.c.
Carburettor	Solex down draft, type 35 F.A.1 with 27 mm. diam. choke and variable main jst.
Spark plug	KLG RC S/4 or R.53

Table No.16

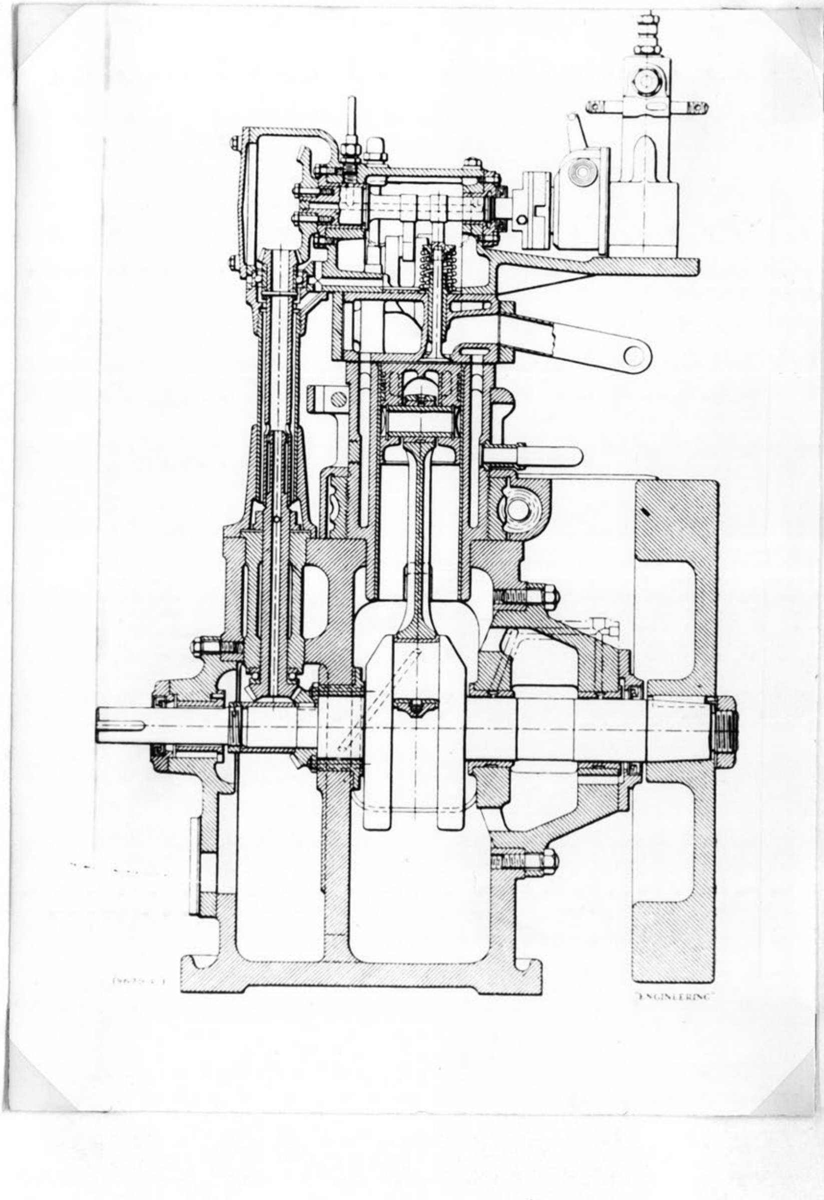


Fig. 7 - Longitudinal-section through Ricardo E-6/S variable compression engine.

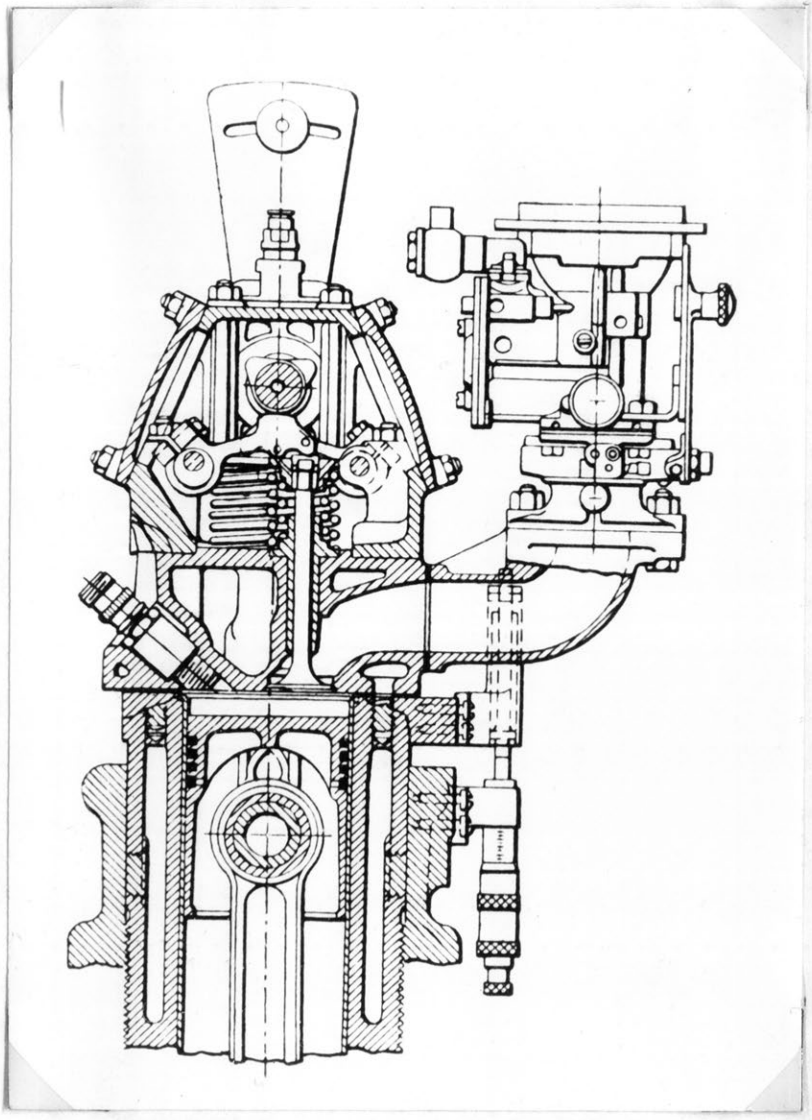


Fig. 8 - Cross-section through Ricardo E-6/S variable compression engine.

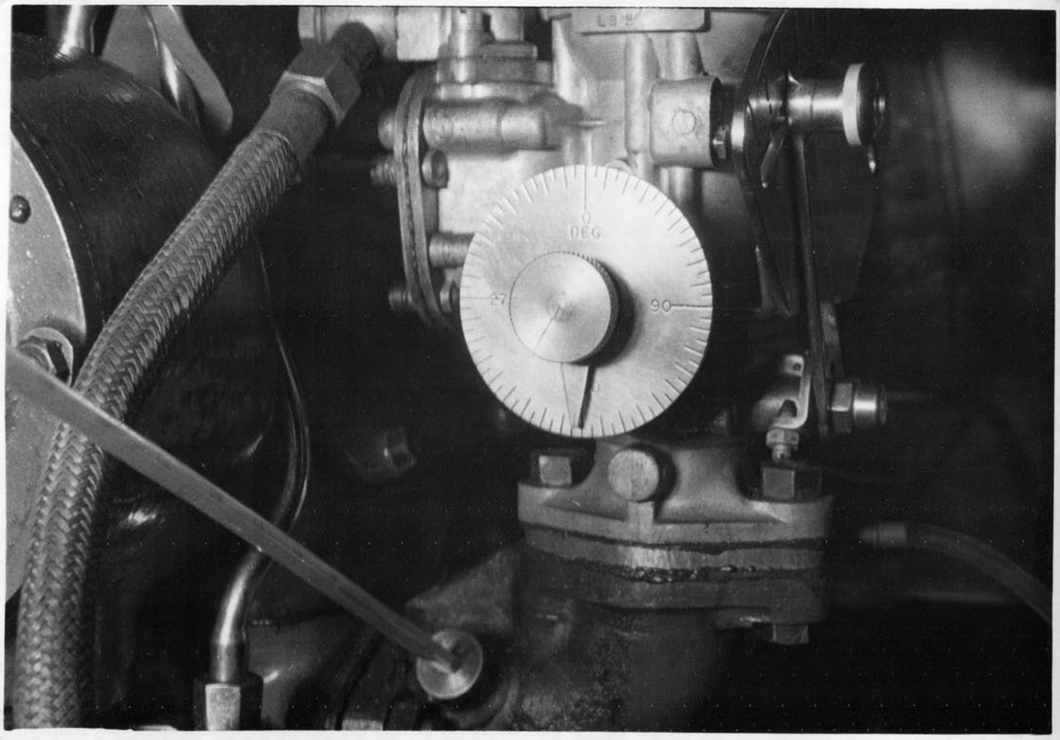


Fig. 9 - Carburettor showing metering device,
inlet manifold thermometer and
plate for water-injection.

The water-injection unit

The water-injection unit consists, in principle, of a water tank and a float-chamber mounted on a stand, (Fig. 10), and two measuring pipettes and a coolant control valve attached to the new panel.

The float-chamber position may be altered to produce any required head and its outlet is connected to the control valve by a plastic tube, (Fig. 11).

This valve, manufactured by George Kent, England, uses a diaphragm instead of a gland to prevent air entering the valve body. The opening of the valve is controlled by the position of a micrometer spindle, which operates a needle valve via the diaphragm.

For high engine speeds it was found that the original needle in the described valve did not permit the required flow of water and a new needle, with an included angle of 7°, had to be made. After cal brating the valve, predertermined water quantities could be discharged and reproduced quite easily.

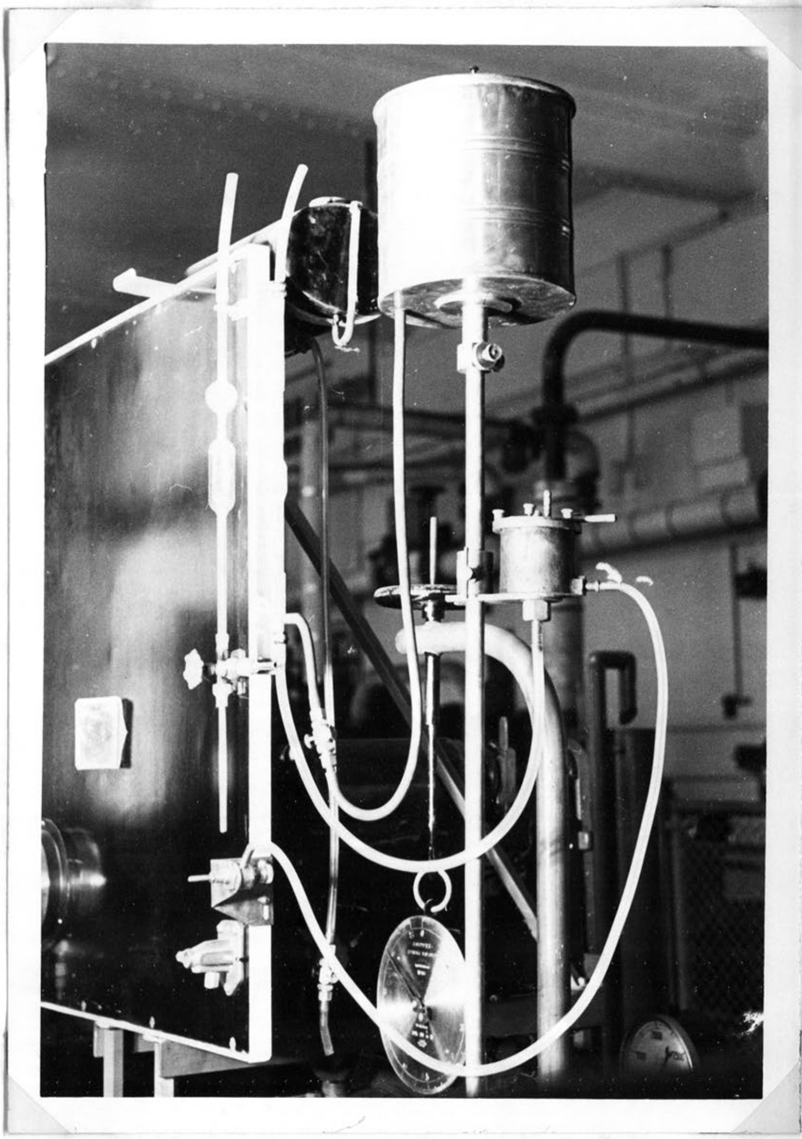


Fig. 10 - Stand with water tank
and float-chamber.

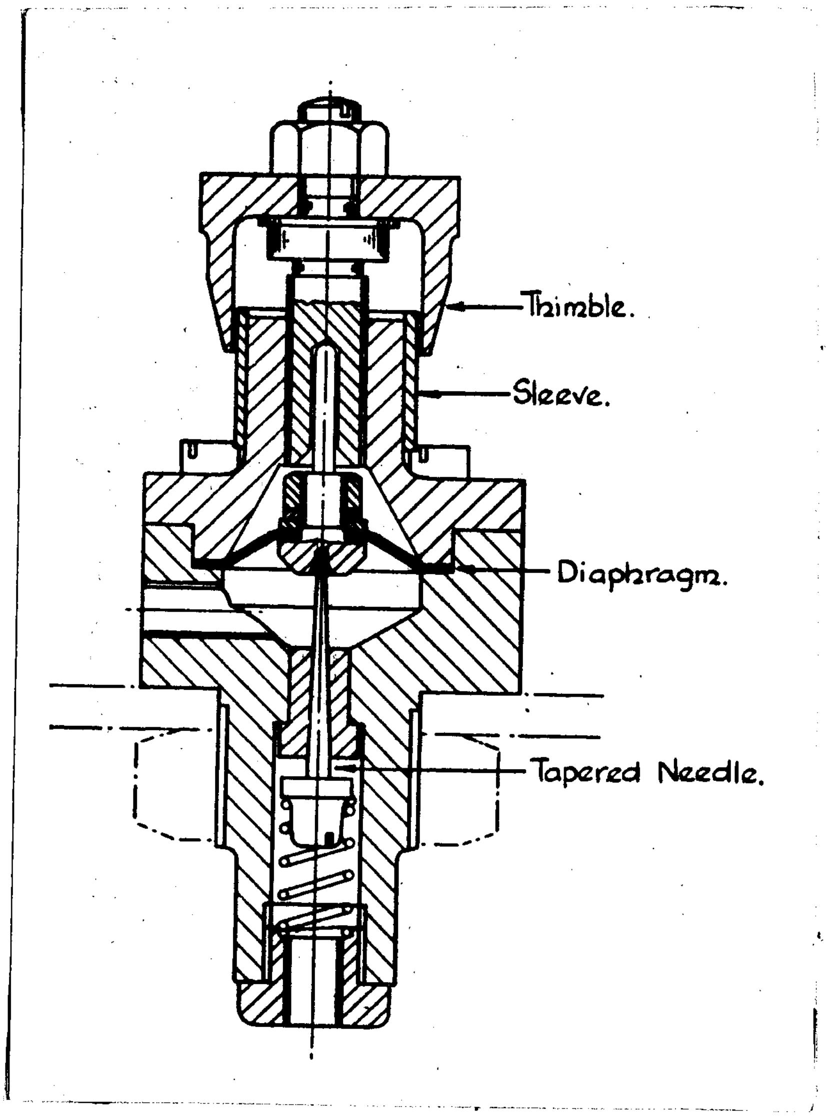


Fig. 11 - Control valve, manufactured by George Kent, England.

From the coolant tank, a hose leads to a T-piece, and water can rise to tank level in the pipette simultaneously filling the float-chamber. An "on-off" valve, situated between tank and T-piece is used to stop the water supply from the tank when consumption tests are carried out. By introducing the float-chamber, a constant head of liquid is guaranteed. From the needle valve outlet water runs to a special injector plate, situated between the inlet manifold and carburettor flange. A plastic hose is used for the connection and may be arranged in the form of a loop to form a sludge trap, (Fig. 12).

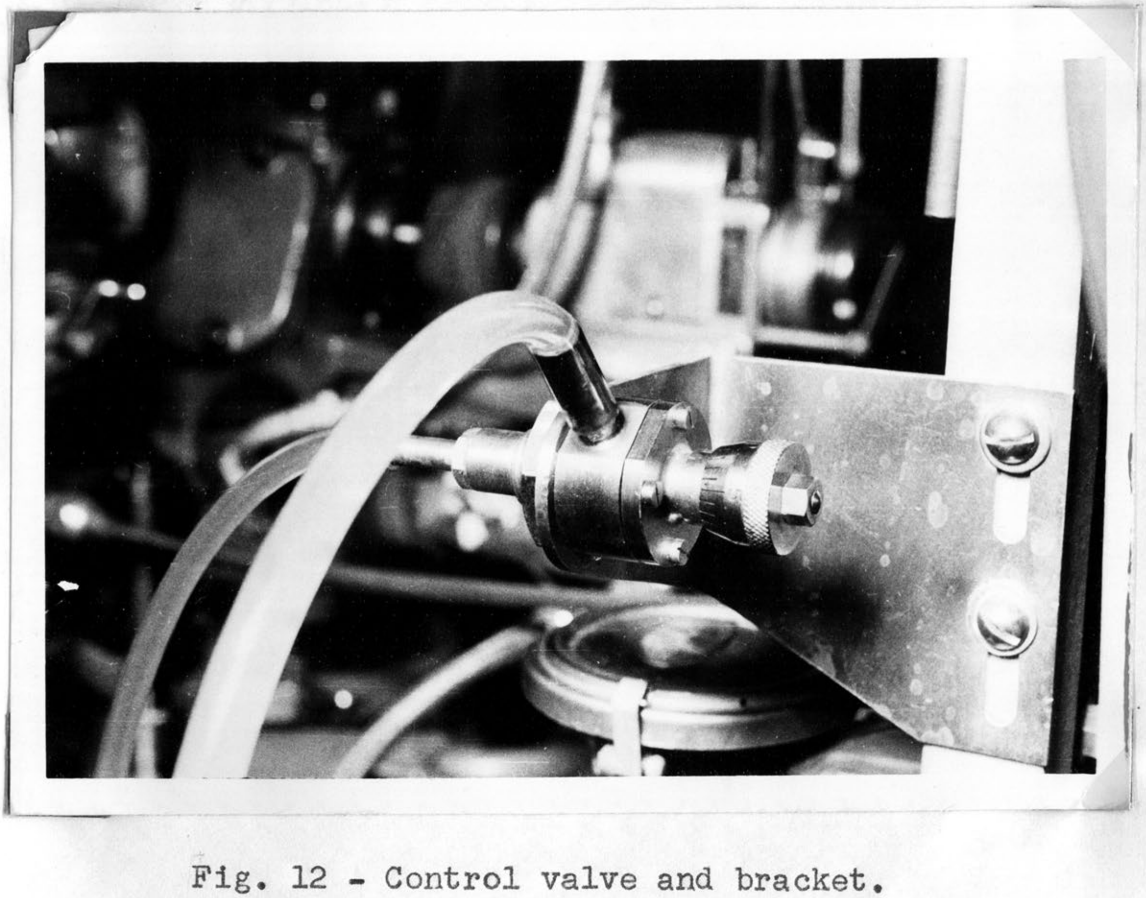


Fig. 12 - Control valve and bracket.

Panel for water-injection equipment

A special panel (Fig. 13) has been made, to carry equipment for the water-injection tests. The board was designed in such a way that it could easily be detached from the Ricardo engine and installed elsewhere.

On the panel, two pipettes are mounted, one to measure 8 ml., the other 25 or 50 ml., whichever is required. Furthermore, the manual control delivery valve, carried in an adjustable bracket, and an automatic water-injection device, manufactured by Kleinig Products Pty Ltd., are fitted to it, (Fig. 10).

To the same panel, a vacuum gauge and pyrometer are attached and room is left for a multi-cavity fuel-tank and a valve chest.

An oval tank, used to hold alcohol-water mixtures, is carried on the back of the board.

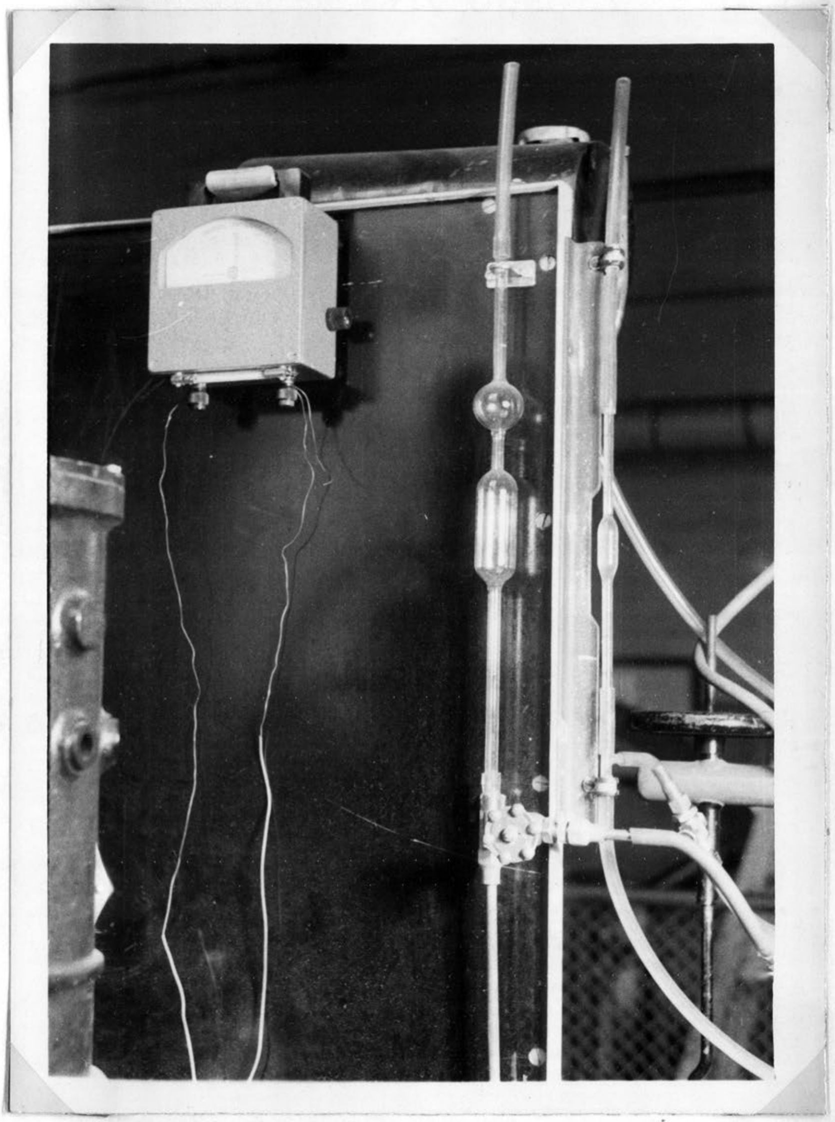


Fig. 13 - Panel for water-injection equipment.

Temperature observation

To be able to record inlet air and exhaust gas temperatures, a thermometer pocket was added to the inlet manifold and a thermocouple installed in the exhaust gas manifold, (Fig. 14). Chromel-alumel 8 gauge wires, recommended for temperatures up to 1000°C, were connected to an Industrial Pyrometer, fastened to the instrument panel. Thus, exhaust gas temperatures could easily be read.

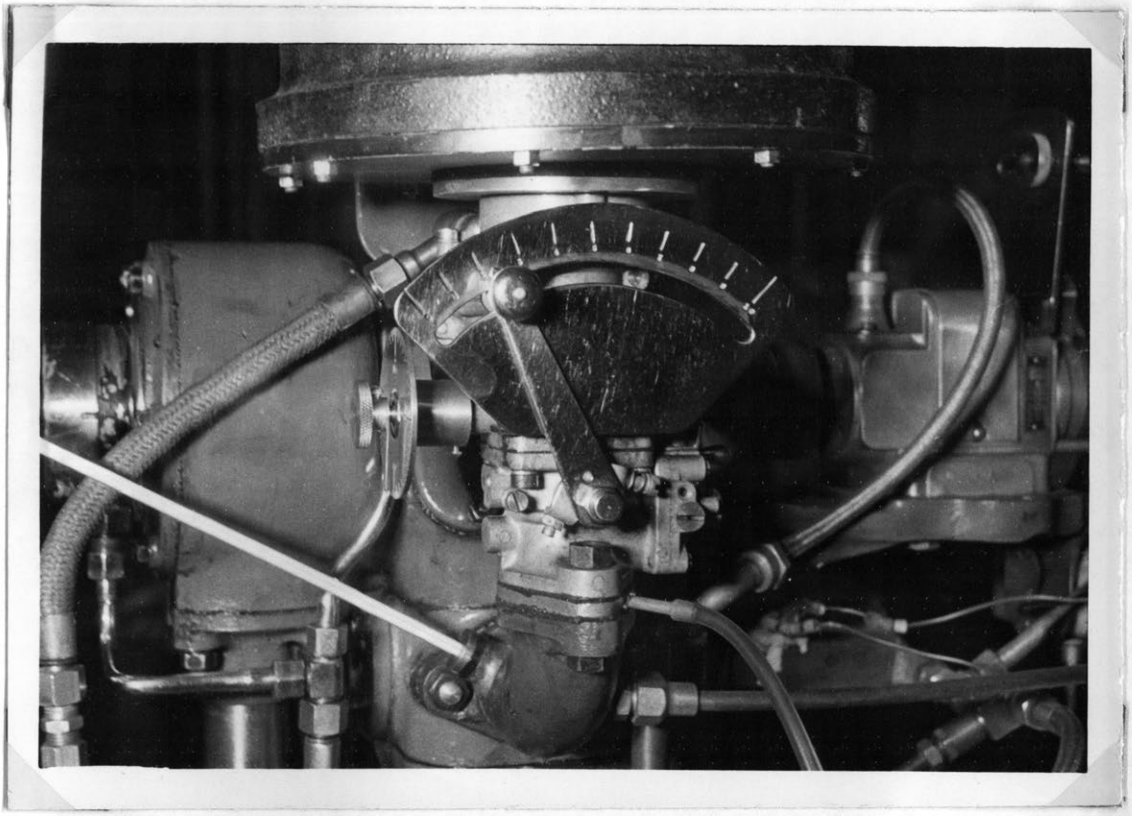


Fig. 14 - Carburettor, showing thermometer pocket and thermocouple connection.

<u>The engine indicator</u>

To be able to run all tests at trace knocking conditions, an electronic high-speed indicator was used. This indicator is a product of the Southern Instruments Ltd., Camberley, Surrey, and is catalogued as Minirack engine indicator ME 109, (Fig. 15).

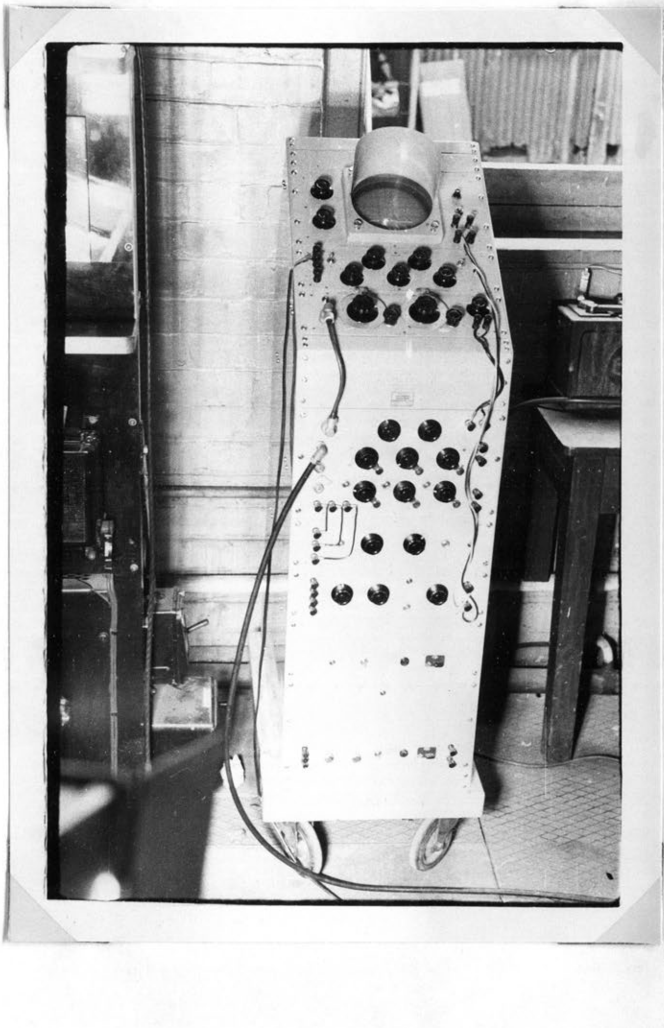


Fig. 15 - The Minirack ME 109 engine indicator.

The "ME 109" is a single-channel, disc-type, trolley-mounted unit and has a cathode-ray tube as a display element. Diagrams of cylinder pressure and pressure rates, can be obtained and measurements made with the help of condenser type pressure gauges.

These pick-ups are extremely sensitive devices but at the same time are able to withstand very severe overloads. They are of the retracted type, the body being of mild steel. The diaphragms used are made of hardened steel or mica, according to pressure range. The latter type has a thickness of about 0.001 of an inch. The bodies of the pick-ups are provided with water jackets and temperatures may be kept constant by circulating cooling water through them, (Fig. 16). The cathode-ray tube produces large diagrams that can be easily seen and accurately measured. Any part of the engine diagram may be expanded, examined and crank-angle degree marks superimposed.

Synchronisation of the time-base of the indicator with the engine speed is performed by means of a magnetic pick-up, supported by a special bracket and a steel ring, having a small knife-edged finger protruding from it, positioned on the started dog of the engine, (Fig. 17).

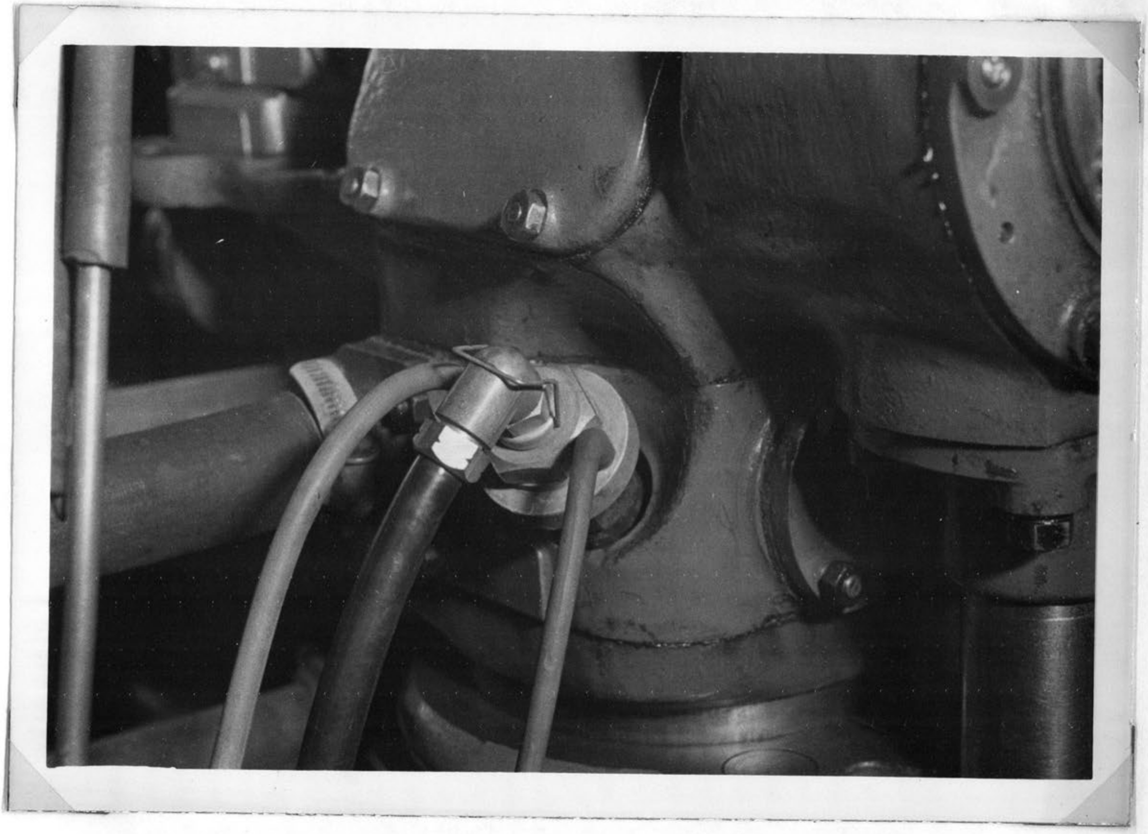


Fig. 16 - Pressure pick-up with water-jacket.

