AN INVESTIGATION OF THE EFFECTS OF MULTIPLE SPARK IGNITION ON PERFORMANCE OF A HIGH SPEED PETROL SPARK-IGNITION ENGINE.

A Thesis
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In Partial Fulfillment
of the Requirements for the Degree
Batchelor of Engineering with Honours (Mechanical)

by
D. L. WALTERS
October 1954.
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CHAPTER 1

THE PROBLEM AND THE OBJECT OF THE INVESTIGATION

At the present time ignition by high tension electric spark is almost universal on high speed reciprocating internal combustion engines other than the compression ignition type. The growing demands for greater efficiency, higher specific power output and increasing speed range with the associated high compression pressures has made greater demands on the ignition system. The modern magneto and coil, having progressed from the early hot wire and hot tube ignition, have been developed to a high stage of efficiency, although others systems have been suggested.¹

Improvements in performance have been noted with dual ignition² and this system is standard on aircraft engines, although in this case improved performance is only a secondary consideration to safety.³


With dual ignition, one set of plugs is connected to the first magneto and the other set to the second magneto, the sparks occurring simultaneously at the two plugs. As a result the explosion is propagated from two points in the cylinder.

The sparking plugs are usually placed as far apart as possible and it was Swaine⁴ who suggested that better results might be obtained with the sparking plugs placed very close together. As far as the author is aware, no experimental work has been carried out along these lines.

1. THE PROBLEM.

Statement of the problem. It was the purpose of the investigation to examine the effects on (a) power output and (b) fuel consumption of a modern petrol spark-ignition engine using a dual coil ignition system with the sparking plugs placed very close together in the combustion chamber. It was also proposed to examine any associated effects on ignition time, peak pressures and range of burning, and where applicable, to compare results with those obtained with the standard ignition system.

Importance of the investigation. Although the days of the high power reciprocating aircraft engine appear to be numbered, the demand for a light, highly efficient and reliable powerplant in light aircraft and automobiles will probably be met, for many years yet, by the spark ignition petrol engine. Intensive development of the engine with increased compression ratios and refinements in design have resulted in higher power outputs per unit weight and improvements in fuel economy. Further improvements may have to be met by improvements in the ignition system especially in the use of low octane fuels and weak mixtures. There are some indications\(^5\) that present day coil ignition systems will be inadequate in the near future if the trend towards greater speed range and higher compression ratios continues.

As it has been stated before, no experimental work has been performed with this type of ignition system, the results may throw some light on one of the least understood phenomena in internal combustion engine design.

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11. OBJECT OF THE INVESTIGATION

The experimental results were of a qualitative nature only, a comparison being made with the performance of the engine fitted with the standard ignition system. This was done because the engine was not designed for experimental work and for this reason a number of complex and variable factors could not be controlled to a high degree of accuracy.

Scope of the investigation. The effects on power output and fuel consumption were of first consideration, the possibilities of using a common circuit breaker and distributor suitable for a multi-cylinder engine without undue complication was also investigated. The effect of increasing the number of plugs per cylinder up to three was determined.

An attempt was made to get some correlation between the igniting power of the spark and the condition of the charge at the point of ignition. The effect of altering the characteristics of the primary coil circuit was also investigated with a theoretical consideration of secondary current and voltage.

Much attention has been paid to combustion in engines at large throttle openings, but in automobile engines, especially those designed for passenger car
installations, low load factors are encountered. This means that the engine may be called to operate at full throttle for only a very small percentage of its total operating time. Investigations into effects on power and fuel consumption were therefore made at a wide range of throttle openings.

Organisation of the remainder of the thesis: The following order has been assumed for the presentation of the remainder of the thesis:

(1) A review of the literature in three parts.
The first section is a summary of the theories of ignition that have been advanced, with a discussion of experimental work leading up to the modern ignition theory. The second section is a discussion of the combustion process in the internal combustion engine with reference to the modern theory of spark-ignition. The third section is a summary of results obtained with dual ignition.

(2) A description of the apparatus and methods of measurement used in the experiments. The technique used in the preliminary tests is given together with that used in special tests devised to check the conclusions reached in the preliminary tests.

(3) A discussion of the results obtained. The effects on power, fuel consumption, ignition time are considered for each individual set-up.
(4) A summary chapter of findings, conclusions and recommendations. Here an attempt is made to relate the results with those forecast by the ignition theory.