Fig. 17 - Magnetic pick-up for synchronisation
of time - base.

4 - 3. Sample calculations

Symbols :

N = Speed in r.p.m.

W = Observed net force on torque arm in lb.

W^1 = Corrected net force in lb.

B.H.P.$_{cor.}$ = Brake horsepower, corrected.

F_p = Petrol consumption in lb./hr.

F_A = Alcohol consumption in lb./hr.

F_s = Specific fuel consumption in lb./B.H.P.-hr.

Q = Air consumption in ft.^{3}/min.

Q^1 = Air consumption in lb/hr.

A/F = Air-fuel ratio.

η_o = Overall thermal efficiency.

L.C.V. = Lower calorific value in B.T.U./lb.

T_p = Time to consume 50 cc. of petrol in sec.

S.G.$_{cor.}$ = Specific gravity of fuel or alcohol, corrected to ambient temperature.

T_c = Time to consume 8 cc. of internal coolant.

p = Atmospheric pressure in lb./in^2.abs.

w = Weight of air at atmospheric conditions in lb./ft^3.

W_p = Weight of 50 cc. of petrol in lb.

W_C = Weight of 8 cc. of internal coolant in lb.

a = Ratio of internal coolant to fuel by weight.

q_c = Weight of coolant consumed per hour.

t_p = Temperature of petrol in ^oC.

Fuel used:

1) Shell Motor Spirit.

Specific gravity at 15°C	0.745
Octane rating (Research)	78 to 80
Additives	1.1 to 1.2 ml. lead per Imp. gallon.
Higher calorific value	19,640 BTU./lb.
Price	3s./8½d. per gal.

2) C.S.R. Special methylated spirit.

Specific gravity at 15°C	0.800
Higher calorific value	12,645 BTU./lb.
Approximate composition	90% ethyl alcohol plus 10% methyl alcohol.
Price	7s./1d. per gal.

Specific gravity figures and higher calorific values were determined by the School of Applied Chemistry of The New South Wales University of Technology. The above values compare favourably with the figures supplied by the makers.

1) Corrected net force

$W^1 = W + C$ lb.

Correction factor "C" makes allowance for torque errors due to the ventilating air flow in the generator.

Speed:	Values of "C"
1,000	0.18 lb.
1,250	0.22 "
1,500	0.26 "
1,750	0.28 "
2,000	0.32 "
2,250	0.35 "
2,500	0.37 "
2,750	0.43 "
3,000	0.50 "

2) Brake horse power

$$B.H.P. = \frac{W^1 \times N}{3500} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4 - 3.1)$$

3) Brake horsepower corrected.

$$B.H.P._{cor.} = B.H.P. \times \left(\frac{29.92}{\text{dry barom. read.}} \times \sqrt{\frac{\text{obs. } {}^{o}R}{520}}\right) =$$

$$B.H.P. \times C_1 \qquad \ldots\ldots\ldots\ldots\ldots \quad (4 - 3.2)$$

In the above formula, the dry barometer reading is, by definition, the atmospheric pressure less the pressure of water vapour, as obtained from wet and dry bulb temperatures.

4) Fuel consumption.

$$F_p = \frac{50x2.2}{1000} xS.G._{cor.} x\frac{3,600}{T_F} = 396x\frac{S.G_{cor.}}{T_F} \text{ lb./hr.} \quad \ldots \quad (4\text{-}3.3)$$

$$S.G._{cor.} = S.G_{15^oC} - (t_F - 15)x0.00085 \quad \ldots\ldots\ldots\ldots\ldots \quad (4\text{-}3.4)$$

5) Specific fuel consumption.

$$F_S = \frac{F}{B.H.P.} \text{-lb/B.H.P.-hr.} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4\text{-}3.5)$$

6) Air consumption.

To find the air consumption an "ALCOCK VISCOUS AIR METER" was used combined with a water-alcohol manometer.

$$Q = h \times f \times C_2 \text{ ft.}^3\text{/min. where} \quad \ldots\ldots\ldots\ldots \quad (4\text{-}3.6)$$

h = manometer reading in cm.

f = calibration factor, depending on the slope of the manometer.

C_2 = Correction factor for ambient temperature taken from Fig. 18.

f = 2.13 for high slope (No. 3)

f = 0.88 for medium slope (No. 2)

f = 0.38 for low slope (No. 1)

$$Q^1 = Q \times w \times 60 \text{ lb./hr. and} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4\text{-}3.7)$$

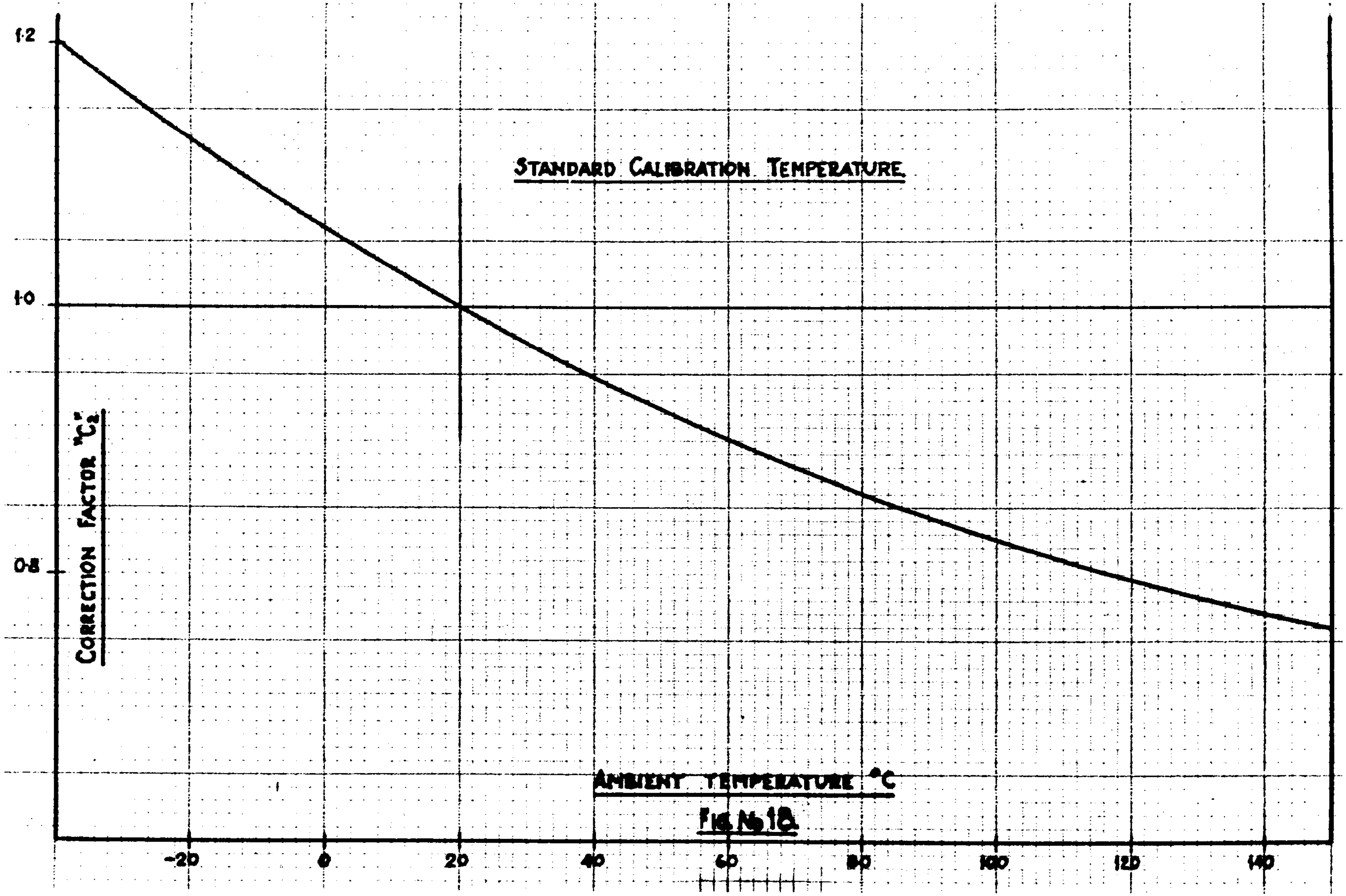
$$w = \frac{144xp}{RxT} \text{ lb./ft.}^3 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4\text{-}3.7a)$$

7) Air-fuel ratio

$$A/F = \frac{Q^1}{F_p} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4\text{-}3.8)$$

8) Overall thermal efficiency

$$\eta_0 = \frac{2545}{F_S xL.C.V.} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4\text{-}3.9)$$


1·2
1·0
0·8
CORRECTION FACTOR "C_2"
STANDARD CALIBRATION TEMPERATURE
AMBIENT TEMPERATURE °C
-20
0
20
40
60
80
100
120
140
FIG. No 18.

If water-alcohol as an internal coolant is used,

$$\eta o_1 = \frac{33,000 \times B.H.P. \times 60}{778(18,140.F_p+11,400.F_A)} = \frac{2.55 \times B.H.P.}{(18.14.F_p+11.4\ F_A)} \quad ..(4\text{-}3.10)$$

According B.S.S. 526/1933,

L.C.V. = H.C.V. - 1,055 x 1.42, for octane: ....(4-3.11)

Using a H.C.V. of 19,640 B.T.U./lb. for Shell Motor Spirit, then L.C.V. becomes 18,140 B.T.U./lb.

For C.S.R. denatured ethyl alcohol, L.C.V. = H.C.V. - 1,055 x 1.173.

Using a H.C.V. of 12,645 B.T.U./lb., the lower calorific value becomes 11,400 B.T.U./lb.

9) Time required to consume 8cc. internal coolant, for a given weight ratio.

$$W_c \times \frac{3600}{T_c} = a \times W_p \times \frac{3600}{T_p} \text{ or,}$$

$$T_c = \frac{W_c}{W_p} \times \frac{T_p}{a} \quad .... (4\text{-}3.12)$$

a) Weight of 8 cc. water.

$$W_c = \frac{8}{1000} \times 2.2 = \underline{0.0176} \text{ lb.}$$

b) Weight of 8 cc. water-alcohol (50 + 50) by volume.

Assuming an average alcohol temperature of 22°C, then for C.S.R. special methylated spirit, $S.G._{cor.} = \underline{0.794}$

and for the mixture (50 + 50), $S.G._{cor.} = \underline{0.897}$

Therefore, $W_c' = \frac{8 \times 2.2}{1000} \times 0.897 = \underline{0.0158 \text{ lb}}.$

c) <u>Weight of 50 cc. of Shell Motor Spirit</u>.

$$W_p = \frac{50 \times 2.2}{1000} \times S.G._{cor.}$$

$S.G._{cor.}$ = <u>0.739</u>, corresponding to an average fuel temperature of 22°C,

therefore, $W_p = \frac{50 \times 2.2 \times 0.739}{1000} = \underline{0.081}$ lb.

d) <u>Consumption time of 8 cc. water for weight ratios of 0.25, 0.50 and 0.75</u>

$$T_{c_1} = \frac{0.0176}{0.081} \times \frac{T_p}{0.25} = \underline{0.868 \times T_p} \quad ....(4\text{-}3.13)$$

$$T_{c_2} = \underline{0.434 \times T_p} \quad ....(4\text{-}3.14)$$

$$T_{c_3} = \underline{0.289 \times T_p} \quad ....(4\text{-}3.15)$$

e) <u>Consumption time of 8cc. water-alcohol (50 + 50) for weight ratios of 0.25 and 0.50</u>.

$$T_{c_1}' = \frac{0.0158}{0.081} \times \frac{T_p}{0.25} = \underline{0.78 \times T_p} \quad ....(4\text{-}3.16)$$

$$T_{c_2}' = \underline{0.39 \times T_p} \quad ....(4\text{-}3.17)$$

f) <u>Weight of water, water-alcohol and alcohol in lb./hr</u>.

$$q_c = \frac{W_c}{T_c} \times 3600 = \frac{0.0176}{T_c} \times 3600 = \frac{63.5}{T_c} \quad ...(4\text{-}3.18)$$

$$q_c' = \frac{0.0158}{T_c'} \times 3600 = \frac{56.8}{T_c'} \quad ....(4\text{-}3.19)$$

$$q_c'' = \frac{0.07 \times 3600}{T_c''} = \frac{25.2}{T_c''} \quad ....(4\text{-}3.20)$$

4 - 4. Test details.

To be able to coordinate test results, standard grade fuel was used for all experiments and the mixture strength was kept constant.

Observations, where possible, were taken over the full speed range of the Ricardo E-6/S engine, and spark advance at each speed was altered to give maximum output at trace knocking conditions.

Compression ratios, as shown in Tables 17 and 18, were changed after completion of each group of tests. The compression ratios were varied between 7:1 and 9:1, the compression ratio 10:1 proving to be too severe for the engine, even with internal cooling.

Preliminary Tests.

Before the final tests were carried out, a considerable number of tests had to be undertaken to investigate the following points:

a) The influence of an internal coolant on the behaviour of an internal combustion engine, when run at various mixture strengths but constant compression ratio.

After comparing results obtained from this group of tests, it was decided to conduct all tests at normal mixture strength. This corresponded to an air-fuel ratio of approximately 13.5:1, determined from an exhaust gas analyser when the engine was running at 2,000 r.p.m., at light load.

To produce this condition, the needle valve had to be unscrewed two and a half turns, the corresponding number being 36.

b) The throttle position for maximum output and for various part-throttle performances:

From Fig. 14, it can be seen that the scale on the segment behind the throttle lever is divided into 10 equal parts. Maximum output was gained when the arm was at No.7 and any further movement of the throttle did not increase the engine output. The air-inlet manifold is, therefore, over-dimensioned when the "Ricardo" works as a petrol engine. For part-throttle operation, No. 3.5 was selected.

c) The flow rate of the needle valve at various speeds and compression ratios:

For the preliminary investigations, a needle valve of standard design was used. It was, however, found that air was getting through the gland into the valve body and delivery line, with the result that variations in discharge time of 100% and more were given for the same needle setting. Thus, predetermined coolant quantities could not be reproduced.

Finally, a needle valve (Fig. 11), manufactured by George Kent, England, originally designed for air and gas flow, was adopted after a slight alteration. Internal coolants, up to a weight ratio of 0.75, could be delivered by this valve for the full speed range.

Calibration tests showed that valve settings for particular weight ratios varied very little with engine speed

As the valve has a micrometer spindle, needle positions could be easily reproduced.

Test procedure.

The engine was started and warmed up according to the maker's instructions. Simultaneously, the electronic indicator was switched on and after 20 minutes was ready for use.

Tests were commenced after cooling water outlet temperature was stable at 70°C and the oil temperature was not less than 65°C.

Atmospheric pressure and wet and dry bulb temperatures were taken at the beginning and end of each test.

All tests were run at full or half-throttle setting, Nos. 7 and 3.5 respectively. Quarter-throttle tests were omitted as earlier investigations have shown that the benefit of an antidetonant injection disappears if an engine runs on less than half-throttle.

The load was increased until the speed dropped to the first test value. The spark was then advanced to give maximum output at trace knocking and the load corrected to keep the speed constant.

Trace knocking conditions were established with the help of the pressure-time or pressure-rate-time diagram on the screen of the cathode-ray tube. Spark advance was altered until the detonating oscillations, typical for a diagram of a knocking engine, almost disappeared.

The following photographs, Figs. 19, 20, 20a and 21 were selected from continuous records, obtained by driving the film vertically across the screen as the indicator diagrams were traced out. This method, however, is responsible for the inclined crank angle axis shown on all photographs.

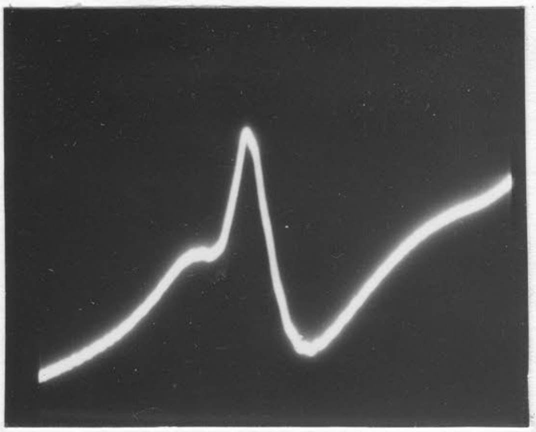


Fig. 19 - Knock free pressure-rate-time diagram, C.R. 7.5:1, throttle position No. 7, water-fuel weight ratio 0, spark advance 18 deg. BTDC, engine speed 2000 r.p.m.

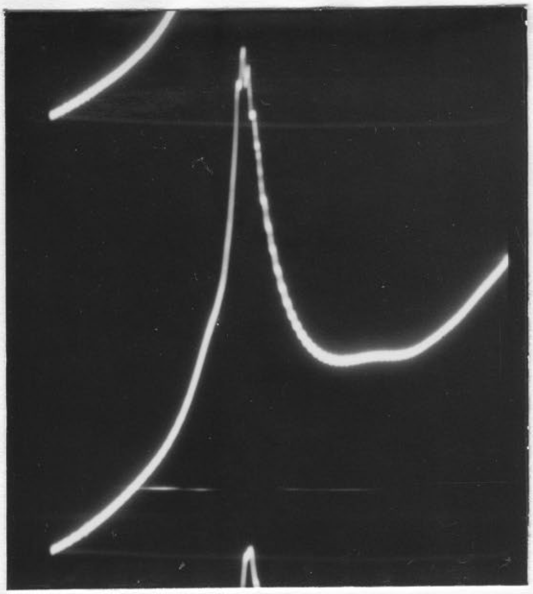


Fig. 20 - Trace knocking pressure-time diagram, C.R. 7.5:1, throttle No. 7, water-fuel weight ratio 0.25, spark advance 28 deg. BTDC., engine speed 2000 r.p.m.

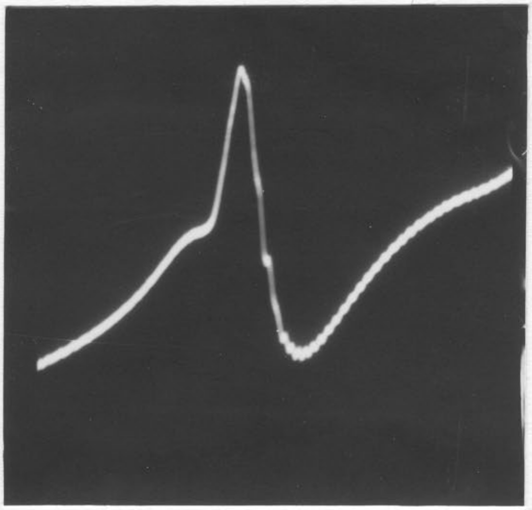


Fig.20a - Trace knocking pressure-rate-time diagram, C.R. 7.5:1, throttle No. 7, water-fuel weight ratio 0.25, spark advance 28 deg. BTDC, engine speed 2000 r.p.m.

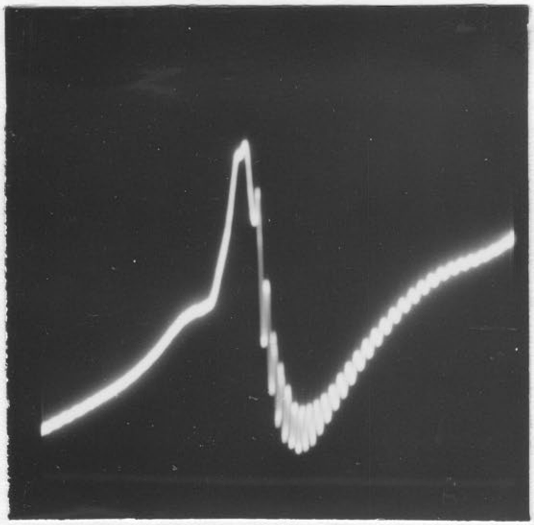


Fig. 21 - Heavy knocking pressure-rate-time diagram, C.R. 7.5:1, throttle position No. 7, water-fuel weight ratio 0, spark advance 28 deg. BTDC, engine speed 2000 r.p.m.

Before any readings were taken, 5 minutes had to elapse to make sure that engine conditions were stable. The first reading of each test was always the fuel consumption time for 50 ml.

This was necessary for the calculation of the coolant flow time for the three selected weight ratios. Stop watches were used to establish fuel and coolant flow times as no automatic measuring devices were available at that time. However, increasing skill of the operators, coupled with multiple readings, reduced observation errors to a minimum.

The following observations were taken during the time to consume 50 ml. of petrol:

a) Cooling water outlet temperature, with the help of a thermometer. Any rise or fall was easily corrected by changing the rate of cooling water flow through the heat exchanger.

b) Lubricating oil temperature, with a thermometer situated in the oil sump. Temperature variations were corrected as for item (a).

c) Air temperature in inlet manifold with a thermometer.

d) Vacuum pressure with a vacuum gauge, located on the water-injection panel. This did not yield any useful results, as, for a single cylinder engine, the fluctuations are too great.

e) Spark advance.

f) Length of column on inclined air manometer.

g) Coolant flow time. If the time did not coincide with the calculated one, the needle valve position was changed by turning the micrometer spindle slightly. Usually a movement of 0.001" was sufficient to rectify conditions.

h) Brake load, as shown on dynamometer dial.

i) Exhaust gas temperature with the help of a special thermocouple and pyrometer.

j) Knock intensity, by watching the diagram on the screen of the cathode ray tube.

To reduce observation errors, and at the same time decrease overall testing time, tests were conducted in the following manner.

At each speed, readings were first taken without injection of internal coolant. These observations provided data for the master curves, shown as heavier lines on the graphs.

Then an internal coolant was injected into the engine, the rate being determined by the weight ratio under consideration. After test readings for all three percentages (25 , 50 and 75) were taken, the load was reduced until the speed was set to the next value. The needle valve was then closed and the procedure repeated as described before.

As an internal coolant, either water or a water-alcohol mixture (50+50 by volume) was used. This mixture was adopted as tests carried out by Colwell and others (13) have shown that when all factors are taken into consideration, a mixture of (50+50) is the best average combination.

Compilation of tests conducted

Test No.	Compr. Ratio	Carbur. needle	Inter. cool 't	Coolant/Fuel Ratio	Throttle Position
1.	7:1	No. 36	None	-	No. 3.5
1a	"	"	water-alc.	0.25 by wt.	" "
1b	"	"	" "	0.50 " "	" "
2	"	"	None	-	No. 7
2a	"	"	water-alc.	0.25 by wt.	" "
2b	"	"	" "	0.50 " "	" "
3	"	"	None	-	No. 3.5
3a	"	"	Water	0.25 by wt.	" "
3b	"	"	"	0.50 " "	" "
3c	"	"	"	0.75 " "	" "
4	"	"	None	-	No. 7
4a	"	"	Water	0.25 by wt.	" "
4b	"	"	"	0.50 " "	" "
4c	"	"	"	0.75 " "	" "
5	7.5:1	"	None	-	No. 3.5
5a	"	"	Water	0.25 by wt.	" "
5b	"	"	"	0.50 " "	" "
5c	"	"	"	0.75 " "	" "
6	"	"	None	-	No. 7
6a	"	"	Water	0.25 by weight	" "
6b	"	"	"	0.50 " "	" "
6c	"	"	"	0.75 " "	" "

Test No.	Compr. Ratio	Carbur. needle	Inter. cool 't	Coolant/Fuel Ratio	Throttle Position
7	7.5:1	No. 36	None	-	No. 3.5
7a	"	"	Water-alc.	0.25 by wt.	" "
7b	"	"	"	0.50 " "	" "
8	"	"	None	-	No. 7
8a	"	"	Water-alc.	0.25 by wt.	-
8b	"	"	"	0.50 " "	" "
9	8:1	"	None	-	No. 7
9a	"	"	Water-alc.	0.25 by wt.	" "
9b	"	"	"	0.50 " "	" "
10	"	"	None	-	No. 7
10a	"	"	Water	0.25 by wt.	" "
10b	"	"	"	0.50 " "	" "
10c	"	"	"	0.75 " "	" "
11	"	"	None	-	No. 3.5
11a	"	"	Water	0.25 by wt.	" "
11b	"	"	"	0.50 " "	" "
11c	"	"	"	0.75 " "	" "
12	9:1	"	None	-	No. 7
12a	"	"	Water	0.25 by wt.	" "
12b	"	"	"	0.50 " "	" "
12c	"	"	"	0.75 " "	" "

Test No.	Compr. Ratio	Carbur. needle	Inter. Cool't	Coolant/Fuel Ratio	Throttle Position
13	9:1	No. 36	None	-	No. 3.5
13a	"	"	Water	0.25 by wt.	" "
13b	"	"	"	0.50 " "	" "
13c	"	"	"	0.75 " "	" "
14	"	"	None	-	No. 7
14a	"	"	Water-alc.	0.25 by wt.	" "
14b	"	"	"	0.50 " "	" "

Table No. 17.

4 - 5. Test Results.

The results, calculated from observations made during tests, are listed in the result sheets Nos. 1 to 14. They provide the material for plotting the following curves, shown on all test sheets:

1) Brake horsepower (corrected) against speed.

These curves were selected to show output variations which may occur after an internal coolant has been injected. The brake horsepower values were corrected to N.T.P. to eliminate any differences in ambient conditions which may have prevailed during tests.

2) Brake thermal efficiency against speed.

These curves were drawn because they took into account the heating value of petrol and alcohol but are independent of the price of each constituent.

3) Petrol consumption against engine speed.

Petrol consumption was expressed in lb./hr. and the equivalent cost in s./hr. (shillings per hour). Petrol consumption was found to be practically independent of any internal coolant and compression ratio and varied only with throttle opening and carburettor settings. A single line, therefore, represents a group of tests on each test sheet.

4) Alcohol consumption against engine speed.

The alcohol consumption was also expressed in lb./hr. and the consumption cost in s./hr., the quantities consumed being dependent on the weight ratio of the coolant injected. Two curves are shown, as weight ratios of 0.25 and 0.50 were only used for water-alcohol as anti-detonant.

5) Combined specific consumption costs against engine speed.

To be able to compare the economy of different tests, the specific consumption costs were expressed in s./B.H.P.-hr. (shillings per brake horsepower-hour).

From curves 3 and 4, the costs of the petrol and alcohol consumption were found and the sums of these figures were divided by the corresponding actual horsepower, to give values of the combined specific consumption costs.

6) <u>Spark advance against engine speed.</u>

Spark advance was expressed in crank-angle degrees before top dead centre. The curves for spark advance are important, because they indicate the knock-suppressing quality of the injected coolant.

To indicate the weight ratios and the relevant coolant used, curves are individually differentiated. The key for them may be found at the bottom of each test sheet.

In the following, an attempt is made to analyse the results of each test and of certain groups of similar experiments. For this purpose, relevant curves are being replotted to facilitate this work.

<u>Test Nos. 1, 1a, 1b. (Fig. No. 22.)</u>

The engine was run at a compression ratio 7:1 and the throttle lever was clamped at position No. 3.5, corresponding to half-throttle opening. The carburettor needle, which controls the mixture strength, was at position No. 36. Water-alcohol (50-50) by volume was used as internal coolant and weight ratios of 0.25 and 0.50 were employed. These two ratios were later used for all the tests employing water-alcohol as anti-detonant.

The maximum brake horsepower (corrected) for tests 1 and 1a was found to be 7.7 at 2750 r.p.m. From 1250 to 2000 r.p.m., test 1a produced a slightly higher output than No.1, but after 2000 r.p.m. the difference became gradually smaller.

The higher output of the engine, when water-alcohol at the 0.25 weight ratio was injected, was brought about by two facts:

1) The knock-suppressing quality of the mixture, which permitted a further spark advance of 3 to 8 crank-angle degrees than in test 1(without any internal coolant).
2) The additional chemical energy of the alcohol, released during the combustion process.

Test 1b, on the other hand, produced only a very small gain in power at low speed, when compared with the master curve. For all speeds higher than 1500 r.p.m., the output of No.1b became even smaller than in case 1.

An explanation of this phenomenon can be found in Chapters 2 and 3. There it has been mentioned that excessive quantities of a coolant have a diluting effect on the combustion mixture. At the same time, a reduction of the maximum cycle temperature takes place, the drop of which depends on the amount and the composition of the internal coolant.

An anti-detonant with a weight ratio of 0.50 will, therefore, reduce the combustion temperature to a greater extent than a 0.25 ratio, with the consequence that overall efficiency and output will fall.

When comparing the curves representing brake thermal efficiency, it can be seen that test 1 has the highest efficiency. The maximum value of 25.5% appears at 1750 r.p.m. Test 1a follows with 24.2% and finally comes 1b with 22.2%, both tests at speeds between 1500 and 1750 r.p.m. The reduction in the overall efficiency of the last two cycles is brought about by the relatively small increase in power from the additional calorific value of the alcohol.

The consumption cost of petrol, which shows practically as a linear function of speed, varies between 1.65 and 2.20 s./hr. over the speed range.

The curve for the alcohol consumption, when using the 0.25 weight ratio, is nearly a straight line, and the running costs vary between 0.31 and 0.44 s./hr. for speeds between 1250 and 3000 r.p.m.

The results for the combined specific fuel costs for the tests under consideration are as follows:
Test 1 is the most economical, reaching its lowest value of 0.273 s./B.H.P.-hr. at 2000 r.p.m., while 1a and 1b have their minimum values of 0.34 and 0.39 s./B.H.P.-hr. respectively at 1750 r.p.m.

When comparing tests 1a and 1, it will be found that the costs to run the engine have increased by about 25% and that the economy becomes even worse for 1b, in which the costs have risen by approximately 43%.

The lines for spark advance show clearly the anti-detonant quality of the coolant, as used in tests 1a and 1b. In the later test, the spark could be advanced to 30 degrees before top dead centre, already at 1250 r.p.m., without any sign of knocking of the engine. At speeds higher than 2500 r.p.m., the curves 1, 1a and 1b approach each other and the relative advantage of spark advance, to give more power, disappears.

Thus, for part-throttle operations, water-alcohol as an anti-detonant, is an uneconomical proposition.

Tests Nos. 3, 3a, 3b, 3c, (Fig.No.23).

The engine settings remained unchanged but water, instead of water-alcohol, was used as the internal coolant. The weight ratios of the injected water were kept at 25, 50 and 75% and these percentages were employed for all further tests with water, independent of throttle opening and compression ratio.

Test 3 should have furnished the same brake horse-power curve as No.1, but a small discrepancy in output may be noted. An explanation for it may be found from the spark advance curves of the above two tests. In test 1, for instance, the spark was advanced 48 degrees while in test 3 only 43 degrees, both at a speed of 2750 r.p.m.

It has been explained previously that the spark, in each test, was advanced as far as possible consistent with freedom from knocking as shown on the screen of the Minirack Indicator. A certain amount of latitude in the interpretation of the indicator diagram, however, was responsible for the variations in spark setting, resulting in slight differences in output.

To avoid such variations in any future tests, they should be carried out as follows:

1) Run the engine with no internal coolant, to produce the master curves.
2) Introduce water as an internal coolant.
3) Change to water-alcohol as an anti-detonant.

This procedure should be repeated at each chosen speed.

To make this possible, it will be necessary, to use two coolant tanks, two float-chambers, two pipettes and a valve chest for quick change-over. By employing such an installation,repeat tests will be avoided and a saving in testing time will be made.

To establish the relative merits of tests 3, 3a, 3b and 3c, however, it is not important that a slight difference should exist between the master curves of Nos. 1 and 3.

When comparing the lines for the brake horsepower (corrected) it may be found that test 3a shows better results than 3 up to 1750 r.p.m., and there the two curves cross. No further improvement in power could be gained by increasing the amount of the injected water, as shown in tests 3b and 3c. The maximum horsepower for these four tests was developed, when running test 3: the equivalent value was 7.43 $B.H.P._{cor.}$ compared with 7.33 for test 3a, both at 2750 r.p.m. Tests 3b and 3c, as mentioned before, produced a still smaller output of 7.22 and 6.95 $B.H.P._{cor.}$ respectively at 2500 r.p.m.

The brake thermal efficiency became a maximum in test 3a with 24.5% at 1750 r.p.m. Between 2000 and 3000 r.p.m., test 3 produced a better efficiency than the remaining tests of Fig. 23.

The four curves of the brake thermal efficiency follow the standard pattern but become more spaced for speeds above 2250 r.p.m.

The lines indicating the specific consumption costs are cramped together and all show the best economy values between 1500 and 2000 r.p.m., the lowest figure being 0.285 s./B.H.P.-hr. for test 3a. The curves of tests 3a and 3b show a small gain in economy at low speeds when compared with No. 3 and No. 3c.

The spark advance curves run nearly parallel, with about 5 crank-angle degrees spacing between them. In other words, each increase in weight ratio of 0.25, makes an additional spark advance of 5 crank-angle degrees possible.