(5) A criticism of the test apparatus with recommendations for future testing.

All design calculations for special test apparatus is given in the appendix together with theoretical cycle calculations. A mathematical treatment for the discharge of an ignition coil under working conditions is also given.
CHAPTER II.

REVIEW OF THE LITERATURE

A tremendous amount of research has been carried out into ignition and combustion problems. From early experimental work two theories arose, namely the thermal and the ionization theory, to explain the phenomena of spark ignition. The latest explanation that has been advanced, the chain reaction theory, has been evolved from further evidence but retains some of the concepts of the earlier theories.

I. THERMAL THEORY OF IGNITION

Statement of the theory. For an explosion to take place in a combustible gas mixture, a certain minimum volume of the gas must be heated to a definite ignition temperature. The minimum ignition volume is determined by the composition and nature of the gas mixture. If the explosion is to be self propagating, the heat liberated by the progressively explosive reaction must be greater than the heat dissipated by conduction to the surrounding cold layers of the gas.

Expressions for a simple quantitative treatment. From a purely thermal consideration, disregarding the effects caused by concentration changes from diffusion
and reaction, if $T$ is the temperature at a distance $r$ from the origin at time $t$ after heating begins, if $k$ is the thermal conductivity, then the equation for heat conduction is:

$$kr^2 \frac{\partial^2 T}{\partial r^2} + \frac{2k}{r} \frac{\partial T}{\partial r} = \frac{\partial T}{\partial t}$$

The solution of this equation will depend on the quantity of heat supplied $Q$, the temperature and type of the source.

From this equation, assuming an ignition temperature, it can be easily indicated what amounts of energy should be added by point sources, or sources lasting over a certain period of time.

Discussion of experimental evidence. Experiments by Coward and Meiter\(^1\) with methane-air mixtures showed that it was possible to bring an appreciable volume of the mixture to combustion by a spark without causing general ignition. Under certain conditions it was possible to send thousands of sparks through a highly combustible mixture without an explosion occurring. Deviations from the chemically correct mixture required greater energy.

greater energy for ignition, and it was shown that the
volume of the gas that could be consumed before ignition
rises rapidly. From these results and others\(^2\), with
methane-air mixtures, Coward and Meiter calculated
minimum ignition volumes, assuming an ignition temperature,
which gave satisfactory results. Coward and Meiter
concluded,

> Nothing in the results of (their) experiments
> suggests the intervention of any electrical effect of
> the spark... other than the normal effect of the
degradation of its electrical energy.\(^3\)

Thornton\(^4\) showed that the thermal energy of a
spark just capable of igniting a mixture varies considerably
with the electrical conditions and put forward the suggestion
that some sort of ionization preceded combustion. This
can be anticipated on thermal considerations alone,
however, as was pointed out by Taylor-Jones, Morgan and
Wheeler.\(^5\)

\(^2\) J.D. Morgan, "Ignition of Combustible Gases by
Electric Sparks," Philosophical Magazine, Volume 45,
p. 968 (1923)

\(^3\) Coward and Meiter, op. cit. p. 399

\(^4\) W.M. Thornton, "Electrical Ignition of Explosive
Mixtures," Proceedings Royal Society (London A) Volume 90,
p. 274, (1914)

\(^5\) E. Taylor-Jones, J.D. Morgan, R.V. Wheeler, "On
the Form of Temperature Wave Spreading by Conduction, from
Point and Spherical Sources; with a Suggested Application
to the Problem of Spark Ignition," Phil. Mag. Volume 43,
p. 359 (1922)
Considering electric sparks of two types:
(1) Exceedingly short (capacity sparks) and (2) Relatively long (inductance sparks) and considering four cases of heating as follows:

(A) Instantaneous at the origin

(B) At the origin at a uniform rate during the time L

(C) Instantaneous over a spherical surface of radius a

(D) Instantaneous through a spherical volume of radius a

If Q is the quantity of heat supplied and c is the thermal capacity, the equation for thermal conduction (p. 8) gives as a solution for Case (A)

\[ T = \frac{Q e^{-r^2/4kt}}{8c (\pi kt)^{3/2}} \]

Similarly, given the boundary conditions, the equation can be solved for the other three cases and the results are shown graphically in Figure 1. It can be seen from the figure that

(1) There is no advantage in raising the source, from a point to a uniformly supplied spherical volume 1 mm. diameter.