Tests Nos. 2, 2a, 2b. (Fig. No. 24).

These tests were conducted to find the variations in output and economy for full-throttle operation, when running the engine initially without any internal coolant and then with water-alcohol as an anti-detonant. The compression ratio and carburettor setting remained the same as in the two previous tests.

Comparing the horsepower curves first, it may be noted that Nos. 2a and 2b are nearly parallel and that 2b departs from the master curve fairly quickly. The maximum output of test 2a being 12.1 $B.H.P._{cor.}$ at 3000 r.p.m. against 11 at the same speed in test 2. The increase in power of 10% is brought about by the characteristics of the coolant.

The brake thermal efficiency of test 2 is higher than the efficiencies of the two other tests of this group. It reaches a maximum of 27.3% at 2250 r.p.m. In test 2a, the highest efficiency is 26.8% at the above speed. In 2b, a still lower efficiency than that of test 2 is found, for reasons explained previously.

The fuel costs vary between 1.75 and nearly 3 s./hr., and the alcohol costs for test 2a vary between 0.35 and 0.55 s./hr. over the speed range.

The curves for the combined specific fuel costs, if compared with each other, are similar in shape, and No. 2 has a minimum of 0.25 s./B.H.P.-hr. at 2250 r.p.m.

For Test 2a, the running costs are slightly higher. The best economy may be noted at 2250 r.p.m., the corresponding value being 0.29 s./B.H.P.-hr. The curve for the 0.5 weight ratio test indicates that the economy deteriorated, and 0.32 at 2750 r.p.m. being the lowest value.

The high price of alcohol is largely responsible for the poor results, when economy alone is considered.

When comparing the results of tests 2 and 2a at the speed of 2500 r.p.m., an increase in the combined specific fuel costs of about 16% may be found, whereas the possible rise in power is 6%. Better results, however, may be noted at 3000 r.p.m. There, an increase in the above costs of 11% is found for an increase in power of 11.1%.

The spark advance curves for tests 2 and 2a are almost parallel and separated by 5 crank-angle degrees. The line for 2b runs parallel with 2a at the beginning, but approaches 2a at 3000 r.p.m.

Tests Nos. 4, 4a, 4b, 4c. (Fig. No. 25).

Engine settings for the tests, represented in Fig.25, were identical with those in test 2, but water was used as an anti-detonant.

When studying the horsepower curves of tests 4 and 4a, it may be noticed that curve 4a runs above 4 and approaches 4 at 3000 r.p.m. The maximum horsepower (corrected) was found to be 11.25 at 2750 r.p.m. for test 4a. The curve of test 4b shows the same engine output as test 4a up to 2500 r.p.m. From thereon, the values of 4b become smaller, reaching 11 B.H.P.$_{cor.}$ at 2750 r.p.m. The curve 4c shows that the engine developed the same power as in test 4a and 4b at low speeds between 1250 and 1750 r.p.m. At 2000 r.p.m., the curve 4c crosses the master curve and then proceeds below it.

The lines for the brake thermal efficiency curve in the usual way. The highest values reached, are between 2250 and 2500 r.p.m. In case 4a, the maximum efficiency being 27.8%, compared with 27.2% for the tests 4, 4b and 4c.

It may be seen that the curve for the specific consumption costs of test 4a has its least value of 0.25 s./B.H.P.-hr. between 2250 and 2500 r.p.m., and the best value of test 4 is only slightly higher than the equivalent figures of 4a and 4b. The line of test 4c shows a deviation from the usual form, the economy over part of the speed range being worse than in tests 4, 4a and 4b.

The spark advance curves indicate that with water as an anti-detonant, the spark could be advanced up to 60 degrees B.T.D.C. at 3000 r.p.m. for the 0.5 and 0.75 weight ratios. Even with such an advance, the engine did not show signs of light knocking. However, firing became intermittent when using the 0.75 water-fuel ratio.

Tests Nos. 2a, 4, 4a. (Fig. No. 26).

Results of these full-throttle tests were replotted to show the variations of output and economy of the engine, when running it initially without internal coolant, then with water and finally with water-alcohol as an anti-detonant. Both internal coolants were kept at a constant weight ratio of 0.25 throughout the tests.

When comparing the graphs of Fig. 26 it may be seen that a gain of power and of economy at the same time can only be achieved, when water is used as a knock-suppressor.

If, however, a substantial increase in power is desired, without regard to cost, water-alcohol is superior to water as an anti-knock medium.

For all speeds higher than 2000 r.p.m., the spark advance possible, when water is injected, is higher than in case of water-alcohol or where no internal coolant is used.

Tests Nos. 5, 5a, 5b, 5c. (Fig. No. 27).

For this group of tests, and the following groups, the compression ratio was increased to 7.5:1. Water was used as internal coolant and the engine run at half-throttle (No. 3.5).

The test results are similar to those of test 1. At speeds up to 1750 r.p.m. the curves of $B.H.P._{cor.}$ indicate that slightly more power was developed by the engine, when water, at the three weight ratios was injected, compared with the run without. At about that speed, the curves 5a, 5b and 5c cross the master curve and a smaller output is indicated. It may be noticed that the maximum power of 7.7 and 7.6 $B.H.P._{cor.}$ of tests 5 and 5a being developed at 2750 r.p.m. By increasing the percentage of the injected water, a reduction of power takes place, for reasons previously explained.

The maximum brake thermal efficiency of test 5a is 25.3% at 1750 r.p.m. and is slightly higher than the corresponding value of test 5.

The graphs showing the combined specific consumption costs fall closely together. The lowest value being 0.276 s./B.H.P.-hr at 2000 r.p.m. for the test 5.

The spark advance curves follow the usual pattern found for water as an internal coolant. They are spaced approximately 5 crank-angle degrees apart and for the 0.75 weight ratio, the spark could be advanced up to 60 degrees at 3000 r.p.m. for trace knocking.

Tests Nos. 7, 7a, 7b. (Fig. No. 28).

All engine variables were kept the same as in test 5, but water-alcohol was injected instead of water.

The horsepower curves for no internal coolant and for the 0.25 water-alcohol are similar in shape, the latter indicating slightly higher output. The maximum values are reached at 2750 r.p.m. with 7.9 and 7.8 $B.H.P._{cor.}$ for tests 7a and 7 respectively. The curve for the 0.5 weight ratio shows that the engine developed less power with this percentage of coolant than before.

The curves representing brake thermal efficiency have a similar configuration and the highest values reached are as follows: For test 7, 24% at 2000 r.p.m., for 7a, 23% at about 1500 r.p.m. and for test 7b, 21.4% at 1750 r.p.m.

The curve for the alcohol consumption shows that the hourly cost change from 0.3 to 0.41 shillings over the speed range.

The curves for the combined specific consumption costs (i.e., the petrol and the alcohol of the anti-detonant mixture) indicate best economy of test 7 at 2000 r.p.m. with 0.29 s./B.H.P.-hr., against 0.34 and 0.39 s./B.H.P.-hr. for test 7a and 7b respectively. The last two figures appear at 1750 r.p.m.

The spark advance lines rise consistently, the greatest advance being at 50 degrees B.T.D.C. for the tests 7a and 7b. Further spark advance would have provoked light knocking.

Tests Nos. 6, 6a, 6b, 6c. (Fig. No. 29).

Full-throttle conditions were restored for these tests, and water was used again as the internal coolant.

The tests 6a and 6b furnished almost identical results for engine output, efficiency and economy. The maximum horsepower found was 11.3 for these tests employing water-injection, compared with 11.2 B.H.P.$_{cor.}$ without internal coolant, both values being at 2750 r.p.m.

The brake thermal efficiency of tests 6a, 6b and 6c became a maximum at 2250 r.p.m., the common value being 28.7%. Test 6 furnished 28.4% at the same speed.

The lines for the specific consumption costs curve as expected. The best economy figure for the tests with water-injection was 0.245 compared with 0.249 s./B.H.P.-hr. for test 6.

The curves for spark advance show again the high anti-knock quality of water as an internal coolant. Spark advance up to 60 degrees B.T.D.C. could be employed at 3000 r.p.m.

Tests Nos. 8, 8a, 8b. (Fig. No. 30).

Engine conditions were kept the same as in test 6 but water-alcohol was substituted as the internal coolant.

The increase in power, derived from the alcohol in the coolant mixture is well demonstrated in these tests. The horsepower curves indicate that up to 2250 r.p.m., practically the same power was produced in tests 8a and 8b but at higher speeds, test 8b yielded slightly more. The maximum output occurred at 3000 r.p.m. and 13.0 respectively 12.5 B.H.P.$_{cor.}$

were developed during tests 8b and 8a. The corresponding value of test 8 was 11.5 $B.H.P._{cor.}$

The brake thermal efficiencies for these tests varied considerably. Test 8a produced 28.2% between 2250 and 2500 r.p.m., while test 8 reached its maximum of 27% at 2250 r.p.m. Test 8b developed its highest efficiency of 26.8% at 2750 r.p.m.

The alcohol consumption cost varied between 0.34 and 0.58 s./hr. over the speed range.

For the combined specific consumption costs, the following minimum figures were reached: 0.26, 0.28 and 0.32 s./B.H.P.-hr. for tests 8, 8a and 8b respectively, all values at 2500 r.p.m.

When comparing the spark advance curves it can be seen that they are parallel for the tests 8 and 8a and separated by about 5 crank-angle degrees. Curve 8b deviates from the other two curves, especially in the region of 2000 to 2500 r.p.m., the spark was given less advance than in tests 8 and 8a.

Tests Nos. 6a, 8, 8a. (Fig. No. 31).

Curves of these tests were redrawn to make possible a comparison of full-throttle tests with and without internal coolants, at a compression ratio of 7.5:1.

As found previously, an increase in power and of economy too, could only be noted when water was used as the anti-detonant. At 2500 r.p.m., the output increased by 4 1/2% and the consumption cost, at the same time, decreased by 4 1/2 % when using the figures of test 8 as the base of the calculations.

Water-alcohol, on the other hand, increased the power output at 2500 r.p.m. by 10%, but the economy decreased by about 8 % at that speed.

Tests Nos. 9, 9a, 9b. (Fig. No. 32).

For these tests the compression ratio was further increased to 8:1 and the engine was run with water-alcohol as the anti-detonant. As engine detonation became heavy at low speed, when no internal coolant was injected, testing had to be commenced at 1750 r.p.m. and even at some higher speeds, a certain amount of light knocking was existing.

Tests 9a and 9b furnished similar horsepower lines and the maximum output obtained was 13.0 $B.H.P._{cor.}$ for test 9b against 11.7 for test 9, both values were taken at 3000 r.p.m.

The brake thermal efficiency curves vary considerably in shape. In test 9a, the best efficiency reached was 28.6 % at 2500, while in 9b a maximum efficiency of 27.8 % between 2500 and 2750 r.p.m. could only be obtained.

The alcohol consumption cost for test 9a varied between 0.4 and 0.58 s./B.H.P.-hr. over the speed range.

The lines for the combined specific fuel costs approached their minimum at 2500 r.p.m., the figures reached for the tests 9, 9a and 9b were as follows: 0.258, 0.274 and 0.308 s./B.H.P.-hr. respectively.

The spark advance curves, when compared, are further apart than in previous tests. At 1500 r.p.m., a difference of 10 crank-angle degrees may be found. At 3000 r.p.m., the curves of tests 9a and 9b almost meet and a difference of 5 degrees exists between the master curve and those two lines.

Tests Nos. 10, 10a, 10b, 10c. (Fig. No. 33).

The engine settings remained the same as in test 9, but water was used instead of water-alcohol as the anti-detonant.

To find values for the master curve, testing began at 1750 r.p.m., as engine detonation proved too severe at lower speeds. For the tests 10a, 10b and 10c, however, testing could be started at 1500 r.p.m. with evidence of light knocking only.

The curves representing engine output are practically parallel and the values of tests 10b and 10c vary so little that they may be represented by a single line. The maximum value for test 10a is 11.45 at 3000 r.p.m. compared with 11.24 B.H.P.$_{cor.}$ for test 10 at 2750 r.p.m.

The maximum thermal efficiencies of the tests 10a, 10b and 10c are 28%, 27.7% and 28.25% at 2250 r.p.m. against

27.6% for curve 10 between 2250 and 2500 r.p.m.

The curves representing the specific consumption cost are nearly parallel, the best economy values of 0.249 and 0.250 s./B.H.P.-hr. occurring at 2250 r.p.m. for tests 10a and 10b.

The spark advance curves conform well with each other. Curve 10c indicates a possible spark advance of 55 degrees at 3000 r.p.m. A spark advance of the order of 60 degrees could not be reached with only trace knocking of the engine.

Tests Nos. 11, 11a, 11b, 11c. (Fig. No. 34).

These tests were carried out under half-throttle conditions with water as the internal coolant.

It may be noted, when comparing the horsepower curves, that very little improvement of output is gained from the use of water-injection in spite of the higher compression ratio. The engine speed, however, could be reduced to 1250 r.p.m. before experiencing trace knocking.

Up to 1750 r.p.m., the curves, indicating the use of water-injection, are slightly higher than the master curve. The latter reaches its best value of 8.15 $B.H.P._{cor.}$ at 3000 r.p.m. compared with 7.95 $B.H.P._{cor.}$ at 2750 r.p.m. for test 11a. The curves for the higher coolant ratios show that the engine output decreased.

The lines of the brake thermal efficiency have a common value of 24.5% at about 1750 r.p.m. and are shaped in the usual way.

The economy graph for the test 11a shows 0.285 s./ B.H.P.-hr. as the lowest value at 1750 r.p.m. against 0.287 at 2000 r.p.m. for test 11.

The spark advance curves for the tests with water as an internal coolant are very similar and with the 0.75 weight ratio, the spark could again be advanced up to 60 degrees.

Tests Nos. 9, 9a, 10a. (Fig. No. 35).

These tests illustrate, again, the improvement of power and economy, when running the engine at full-throttle with water and water-alcohol as internal coolants at the 0.25 weight ratio.

For the compression ratio 8:1, test 9a, as far as power is concerned, furnished the best results.
The maximum output was 12.9 $\text{B.H.P.}_{\text{cor.}}$ at 3000 r.p.m.
In test 9, only 11.65 $\text{B.H.P.}_{\text{cor.}}$ was developed at the same speed. At 2500 r.p.m., the increase of power between tests 9 and 9a was calculated as 11.5%.
At 2250 r.p.m., test 10a furnished 5% more power than No. 9.

The brake thermal efficiency was best for test 9a. At 2500 r.p.m., 28.8% was reached, a very high figure for a single cylinder petrol engine.

The cost of the alcohol consumption varied between

0.4 and 0.58 s./hr. over the speed range.

The specific consumption cost was found lowest for test 10a, followed by 9 and 9a. At 2500 r.p.m., the equivalent costs were: 0.251, 0.259 and 0.271 s./B.H.P.-hr. respectively.

The spark advance settings for the tests with water and water-alcohol were identical up to 2250 r.p.m. Above this speed, the spark in test 10a could be further advanced than in 9a, due to the greater cooling effect of water in comparison with water-alcohol.

Tests Nos. 12, 12a, 12b, 12c. (Fig. No. 36).

In the last group of tests the compression ratio was further increased to 9:1 and full-throttle operations with water as the internal coolant were investigated.

Without the internal coolant, the engine could not be run at speeds lower than 2250 r.p.m., knocking becoming too severe. Practically no difference in output was experienced when injecting water as the knock suppressor in any of the three weight ratios up to 2250 r.p.m. The maximum value was 12.25 $B.H.P._{cor.}$ at 3000 r.p.m. for test 12.

The brake thermal efficiency became a maximum of 29.25% at 2250 r.p.m. for test 12c, compared with 28.4% for operation without. 29.25% was found to be the highest thermal efficiency of all tests conducted by the author.

The specific consumption costs for the tests with water-injection reached a minimum of 0.240 s./B.H.P.-hr. at 2250 r.p.m. compared with 0.246 for test 12 at 2500 r.p.m.

The maximum possible spark advance for test 12c was 55 degrees B.T.D.C. at 3000 r.p.m. As in previous tests, the gain from high spark advance was cancelled out by the reduction of cycle temperature and therefore of output.

Tests Nos. 14, 14a, 14b. (Fig. No. 37).

These tests were conducted at full-throttle with water-alcohol as the internal coolant.

From Fig. 37 it can be seen that the horsepower curves are parallel for tests 14a and 14b, the maximum outputs being 13 and 12.6 $B.H.P._{cor.}$ respectively, both values at 2750 r.p.m. The horsepower curve for test 14 (no internal coolant used) is indicated only on the graph, because knocking interfered very badly at lower speeds and to a certain extent at higher speeds. The output of test 14 is, therefore, much smaller than in previous tests having lower compression ratios and no internal coolant.

The maximum brake thermal efficiency for test 14a is 28.2% and for 14b, 26.5%, both values at about 2500 r.p.m.

The curves of the spark advance are almost parallel.

It may be noticed that the rate of rise of these lines over the speed range is much smaller than in previous tests. This is necessary to prevent knocking at the compression ratio of 9:1, even though an anti-detonant is being used.

Tests Nos. 13, 13a, 13b, 13c (Fig. No. 38).

To conclude the tests on the Ricardo engine, half-throttle operations were carried out at the compression ratio 9:1, using water as internal coolant.

The injection of water gave no increase in power and little in economy in this series. The maxima of the horsepowers were developed at about 2750 r.p.m. Test 13 topped the others with 8.55 $B.H.P._{cor.}$; next came test 13a with 8.45 $B.H.P._{cor.}$

The brake thermal efficiency curves do not vary a great deal. In tests 13a and 13, 25.7% and 25.3% respectively, were obtained at 2250 r.p.m.

The fuel costs for all the tests of this group are practically the same and vary from 1.87 to 2.37 s./hr. over the speed range.

The most economical test, it may be noted, is No. 13a. The lowest specific fuel cost was 0.277 s./B.H.P.-hr. at 2250 r.p.m.

The spark advance curves are similar to each other in shape and reach values ranging from 40 to 55 degrees at top speed.

Tests Nos. 12, 12a, 14a. (Fig. No. 39).

Graph 39 was drawn to illustrate the difference in results obtained when running the engine with, and without, internal coolants at full-throttle and a compression ratio of 9:1.

Without coolant, the engine could only be used over the upper section of the speed range, as detonation became so heavy that damage to the engine may have occurred.

The output reached a maximum between 2750 and 3000 r.p.m. in test 14a, when the engine developed 12.75 $B.H.P._{cor.}$

The brake thermal efficiency for test 12a, with water as anti-detonant, became the highest for this group with a value of 29.25%. The efficiencies for tests 12 and 14a were the same at 28.3% and all three tests had their maxima at 2500 r.p.m.

The spark advance curves again show the higher anti-knock quality of water. At 3000 r.p.m., the spark in test 12a could be advanced 40 degrees compared with 30 degrees for the other two.

To be able to compare, more readily, the results of the various tests, four tables have been arranged.

The first two, Nos. 18 and 18a, show figures for the brake horsepower corrected and the specific fuel costs at speeds of 2000, 2250 and 2500 r.p.m. These values cover the average operating speeds of an internal combustion engine. Results at other speeds, if desired, may be taken from the graphs of the corresponding tests.

The results are compiled for all tests involving no internal coolant and also for those employing coolants at the 0.25 weight ratio. The findings of the other tests, however, are omitted, because the test graphs indicate that no, or very little, gain in power and economy are obtained by increasing the amount of the internal coolant above the 0.25 weight ratio.

The appropriate results are tabulated in two vertical columns, showing absolute and per cent values. The latter are given as percentages of the equivalent values for tests without internal coolants. Thus, the relative merits of the tests with anti-detonant injection may be seen immediately.

In tables 19 and 19a, results from the previous two tables are rearranged in such a way that they form groups according to the throttle opening and the internal coolant used.

The findings are again tabulated in two vertical columns, showing absolute and per cent values. The results for $B.H.P._{cor.}$ and specific fuel costs are given as percentages of the equivalent values for tests without internal coolant and a compression ratio 7:1.

The tables 19 and 19a are very important because they convey the improvements in power and economy, when increasing the compression ratio from 7:1 to 9:1 in an internal combustion engine which uses standard grade fuel and water or water-alcohol as an anti-detonant.

The engine, which showed signs of heavy detonation at full throttle opening and compression ratios above 7.5:1, would run smoothly after the injection of one of the two coolants, and detonation could be kept at trace knocking.

Summary of power and economy results at speeds of 2000, 2250 and 2500 r.p.m.

The weight ratio (coolant to petrol) for these tests is 0.25.

Values of brake horsepower corrected and specific fuel costs are given as a percentage of the equivalent values for tests without an internal coolant.

			Brake horsepower corrected						Specific fuel costs, s./B.H.P.-hr.							
Engine speed			2000		2250		2500		2000		2250		2500			
Test No.	Fig. No.	C.R.	B.H.P. cor.	%	B.H.P. cor.	%	B.H.P. cor.	%	F_s	%	F_s	%	F_s	%	Internal coolant	Throttle position
1	22	7:1	7.30	100	7.50	100	7.68	100	0.273	100	0.275	100	0.285	100	none	No. 3.5
1a	"	"	7.38	101	7.57	101	7.68	100	0.340	124.5	0.345	125.5	0.350	123	water-alc.(50+50)	"
3	23	"	7.10	100	7.25	100	7.40	100	0.290	100	0.293	100	0.299	100	none	"
3a	"	"	7.08	99.8	7.20	99.3	7.28	98.4	0.285	98.2	0.291	99.4	0.297	99.3	water	"
2	24	"	9.15	100	10.00	100	10.60	100	0.255	100	0.250	100	0.255	100	none	No. 7
2a	"	"	9.50	104	10.50	105	11.25	106	0.295	115.6	0.290	116	0.295	115.6	water-alc.(50+50)	"
4	25	"	9.08	100	9.88	100	10.50	100	0.262	100	0.256	100	0.255	100	none	"
4a	"	"	9.25	102	10.08	102	10.70	102	0.255	97.5	0.252	98.5	0.252	99	water	"
5	27	7.5:1	7.40	100	7.60	100	7.70	100	0.276	100	0.280	100	0.285	100	none	No. 3.5
5a	"	"	7.34	99.2	7.50	98.7	7.60	98.7	0.278	100.5	0.282	101	0.288	101	water	"
7	28	"	7.40	100	7.62	100	7.77	100	0.294	100	0.295	100	0.299	100	none	"
7a	"	"	7.45	100.5	7.68	101	7.83	101	0.340	115.5	0.345	117	0.351	117	water-alc.(50+50)	"
6	29	"	9.50	100	10.40	100	10.95	100	0.250	100	0.249	100	0.252	100	none	No. 7
6a	"	"	9.80	103	10.50	101	11.10	101.5	0.248	99	0.245	98.5	0.248	97.8	water	"

Table No. 18.

			Brake horsepower corrected						Specific fuel costs, s./B.H.P.-hr.							
Engine speed			2000		2250		2500		2000		2250		2500			
Test No.	Fig. No.	C.R.	B.H.P. cor.	%	B.H.P. cor.	%	B.H.P. cor.	%	F_s	%	F_s	%	F_s	%	Internal Coolant	Throttle position
8	30	7.5:1	9.00	100	9.83	100	10.50	100	0.262	100	0.260	100	0.260	100	none	No. 7
8a	"	"	10.00	111	11.00	112	11.70	111.2	0.287	109.6	0.280	107.8	0.280	107.8	water-alc.(50+50)	"
9	32	8:1	9.20	100	10.00	100	10.70	100	0.264	100	0.260	100	0.257	100	none	"
9a	"	"	10.10	109.8	11.20	112	12.00	112	0.281	106.5	0.276	106	0.274	106.5	water-alc.(50+50)	"
10	33	"	9.70	100	10.50	100	11.00	100	0.253	100	0.252	100	0.257	100	none	"
10a	"	"	9.62	99.1	10.50	100	11.15	101.5	0.250	99	0.249	98.7	0.252	98	water	"
11	34	"	7.60	100	7.88	100	8.10	100	0.287	100	0.287	100	0.291	100	none	No. 3.5
11a	"	"	7.55	99.4	7.75	98.5	7.88	97.5	0.285	99.3	0.287	100	0.293	100.8	water	"
12	36	9:1	-	-	10.25	100	11.20	100	-	-	0.251	100	0.246	100	none	No. 7
12a	"	"	9.50	-	10.50	102.5	11.30	101	0.245	-	0.241	96	0.242	98.5	water	"
14	37	"	-	-	-	-	10.10	100	-	-	-	-	-	-	none	"
14a	"	"	9.90	-	11.05	-	12.00	119	0.284	-	0.282	-	0.283	-	water-alc.(50+50)	"
13	38	"	7.75	100	8.20	100	8.45	100	0.282	100	0.278	100	0.279	100	none	No. 3.5
13a	"	"	7.70	99.4	8.20	100	8.40	99.5	0.280	99.2	0.277	99.5	0.278	99.5	water	"

Table No. 18a.

(continuation of Table 18)

Summary of power and economy results at speeds of 2000, 2250 and 2500 r.p.m.

The weight ratio (coolant to petrol) for these tests is 0.25.

Values of brake horsepower corrected and specific fuel costs are given as a percentage of the equivalent values for tests without an internal coolant and a compression ratio 7:1.

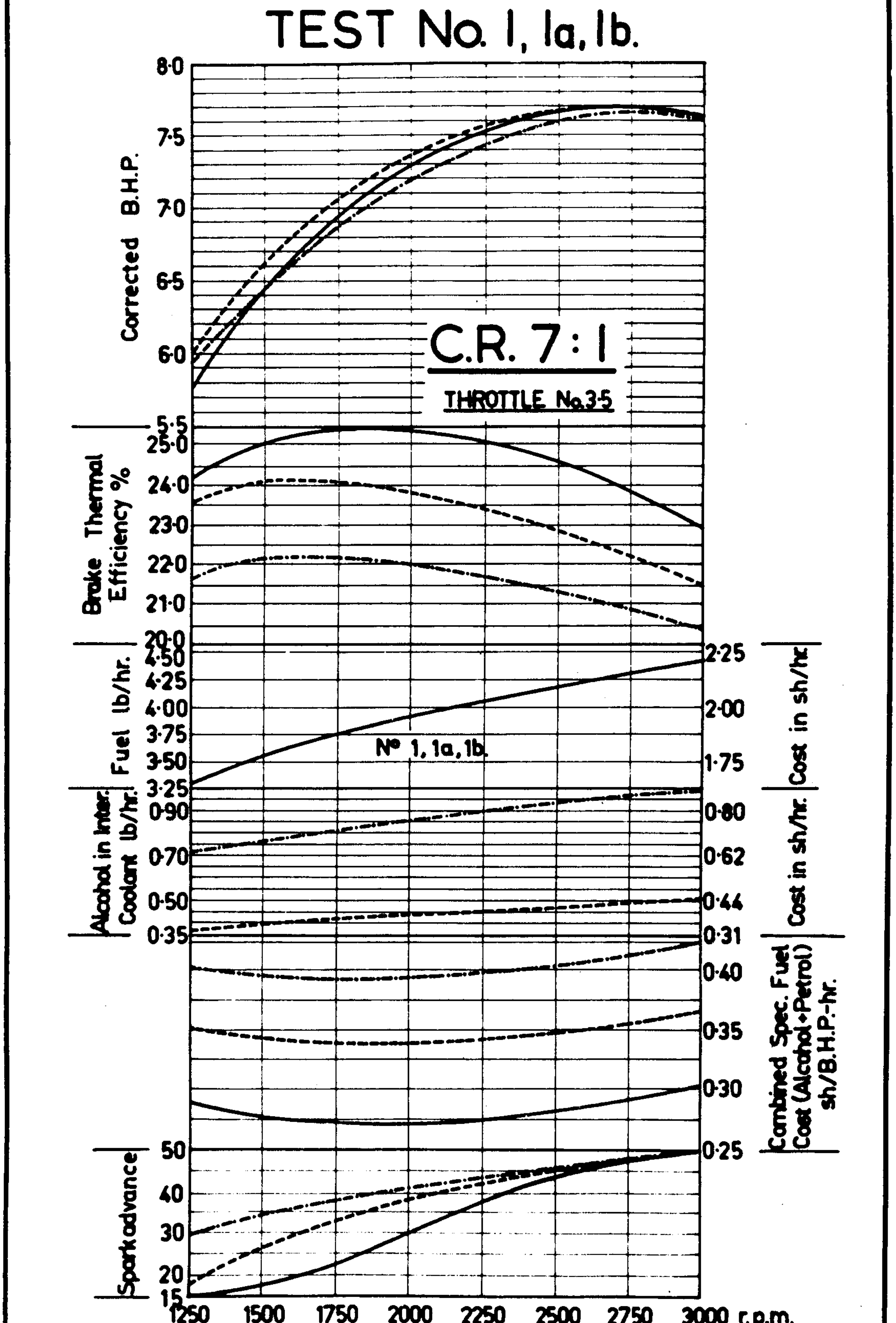
			Brake horsepower corrected						Specific fuel costs, s./B.H.P.-hr.							
Engine speed			2000		2250		2500		2000		2250		2500			
Test No.	Fig. No.	C.R.	B.H.P. cor.	%	B.H.P. cor.	%	B.H.P. cor.	%	F_s	%	F_s	%	F_s	%	Internal Coolant	Throttle position
4	25	7:1	9.08	100	9.88	100	10.50	100	0.262	100	0.256	100	0.255	100	none	No. 7
4a	"	"	9.25	102	10.08	102	10.70	102	0.255	97.3	0.252	98.5	0.252	99	water	"
6a	29	7.5:1	9.80	108	10.50	106.5	11.10	105.8	0.248	94.8	0.245	95.7	0.248	97.2	"	"
10a	33	8:1	9.62	106	10.50	106.5	11.15	106	0.250	95.5	0.249	97	0.252	98.8	"	"
12a	36	9:1	9.50	104.5	10.50	106.5	11.30	107.5	0.245	93.5	0.241	94.2	0.242	95	"	"
2	24	7:1	9.15	100	10.00	100	10.60	100	0.255	100	0.250	100	0.255	100	none	No. 7
2a	"	"	9.50	104	10.50	105	11.25	106	0.295	115.6	0.290	116	0.295	115.6	water-alc.(50+50)	"
8a	30	7.5:1	10.00	109.2	11.00	110	11.70	110	0.287	112.5	0.280	112	0.280	110	"	"
9a	32	8:1	10.10	110.5	11.20	112	12.00	113	0.281	110	0.276	110	0.274	107.5	"	"
14a	37	9:1	9.90	108	11.05	110.5	12.00	113	0.284	111	0.282	112.5	0.283	111	"	"

Table No. 19.

			Brake horsepower corrected						Specific fuel costs, s./B.H.P.-hr.							
Engine speed			2000		2250		2500		2000		2250		2500			
Test No.	Fig. No.	C.R.	B.H.P. cor.	%	B.H.P. cor.	%	B.H.P. cor.	%	F_s	%	F_s	%	F_s	%	Internal Coolant	Throttle position
3	23	7:1	7.10	100	7.25	100	7.40	100	0.290	100	0.293	100	0.299	100	none	No. 3.5
3a	"	"	7.08	99.8	7.20	99.3	7.28	98.5	0.285	98.5	0.291	99.2	0.297	99.3	water	"
5a	27	7.5:1	7.34	103.2	7.50	103.2	7.60	102.5	0.278	96	0.282	96.2	0.288	96.5	"	"
11a	34	8:1	7.55	106.2	7.75	106.8	7.88	106.5	0.285	98.5	0.287	98	0.293	98	"	"
13a	38	9:1	7.70	108.4	8.20	113	8.40	113.5	0.280	96.5	0.277	94.5	0.278	93	"	"
1	22	7:1	7.30	100	7.50	100	7.68	100	0.273	100	0.275	100	0.285	100	none	No. 3.5
1a	"	"	7.38	101	7.57	101	7.68	100	0.340	124.5	0.345	125.5	0.350	123	water-alc.(50+50)	"
7a	28	7.5:1	7.45	102	7.68	102.2	7.83	102	0.340	124.5	0.345	125.5	0.351	123	"	"
Tests were discontinued as specific fuel costs became too high.																

Table No. 19a.

(continuation of table 19)

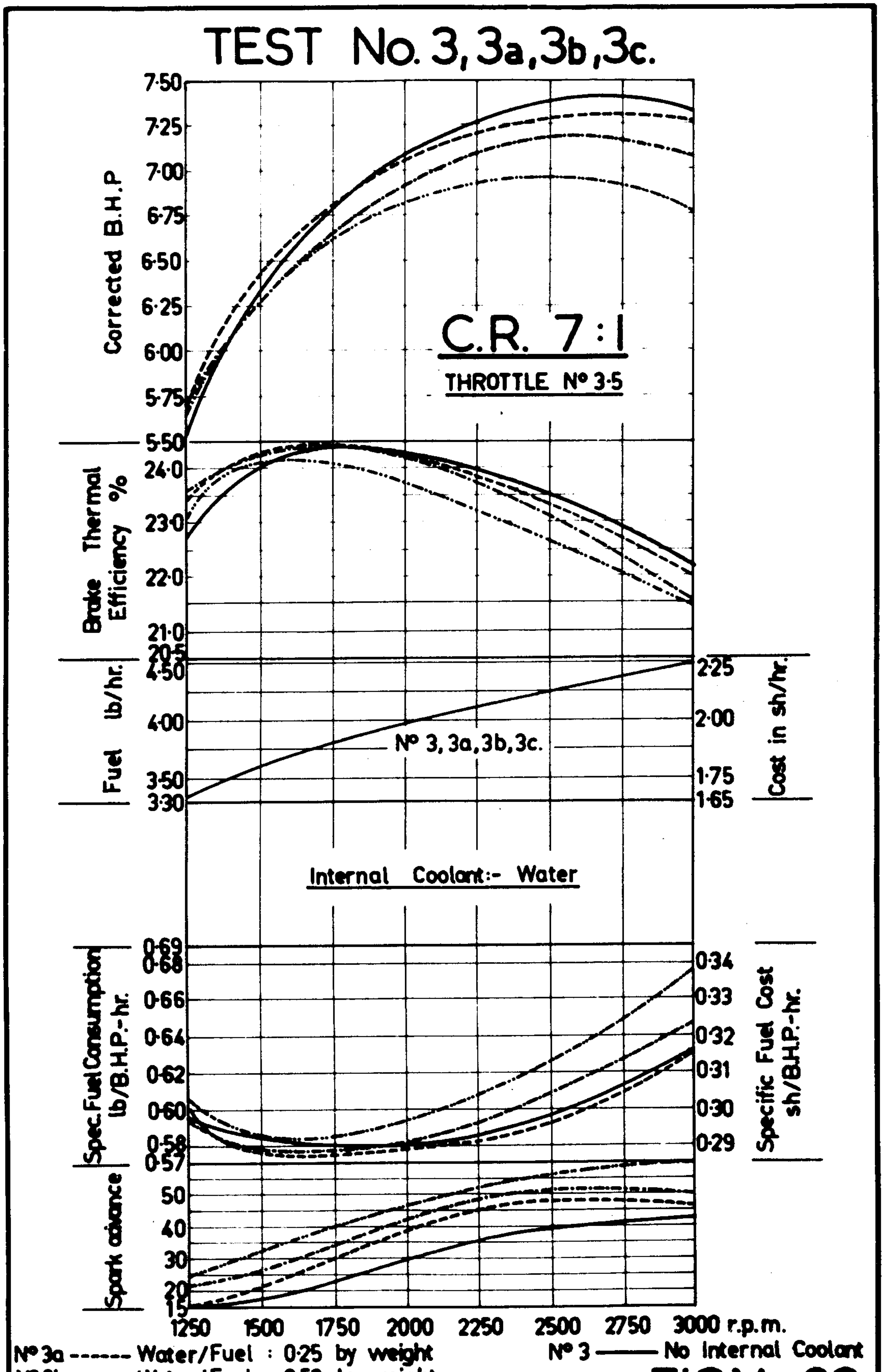


N° 1 ——— No Internal Coolant

N° 1a ------(Water + Alcohol)/Fuel : 0·25 by weight

N° 1b -·-·-·-(Water + Alcohol)/Fuel : 0·50 by weight

FIG. No. 22


TEST No. 3, 3a, 3b, 3c.
C.R. 7:1
THROTTLE Nº 3·5
Corrected B.H.P
Brake Thermal Efficiency %
Fuel lb/hr.
Cost in sh/hr.
Nº 3, 3a, 3b, 3c.
Internal Coolant:- Water
Spec. Fuel Consumption lb/B.H.P.-hr.
Specific Fuel Cost sh/B.H.P.-hr.
Spark advance
1250 1500 1750 2000 2250 2500 2750 3000 r.p.m.
Nº 3a ------ Water/Fuel : 0·25 by weight
Nº 3b -·-·-·- Water/Fuel : 0·50 by weight
Nº 3c -··-··- Water/Fuel : 0·75 by weight
Nº 3 ——— No Internal Coolant
FIG. No. 23

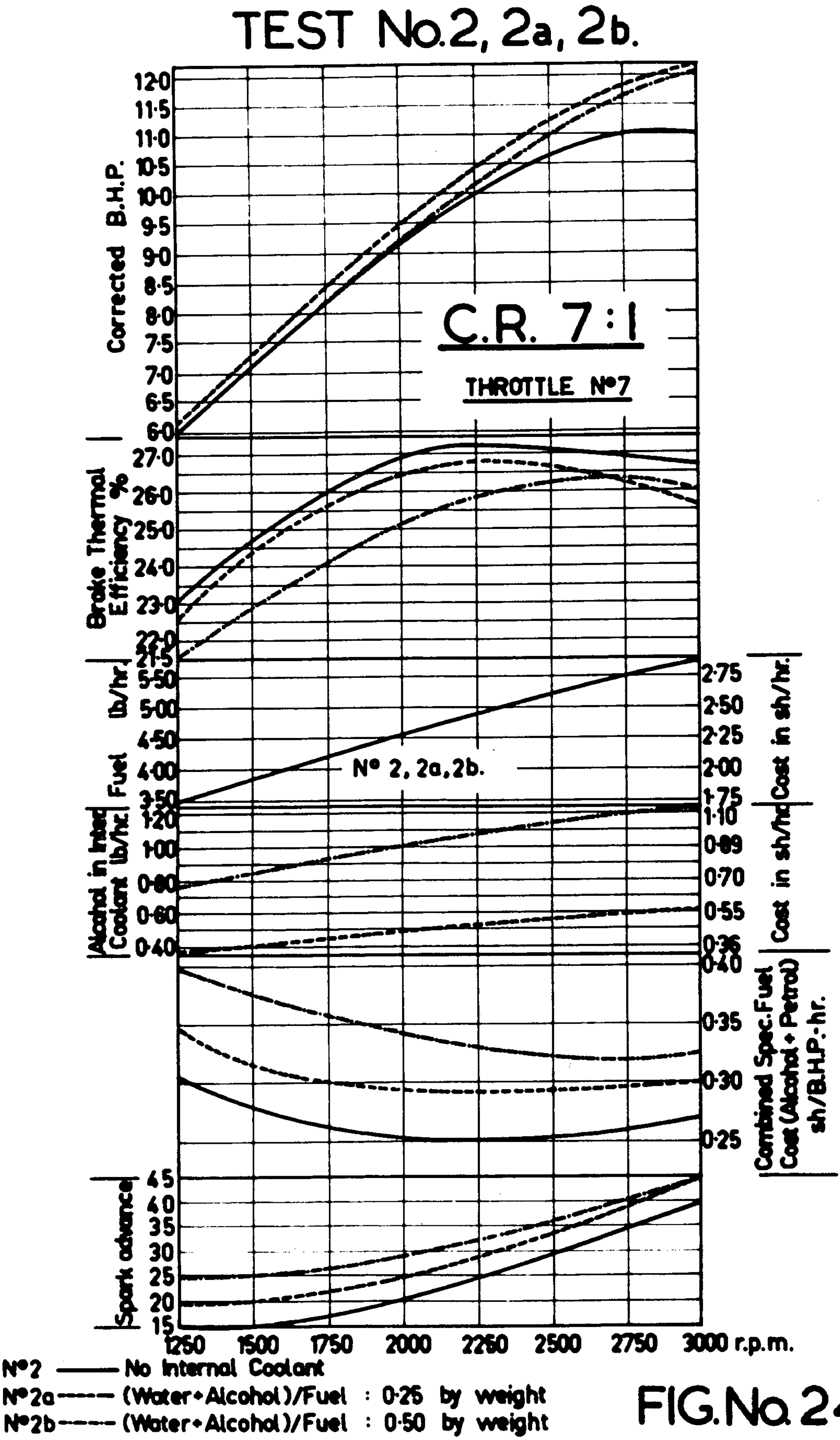
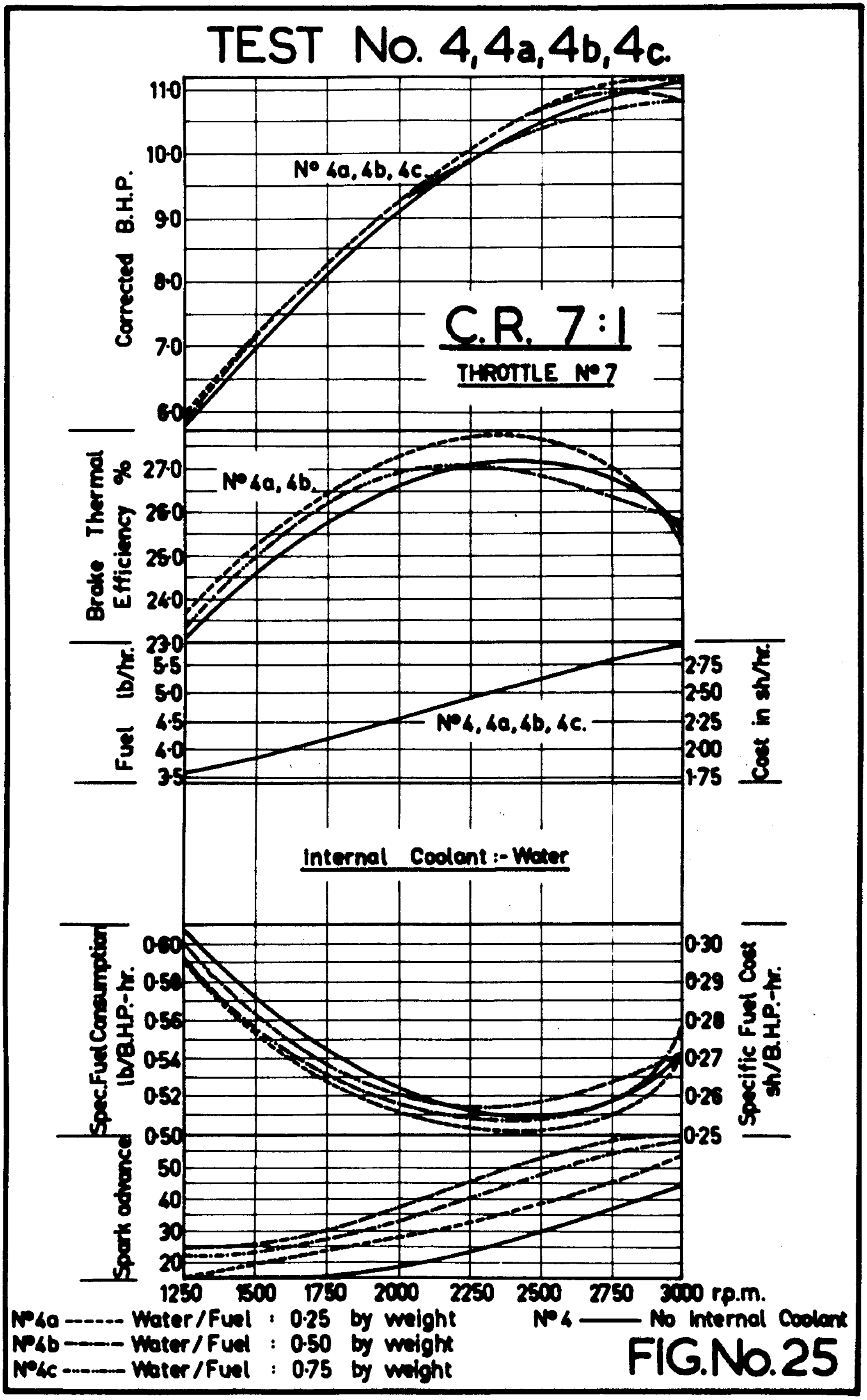
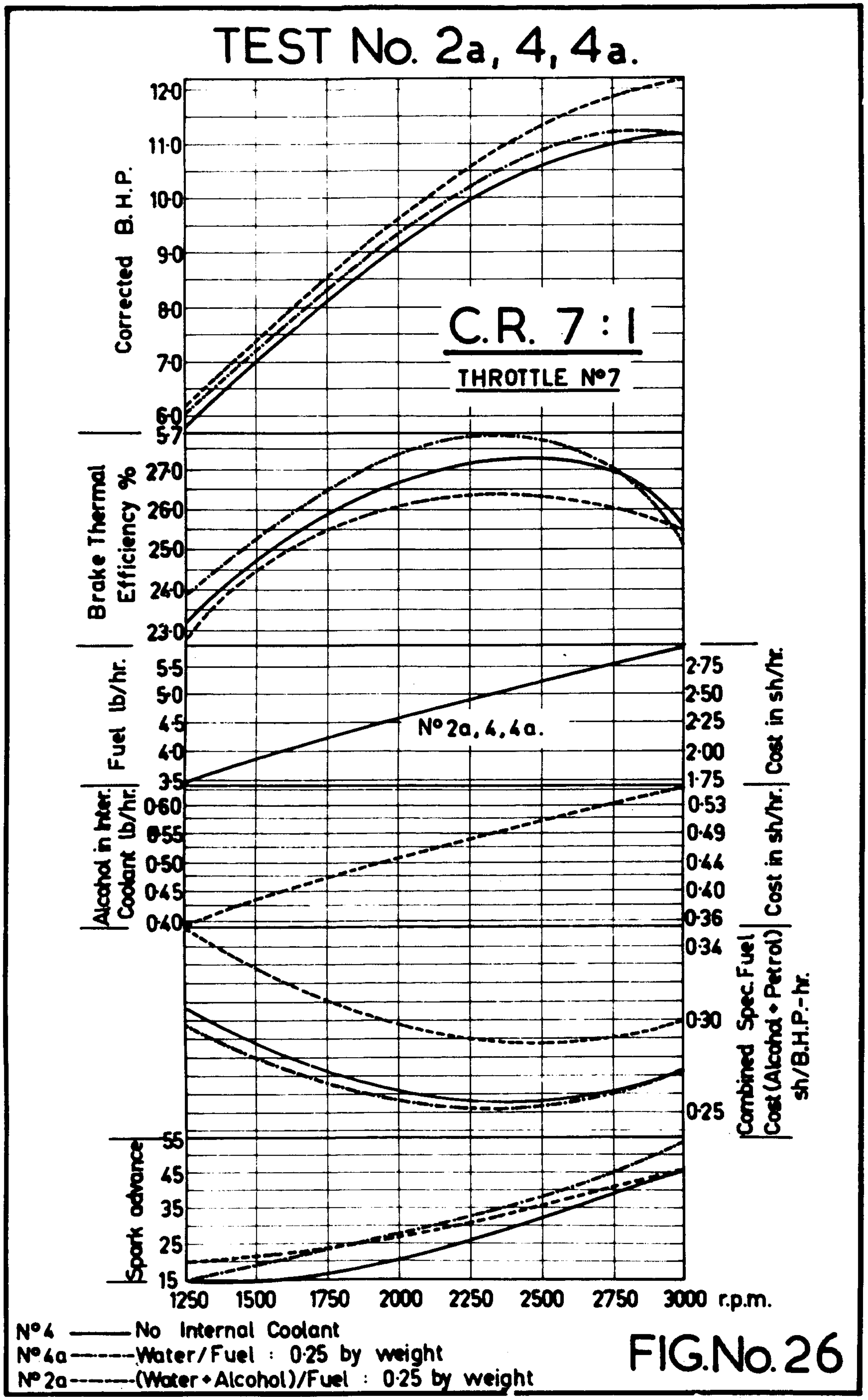

TEST No.2, 2a, 2b.
C.R. 7:1
THROTTLE N°7
Corrected B.H.P.
12·0
11·5
11·0
10·5
10·0
9·5
9·0
8·5
8·0
7·5
7·0
6·5
6·0
Brake Thermal Efficiency %
27·0
26·0
25·0
24·0
23·0
22·0
21·5
Fuel lb/hr.
5·50
5·00
4·50
4·00
3·50
N° 2, 2a, 2b.
Cost in sh/hr.
2·75
2·50
2·25
2·00
1·75
Alcohol in Inter Coolant lb/hr.
1·20
1·00
0·80
0·60
0·40
Cost in sh/hr.
1·10
0·89
0·70
0·55
0·36
Combined Spec. Fuel Cost (Alcohol + Petrol) sh/B.H.P.-hr.
0·40
0·35
0·30
0·25
Spark advance
45
40
35
30
25
20
15
1250
1500
1750
2000
2250
2500
2750
3000 r.p.m.
N°2 —— No Internal Coolant
N°2a ---- (Water+Alcohol)/Fuel : 0·25 by weight
N°2b -·-·- (Water+Alcohol)/Fuel : 0·50 by weight

FIG. No. 24



TEST No. 4, 4a, 4b, 4c.
Corrected B.H.P.
11·0
10·0
9·0
8·0
7·0
6·0
N° 4a, 4b, 4c.
C.R. 7 : 1
THROTTLE N° 7
Brake Thermal Efficiency %
27·0
26·0
25·0
24·0
23·0
N° 4a, 4b.
Fuel lb/hr.
5·5
5·0
4·5
4·0
3·5
N° 4, 4a, 4b, 4c.
Cost in sh/hr.
2·75
2·50
2·25
2·00
1·75
Internal Coolant :- Water
Spec. Fuel Consumption lb/B.H.P.-hr.
0·60
0·58
0·56
0·54
0·52
0·50
Specific Fuel Cost sh/B.H.P.-hr.
0·30
0·29
0·28
0·27
0·26
0·25
Spark advance
50
40
30
20
1250
1500
1750
2000
2250
2500
2750
3000 r.p.m.
N° 4a ------ Water/Fuel : 0·25 by weight
N° 4b —·—·— Water/Fuel : 0·50 by weight
N° 4c —··—··— Water/Fuel : 0·75 by weight
N° 4 ——— No Internal Coolant

FIG. No. 25



TEST No. 2a, 4, 4a.
C.R. 7 : 1
THROTTLE N°7
Corrected B.H.P.
Brake Thermal Efficiency %
Fuel lb/hr.
Cost in sh/hr.
N° 2a, 4, 4a.
Alcohol in Inter. Coolant lb/hr.
Cost in sh/hr.
Combined Spec. Fuel Cost (Alcohol + Petrol) sh/B.H.P.-hr.
Spark advance
1250 1500 1750 2000 2250 2500 2750 3000 r.p.m.
N°4 —— No Internal Coolant
N°4a ------ Water/Fuel : 0·25 by weight
N°2a -·-·-·- (Water + Alcohol)/Fuel : 0·25 by weight

FIG. No. 26

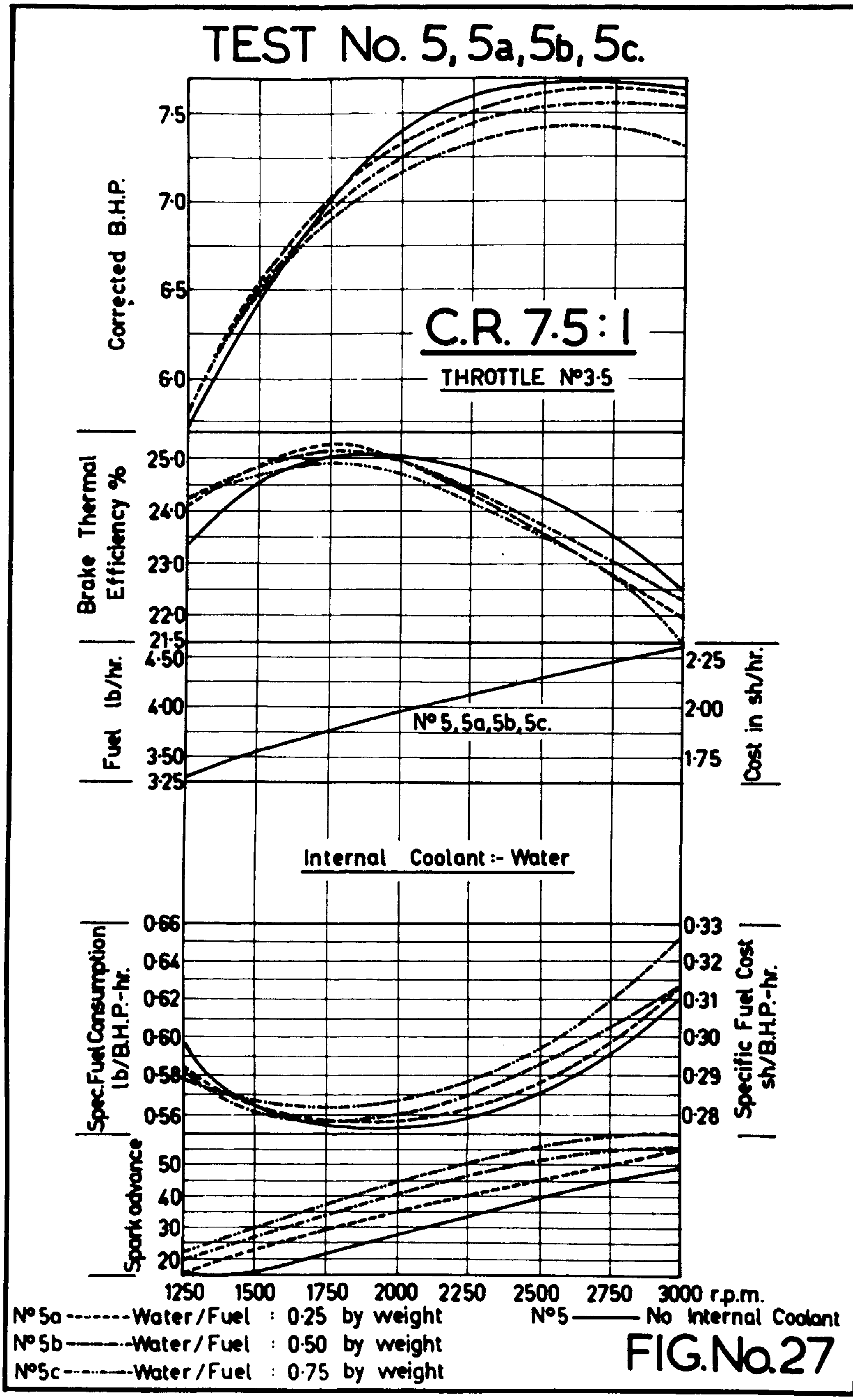

TEST No. 5, 5a, 5b, 5c.
C.R. 7·5 : 1
THROTTLE No 3·5
Corrected B.H.P.
7·5
7·0
6·5
6·0
Brake Thermal Efficiency %
25·0
24·0
23·0
22·0
21·5
Fuel lb/hr.
4·50
4·00
3·50
3·25
No 5, 5a, 5b, 5c.
2·25
2·00
1·75
Cost in sh/hr.
Internal Coolant :- Water
Spec. Fuel Consumption lb/B.H.P.-hr.
0·66
0·64
0·62
0·60
0·58
0·56
0·33
0·32
0·31
0·30
0·29
0·28
Specific Fuel Cost sh/B.H.P.-hr.
Spark advance
50
40
30
20
1250
1500
1750
2000
2250
2500
2750
3000 r.p.m.
No 5a -------Water/Fuel : 0·25 by weight
No 5b ——-——-Water/Fuel : 0·50 by weight
No 5c -·-·-·——Water/Fuel : 0·75 by weight
No 5 ———— No Internal Coolant

FIG. No. 27

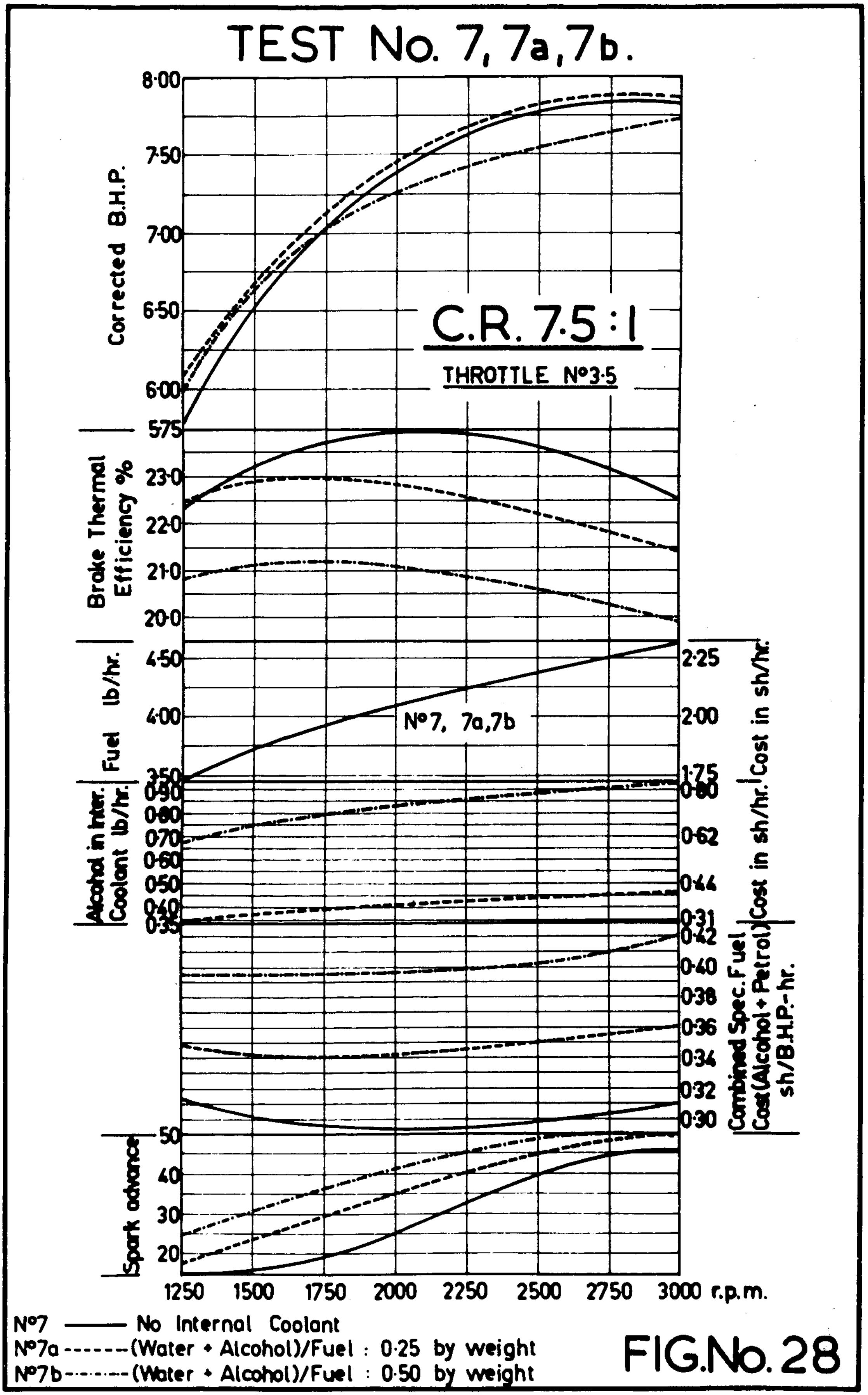

TEST No. 7, 7a, 7b.
Corrected B.H.P.
C.R. 7·5 : 1
THROTTLE N°3·5
Brake Thermal Efficiency %
Fuel lb/hr.
N°7, 7a, 7b
Cost in sh/hr.
Alcohol in Inter. Coolant lb/hr.
Cost in sh/hr.
Combined Spec. Fuel Cost (Alcohol + Petrol) sh/B.H.P.-hr.
Spark advance
1250 1500 1750 2000 2250 2500 2750 3000 r.p.m.
N°7 ——— No Internal Coolant
N°7a ------ (Water + Alcohol)/Fuel : 0·25 by weight
N°7b -·-·-·- (Water + Alcohol)/Fuel : 0·50 by weight

FIG.No. 28

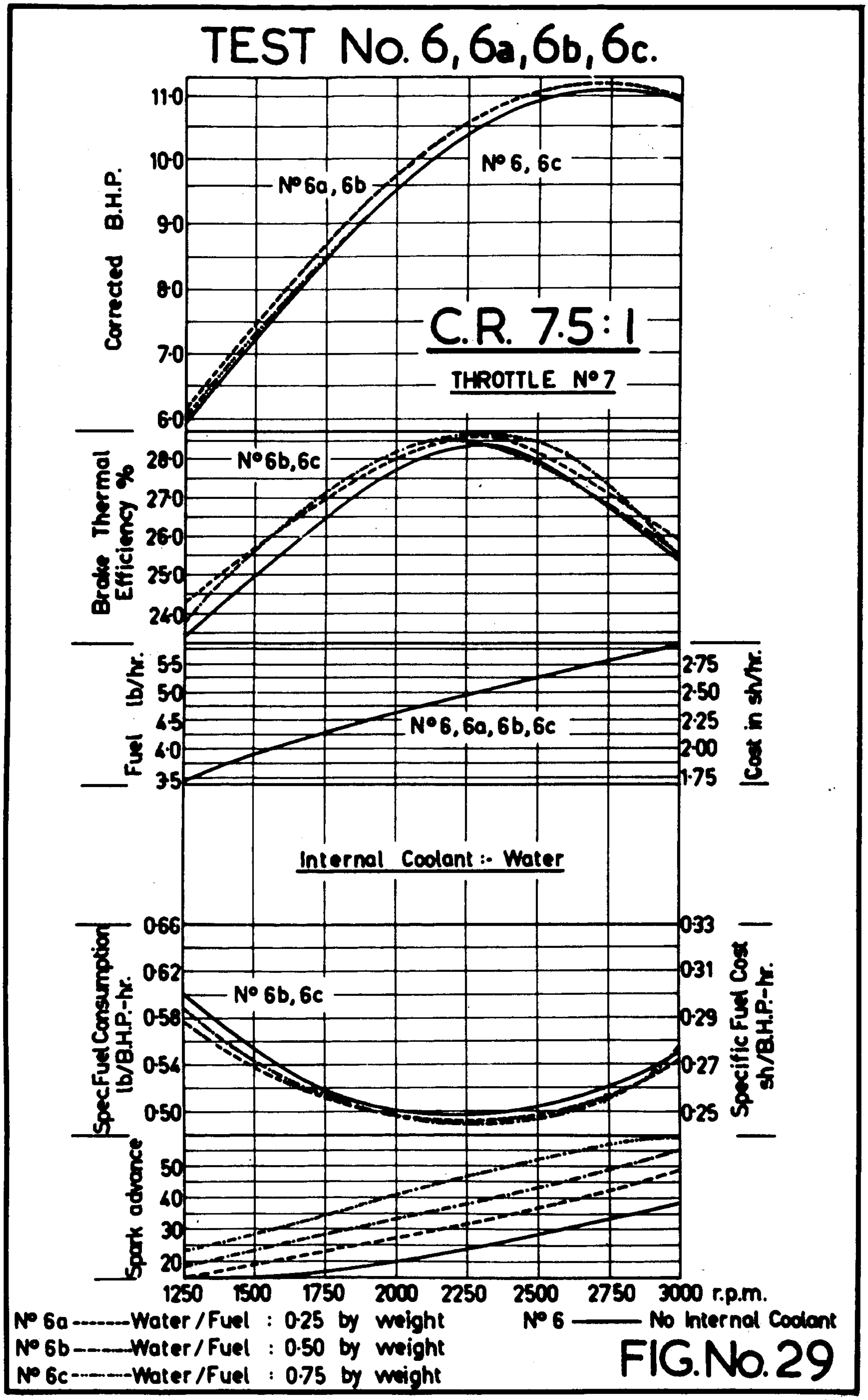

TEST No. 6, 6a, 6b, 6c.
Corrected B.H.P.
11·0
10·0
9·0
8·0
7·0
6·0
Nº 6a, 6b
Nº 6, 6c
C.R. 7·5 : 1
THROTTLE Nº 7
Brake Thermal Efficiency %
28·0
27·0
26·0
25·0
24·0
Nº 6b, 6c
Fuel lb/hr.
5·5
5·0
4·5
4·0
3·5
Cost in sh/hr.
2·75
2·50
2·25
2·00
1·75
Nº 6, 6a, 6b, 6c
Internal Coolant :- Water
Spec. Fuel Consumption lb/B.H.P.-hr.
0·66
0·62
0·58
0·54
0·50
Specific Fuel Cost sh/B.H.P.-hr.
0·33
0·31
0·29
0·27
0·25
Nº 6b, 6c
Spark advance
50
40
30
20
1250 1500 1750 2000 2250 2500 2750 3000 r.p.m.
Nº 6a ------- Water / Fuel : 0·25 by weight
Nº 6b ------- Water / Fuel : 0·50 by weight
Nº 6c ------- Water / Fuel : 0·75 by weight
Nº 6 ——— No Internal Coolant