(2) The two sources mentioned above will have the same effect as a spherical surface of 2 mm. diameter.
TEMPERATURE WAVE FORMS OF FOUR METHODS
OF HEATING AIR [Taylor-Jones, Morgan & Wheeler]

A. Instantaneous Point Source  \( t = 0.00275 \text{ sec.} \)
B. Continued Point Source  \( t = 0.006 \text{ sec.} \)  \( L = 0.005 \text{ sec.} \)
C. Instantaneous Spherical Surface  \( a = 0.1 \text{ cm.} \)  \( t = 0.0005 \text{ sec.} \)
D. Instantaneous Spherical Volume  \( a = 0.05 \text{ cm.} \)  \( t = 0.002 \text{ sec.} \)
(3) An increase in the size of the source, however small, will give better heat conduction.

These results will be considered again later in the general discussion on ignition theories.

II. IONIZATION THEORY OF IGNITION

General Statement of the Theory. Ignition in combustible gas mixtures by means of a spark discharge is in some way associated with the ionization process. The type of electrical discharge is therefore of fundamental importance in the initiating of the explosion.

Investigations in the ignition properties of various types of spark. Early investigations showed that the electrical conditions affected the igniting properties of the spark discharge. Thornton found that it was possible to have brilliant sparks which did not cause ignition of the most inflammable mixtures. Thornton suggested, "That a gas has a particular temperature of inflammation may mean that ionization begins at this temperature."

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A series of experiments by Finch and Cowen\textsuperscript{7} showed that a high tension arc dissipating energy at a surprisingly high rate could be maintained in explosive gaseous mixtures without causing ignition. In the case of hydrogen-oxygen mixtures it was found

(1) Ignition occurs without lag immediately on the attainment of a certain limiting current,

(2) \( p_z i_z = \) constant over a considerable range. Where \( p_z \) is the pressure and \( i_z \) is the minimum ignition current.

It was concluded that as the concentration or ions, or of molecules or atoms excited to any particular state, is also approximately a hyperbolic function of the gas pressure in which the potential drop across the discharge is constant, it followed that the ignition was determined by the attainment of a certain definite concentration of suitably excited molecules or atoms. (Note that this conclusion does not specify ionisation but 'suitably excited atoms or molecules'. As it is seen later, the modern theory of spark-ignition uses a similar concept).

A further series of experiments were made with carbon monoxide-air mixtures ignited by condenser discharges of known oscillation frequency. Igniting powers were determined by the minimum ignition pressure. The effect of frequency could be studied independently of

(a) The total amount of energy dissipated
(b) The rate of energy dissipation

The results of the experiments showed

(1) The igniting power of the spark was independent of the value of the peak current.
(2) The igniting power was determined by the natural frequency of the circuit to such an extent that a suitable decrease in frequency could outweigh the effect on any possible reduction in igniting power due to a decreased amount of rate of energy dissipation, or both, either by the first half oscillation of the spark or the whole discharge.

These results, shown graphically in Figure 2, conflict with the thermal theory but consistent with an excitation theory as a high frequency spark is a rich source of ionization.

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Influence of Frequency and Maximum Current of a Condenser Discharge on Ignitability

[Bradford & Finch]

Carbon Monoxide - Oxygen Mixture
0.041 \mu F Capacitance

![Graph showing the relationship between frequency and igniting power. The graph indicates a decrease in igniting power with increasing frequency, while the peak current increases with frequency.](image-url)
III. CHAIN REACTION THEORY OF IGNITION.

General statement of the theory. In a gas mixture ignited by an electric spark, a small spherical volume of the gas is heated instantaneously, and at the same time a small quantity of active particles is created. It is of no importance whether these particles are ions, atoms or molecules; nor is it important to state how such particles are created. It is merely sufficient to state that a heat generating reaction takes place at a rate proportional to the concentration of the active particles, and that this reaction varies in intensity as the active particles diffuse through the gas and increase at a rate proportional to their local concentration. Now the temperature at the centre of the sphere tends to fall as heat is conducted away and rise as heat is generated. Thus, for the reaction to be self sustaining, we must assume a reaction rate such that this temperature never decreases during the combustion process.