FIG. No. 29

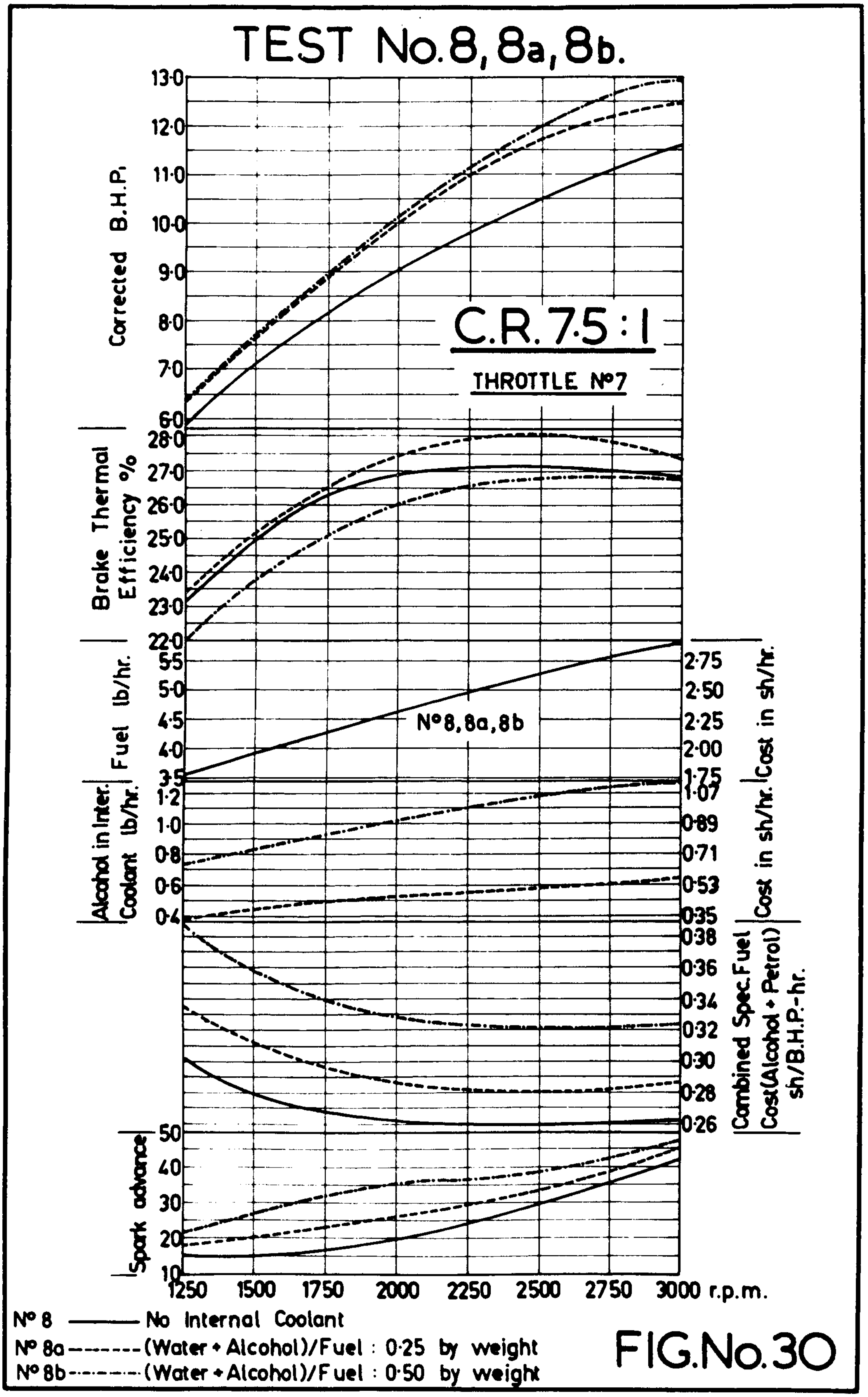
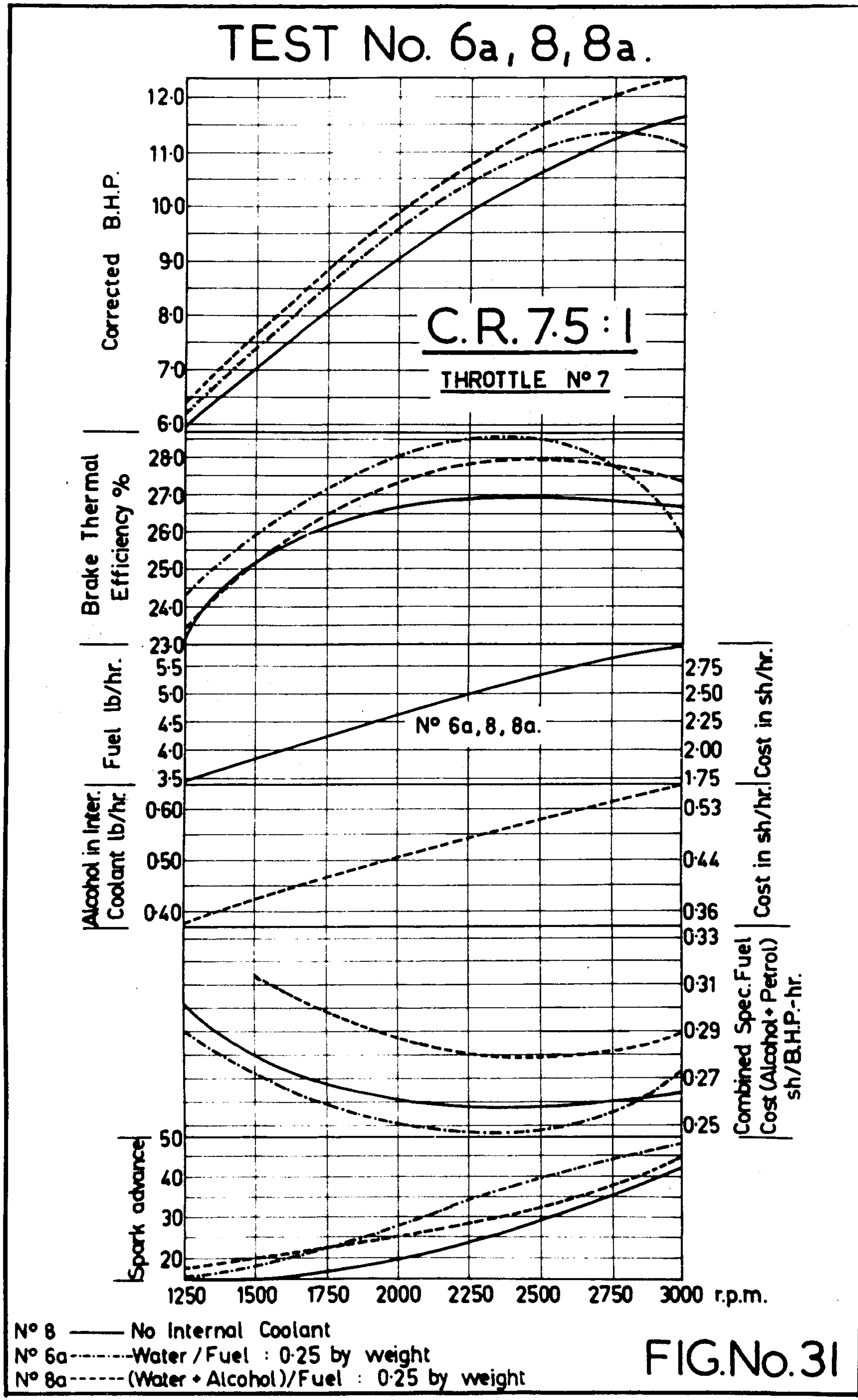

TEST No.8, 8a, 8b.
Corrected B.H.P.
C.R. 7.5 : 1
THROTTLE Nº7
Brake Thermal Efficiency %
Fuel lb/hr.
Cost in sh/hr.
Nº8, 8a, 8b
Alcohol in Inter. Coolant lb/hr.
Cost in sh/hr.
Combined Spec. Fuel Cost (Alcohol + Petrol) sh/B.H.P.-hr.
Spark advance
r.p.m.
Nº 8 ——— No Internal Coolant
Nº 8a ------ (Water + Alcohol)/Fuel : 0·25 by weight
Nº 8b -·-·-·- (Water + Alcohol)/Fuel : 0·50 by weight

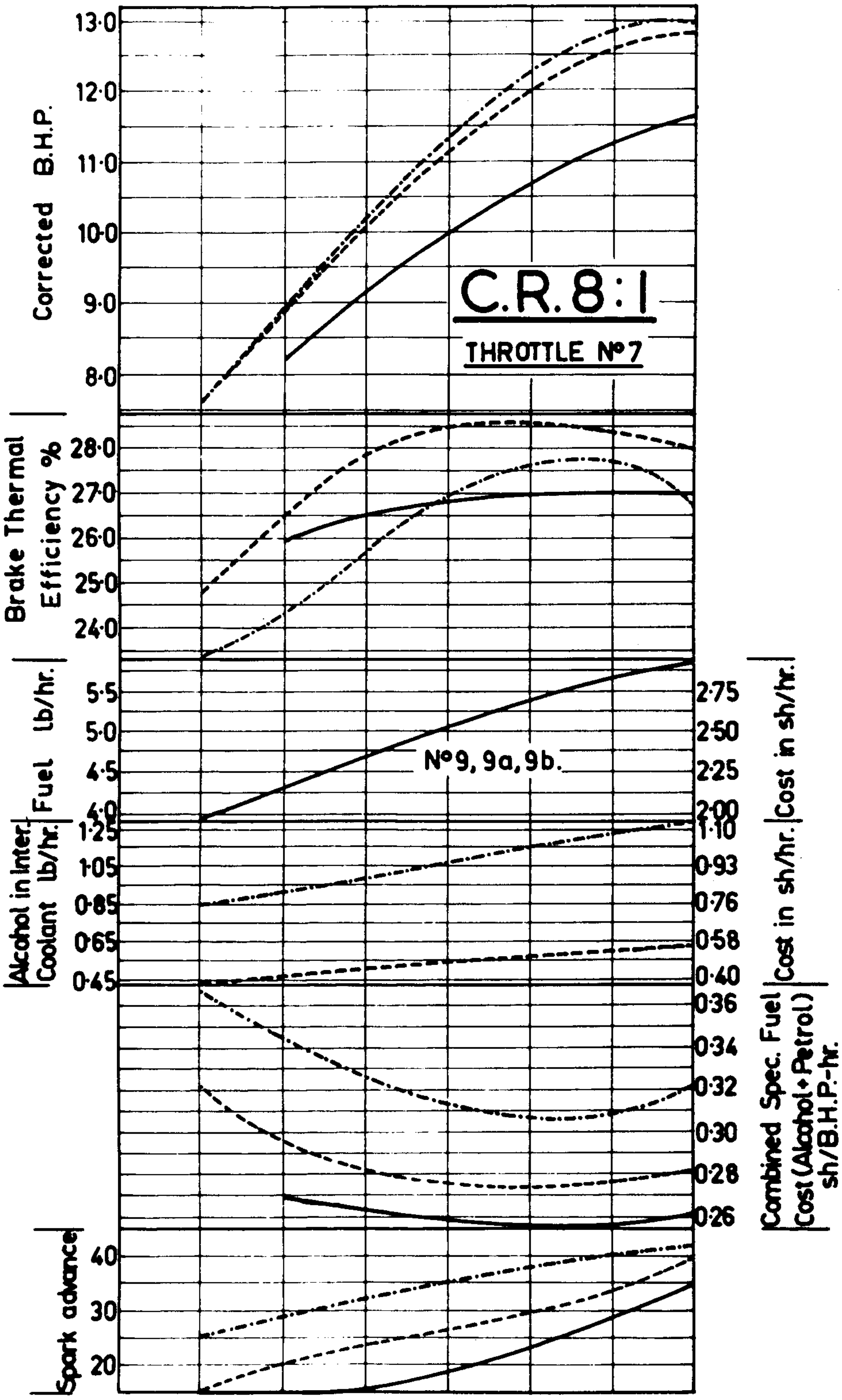
FIG. No. 30



TEST No. 6a, 8, 8a.
Corrected B.H.P.
12·0
11·0
10·0
9·0
8·0
7·0
6·0
C.R. 7·5 : 1
THROTTLE Nº 7
Brake Thermal Efficiency %
28·0
27·0
26·0
25·0
24·0
23·0
Fuel lb/hr.
5·5
5·0
4·5
4·0
3·5
Nº 6a, 8, 8a.
Cost in sh/hr.
2·75
2·50
2·25
2·00
1·75
Alcohol in Inter. Coolant lb/hr.
0·60
0·50
0·40
Cost in sh/hr.
0·53
0·44
0·36
Combined Spec. Fuel Cost (Alcohol + Petrol) sh/B.H.P.-hr.
0·33
0·31
0·29
0·27
0·25
Spark advance
50
40
30
20
1250
1500
1750
2000
2250
2500
2750
3000 r.p.m.
Nº 8 —— No Internal Coolant
Nº 6a -·-·-·- Water / Fuel : 0·25 by weight
Nº 8a ------ (Water + Alcohol)/Fuel : 0·25 by weight

FIG. No. 31

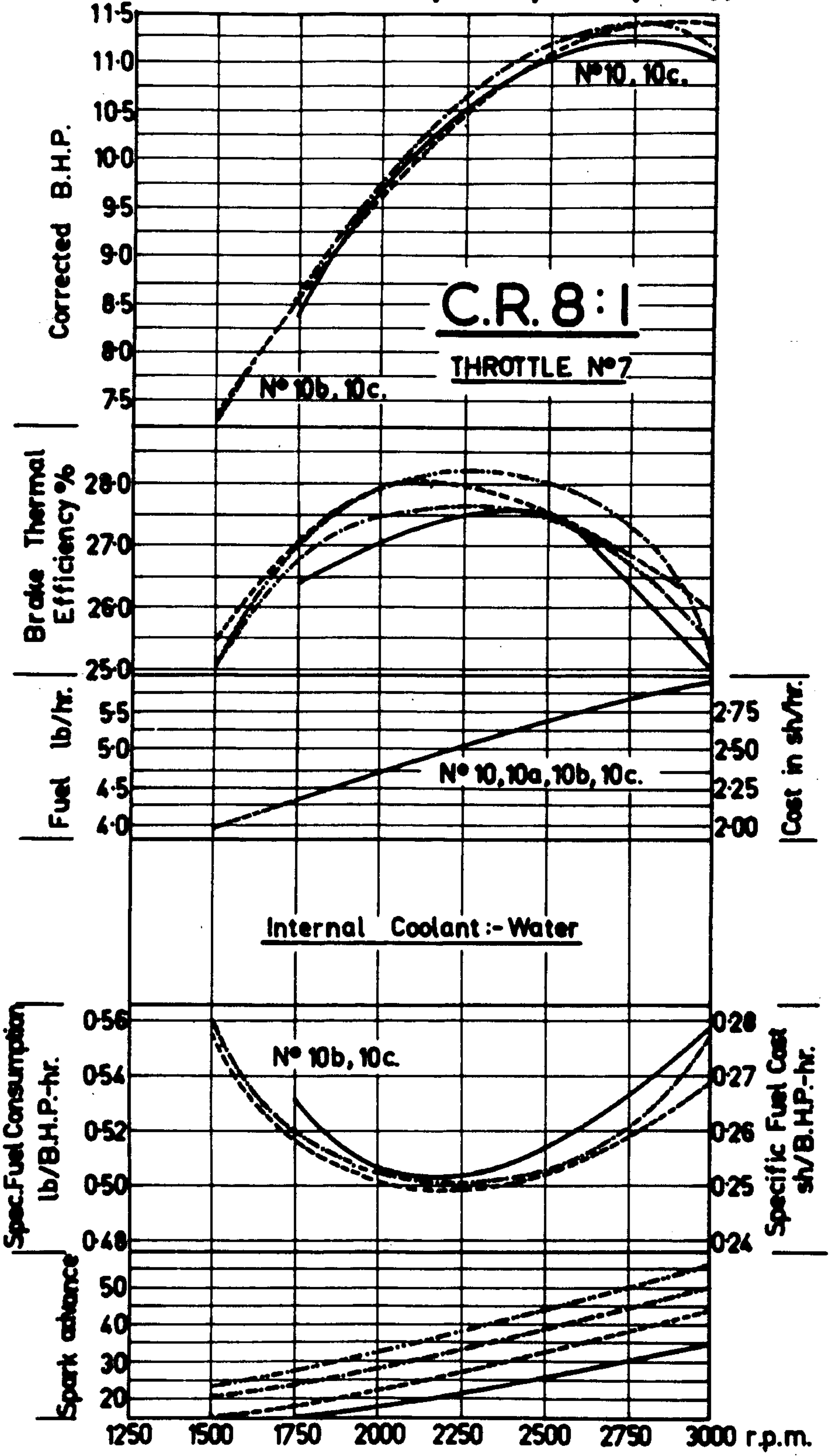
TEST No. 9, 9a, 9b.



Nº 9 ——— No Internal Coolant
Nº 9a ------- (Water + Alcohol)/Fuel : 0·25 by weight
Nº 9b -·-·-·- (Water + Alcohol)/Fuel : 0·50 by weight

FIG. No. 32

TEST No. 10, 10a, 10b, 10c.



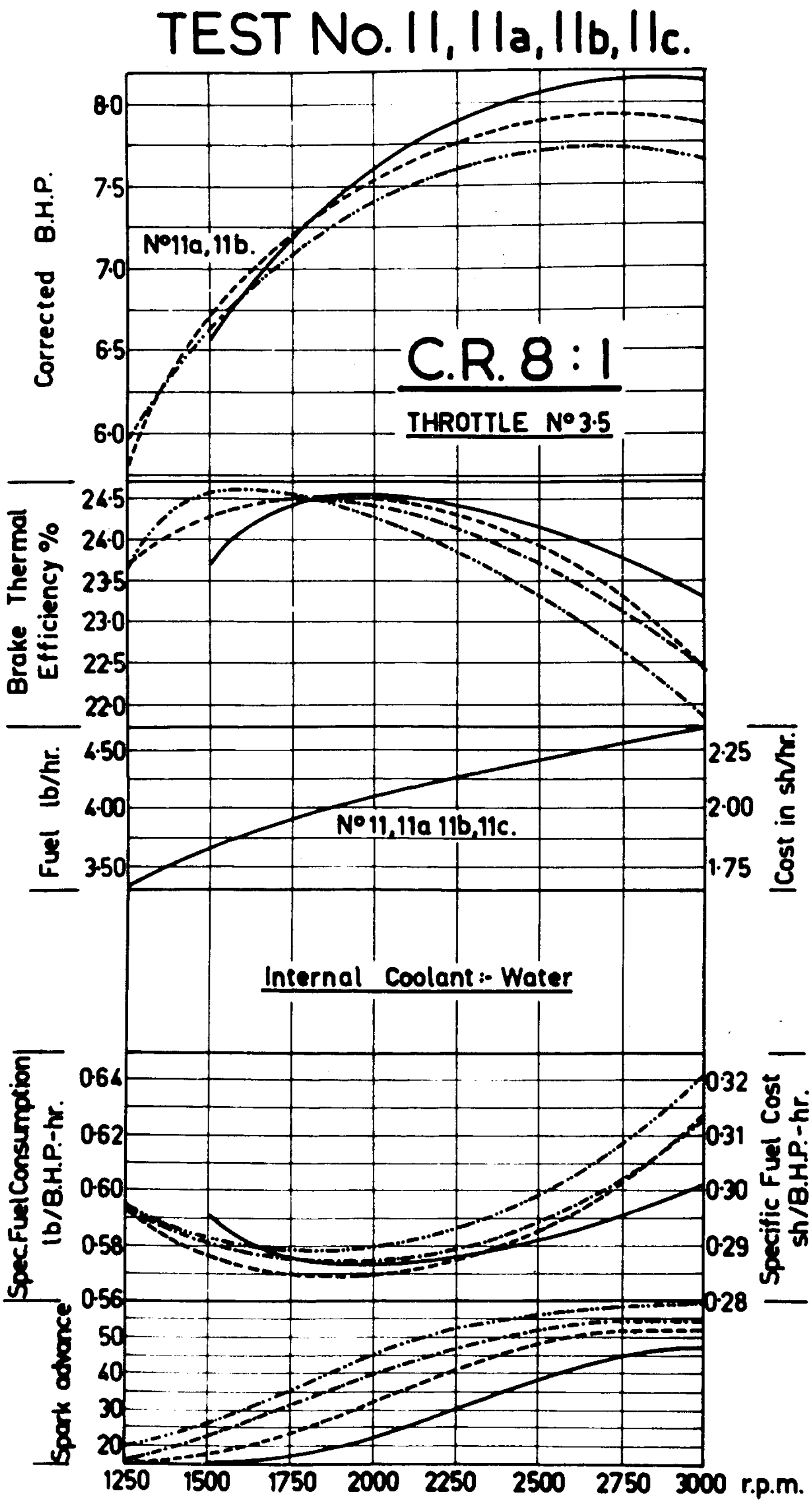
Nº 10a ------- Water/Fuel : 0·25 by weight

Nº 10b -·-·-·- Water/Fuel :- 0·50 by weight

Nº 10c -··-··- Water/Fuel :- 0·75 by weight

Nº 10 ——— No Internal Coolant

FIG. No. 33



N° 11a -------- Water / Fuel : 0·25 by weight
N° 11b -·-·-·--- Water / Fuel : 0·50 by weight
N° 11c -··-··--- Water / Fuel : 0·75 by weight
N° 11 ——— No Internal Coolant

FIG. No. 34

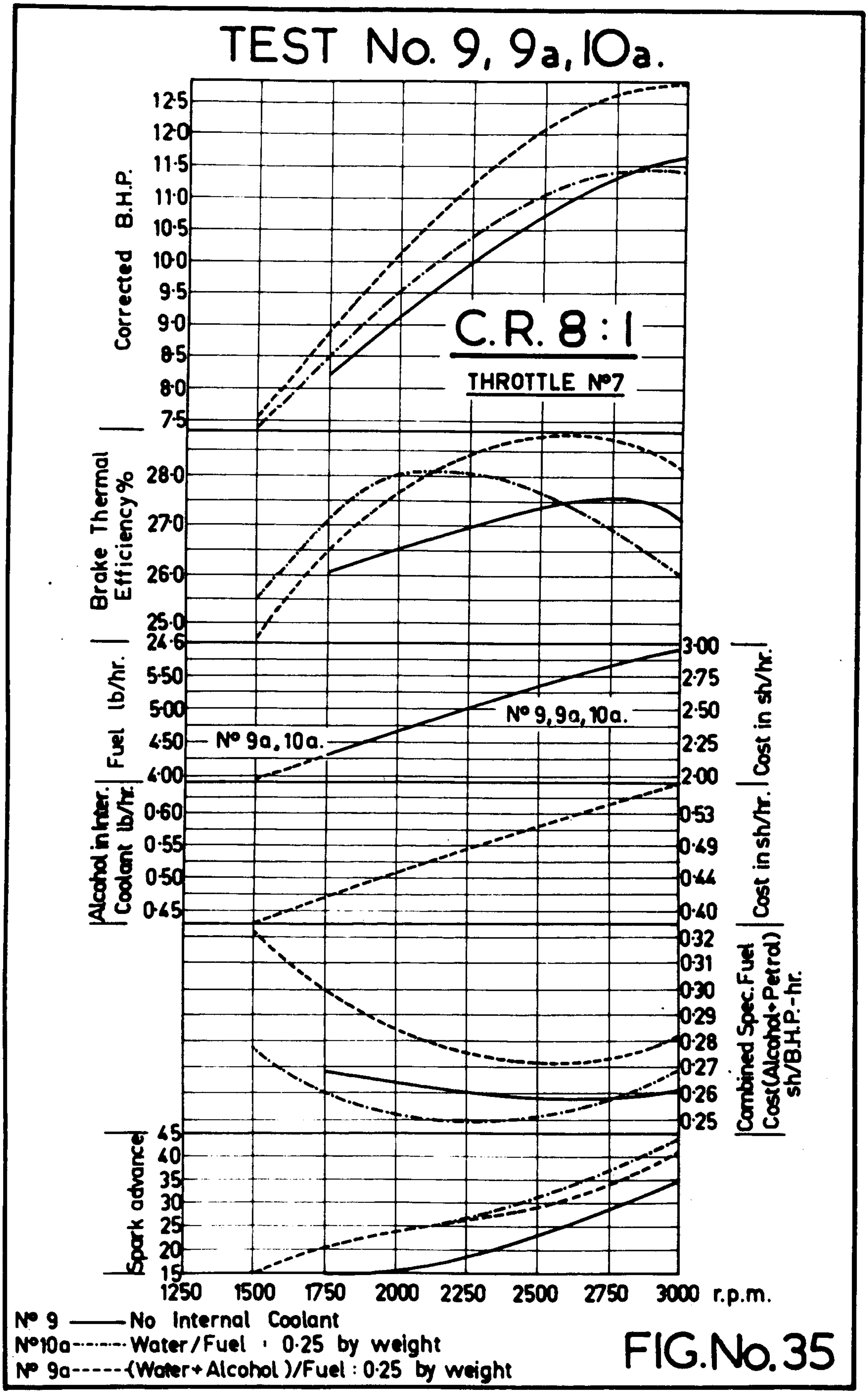

TEST No. 9, 9a, 10a.
Corrected B.H.P.
12·5
12·0
11·5
11·0
10·5
10·0
9·5
9·0
8·5
8·0
7·5
C.R. 8 : 1
THROTTLE Nº7
Brake Thermal Efficiency %
28·0
27·0
26·0
25·0
24·6
Fuel lb/hr.
5·50
5·00
4·50
4·00
Nº 9a, 10a.
Nº 9, 9a, 10a.
Cost in sh/hr.
3·00
2·75
2·50
2·25
2·00
Alcohol in Inter. Coolant lb/hr.
0·60
0·55
0·50
0·45
Cost in sh/hr.
0·53
0·49
0·44
0·40
Combined Spec. Fuel Cost (Alcohol+Petrol) sh/B.H.P.-hr.
0·32
0·31
0·30
0·29
0·28
0·27
0·26
0·25
Spark advance
45
40
35
30
25
20
15
1250
1500
1750
2000
2250
2500
2750
3000 r.p.m.
Nº 9 ——— No Internal Coolant
Nº10a -·-·-·- Water/Fuel : 0·25 by weight
Nº 9a - - - - - (Water + Alcohol)/Fuel : 0·25 by weight

FIG. No. 35

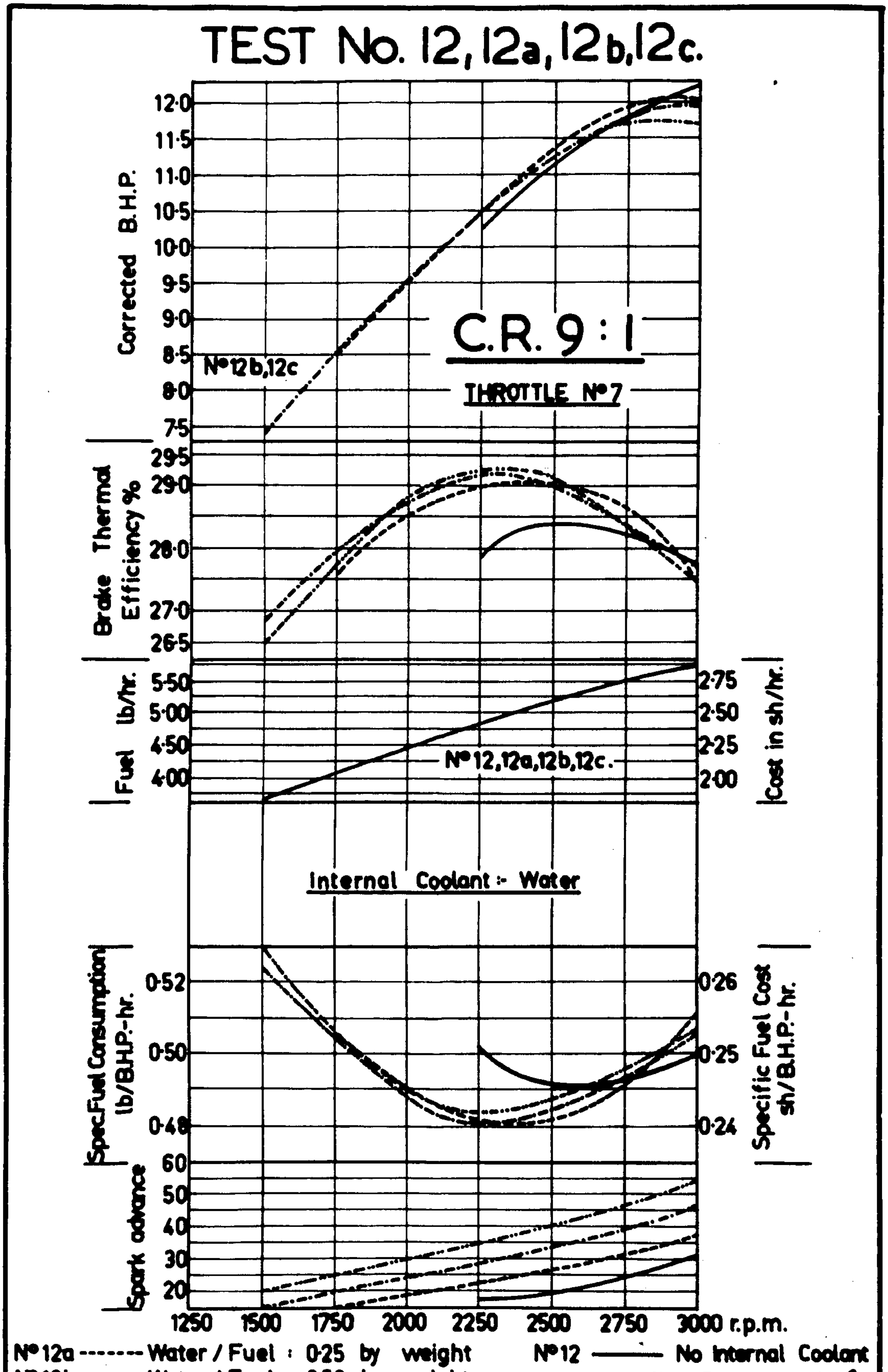
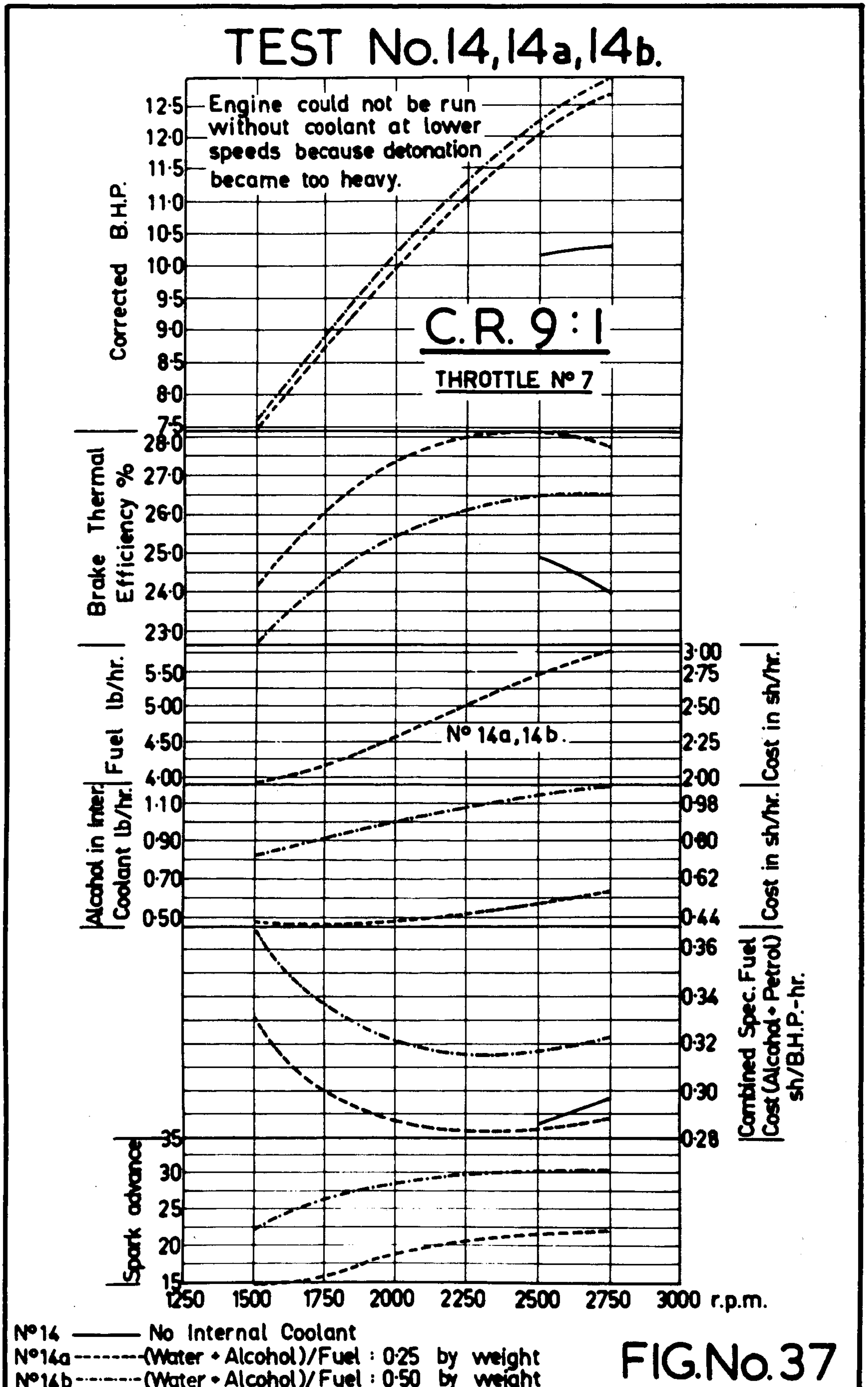

TEST No. 12, 12a, 12b, 12c.
Corrected B.H.P.
12·0
11·5
11·0
10·5
10·0
9·5
9·0
8·5
8·0
7·5
Nº 12b, 12c
C.R. 9 : 1
THROTTLE Nº 7
Brake Thermal Efficiency %
29·5
29·0
28·0
27·0
26·5
Fuel lb/hr.
5·50
5·00
4·50
4·00
Cost in sh/hr.
2·75
2·50
2·25
2·00
Nº 12, 12a, 12b, 12c.
Internal Coolant :- Water
Spec. Fuel Consumption lb/B.H.P.-hr.
0·52
0·50
0·48
Specific Fuel Cost sh/B.H.P.-hr.
0·26
0·25
0·24
Spark advance
60
50
40
30
20
1250
1500
1750
2000
2250
2500
2750
3000 r.p.m.
Nº 12a ------ Water / Fuel : 0·25 by weight
Nº 12b -·-·-·- Water / Fuel : 0·50 by weight
Nº 12c -··-··- Water / Fuel : 0·75 by weight
Nº 12 ——— No Internal Coolant

FIG. No. 36



TEST No. 14, 14a, 14b.
Engine could not be run without coolant at lower speeds because detonation became too heavy.
C.R. 9:1
THROTTLE N° 7
Corrected B.H.P.
Brake Thermal Efficiency %
Fuel lb/hr.
N° 14a, 14b.
Cost in sh/hr.
Alcohol in Inter. Coolant lb/hr.
Cost in sh/hr.
Combined Spec. Fuel Cost (Alcohol + Petrol) sh/B.H.P.-hr.
Spark advance
1250 1500 1750 2000 2250 2500 2750 3000 r.p.m.
N° 14 ——— No Internal Coolant
N° 14a - - - - - (Water + Alcohol)/Fuel : 0·25 by weight
N° 14b -·-·-·- (Water + Alcohol)/Fuel : 0·50 by weight

FIG. No. 37

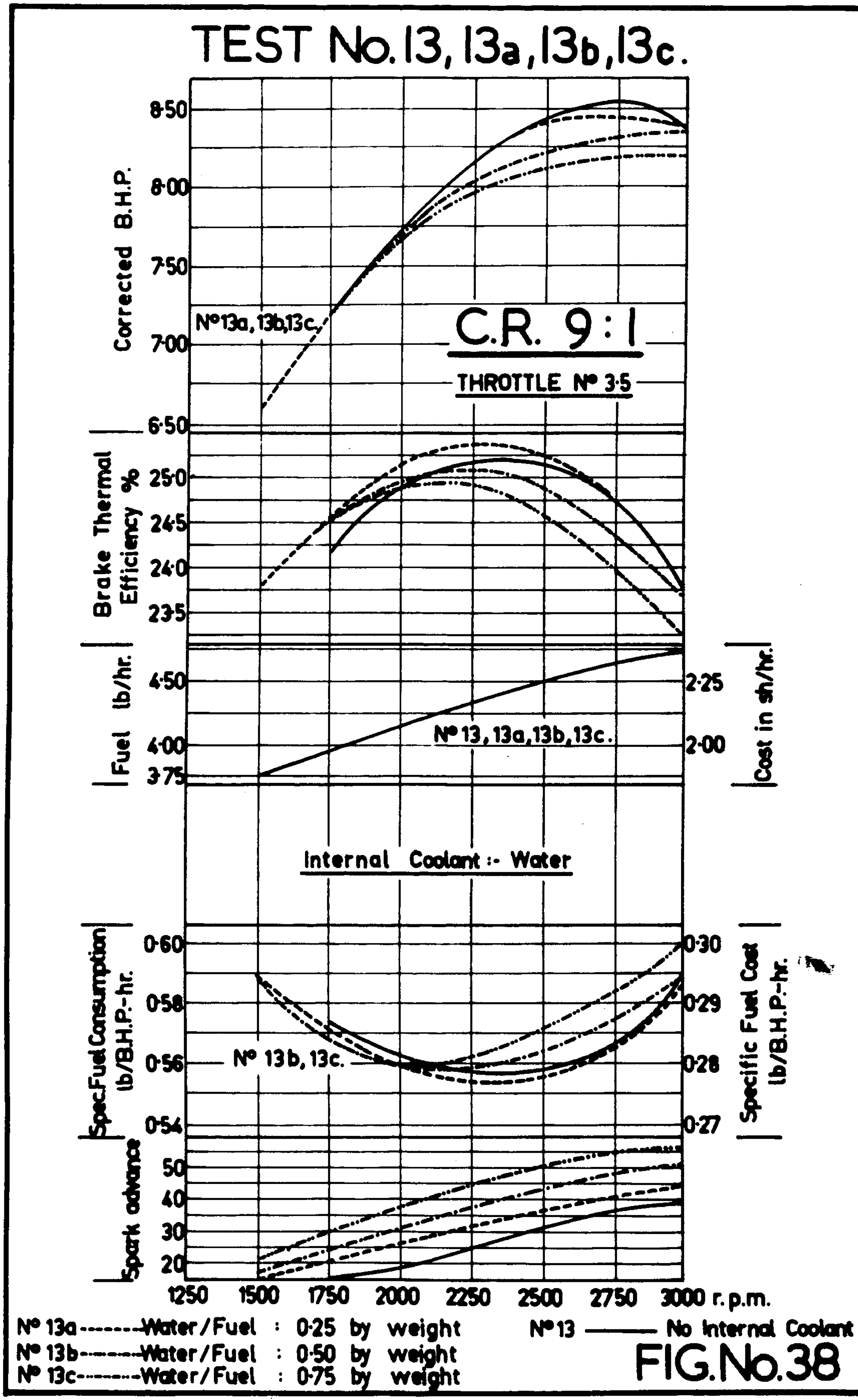

TEST No. 13, 13a, 13b, 13c.
Corrected B.H.P.
8·50
8·00
7·50
7·00
6·50
N° 13a, 13b, 13c.
C.R. 9 : 1
THROTTLE N° 3·5
Brake Thermal Efficiency %
25·0
24·5
24·0
23·5
Fuel lb/hr.
4·50
4·00
3·75
2·25
2·00
Cost in sh/hr.
N° 13, 13a, 13b, 13c.
Internal Coolant :- Water
Spec. Fuel Consumption lb/B.H.P.-hr.
0·60
0·58
0·56
0·54
0·30
0·29
0·28
0·27
Specific Fuel Cost lb/B.H.P.-hr.
N° 13b, 13c.
Spark advance
50
40
30
20
1250
1500
1750
2000
2250
2500
2750
3000 r.p.m.
N° 13a ------- Water/Fuel : 0·25 by weight
N° 13b -·-·-·- Water/Fuel : 0·50 by weight
N° 13c -··-··- Water/Fuel : 0·75 by weight
N° 13 ——— No Internal Coolant

FIG. No. 38

TEST No. 12, 12 a, 14 a.

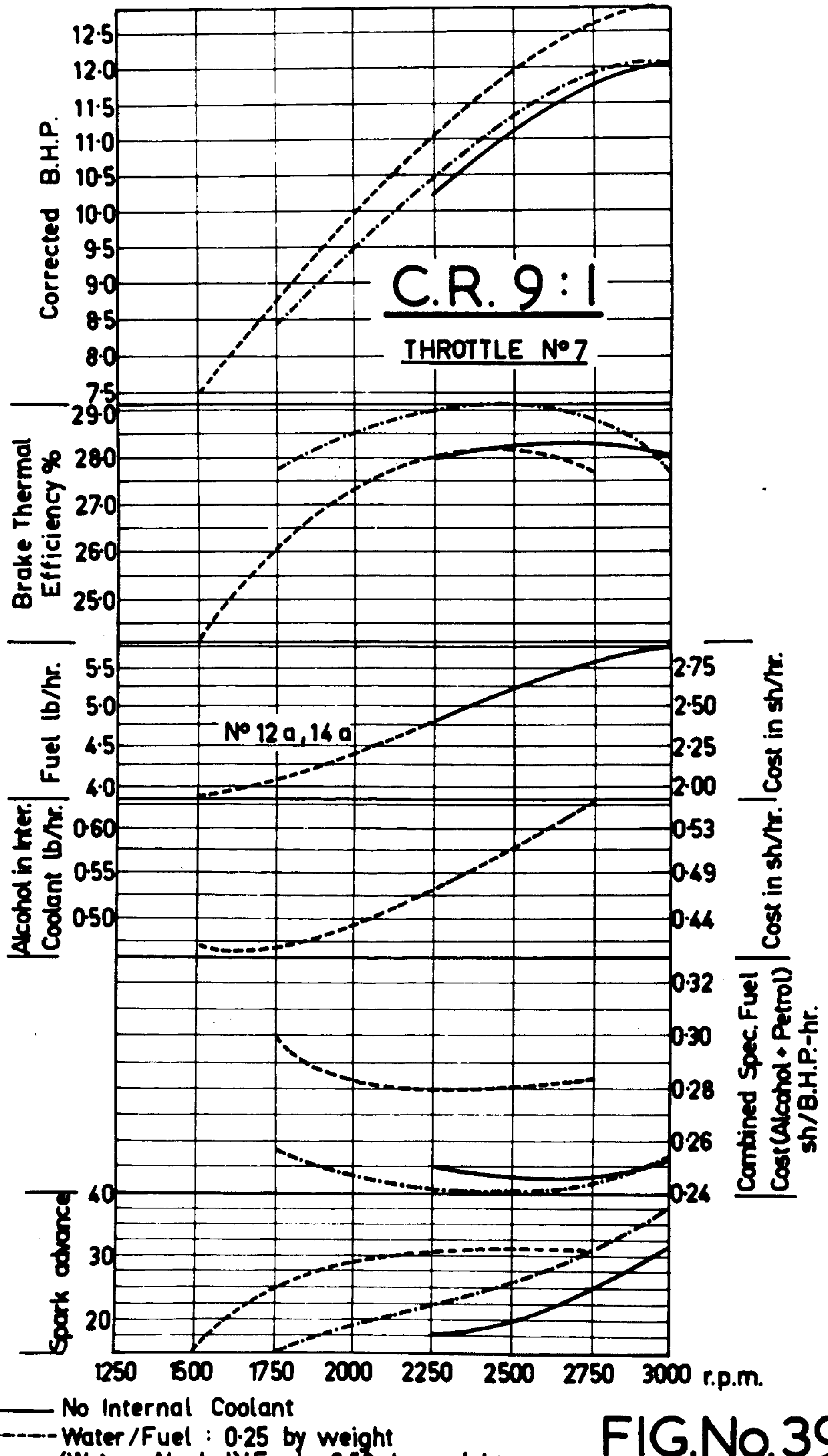


FIG. No. 39

4 - 6. Accuracy of test results.

Calibration of the dynamometer and of other test equipment had been previously carried out by Mr. H. Vainomae, as part of a B.E. degree research assignment under the author's direction.

He found that the friction in the trunnion bearings of the dynamometer was negligible and, therefore, no correction for brake horsepower calculations was necessary in this respect. On the other hand, torque errors due to the ventilating air-flow through the dynamometer housing had to be taken into account. The correction factors for torque, which increased over the speed range, were used for the calculation of power output.

The Budenberg Tachometer, used on the Ricardo engine, was calibrated by Mr. Vainomae against a Smith's Tachometer, which had been calibrated at the National Standards Laboratory (C.S.I.R.O.). The values of the first tachometer, when compared with the true speed values, varied between 0 and plus 8 r.p.m. over the major part of the speed range. As these variations are smaller than the probable reading errors, corrections for speed variation were neglected.

Spark advance values, marked in crank angle degrees on the quadrant of the magneto, were checked against the graduation on the periphery of the flywheel. The readings on the quadrant were found to be consistently

$1\frac{1}{2}^{\circ}$ less than the true spark advance.

Finally, the 10 lb. spring balance of the dynamometer was checked periodically by the author and adjustments made to bring the pointer back to zero when unloaded. After the conclusion of the water-injection tests, the pointer showed readings about plus 0.05 lb. unloaded.

Taking into account the corrections derived from the calibration tests, the accuracy of the test results depends on the precision of the various readings and on the influence of engine variables, such as the temperature of the cooling water discharged and the temperature of the lubricating oil.

In the chapter dealing with "Test procedure", it was pointed out that the engine conditions should be stabilised before taking readings. By maintaining stable engine conditions, balance and tachometer values showed practically no variation during each of the observation periods.

To indicate the accuracy of the test results, the maximum errors and the percentage errors are calculated for one of the full-throttle and one of the half-throttle tests as follows:

1. Calculation of the maximum and percentage errors in brake horsepower.

The relevant data were taken from observations during tests Nos. 5 and 6.

Observed brake net force

$$W = Q - S$$

Speed r.p.m.	Test No.5, C.R.7.5:1, Throttle No.3.5	Test No.6, C.R.7.5:1, Throttle No.7
1250	15.30 = 20 - 4.70	15.85 = 20 - 4.15
1500	14.45 = 20 - 5.55	16.30 = 20 - 3.70
1750	13.40 = 20 - 6.60	16.00 = 20 - 4.00
2000	12.20 = 20 - 7.80	15.75 = 20 - 4.25
2250	11.15 = 15 - 3.85	15.50 = 20 - 4.50
2500	9.90 = 15 - 5.10	14.45 = 20 - 5.55
2750	9.05 = 15 - 5.95	13.25 = 20 - 6.75
3000	8.15 = 15 - 6.85	11.90 = 15 - 3.10

Table No. 20.

In the above formula " Q " represents the brake anchor load and " S " the spring balance reading in pounds. For the brake anchor loads of 20 lb., an error of $\pm$ 0.1 may be estimated and for 15 lb. a corresponding error of $\pm$ 0.075 lb.

The error for the spring balance readings, in the case of a single observation, may be taken as $\pm$ 0.05 lb, which is equal to the smallest division.

Finally, the error for the tachometer readings " N " was estimated as $\pm$ 10 r.p.m., which represents one fifth of the smallest division.

In respect to engine speeds of 1250 and 3000 r.p.m., the following maximum errors can be tabulated for tests Nos. 5 and 6:

Speed r.p.m.	Test No.5, C.R.7.5:1, Throttle No.3.5	Test No.6, C.R.7.5:1, Throttle No.7
1250	Q = (20 ± 0.1) lb.	Q = (20 ± 0.1) lb.
"	S = (4.70 ± 0.05) lb.	S = (4.15 ± 0.05) lb.
"	N = (1250 ± 10)r.p.m.	N = (1250 ± 10)r.p.m.
3000	Q = (15 ± 0.075) lb.	Q = (15 ± 0.075) lb.
"	S = (6.85 ± 0.05) lb.	S = (3.10 ± 0.05) lb.
"	N = (3000 ± 10)r.p.m.	N = (3000± 10)r.p.m.

Table No. 21.

Thus the maximum and percentage errors of "W" are:

Test No.	Speed r.p.m.	Maximum Error	% Error
5	1250	0.1+0.05=0.15 lb.	$\frac{0.15}{15.30} \times 100 = 1\%$
5	3000	0.075+0.05=0.125 lb.	$\frac{0.125}{8.15} \times 100 = 1.5\%$
6	1250	0.1+0.05=0.15 lb.	$\frac{0.15}{15.85} \times 100 = 1\%$
6	3000	0.075+0.05=0.125 lb.	$\frac{0.125}{11.90} \times 100 = 1\%$

Table No. 22.

The percentage errors for the corresponding speeds are: $\frac{10}{1250} \times 100 = 0.8$, say 1% and

$\frac{10}{3000} \times 100 = 0.33$, say 0.5%

As the maximum percentage error of the brake horsepower is the sum of the individual errors, it can be tabulated as:

$$B.H.P. = \frac{W \cdot N}{3500}$$

Test No.	Speed r.p.m.	Maximum % Error
5	1250	1 + 1 = 2 %
5	3000	1.5 + 0.5 = 2 %
6	1250	1 + 1 = 2 %
6	3000	1 + 0.5 = 1.5 %

Table No. 23.

The probable errors, however, will be less than the maximum errors and will not be more than 1.5%.

2) Calculation of the maximum and percentage errors for the fuel consumption.

The time for the consumption of 50 ml. petrol was found by a stopwatch. Several readings were made at each speed and the mean values calculated.

Test 5 furnished the following readings: 89.25, 89.50, 90.3, 90.8 sec. at 1250 r.p.m. and 64.25, 64.30, 64.50, 64.0 sec. at 3000 r.p.m.

The corresponding mean values are 89.96 and 64.25 sec. and the maximum errors therefore:

90.80 - 89.96 = <u>0.84</u> sec. and 64.50 - 64.25 = <u>0.25</u> sec.

The maximum percentage error, therefore, is:

$$\frac{0.84}{89.96} \times 100 = \underline{1\%} \quad \text{at 1250 r.p.m. and}$$

$$\frac{0.25}{64.25} \times 100 = \underline{0.5\%} \text{ at 3000 r.p.m.}$$

The probable errors will again be smaller than the above values.

The calculated values of the hourly petrol consumption will include errors of the above magnitude.

Errors of the same order may be expected in the consumption time of the water and water-alcohol.

3) Influence of different oil temperatures on output figures.

To investigate the influence of different oil temperatures on the developed brake horsepower, a number of brake load readings were made during a full-throttle test at 2000 r.p.m. Every engine variable was kept constant with the exception of the oil temperature, which was varied by changing the rate of cooling water through its own heat exchanger. Brake load readings were taken at 60, 65 and 70°F oil temperature, while water outlet temperature was kept at 70°F.

The three readings showed negligible differences of output and as all tests with water-injection were run at oil temperatures between 65° and 73°F, correction for variations in oil temperatures were neglected.

4 - 7. Internal condition of the engine.

To investigate the influence of water or water-alcohol injection on the formation of carbon deposits and valve life, the cylinder head of the Ricardo engine was removed before the main tests commenced and the amount of carbon on the piston crown and liner top were recorded by taking photographs. (Figs. 40 and 41)

The cylinder head then was thoroughly decarbonized and the valves were ground and lapped. The carbon deposits on the piston crown and liner top, however, were left untouched and the engine reassembled. This procedure was adopted to investigate claims that internal coolants, when injected, will remove existing carbon deposits after a certain running time.

The main tests then were carried out and the engine was in operation for over 170 hours. After this period, the cylinder head was removed again and a close investigation of the internal condition of the engine conducted.

It was found that the piston crown was practically free of carbon, which proved that the carbon layer, purposely left on the crown, was disposed of and no other carbon deposited during the 170 hours running. The liner top and the cylinder head were also extremely clean. The exhaust valve, although perfectly clean on the head, showed a slight amount of deposits on the face, but otherwise was in very good condition.

This was an important result, taking into account the fact that the engine was run on full throttle for about 60% of the test period, the remainder being on half throttle.

After the tests, for purpose of comparison, two more photographs were taken, Figs. 42 and 43, one showing the piston at top dead centre and the other the combustion chamber, immediately the head had been removed.

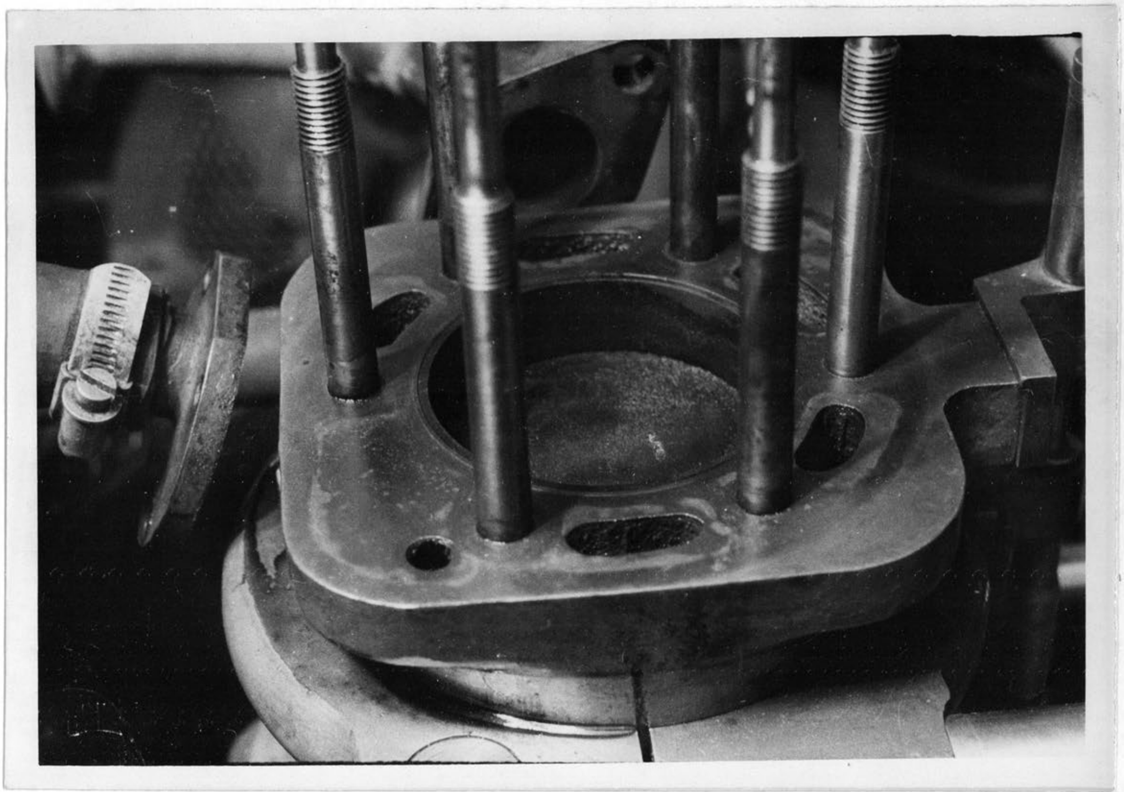


Fig. 40 - Carbon deposits on piston crown and liner top before main tests commenced.

Fig. 41 - Carbon deposits on piston crown before main tests commenced.

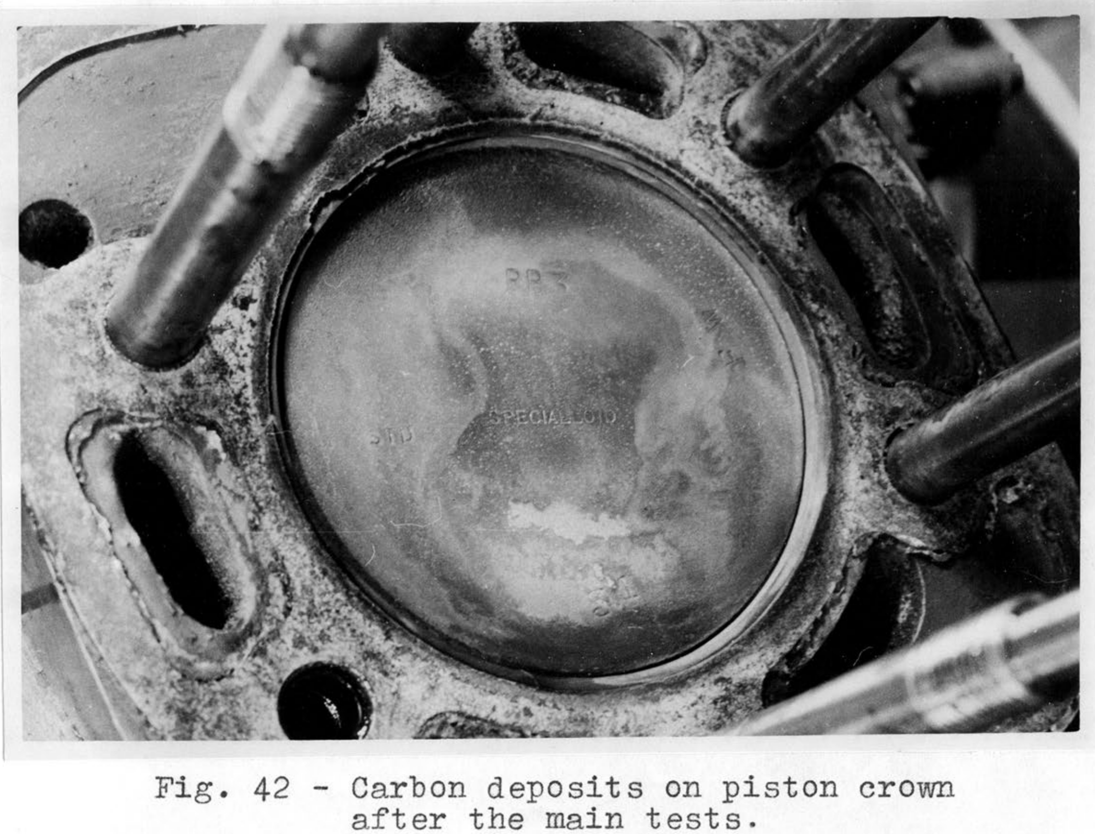


Fig. 42 - Carbon deposits on piston crown after the main tests.

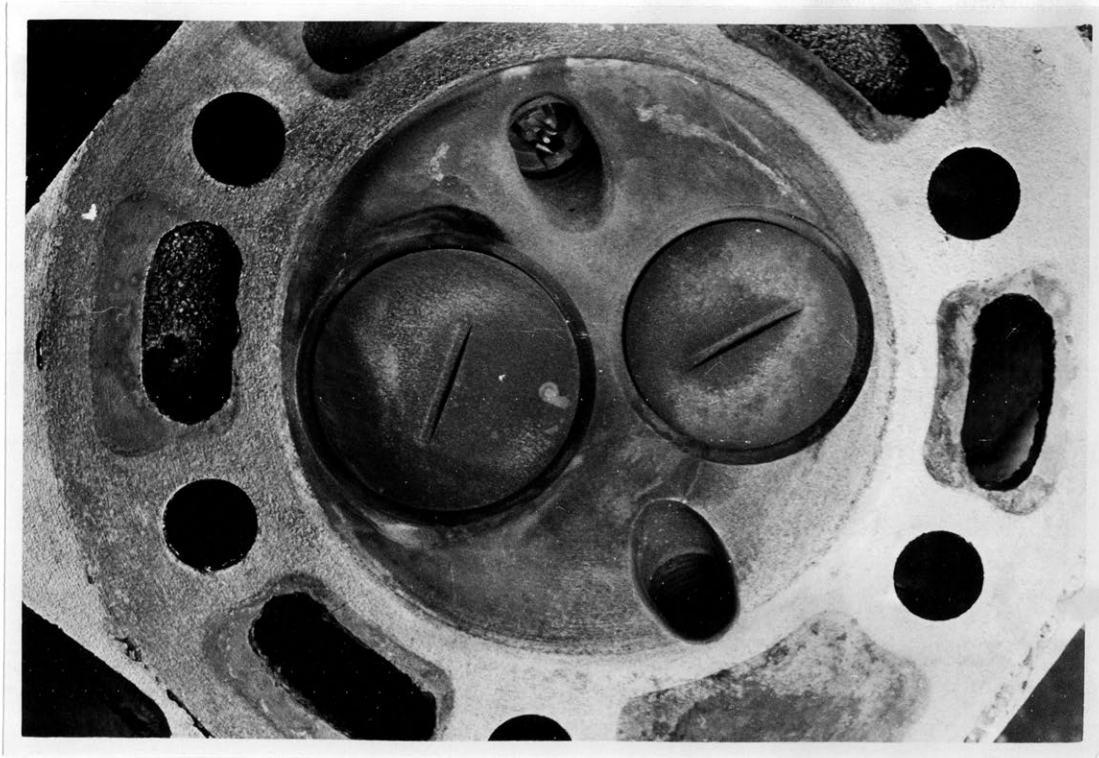


Fig. 43 - Combustion chamber, practically free of carbon deposits, after the main test.

4 - 8. Reduction in Mean Cycle Temperature, when using Water as an Internal Coolant.

After completion of the thesis, a suggestion was made by Professor J.F.D. Wood that the reduction in the mean cycle temperature of a petrol engine, when using water as an internal coolant, should be measured as a means of comparing the drop in maximum cycle temperature, as calculated in chapter 3.

For this purpose, the author conducted a number of additional tests on the Ricardo E-6/S variable compression engine. The spark ignition cylinder head of this engine contains two tapped openings, each of them may be used to accomodate a spark plug or a pick-up for an indicator. It was contemplated to remove the pick-up and to insert instead a thermocouple capable of measuring the desired temperatures.

The problem which confronted the author was to obtain a thermocouple which could be sealed in the combustion chamber and would stand up to the high pressures and temperatures of the combustion gases. Two ideas were followed up. The first one was to use a standard spark plug with specially insulated chromel-alumel wires cemented in the porcelain instead of the centre electrode. S. Smith and Sons (Aust) Pty. Ltd. took up this idea and provided the author with modified spark plugs. The thermocouple wires used were fibre-glass insulated and the ends were fused about three quarters of an inch from the point where they penetrated the porcelain. The second idea was to procure a standard shielded thermocouple and to machine the body of a Ricardo spark plug to provide room for such a thermocouple. The necessary work was carried out in the workshop of the School of Mechanical Engineering and a commercial thermocouple was brazed into the spark plug to form a gas-tight joint.

Preliminary tests were carried out with both types of thermocouples at identical engine settings, but it was found that the readings taken with the thermocouples made by S. Smith and Sons were much higher than with the second unit. The difference in readings may have resulted from the fact that the tube carrying the thermocouple wires of type two did not have room to protrude sufficiently deep into the combustion chamber to give correct readings.

It was, therefore, decided to use the thermocouples made by S. Smith and Sons and two tests were carried out; one at a compression ratio 7:1 and the second at 7.5:1. Temperature readings were taken at speeds, ranging from 1500 to 2750 r.p.m., first without internal coolant and then with water added at weight ratios 0.25 and 0.50. The spark was adjusted at each point of the speed range to give maximum output without detonating of the engine and the spark remained at this setting when water was injected. This procedure was carried out to obtain mean cycle temperature readings under conditions similar to those assumed for the calculation of the maximum cycle temperature in chapter 3.

In Fig.40, mean cycle temperatures of the engine, when running at a compression ratio 7:1 and a throttle opening No. 7, were plotted against engine speed. It may be seen that water as an anti-detonant when injected at the weight ratio 0.25, is responsible for a temperature drop of about 40°C which is practically constant over the speed range. Water when injected at the weight ratio 0.50, reduces the mean cycle temperature by amounts ranging between 60°C and 75°C according to engine speed. Similar results were obtained from the test at a compression ratio 7.5:1.

Finally, a test was carried out to investigate the influence of an additional spark advance on the mean cycle temperature with water injection. In this test, spark timings were set at the same values as shown on graph No. 25. This test proved that by increasing the spark advance until trace knocking occurred, the mean cycle temperatures remained the same as obtained when no water was used. Thus the increase in mean cycle temperature, to be anticipated by further spark advance, must have been prevented by the use of water as an internal coolant.

To be able to compare the results of Fig. 40 with the calculated results in chapter 3, it has to be remembered that those calculations were based on an ideal Otto cycle, using octane as fuel at a chemically correct air-fuel ratio and water as an anti-detonant at a weight ratio 0.25. The reduction of the theoretical maximum cycle temperature was found with the help of the appropriate Hottel charts and the tabulation method of the author and heat losses to cooling water were not taken into account. The measured drop in mean cycle temperature of 40°C or ~~104~~ 72 $^{\circ}$F, when using water as an internal coolant at the weight ratio 0.25, seems to be in close agreement with the theoretical result of 100°F.

In conclusion, it may be said that in problems connected with reduction of the maximum cycle temperature due to the injection of an anti-detonant, the measured drop in mean cycle temperature may be used as a good indication of the drop in maximum cycle temperature. To find the actual reduction, however, an instrument for instantaneous temperature readings would be needed. It is believed that such instruments are available in U.S.A.

CHAPTER 5.

CONCLUSION.

The investigations carried out to establish the effect of water or water-alcohol injection on the performance of an internal combustion engine have furnished the following results:

1) Water, as has been known for many years, can successfully be used as an internal coolant. From the three weight ratios, namely, 0.25, 0.50 and 0.75, the first one yields the most favourable results. Further research, however, will be necessary to find the optimum weight ratio of water to petrol which will give the maximum engine output and efficiency at various speeds of the speed range.

2) The theoretical combustion temperature of an Otto cycle will fall by about 2%, when water at the 0.25 weight ratio is used as an anti-detonant and the drop of this temperature depends on the amount of water injected. The reduction of the combustion temperature of 2% is sufficient to bring the temperature of the end-gas below its critical value and to reduce the rate of pressure rise to such an extent that knocking will not occur.

3) Water, as an internal coolant, makes it possible to increase the compression ratio by at least two numbers

when standard grade petrol is used. Detonation in the lower speed range, where knocking usually takes place, disappears after water is injected, even at compression ratios of about 9:1.

Alternatively, fuels having a much lower octane rating than standard grade petrol may be used in conjunction with water injection, and satisfy the octane requirements of an engine with a compression ratio of about 7:1.

4) At full throttle operation, using water as the anti-detonant, an increase in power of up to 7 1/2% at 2500 r.p.m. may be obtained by increasing the compression ratio from 7:1 to 9:1.

At the compression ratio 7:1, an improvement in power of 2% may be reached, when running the engine between 2000 and 2500 r.p.m., after water is injected.

5) When operating the engine at half-throttle and a compression ratio of 9:1, a considerable gain in power and economy may be obtained, when using water injection. At 2500 r.p.m. the increases in power and economy are 13.5% and 7% respectively. On the other hand, at the compression ratio 7:1, no gains in power or economy are found when using water as a knock suppressor.

6) If a water-alcohol mixture (50+50) by volume is used as an anti-detonant, at a weight ratio of 0.25, considerable gain in power may be obtained. The economy, however, deteriorates, due to the high price of alcohol in Australia.

Values, when running the engine with a fully open throttle at 2500 r.p.m. and a compression ratio of 9:1 are as follows:

Increase in power 13%,

Increase in specific fuel costs 11%.

7) If water-alcohol (50+50), at the weight ratio 0.25, is used at half-throttle operations, the power shows very little improvement and the specific fuel costs become even worse than in the previous case. Tables 19 and 19a give the equivalent figures.

In summing up the results, it may be said that water, as an internal coolant, has greater possibilities in Australia than water-alcohol mixtures, unless the price of the latter is reduced considerably.

Water-injection provides the means of introducing to the customer new engines with compression ratios up to 10:1, which may be run on standard grade petrol with practically no signs of knocking. It also makes possible the conversion of engines to a higher compression ratio, either by machining of the existing cylinder heads, or by purchasing special high compression heads.

Thus not only an increase in power and efficiency will result, depending on the selected compression ratio, but also from the difference in price between standard and high grade petrols. This amounts to a further gain in economy of about 5½% in Australia.

Where an increase of power is wanted without consideration of specific fuel costs, water-alcohol (50+50 by volume) is the correct mixture to inject. This gives an increase in power due to the additional heat value of the alcohol. A further increase in power may also be obtained by raising the compression ratio.

Finally, water or water-alcohol injection will prevent the building up of carbon deposits on the piston crown and the combustion chamber, because carbon, under the influence of an internal coolant, will not adhere to these surfaces. Thus, the carbon particles are carried away by the exhaust gases.

Furthermore, the reduction of the combustion temperature, brought about by the injected coolant, and the absence of hard carbon particles, will result in less burning, pitting and distortion of the valves. This, in addition, will prolong the interval between valve grindings and the life of the valves will be increased. Thus a further economy will be effected.

On the other hand, the cost of a water-injector, complete with a copper tank and fittings, amounts to £A18-0-0. This outlay can be amortized in two years, if it is assumed that for an ordinary car engine, decarbonisation should be made every 7,500 miles, compared with 20,000 to 25,000 mls. for an engine fitted with a water-injector.

If the cost of decarbonizing is estimated at £A10-0-0, then £A20-0-0 will be saved by the owner of the car with the water-injector after 25,000 miles.

Suppose that two cars do 7,500 mls. each per year and travel 30 mls. to the gallon, but one car is using super grade petrol while the other, having a water-injector fitted, uses standard grade petrol, then a saving in fuel cost of about £A8-0-0 will be made by the owner of the latter car in three years.

The total savings, derived from the use of the cheaper petrol and less frequent decarbonization, not taking into account any other gains in economy, will amount to £A28-0-0 over a period of three years or to about £A18-10-0 in two years. This latter sum is equivalent to the purchasing price of a complete water-injection unit.

Thus, water as an internal coolant not only helps to improve the running condition but, after amortizing of the necessary equipment, will also improve the economy of the car by:

1) Allowing the use of standard grade fuel;
2) Permitting much longer intervals between decarbonising and valve grindings;
3) Giving a better overall efficiency, especially during full-throttle operations.

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OBSERVATION SHEET No. 5

		a	b	c		a	b	c
Observation No.:	1	2	3	4	5	6	7	8
Speed ... r.p.m.	1250	constant			1500	constant.		
Compr. Ratio	7.5:1		"		7.5:1		"	
Carburettor Needle Valve Position ....No.	36		"		36		"	
Cooling Water Outlet Temp. ... °C	67	70	69	67	68	72	71	70
Oil Temp. ..... °C	67	65	66	66	65	64	64	65
Manifold-Temp. °C	22	21	20	20	21.5	20	19	19
Vacuum .....in.Hg.	0	constant			0	constant		
Ignition: Spark-Advance ..... °E.	15	15	20	22	16	22	25	30
Main Fuel Cons. sec/50cc.	89.25	89.50	90.3	90.8	82	82.25	82.75	83
Air Consumption Manom. Slop No .2 cm.	9.8	9.8	9.7	9.7	11.2	11.2	11.1	11.1
Water ~~and/or Alcoh.~~ No.		67	57	49.5		68	57	49.5
Cons. Sec/8 25cc.	-	77	39	26	-	71	35.75	23.25
Observed Brake Load ... lb.	153.0	156.0	156.0	155.0	144.5	145.5	145.5	145.0
Exhaust Gas Temp. ... °C.	530	550	540	530	590	570	560	540
Knock Intensity	M.K.	L.K.	T.K.	T.K.	L.K.	T.K.	T.K.	T.K.
Smoke Density A/F	12.9	As per Exhaust Gas Analyser.						

Remarks: Throttle setting No. 3.5 constant.
Ignition set for maximum output. at trace knocking.

Date ......27..11..1956..

Barometer reading ......30.11......in.Hg.

Wet bulb temp. ......66...... °F.

Dry " " ......76...... °F.

Ambient temp. air cleaner ..26...... /~~°F~~. C.

Heat input 550W to air intake:
used
not used

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OBSERVATION SHEET No. .5a

		a	b	c		a	b	c
Observation No.:	1	2	3	4	5	6	7	8
Speed ... r.p.m.	1750	constant			2000	constant		
Compr. Ratio	7.5:1		"		7.5:1		"	
Carburettor Needle Valve Position ....No.	36		"		36		"	
Cooling Water Outlet Temp. ... °C	70	72	69	68	69	70	71	70
Oil Temp. ..... °C	66	68	69	69	68	69	70	70
Manifold-Temp. °C Vacuum .....in.Hg.	23 0	21 constant	20	20	25	22 0	21 constant	20
Ignition: Spark-Advance ..... °E.	24	29	35	39	30	36	42	49
Main Fuel Cons. sec/50cc.	78	78.25	78.4	78.5	74	74.5	75.25	75
Air Consumption Manom. Slop No 2 cm.	12.1	12	11.9	11.9	13	12.9	12.8	12.70
Water ~~and/or Alcoh.~~ No Cons. Sec/~~25~~ 8 cc.	-	69 68	58.5 34	51.5 22.5	-	68.5 64	58.5 32	51.5 21.2
Observed Brake Load ... lb.	13.4	13.4	13.25	13.10	12.2	12.1	12	11.80
Exhaust Gas Temp. ... °C.	610	600	580	590	630	620	600	590
Knock Intensity	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.
Smoke Density A/F	13.5	As per Exhaust Gas Analyser.						

Remarks:

Conditions as per Sheet No. 5.

Date ...27. .11.. .1956......

Barometer reading ......30.11....... in.Hg.

Wet bulb temp. ....67.......... °F.

Dry " " ....77.......... °F.

Ambient temp. air cleaner ...26...... /°F. C.

Heat input 550W to air intake:

~~used~~

not used

U.T. The New South Wales University of Technology

OBSERVATION SHEET No. 5b

		a	b	c		a	b	c
Observation No.:	1	2	3	4	5	6	7	8
Speed ... r.p.m.	2250	constant			2500	constant		
Compr. Ratio	7.5:1		"		7.5:1		"	
Carburettor Needle Valve Position ....No.	36		"		36		"	
Cooling Water Outlet Temp. ... °C	73	70	68	67	65	72	72	71
Oil Temp. ..... °C	72	73	75	73	64	66	69	75
Manifold-Temp. °C	27	25	23	21	27	23	21	21
Vacuum .....in.Hg.	0	constant			0	constant		
Ignition: Spark-Advance ..... °E.	35	45	45	50	40	44	50	55
Main Fuel Cons. sec/50cc.	71.7	72	71.50	71.75	68.3	68.6	68.75	69.3
Air Consumption Manom. Slop No .2 cm.	13.7	13.6	13.5	13.4	14.5	14.3	14.2	14.2
Water ~~and/or Alcoh.~~ No.		70	59	52.5		69	58.5	52
Cons. Sec/~~25~~ 8 cc.	-	62.5	31.5	21.2	-	59.5	29	19.75
Observed Brake Load ... lb.	11.15	11	10.85	10.70	9.90	9.85	9.80	9.65
Exhaust Gas Temp. ... °C.	650	630	620	610	670	650	650	640
Knock Intensity	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.
Smoke Density A/F	13.7				13.8			

Remarks:

Conditions as per sheet No. 5.

Date ...27.11.56.......

Barometer reading ...30.13.......... in.Hg.

Wet bulb temp. ...67............ °F.

Dry " " ...78............ °F.

Ambient temp. air cleaner ...27...... /~~°F~~. °C

Heat input 550W to air intake:

~~used~~

not used

U.T. The New South Wales University of Technology

OBSERVATION SHEET No. 5c

		a	b	c		a	b	c
Observation No.:	1	2	3	4	5	6	7	8
Speed ... r.p.m.	2750	constant			3000	constant		
Compr. Ratio	7.5:1		"		7.5:1		"	
Carburettor Needle Valve Position ....No.	36		"		36		"	
Cooling Water Outlet Temp. ... °C	70	71	71	69	70	73	70	69
Oil Temp. ..... °C	71	67	66	73	72	73	72	72
Manifold-Temp. °C	28	23	21.5	21	28	24	22	21
Vacuum .....in.Hg.	0	constant			0	constant		
Ignition: Spark-Advance ..... °E.	47	50	57	60	50	55	55	60
Main Fuel Cons. sec/50cc.	66.8	67	67	67	64.25	64.30	64.50	64.0
Air Consumption Manom. Slop No 2. cm.	15.2	15.1	15	15	16.1	15.9	15.9	15.8
Water ~~and/or Alcoh.~~ No.	-	70	57	51.5	-	70	59	51.5
Cons. Sec/8 25cc.		58	29	19		55.5	28	17.5
Observed Brake Load ... lb.	9.05	9	8.90	8.75	8.15	8.10	8.05	775
Exhaust Gas Temp. ... °C.	680	670	660	650	690	680	680	670
Knock Intensity	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.
Smoke Density A/F	13.9				13.6			

Remarks: Conditions as per Sheet No. 5.

Date ..27. 11. 1956.......

Barometer reading ......30.11........in.Hg.

Wet bulb temp. ......67.......... °F.

Dry " " ......79.......... °F.

Ambient temp. air cleaner ...27..... /~~°F~~. C.

Heat input 550W to air intake:

~~used~~

not used

U.T. The New South Wales University of Technology

OBSERVATION SHEET No. 6..

	a	b	c		a	b	c
Observation No.:	1	2	3	4	5	6	7	8

Observation No.:	1	2	3	4	5	6	7	8
		a	b	c		a	b	c
Speed ... r.p.m.	1250	constant			1500	constant		
Compr. Ratio	7.5:1		"		7.5:1		"	
Carburettor Needle Valve Position ....No.	36		"		36		"	
Cooling Water Outlet Temp. ... °C	73	68	65	65	67	71	69	67
Oil Temp. ..... °C	65	65	65	65	65	65	66	66
Manifold-Temp. °C	20	19	19	19	21	18.5	18.5	19
Vacuum .....in.Hg.	0	constant			0	constant		
Ignition: Spark-Advance ..... °E.	15	15	20	24	15	19	23	27
Main Fuel Cons. sec/50cc.	85.6	85.4	84.25	85.25	75	75.7	75.3	75.75
Air Consumption Manom. Slop No .2 cm.	10.2	10.2	10.3	10.1	12.4	12.2	12.3	12.2
Water ~~and/or Alcoh.~~ No.	-	61	50	40		59.5	48	37
Cons. Sec/8 ~~25~~cc.		74.25	36.80	25		65	32.8	22
Observed Brake Load ... lb.	15.85	16.65	16.50	16.20	16.30	16.40	16.55	16.40
Exhaust Gas Temp. ... °C.	510	550	540	520	620	610	580	570
Knock Intensity	M.K.	T.K.	T.K.	T.K.	L.K.	T.K.	T.K.	T.K
Smoke Density A/F	12.7	Exhaust Gas Analyser Readings.						

Remarks: Throttle setting No. 7 constant. Ignition set for maximum putput at trace knocking.

Date ..28. 11. 56.........

Barometer reading ...30.06..........in.Hg.

Wet bulb temp. ...66............... °F.

Dry " " ...76............... °F.

Ambient temp. air cleaner ..26....... /°~~F~~.C.

Heat input 550W to air intake:

~~used~~

not used

U.T. The New South Wales University of Technology

OBSERVATION SHEET No. 6a

	a	b	c		a	b	c	
Observation No.:	1	2	3	4	5	6	7	8
Speed ... r.p.m.	1750	constant			2000	constant		
Compr. Ratio	7.5:1		"		7.5:1		"	
Carburettor Needle Valve Position ....No.	36		"		36		"	
Cooling Water Outlet Temp. ... °C	70	73	71	68	72	70	67	68
Oil Temp. ..... °C	64	66	67	67	70	69	67	67
Manifold-Temp. °C Vacuum .....in.Hg.	20 0	18 constant	18	18	21 0	18.5 constant	18	18
Ignition: Spark-Advance ..... °E.	18	23	28	33	23	28	33	38
Main Fuel Cons. sec/50cc.	68.6	69.5	69.25	69.75	63.5	63.5	63.75	64.75
Air Consumption Manom. Slop No 2 cm.	14.2	14.1	14.1	13.9	16.1	16.1	16.0	16.1
Water ~~and/or Alcoh.~~ No. Cons. Sec/~~25~~ 8 cc.	-	59.5 59.5	46.5 30	34 19.8	-	59 55.5	46 27.75	34 19
Observed Brake Load ... lb.	16.00	16.10	16.25	16.15	15.75	16.00	16.00	15.85
Exhaust Gas Temp. ... °C.	640	640	610	600	670	660	630	
Knock Intensity	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	
Smoke Density A/F	13.4				13.7			

Remarks: Conditions as per Sheet No. 6.

Date ....28..11..56.....

Barometer reading ....in.Hg.

Wet bulb temp. .....67............ °F.

Dry " " ......77........... °F.

Ambient temp. air cleaner ....26..... /°~~F~~.C.

Heat input 550W to air intake:

~~used~~

not used

U.T. The New South Wales University of Technology

OBSERVATION SHEET No. 6b.

		a	b	c		a	b	c
Observation No.:	1	2	3	4	5	6	7	8
Speed ... r.p.m.	2250	constant			2500	constant		
Compr. Ratio	7.5:1		"		7.5:1		"	
Carburettor Needle Valve Position ....No.	36		"		36		"	
Cooling Water Outlet Temp. ... °C	68	72	71	70	71	74	73	72
Oil Temp. ..... °C	69	71.5	69	68	70	70	70	70
Manifold-Temp. °C	20	18.5	18	18	20	18.5	18.5	18.5
Vacuum .....in.Hg.	0	constant			0	constant		
Ignition : Spark-Advance ..... °E.	26	33	37	45	30	39	45	50
Main Fuel Cons. sec/50cc.	58.75	58.90	59.25	59.80	55.70	55.50	56	56
Air Consumption Manom. Slop No 2. cm.	18	17.9	17.8	17.7	19.6	19.7	19.4	19.2
Water ~~and/or Alcoh.~~ No.		59	45	33		59	45	33
Cons. Sec/8 ~~25~~ cc.	-	51	25.4	18	-	48.5	24	16.75
Observed Brake Load ... lb.	15.50	15.60	15.45	15.25	14.45	14.75	14.70	14.40
Exhaust Gas Temp. ... °C.	670	660	650	630	680	650	650	640
Knock Intensity	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.
Smoke Density A/F	13.8							

Remarks: Conditions as per sheet No. 6.

Date ....28. 11. 56.........

Barometer reading ....30.02.......... in.Hg.

Wet bulb temp. ....68........ °F.

Dry " " ....78....... °F.

Ambient temp. air cleaner ..26....... /°~~F~~.C.

Heat input 550W to air intake:

~~used~~

not used

U.T. The New South Wales University of Technology

OBSERVATION SHEET No. 6c.

		a	b	c		a	b	c
Observation No.:	1	2	3	4	5	6	7	8
Speed ... r.p.m.	2750	constant			3000	constant		
Compr. Ratio	7.5:1		"		7.5:1		"	
Carburettor Needle Valve Position ....No.	36		"		36		"	
Cooling Water Outlet Temp. ... °C	77	71	69	69	71	71	73	70
Oil Temp. ..... °C	67	72	69	71	71	71	73	73
Manifold-Temp. °C	21	19	18	18	21	19	19	19
Vacuum .....in.Hg.	0	constant			0	constant		
Ignition: Spark-Advance ..... °E.	32	45	55	60	40	48	55	60
Main Fuel Cons. sec/50cc.	52.75	52.40	53	52.50	50.6	50.3	50.65	50.5
Air Consumption Manom. Slop No .2 cm.	21.1	21.3	21	21.3	22.7	22.7	22.7	22.8
Water ~~and/or Alcoh.~~ No. Cons. Sec/~~25~~ 8 cc.		54.5 45.75	40 22.75	27 15	-	58 44	44 22	37.5 18
Observed Brake Load ... lb.	13.25	13.40	13.45	13.30	11.90	12.9	11.95	1200
Exhaust Gas Temp. ... °C.	690	660	640	630	670	660	650	640
Knock Intensity	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.	T.K.
Smoke Density A/F	13.8				13.8			

Remarks: Conditions as per Sheet No. 6

Date ....29. 11. 56.

Barometer reading .....29.87........in.Hg.

Wet bulb temp. .....62.......... °F.

Dry " " .....77.......... °F.

Ambient temp. air cleaner ..27........ /~~°F~~. C.

Heat input 550W to air intake:

used

not used

THE NEW SOUTH WALES UNIVERSITY OF TECHNOLOGY.

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RESULT SHEET No. ..5..

Observation No.	1	2	3	4	5	6	7	8
Speed r.p.m.	1250	1500	1750	2000	2250	2500	2750	3000
Corrected Brake Load lb.	15.52	14.71	13.68	12.52	11.50	10.27	9.48	8.65
Brake Horse Power corrected.	5.72	6.5	7.04	7.40	7.64	7.58	7.66	7.62
Torque lb-ft.	24.1	22.8	21.1	19.5	17.9	15.9	14.7	13.35
Air Cons. lb./hr.	38.2			50.6				62.7
Fuel Cons. lb./hr.	3.32	3.61	3.79	4.0	4.13	4.34	4.43	4.61
Fuel cost s./hr.	1.66	1.82	1.89	2.0	2.07	2.17	2.22	2.32
Alcohol Cons lb./hr.								
s./hr.								
Int. Coolant / Total Fuel % by wt	24.8			24.8				25.2
Spec. Fuel lb/BHP-hr	.60	.577	.555	.558	.557	.588	.595	.624
Consump. s./BHP-hr.	.30	.289	.278	.279	.279	.294	.298	.312
Brake Therm. Effic.%	23.3	24.3	25.2	25.1	24.6	23.8	23.5	22.5
Air/Fuel Ratio	11.5			12.65				13.6
Remarks:	No internal coolant Throttle setting No. 3.5 C.R. 7.5:1 constant.							

Fuel: Shell Motor Spirit
L.H.V.: 18,140 B.T.U./lb.
S.G. ~~Density~~: 0.745 at 15°C
Price: 0.5 s/lb.

Alcohol:
L.H.V:
Density:
Price:

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RESULT SHEET No. 5a

Observation No.	1	2	3	4	5	6	7	8
Speed r.p.m.	1250	1500	1750	2000	2250	2500	2750	3000
Corrected Brake Load lb.	15.82	14.81	13.68	12.42	11.35	10.22	9.43	8.60
Brake Horse Power corrected.	5.82	6.55	7.04	7.32	7.52	7.54	7.60	7.57
Torque lb-ft.	24.5	22.9	21.1	19.3	17.6	15.8	14.6	13.25
Air Cons. lb./hr.	38.2			50.4				62
Fuel Cons. lb./hr.	3.3	3.6	3.78	3.97	4.12	4.32	4.42	4.60
Fuel cost s./hr.	1.65	1.8	1.89	1.99	2.06	2.16	2.21	2.30
Alcohol Cons lb./hr.								
s./hr.								
Int. Coolant / Total Fuel % by wt	24.9			25				24.8
Spec. Fuel lb/BHP-hr	.585	.566	.554	.56	.564	.59	.595	.625
Consump. s./BHP-hr.	.293	.283	.277	.28	.282	.295	.298	.313
Brake Therm. Effic.%	23.9	24.7	25.3	25	24.8	23.7	23.5	22.4
Air/Fuel Ratio	11.55			12.7				13.5
Remarks:	Internal coolant water Weight ratio approx. 25% Throttle setting No. 3.5 C.R. 7.5:1 constant.							

Fuel: Shell Motor Spirit.
L.H.V.: 18,140 B.T.U/lb.
~~Density~~ S.G. 0.745 at 15°C
Price: 0.5 s./lb.

Alcohol:
L.H.V:
Density:
Price:

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RESULT SHEET No. .5b.

Observation No.	1	2	3	4	5	6	7	8
Speed r.p.m.	1250	1500	1750	2000	2250	2500	2750	3000
Corrected Brake Load lb.	15.82	14.81	13.53	12.32	11.20	10.17	9.33	8.55
Brake Horse Power corrected.	5.82	6.55	6.97	7.26	7.44	7.50	7.57	7.55
Torque lb-ft.	24.5	22.9	20.9	19.1	17.4	15.7	14.5	13.2
Air Cons. lb./hr.	37.9			50				62
Fuel Cons. lb./hr.	3.28	3.58	3.77	3.99	4.15	4.31	4.42	4.60
Fuel cost s./hr.	1.64	1.79	1.89	2.0	2.08	2.16	2.21	2.30
Alcohol Cons lb./hr.								
s./hr.								
Int. Coolant / Total Fuel % by wt.	49.5			49.6				49.3
Spec. Fuel lb/BHP-hr	.58	.565	.556	.565	.575	.592	.60	.628
Consump. s./BHP-hr.	.29	.283	.278	.283	.288	.296	.30	.314
Brake Therm. Effic.%	24.1	24.8	25.2	24.8	24.4	23.7	23.3	22.3
Air/Fuel Ratio	11.5			12.5				13.5
Remarks:	Internal coolant water Weight ratio approx. 50% Throttle setting No 3.5 C.R. 7.5:1 constant.							

Fuel: Shell Motor Spirit

L.H.V.: 18,140 B.T.U./lb.

S.G. ~~Density~~: 0.745 at 15°C

Price: 0.5 s./lb.

Alcohol:

L.H.V:

Density:

Price:

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RESULT SHEET No. 5c.

Observation No.	1	2	3	4	5	6	7	8
Speed r.p.m.	1250	1500	1750	2000	2250	2500	2750	3000
Corrected Brake Load lb.	15.75	14.76	13.38	12.12	11.05	10.0	9.18	8.25
Brake Horse Power corrected.	5.8	6.52	6.88	7.16	7.33	7.35	7.41	7.25
Torque lb-ft.	24.4	22.8	20.6	18.9	17.2	15.4	14.2	12.7
Air Cons. lb./hr.	37.9			49.6				61.7
Fuel Cons. lb./hr.	3.25	3.57	3.77	3.95	4.13	4.27	4.42	4.62
Fuel cost s./hr.	1.63	1.79	1.89	1.98	2.07	2.14	2.21	2.31
Alcohol Cons lb./hr.								
s./hr.								
Int. Coolant / Total Fuel % by wt.	75			76				78.5
Spec. Fuel lb/BHP-hr.	.578	.565	.565	.568	.58	.60	.612	.654
Consump. s./BHP-hr.	.289	.283	.283	.284	.29	.30	.306	.327
Brake Therm. Effic.%	24.2	24.8	24.8	24.6	24.2	23.3	22.9	21.4
Air/Fuel Ratio	11.7			12.6				13.4
Remarks:	Internal coolant water Weight ratio approx. 75% Throttle setting No. 35. C.R. 7.5:1 constant.							

Fuel: Shell Motor Spirit — Alcohol:

L.H.V.: 18,140 B.T.U./lb. — L.H.V:

S.G. ~~Density~~ 0.745 at 15°C. — Density:

Price: 0.5 s./lb. — Price:

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RESULT SHEET No. 6

Observation No.	1	2	3	4	5	6	7	8
Speed r.p.m.	1250	1500	1750	2000	2250	2500	2750	3000
Corrected Brake Load lb.	16.07	16.56	16.28	16.07	15.85	14.82	13.68	12.4
Brake Horse Power corrected.	5.92	7.33	8.4	9.46	10.53	10.95	11.1	10.95
Torque lb-ft.	25	25.6	25.2	24.9	24.7	23	21.3	19.2
Air Cons. lb./hr.	38.6			62				88
Fuel Cons. lb./hr.	3.45	3.95	4.31	4.66	5.04	5.31	5.6	5.85
Fuel cost s./hr.	1.73	1.98	2.16	2.33	2.52	2.66	2.8	2.93
Alcohol Cons lb./hr.								
s./hr.								
Int. Coolant / Total Fuel % by wt.								
Spec. Fuel lb/BHP-hr.	.601	.556	.53	.508	.493	.502	.521	.553
Consump. s./BHP-hr.	.30	.278	.265	.254	.247	.251	.261	.277
Brake Therm. Effic.%	23.3	25.1	26.4	27.5	28.4	27.9	26.8	25.3
Air/Fuel Ratio	11.2			13.3				15
Remarks:	No internal coolant. Throttle setting No. 7 constant. C.R. 7.5:1.							

Fuel: Shell Motor Spirit.

L.H.V.: 18,140 B.T.U./lb.

S.G. ~~Density~~: 0.745 at 15°C.

Price: 0.5 s./lb.

Alcohol:

L.H.V:

Density:

Price:

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SCHOOL OF MECHANICAL ENGINEERING.

RESULT SHEET No. 6a.

Observation No.	1	2	3	4	5	6	7	8
Speed r.p.m.	1250	1500	1750	2000	2250	2500	2750	3000
Corrected Brake Load lb.	16.87	16.66	16.38	16.32	15.95	15.12	13.85	12.55
Brake Horse Power corrected.	6.22	7.36	8.45	9.65	10.60	11.15	11.25	11.1
Torque lb-ft.	26.2	25.75	25.3	25.3	24.8	23.4	21.6	19.5
Air Cons. lb./hr.	38.6			62				88
Fuel Cons. lb./hr.	3.47	3.91	4.25	4.66	5.03	5.33	5.65	5.88
Fuel cost s./hr.	1.74	1.96	2.13	2.33	2.52	2.67	2.83	2.94
Alcohol Cons lb./hr.								
s./hr.								
Int. Coolant / Total Fuel % by wt.	24.6	24.6	25.1	24.6	24.8	24.6	24.6	24.6
Spec. Fuel lb/BHP-hr.	.577	.546	.518	.499	.49	.494	.518	.54
Consump. s./BHP-hr.	.288	.273	.259	.25	.245	.247	.259	.27
Brake Therm. Effic.%	24.3	25.6	27	28.1	28.6	28.4	27	25.9
Air/Fuel Ratio	11.1			13.3				14.9

Remarks:

Internal coolant water.

Weight approx. 25%.

Throttle setting No. 7 constant.

C.R. 7.5:1.

Fuel: Shell Motor Spirit.

L.H.V.: 18,140 B.T.U./lb.

S.G. ~~Density~~: 0.745 at 15°C.

Price: 0.5 s./lb.

Alcohol:

L.H.V:

Density:

Price:

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RESULT SHEET No. . .6h.

Observation No.	1	2	3	4	5	6	7	8
Speed r.p.m.	1250	1500	1750	2000	2250	2500	2750	3000
Corrected Brake Load lb.	16.72	16.8	16.53	16.32	15.80	15.07	13.8	12.45
Brake Horse Power corrected.	6.18	7.44	8.55	9.65	10.52	11.1	11.25	11
Torque lb-ft.	26	26	25.7	25.4	24.6	23.2	21.5	19.3
Air Cons. lb./hr.	38.6			62				88
Fuel Cons. lb./hr.	3.51	3.93	4.27	4.64	4.99	5.28	5.58	5.85
Fuel cost s./hr.	1.76	1.97	2.14	2.32	2.50	2.64	2.79	2.93
Alcohol Cons lb./hr.								
s./hr.								
Int. Coolant / Total Fuel % by wt.	49.2	49.3	49.6	49.3	50	50	50	49.5
Spec. Fuel lb/BHP-hr.	.587	.545	.516	.496	.49	.492	.512	.55
Consump. s./BHP-hr.	.294	.273	.258	.248	.245	.246	.256	.275
Brake Therm. Effic.%	23.8	25.7	27.1	28.2	28.5	28.5	27.4	25.5
Air/Fuel Ratio	11			13.35				15
Remarks:	Internal coolant water. Weight ratio approx. 50% Throttle setting No. 7 constant. C.R. 7.5:1.							

Fuel: Shell Motor Spirit
L.H.V.: 18,140 B.T.U./lb.
S.G. ~~Density~~: 0.745 at 15°C.
Price: 0.5 s./lb.

Alcohol:
L.H.V:
Density:
Price:

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RESULT SHEET No. .6c.

Observation No.	1	2	3	4	5	6	7	8
Speed r.p.m.	1250	1500	1750	2000	2250	2500	2750	3000
Corrected Brake Load lb.	16.44	16.65	16.43	16.17	15.60	14.77	13.73	12.50
Brake Horse Power corrected.	6.05	7.37	8.5	9.55	10.35	10.9	11.15	11.05
Torque lb-ft.	25.5	25.8	25.5	25.2	24.2	22.9	21.4	19.3
Air Cons. lb./hr.	39.2			62				88.5
Fuel Cons. lb./hr.	3.47	3.90	4.24	4.57	4.95	5.28	5.64	5.86
Fuel cost s./hr.	1.74	1.95	2.12	2.29	2.48	2.64	2.82	2.93
Alcohol Cons lb./hr.								
s./hr.								
Int. Coolant / Total Fuel % by wt.	73	74	75.4	73.2	71.5	72	75	72
Spec. Fuel lb/BHP-hr	.59	.546	.516	.494	.492	.50	.522	.547
Consump. s./BHP-hr.	.295	.273	.258	.247	.246	.25	.261	.274
Brake Therm. Effic.%	23.7	25.6	27.1	28.4	28.4	28.0	26.8	25.6
Air/Fuel Ratio	11.3			13.5				15.1

Remarks:

Internal coolant water.
Weight ratio approx. 75%.
Throttle Setting No. 7 constant.
C.R. 7.5:1.

Fuel: Shell Motor Spirit.
L.H.V.: 18,140 B.T.U./lb.
S.G. ~~Density~~: 0.745 at 15°C.
Price: 0.5 s./lb.

Alcohol:
L.H.V:
Density:
Price:

Reprinted from AUTOMOBILE ENGINEER (LONDON) *October 1955*

BLENDED FUEL EXHAUST GAS

Determinations with the Thermal Conductivity Analyser

Felix Gutman and K. Weiss*

THE establishment and means of adjustment of the air-fuel ratio are of vital importance for the satisfactory operation of petrol engines. Complete stoichiometric combustion of straight hydrocarbon fuels corresponds to a theoretical air-fuel ratio of 14·3:1; below this ratio, in the region of rich mixtures, combustion is incomplete due to lack of oxygen: unburned fuel, mainly hydrogen and carbon monoxide, appears in the exhaust gas. In the lean region, at air-fuel ratios in excess of 14·3:1, combustion is complete: no hydrogen, very little methane, only very small amounts of CO plus a decreasing concentration of CO_2 appear in the exhaust gas, in addition to unconsumed oxygen.[1]

The analysis of exhaust gases by means of the "thermal conductivity gas analyser" is a well established method.[2] It is based upon the fact that the thermal conductivity of hydrogen is about 7 times that of nitrogen or oxygen, while the thermal conductivity of CO_2 is about two-thirds that of N_2. Thus, even traces of H_2 or CO_2 in air are readily measurable. The apparatus[3] consists, in principle, of a platinum filament which is electrically heated at a constant rate and exposed to the dry gas in a closed chamber. The filament eventually attains an equilibrium temperature dependent on the rate at which heat is applied to and removed from it; since the heat input is kept constant its temperature depends only on the rate of heat loss, the gas being assumed to be at ambient temperature. If heat is removed from the filament purely by convection, then the equilibrium temperature will depend only on the thermal conductivity of the gas. Since the electrical resistance of the filament is very nearly a linear function of its temperature, a comparison of the resistance of the filament when surrounded by exhaust gas or by a standard gas (usually air) yields a means, through an empirical calibration, of estimating the thermal conductivity of the exhaust gas.

It is well known[1,4] that for straight hydrocarbon fuels, the air-fuel ratio may be uniquely determined by the estimation of either H_2 or CO in the rich range and by that of CO_2 or O_2 in the lean range. Thus the electrical thermal conductivity gas analyser represents a convenient means of measuring the air-fuel ratio and large numbers of such instruments are in use.

It was the purpose of this work to investigate the response of the thermal conductivity analyser to blended fuels used in high performance engines and to study the performance of the method.

Experimental equipment

The measurements were carried out with the aid of a Ricardo E6, single cylinder, 4-stroke variable compression engine, coupled to a swinging field electrical dynamometer. The carburettor was a Solex type F.A.1 fitted with a 27 mm choke. A variable main jet allowed variation of the fuel mixture during tests by means of a tapered needle valve. A 2-gallon tank as main supply, a small sampling tank and a calibrated measuring burette comprised the fuel measuring apparatus. The time for 50 c.c. of fuel to be burned was then measured with a stop-watch, while the air consumption was determined by means of an "Alcock" viscous flow air meter. The engine speed was kept constant at 2,750 r.p.m.

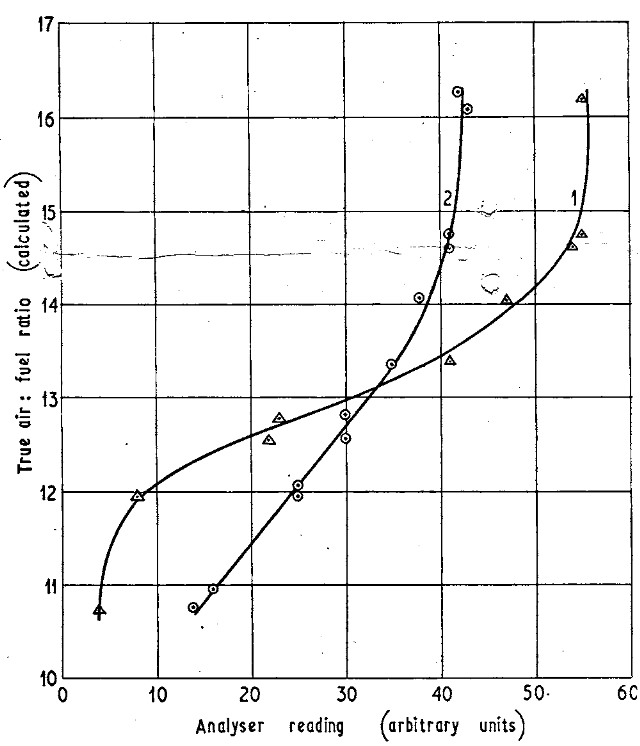

Fig. 1. Calibration of two thermal conductivity type gas analysers on straight hydrocarbon commercial petrols. Curve 1 refers to an analyser operating at 200 mA bridge current, and Curve 2 to an identical analyser with 150 mA bridge current

Readings were taken of barometric pressure and wet and dry bulb temperatures. Air-fuel ratios were calculated from the observed values for fuel and air consumed, and corrected for humidity. Since water vapour has a thermal conductivity of about three-quarters of that of air, a water trap was employed to condense and thus remove it from the exhaust gas, which was allowed to cool to ambient temperature.

The exhaust gases were chemically analysed by means of an Orsat gas analysis apparatus which received its samples immediately after the electrical analyser. These analyses were used as a check on the air-fuel ratios indicated, employing the relationship between this quantity and chemical composition given by D'Alleva and Lovell[1] and others[4].

Four analysers of three different designs were used, the circuit in all cases being a constant current Wheatstone bridge. A preliminary setting standardizes the bridge current and thus the filament equilibrium temperature when surrounded by air. Platinum filaments of 9 ohm cold resistance operated at 100 mA current with a 400 microamperes full scale deflection meter of 30 ohm internal resistance produced the best results; Pt-Ir filaments of 17·1 ohm cold resistance operated at 150 mA bridge current with a 500 μA meter of 50 ohm resistance gave only slightly inferior results. The operating temperature of the Pt elements (100 mA bridge current) was calculated to be about 60 deg C above ambient temperature and that of the Pt-Ir filaments (150 mA) about 70 deg C above ambient temperature. An attempt was made to raise the sensitivity of one of the instruments by increasing the bridge current and thus the filament temperature. This, however, rendered the instrument sensitive to the velocity of flow of the gas and introduced another element of non-linearity between its readings and the air-fuel ratio. This is shown in Fig. 1, which represents the calibration curves of two identical analysers, one operating with 150 mA (Curve 2) and the other with 200 mA bridge current (Curve 1).

The maximum permissible bridge

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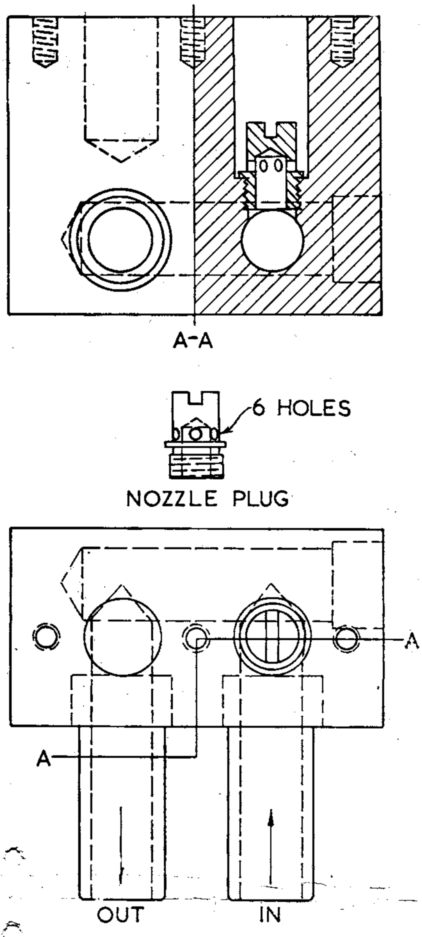

Fig. 2. Design of sampling chamber and nozzle plug to allow access of the exhaust gas to the filaments by diffusion only, yielding readings independent of velocity of gas flow

current is closely connected to the design of the sampling chamber; a design which has proved satisfactory is given in Fig. 2. It is essential that the gas has access to the filaments by diffusion only, otherwise the readings again become dependent on flow velocity; a special nozzle plug, of 8 mm diameter and screwed into the inlet port of the sampling chamber, was found to give satisfactory operation.

The location of the sampling tube within the exhaust gas stream is somewhat critical: if, for example, the sampling tube is located in a direction perpendicular to the direction of gas flow, the resulting disturbance of the main flow of gas tends to produce a partial vacuum in the sampling chamber, resulting in quite erroneous readings. However, if the sampling tube is mounted coaxial with the exhaust pipe with its opening facing up-stream, then a small positive pressure is produced which yields consistent, accurate and well-reproducible results.

Air-fuel Ratio for Maximum Power Output at 2,750 r.p.m. for various kinds of fuel

Fuel	A-F Ratio	Compression Ratio
Petrol "A"	12·25:1	7:1
Petrol "B"	13·13:1	7:1
Petrol "C"	13·38:1	7:1
Fuel "A"	9·71:1	11:1
Fuel "B"	8·11:1	11:1
Fuel "C"	10·6:1	9·5:1
Fuel "D"	7·54:1	12:1
	6·93:1	11:1
Fuel "E"	5·2:1	11:1
Fuel "F"	10·8:1	12:1
	7·6:1	10·5:1

Results

The following fuels were tested:

Fuel A, containing:
50 per cent ethanol,
25 per cent benzol,
25 per cent octane,
to which were added
6·5 per cent nitrobenzene, producing a fuel of specific gravity of 0·826.

Fuel B, containing:
60 per cent methanol,
20 per cent octane,
20 per cent benzol,
and having a specific gravity of 0·789.

Fuel C: Shell Racing Fuel Type "X."

Fuel D: Shell Racing Fuel Type 8/11, spec. gr. 0·790.

Fuel E: Shell Racing Fuel Type A, spec. gr. 0·792.

Fuel F: A mixture of equal parts of Shell Racing Fuels Types TT and M.

Tests were made also with two brands of straight, hydrocarbon, commercial, petrol of specific gravity 0·715 and 0·720, respectively. Plots of measurements on these two hydrocarbon fuels, using a compression ratio of 7:1, are shown in Fig. 1. Curve 2 refers to correct operating conditions. The abscissæ give analyser readings in arbitrary units on a 100 division scale. A single curve is seen to fit the readings for both petrols.

The response of the instrument is almost entirely determined by the hydrogen concentration within the rich range and by that of CO_2 in the lean region. Both responses are approximately linear, but have different slopes. If thus the air-fuel ratio is slowly increased towards 14·3:1, the ratio of complete combustion, the response of the analyser becomes non-linear in the vicinity of this value. Curve 2, it will be seen, follows a fairly good straight line in the rich mixture range, but its curvature becomes appreciable in the lean region.

A kind of saturation effect is evident at higher values of air-fuel ratio, and the readings virtually cease to increase beyond 14·8. This is probably caused by the rapid decrease in CO_2 concentration at air-fuel ratios above the stoichiometric ratio, and to the higher concentrations of oxygen then present: the thermal conductivity of the exhaust gas then tends to approach that of air. This effect is present with all analysers tested. It should therefore be stressed that this type of exhaust gas analyser yields results which are far less accurate for lean than for rich mixtures; in fact, the instrument can hardly be used at mixtures leaner than about 14·5:1. The analyser readings are practically

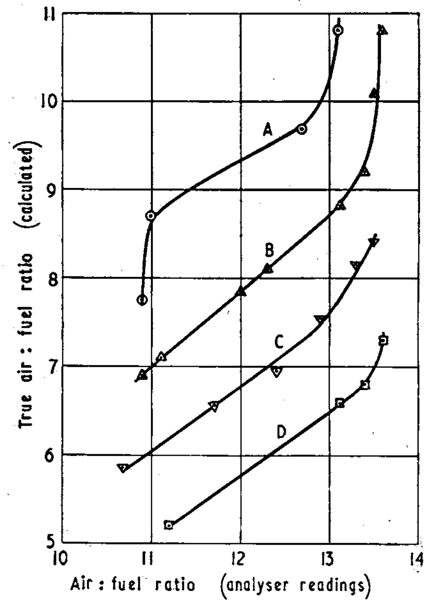

Fig. 3. Analyser readings obtained with 4 different kinds of blended fuels

independent of the compression ratio between 7:1 and 12:1.

Fig. 3 shows typical calibration curves for some of the blended fuel mixtures. The abscissæ are taken from the analyser scales which previously had been accurately calibrated and checked against values obtained with straight hydrocarbon fuels (commercial petrols). It will be seen that very large deviations are observed for all the blended fuels tested, the analysers always reading far too lean. This is due to the fact that these blended fuels contain large proportions of oxygen in their constituents, and therefore require less atmospheric oxygen for complete combustion. Thus fuel "B," containing methanol which has a stoichiometric air-fuel ratio of 6·44:1, produces a much larger shift in the analyser readings than, for example, fuel "A," containing ethanol requiring an air-fuel ratio of 8·84:1 for complete combustion. That the curves are rather more non-linear than those obtained with straight hydrocarbons, is probably due to the complexity of the combustion products which contribute to the thermal conductivity of the exhaust gases. The curve for fuel "A," containing nitrobenzene, shows the greatest curvature.

The average slope changes but little from fuel to fuel, but the whole curves are shifted towards lower values of calculated air-fuel ratio, very nearly parallel to themselves. Such a behaviour would be expected if it is the availability of chemically bound O_2 from within the fuel itself which is responsible for these changes. There is no correlation between the analyser readings and the specific gravity of the fuel. There is evidence of a "saturation effect" in the lean region similar to that observed with straight hydrocarbon fuel. The blended fuels permit the engine to run on much richer mixtures than does straight petrol, and again the analyser readings are far more accurate in the rich than in the lean region.

It is of interest to compare the air-fuel ratio at which the fuels used deliver maximum power output: it is seen from the accompanying table that appreciable differences exist even between commercially available straight petrols. The air-fuel ratio for maximum power also depends on the compression ratio. Fuel "F," for example, yields maximum power at a/f ratio 10·8:1 at a compression ratio of 12:1, compared to 7·6:1 at a compression of 10·5:1; fuel "D" likewise produces maximum power output at an air-fuel ratio of 7·54:1 at a compression of 12:1, compared to 6·93:1 at a compression of 10·5:1. A higher compression ratio always produces maximum power with a leaner mixture.

The authors are indebted to Mr. E. C. Martin of the Analytical Department of the New South Wales University of Technology, for assistance with the Orsat apparatus, and to Mr. A. Vainomae who helped run the test engine.

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Reprinted from AUTOMOBILE ENGINEER October 1956

CALCULATION OF THERMODYNAMIC CHARTS

A New Method of Solving Equilibrium Equations Assists Evaluation of Ideal Efficiencies

K. WEISS

THERMODYNAMIC charts such as developed by Hershey, Eberhardt and Hottel are of the greatest help when the problem of determining the ideal efficiency of an internal combustion engine arises. Unfortunately, these charts are constructed for three mixture-strengths of octane and air and certain octane mixtures only. If fuels other than octane are used, or special air-fuel ratios are required, new charts have to be calculated, based on the thermal data available.

A. S. Leah* has shown how to construct the necessary lines to find the ideal efficiency and mean effective pressure for an engine, following the Otto cycle using ethyl alcohol as fuel. To plot graphs of internal-energy and heat released during reaction against temperature, for the products of combustion, it would be necessary to solve the equilibrium equations in the neighbourhood of the maximum temperature of the cycle. This can be done by using the method of mathematical iteration, which is a slow method when the number of unknowns is greater than four, as in the case of total dissociation. A new tabular method will be described, which, in the author's opinion, takes less time than the methods used so far.

Calculation of the dissociation quantities

To calculate the dissociation quantities at a certain temperature, the following dissociation equations have to be solved:

$$\frac{[CO]^2 \times [O_2]}{[CO_2]^2} = K_1 \qquad (1)$$

$$\frac{[CO] \times [H_2O]}{[CO_2] \times [H_2]} = K_2 \qquad (2)$$

$$\frac{[OH]^2 \times [H_2]}{[H_2O]^2} = K_3 \qquad (3)$$

$$\frac{[H]^2}{[H_2]} = K_4 \qquad (4)$$

$$\frac{[O]^2}{[O_2]} = K_5 \qquad (5)$$

$$\frac{[N]^2}{[N_2]} = K_6 \qquad (6)$$

$$\frac{[NO]^2}{[O_2][N_2]} = K_7 \qquad (7)$$

The quantities in brackets are the partial pressures of the constituents, and K_1 to K_7 the equilibrium constants for an assumed temperature, "T".

If combustion takes place at constant volume, as shown in Fig. 1, the maximum pressure "p" for an absolute temperature "T" may be found as follows:

$$144\, p\, V_2 = m_p\, GT \qquad (8)$$

$$144\, p_1 V_1 = m_m\, GT_1 \qquad (9)$$

therefore:

$$p = p_1 \times \frac{V_1}{V_2} \times \frac{T}{T_1} \times \frac{m_p}{m_m} = p_1 \times r \times \frac{T}{T_1} \times \propto \qquad (10)$$

In the above equations "p" and "T" represent the pressure and temperature of the products of combustion, without regard to the true final value p_3 and T_3, "r" the compression ratio and "$\propto$" the molecular ratio. This is the ratio of the number of molecules of the products of combustion to the number of the molecules of the mixture before combustion.

If "m_p" is replaced by the number of mols of a constituent, then "p" in equation (10), becomes the partial pressure of it and its value can be inserted in the various equilibrium equations. In cycle calculation for petrol engines it is usual to make the mixture "m_m" one lb-mol, while in Diesel engines the fresh air and the residual gases are kept as one lb-mol. Thus "m_m" before combustion will be higher, corresponding to the amount of fuel injected. Equation (1) can, therefore, be written:

$$\frac{\left[p_1 \times r \times \frac{T}{T_1} \times \frac{m_{co}}{m_m}\right]^2 \times \left[p_1 \times r \times \frac{T}{T_1} \times \frac{m_{o2}}{m_m}\right]}{\left[p_1 \times r \times \frac{T}{T_1} \times \frac{m_{co2}}{m_m}\right]^2} = K_1$$

$$\text{or } \frac{[m_{co}]^2}{[m_{co2}]^2} \times p_1 \times r \times \frac{T}{T_1} \times \frac{m_{o2}}{m_m} = K_1 \qquad (11)$$

In equation (11) m_{co}, m_{co2}, m_{o2}, are the number of lb-mols of CO, CO_2 and O_2 in the products of combustion

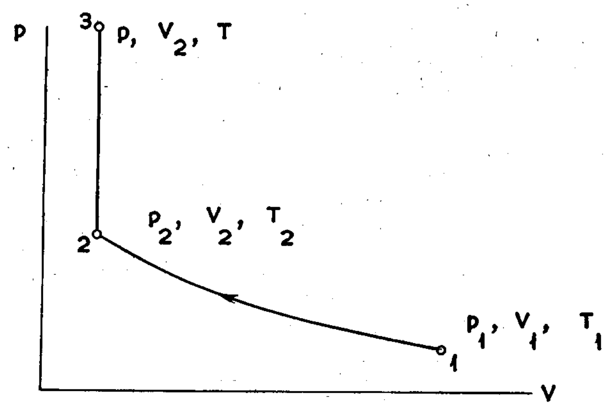

Fig. 1. Curve for combustion and constant volume

at the assumed temperature "T", and "p_1" the inlet pressure in atmospheres. Let x, y, z, 2p, 2q, 2s and t be the number of lb-mols of CO_2, H_2O, OH, H, O, N and NO in the products of combustion and a, b, c, and d represent the number of lb-mols CO_2, O_2 H_2 and N_2 if no dissociation were to occur, then the products of the combustion gas at point 3 in figure 1 will be:

$$xCO_2 + yH_2O + zOH + 2pH + 2qO + 2sN + tNO +$$
$$(a - x)CO + \left(b - \frac{x + y + z + t + 2q}{2}\right)O_2 +$$
$$\left[c - \left(y + \frac{z}{2} + p\right)\right]H_2 + \left(d - \frac{t + 2s}{2}\right)N_2 \qquad (12)$$

and "m_p" then becomes

$$\left(a + b + c + d - \frac{x}{2} - \frac{y}{2} + p + q + s\right) \text{lb-mols.}$$

* A. S. Leah. *The Ideal Efficiency of Petrol Engines.* T.I.M.E. Proceedings 1949 Vol. 161

The equilibrium equation (11) can now be written as:

$$\left(\frac{a-x}{x}\right)^2 \times \left(b - \frac{x+y+z+t+2q}{2}\right) \times \left(p_1 \times r \times \frac{T}{T_1} \times \frac{1}{m_m}\right) = K_1 \quad \dots \quad \dots \quad \dots \quad (13)$$

The product

$$\left(p_1 \times r \times \frac{T}{T_1} \times \frac{1}{m_m}\right)$$

is a constant as the values p_1, r, T, and m_m are given and should be called C_1.

The equilibrium equations can then be written in their final form:

$$\left(\frac{a-x}{X}\right)^2 \left(b - \frac{x+y+z+t+2q}{2}\right) = \frac{K_1}{C_1} = A \quad (14)$$

$$\left(\frac{a-x}{X}\right) \frac{y}{\left[c - \left(y + \frac{z}{2} + p\right)\right]} = K_2 \quad (15)$$

$$\frac{z^2}{y^2}\left[c - \left(y + \frac{z}{2} + p\right)\right] = \frac{K_3}{C_1} = B \quad (16)$$

$$\frac{4p^2}{\left[c - \left(y + \frac{z}{2} + p\right)\right]} = \frac{K_4}{C_1} = C \quad (17)$$

$$\frac{4q^2}{\left[b - \frac{x+y+z+t+2q}{2}\right]} = \frac{K_5}{C_1} = D \quad (18)$$

$$\frac{4s^2}{d - \frac{t}{2} - s} = \frac{K_6}{C_1} = E \quad (19)$$

$$\frac{t^2}{\left(b - \frac{x+y+z+t+2q}{2}\right)\left(d - \frac{t}{2} - s\right)} = K_7 \quad (20)$$

To solve equation (14)–(20) by the new method, one has to estimate firstly the number of lb-mols of CO_2 in the products of combustion represented as "x" in above formulae. As dissociation increases with increase of "T", a value of "x" may be selected within close limits of the actual value.

Assume for the first calculation the number of lb-mols of CO_2 and insert this value, called "x_1", in the seven equilibrium equations, then they may be written as:

$$\left(b - \frac{x+y+z+t+2q}{2}\right) = A\left(\frac{x_1}{a-x_1}\right)^2 = F \quad (21)$$

$$\frac{y}{\left[c - \left(y + \frac{z}{2} + p\right)\right]} = K_2\left(\frac{x_1}{a-x_1}\right) = G \quad (22)$$

$$\text{therefore: } \left[c - \left(y + \frac{z}{2} + p\right)\right] = \frac{Y}{G} \quad \dots \quad \dots \quad (23)$$

$$\text{but } \left[c - \left(y + \frac{z}{2} + p\right)\right] = B \times \frac{y^2}{z^2} \text{ (equ. 16)} \quad \dots \quad (24)$$

$$\text{and } \frac{y}{G} = B\frac{y^2}{z^2} \qquad \therefore y = \frac{z^2}{BG} \quad \dots \quad \dots \quad (25)$$

$$4p^2 = C\left[c - \left(y + \frac{z}{2} + p\right)\right] = C\frac{y}{G} = C\frac{z^2}{BG^2} \quad (26)$$

$$2p = \frac{z}{G}\sqrt{\frac{C}{B}} \quad \dots \quad \dots \quad \dots \quad \dots \quad \dots \quad (27)$$

Equation (23) can now be expressed in terms of "z" as

$$c - \frac{z^2}{BG} - \frac{z}{2} - \frac{z}{2G}\sqrt{\frac{C}{B}} = \frac{y}{G} = \frac{z^2}{BG^2}$$

$$\text{or } z^2\left(\frac{1}{BG^2} + \frac{1}{BG}\right) + \frac{z}{2}\left(1 + \frac{1}{G}\sqrt{\frac{C}{B}}\right) - c = 0$$

$$z^2\frac{(1+G)}{BG^2} + \frac{z}{2}\left(1 + \frac{1}{G}\sqrt{\frac{C}{B}}\right) - c = 0$$

$$z^2 + \left(\frac{BG}{1+G}\right)\left(G + \sqrt{\frac{C}{B}}\right)\frac{z}{2} - cG\left(\frac{B G}{1+G}\right) = 0$$

$$\text{if } \frac{BG}{1+G} = H \quad \dots \quad \dots \quad \dots \quad \dots \quad (28)$$

$$\text{then: } z^2 + H\left(G + \sqrt{\frac{C}{B}}\right)\frac{z}{2} - cGH = 0 \quad \dots \quad (29)$$

$$\text{and } z_{1,2} = \frac{-H\left(G + \sqrt{\frac{C}{B}}\right)}{4} \pm \sqrt{\frac{H^2\left(G + \sqrt{\frac{C}{B}}\right)^2}{16} + cGH} \quad \dots \quad \dots \quad (30)$$

$$\text{as: } 4q^2 = D\ F, \text{ equ. (18)} \therefore 2q = \sqrt{DF} \quad \dots \quad (31)$$

Using the positive root value only "y" and "p" can now be found by inserting "z_1" in equations (25) and (27).

From equation (21) "t" can then be found:

$$b - \frac{x_1 + y + z + t + 2q}{2} = F$$

$$\text{and } t = 2b - 2F - x_1 - y - z - 2q \quad \dots \quad \dots \quad (32)$$

after finding the value of "t", "s" can be calculated using equations (19)

$$4s^2 = Ed - \frac{E}{2}t - Es = 0$$

$$s_{1,2} = -\frac{E}{2}\left(\frac{1}{4} \pm \sqrt{\frac{1}{16} + \frac{d - \frac{t}{2}}{E}}\right) \quad \dots \quad (33)$$

$$\text{and then } K_7 = \frac{t^2}{F \times \left(d - \frac{t}{2} - s\right)} \quad \dots \quad (34)$$

In most cases, the value of "s" may be neglected, being too small to show any real heat losses due to dissociation of "N_2". Finally by inserting the values "t" and "s" into equation (34) the value of "K_7" may be found. If it does not correspond with the original value, a new value "x_2" has to be selected and the calculations repeated.

Using a tabular method, one quickly approaches the value of "K_7" and the corresponding values x, y, z, t, 2p, 2q and s should satisfy the seven equilibrium equations.

Example

A port-controlled two-stroke Diesel engine, with blower uses 46% excess air at full load. At the beginning of compression, the charge may consist of 85% fresh air and 15% residual gases; the latter have a ratio of 70% combustion gases and 30% air. All ratios are per weight and the air in the residual gases does not take part in the following combustion process.

Neglecting the small difference in molecular weight between air and the exhaust gases, the combustion equations for this problem using "$C_{12}H_{26}$" as fuel can be written as:

$$0{\cdot}0066C_{12}H_{26} + 0{\cdot}85 \text{ Air} + (0{\cdot}0133CO_2 + 0{\cdot}0143H_2O + 0{\cdot}0094O_2 + 0{\cdot}1130N_2 = 0{\cdot}0925CO_2 + 0{\cdot}1001H_2O + 0{\cdot}0655\ O_2 + 0{\cdot}7830\ N_2 \quad \dots \quad \dots \quad \dots \quad (35)$$

The above quantities are the number of lb-mols of CO_2, H_2O, O_2 and N_2 if no dissociation were to take place.

To find the dissociation quantities at 5000°R, the following data are used:

The compression ratio r = 14:1 (this is ratio of the remaining total volume after the piston has just covered the exhaust port to the clearance volume); the inlet temperature T_1 = 700°R, which is the temperature of fresh air, mixed with the residual gases, the inlet pressure p_1 = 1·155 atmospheres, and the dissociation coefficients at 5000 deg R as follows:

Values of "K" at 5000 deg R.

$K_1 = 1{\cdot}995 \times 10^{-2}$
$K_2 = 6{\cdot}546$
$K_3 = 1{\cdot}738 \times 10^{-3}$
$K_4 = 5{\cdot}888 \times 10^{-3}$
$K_5 = 2{\cdot}692 \times 10^{-3}$
$K_6 = 2{\cdot}089 \times 10^{-7}$
$K_7 = 8{\cdot}710 \times 10^{-3}$

Equation (12) can be written as:

$$xCO_2 + yH_2O + zOH + 2pH + 2qO + 2sN + tNO + (0{\cdot}0925 - x)CO + \left(0{\cdot}0655 + 0{\cdot}04625 + 0{\cdot}05005 - \frac{x+y+z+2q+t}{2}\right)O_2 + \left[0{\cdot}1001 - \left(y + \frac{z}{2} + p\right)\right]H_2 + \left(0{\cdot}7830 - \frac{2s+t}{2}\right)N_2 =$$

$$xCO_2 + yH_2O + zOH + 2pH + 2qO + 2sN + tNO + (0{\cdot}0925 - x)CO + \left(0{\cdot}1619 - \frac{x+y+z+2q+t}{2}\right)O_2 + \left[0{\cdot}1001 - \left(y + \frac{z}{2} + p\right)\right]H_2 + \left(0{\cdot}7830 - \frac{2s+t}{2}\right)N_2 \quad \ldots \ldots \quad (36)$$

Therefore:

$a = 0{\cdot}0925$
$b = 0{\cdot}1619$
$c = 0{\cdot}1001$
$d = 0{\cdot}7830$

$$A = K_1 \times \frac{T_1}{T} \times \frac{m_m}{p_1} \times \frac{1}{r} = \frac{1{\cdot}955}{10^2} \times \frac{700}{5000} \times \frac{1{\cdot}0066}{1{\cdot}155} \times \frac{1}{14} = 1{\cdot}7387 \times 10^{-4}$$

$$B = \frac{K_3}{C_1} = 1{\cdot}5207 \times 10^{-5}$$

$$C = \frac{K_4}{C_1} = 5{\cdot}152 \times 10^{-5}$$

$$D = \frac{K_5}{C_1} = 2{\cdot}3555 \times 10^{-5}$$

$$E = \frac{K_6}{C_1} = 1{\cdot}8278 \times 10^{-9}$$

Assume for the first calculation a value of "x_1" then "K_7" can be calculated after finding the corresponding values of y, z, t, and 2q only as "s" in most cases can be neglected.

The accompanying table gives values of "K_7" as derived after selecting various values of "X_1".

Conclusion

The tabular method gives a fairly quick way of finding the dissociation quantities, especially if a calculating machine is available. The values of a, b, c and d are constant for any temperature "T" and the remaining constants A, B, C, D and E remain unchanged for each temperature value of "T".

If the dissociation values for different temperatures "T" have to be calculated, it has to be remembered that with increase of "T" the quantity of "CO" increases therefore "x" has to be decreased and a new table can then be calculated. For most problems it is sufficient to stop calculations, if a value of "K_7" has been found which does not differ much from the original figure given in the tabulation at the top of this page.

VALUES USED IN FINDING VALUES OF K_7

	1st Trial	2nd Trial	3rd Trial	Final Trial
1. x_1	0·0870	0·0875	0·0876	0·08758
2. $a - x_1$	0·0055	0·0050	0·0049	0·00492
3. $\frac{x_1}{a - x_1}$	15·818	17·5000	17·8775	17·800
4. $\left(\frac{x_1}{a - x_1}\right)^2$	250·2091	306·25	319·605	316·843
5. $F = A \times \left(\frac{x_1}{a - x_1}\right)^2$	$4{\cdot}3326 \times 10^{-2}$	$5{\cdot}3247 \times 10^{-2}$	$5{\cdot}5569 \times 10^{-2}$	$5{\cdot}5089 \times 10^{-2}$
6. $G = K_2\left(\frac{x_1}{a - x_1}\right)$	103·5446	114·555	117·026	116·519
7. $H = \frac{B \cdot G}{1 + G}$	$1{\cdot}5061 \times 10^{-5}$	$1{\cdot}5061 \times 10^{-5}$	$1{\cdot}5078 \times 10^{-5}$	A $1{\cdot}50775 \times 10^{-5}$
8. z as per eq. (30)	0·01209	0·01270	·001285	0·01283
9. $y = \frac{z^2}{B \cdot G}$	0·09289	0·09258	0·09278	0·092899
10. $p = \frac{z}{2 \cdot G}\sqrt{\frac{C}{B}}$	not cal.	n.c.	0·000101	0·0001036
11. $2q = \sqrt{F.D.}$	0·00101	0·00119	0·00114	0·001139
12. $t = 2b - 2F - x - y - z - 2q$	0·0445	0·0236	0·0185	0·019
13. s = as per eq. (33)	n.c.	n.c.	n.c.	0·000019
14. $K_7 = \frac{t_o^2}{F \times \left(d - \frac{t}{2} - s\right)}$	$6{\cdot}0118 \times 10^{-2}$	$1{\cdot}357 \times 10^{-2}$	$7{\cdot}961 \times 10^{-3}$	$8{\cdot}652 \times 10^{-3}$

Note: The correct value of K^7 at 5000 deg R taken from Table 1 is 8·710 × 10P³