Summary of the theories of spark ignition. A similarity between the chain reaction theory and the thermal theory is noticed, but it is obvious that ignition cannot be understood exclusively as a thermal process, simply raising the temperature of a combustible gas mixture will not make it burn -- a certain type of chemical activity peculiar to the flame must be initiated.
Jost\textsuperscript{9} is emphatic in his statement that activation of ions and molecules occurs before the spark energy depreciates to heat and is, therefore, of prime importance in the ignition process. Jost suggests, however,

\ldots that with a theory not purely thermal in character, diffusion, perhaps especially that of free atoms and radicals, comes into play. Since diffusion follows the same laws as those for heat conduction, it is completely conceivable that the concepts of a pure thermal theory can be retained.\textsuperscript{10}

It is possible to conclude that the igniting power of the spark is dependent mainly on its molecular stimulation and eventual dissociation, partly through its thermal effect and only very slightly through its ionizing effect. The ionization is probably a supplementary rather than a fundamental mechanism of ignition.

**General characteristics of spark ignition as indicated by the theory.** It is possible to reach some general conclusions from the evidence.

(1) For a given spark energy, the explosion will occur within certain limits of composition. More energy is required to initiate the explosion near these composition limits, the amount of energy being required rising rapidly

\begin{flushright}
\textsuperscript{10} Ibid. p. 61.
\end{flushright}
Minimum Ignition Currents (in primary)
for Hydrocarbon - Air Mixtures

[Jos1]
as the limits are approached.

(2) For any gas mixture, there is a limiting ignition pressure and energy, the ignitability decreases as the pressure is lowered.

(3) Cooling effects of boundary walls and electrodes may have considerable influence in determining whether or not the flame will propagate.

IV. PROPAGATION OF THE EXPLOSION

In engine combustion, we are dealing with a "progressive explosive reaction," that is, the reaction velocity keeps increasing and the only limiting factor is the complete consumption of the charge. Although a number of factors prevent the results being applied directly, results with this type of reaction with hydrocarbon fuels do give a good indication of the fundamentals of the process.

Observations with bomb explosions. Explosions in closed vessels show that there is always a noticeable interval of time after the passage of the spark to when there is a perceptible rise of pressure.11 This interval or induction period, after which the reaction begins to accelerate

was originally thought to be "ignition lag," that is, the time required for a self propagating nucleus of flame to be built up. Flame photographs\textsuperscript{12}, however, clearly show that a considerable volume of combustion takes place during this period and ignition lag accounts for only a minute fraction of it.

The effect of turbulence was investigated by David\textsuperscript{13} who fitted a fan inside the combustion chamber which could be run at various speeds. It was found that turbulence had very little effect on the delay period but a marked effect on the subsequent propagation of the flame. It has been subsequently shown\textsuperscript{14} that during the delay period, the early pressure rise is very slow but nevertheless present, (For a specific case, the pressure rise over the first 20\% of flame travel = pressure rise over the last 0.0107\%).

\textbf{MEASUREMENT OF FLAME VELOCITY.} Attempts to measure the velocity of flame propagation in a combustion bomb are complicated by the fact that the pressure rises and the progress of the explosion is affected by the compression of the fresh gas by the combustion gases. To overcome this


difficulty, Flock and Roeder\textsuperscript{15} initiated the explosion at the centre of a soap bubble containing a combustible gas mixture. The bubble was photographed through a narrow slit which left only its horizontal diameter visible. The film was carried on a drum rotating at a known constant speed about an axis parallel to the slit so that as the diameter of the flame increased, the lengthening image moved along the screen and produced a V - shaped trace which constituted a time - displacement record of the flame front.

From this trace the velocity of combustion could be calculated directly. Results showed that the flame speeds are a maximum at approximately chemically correct mixture strength but decreases as the mixture becomes too rich or too lean.

\textbf{V. COMBUSTION IN THE ENGINE}

Combustion in the internal combustion engine is a highly complicated process and is still not fully understood. Considering the normal combustion process and disregarding effects such as detonation, the problem is still far from easy as in the combustion chamber the mixture is turbulent, the combustion volume is continually changing, and the

PRESSURE RISE DURING COMBUSTION IN A CLOSED COMBUSTION BOMB WITH TURBULENCE

[David]
incoming charge is heated by conduction from the cylinder walls. Using hydrocarbon fuels, this heating of the charge can cause dissociation and chain branching, the charge may also be diluted with quantities of hot residual exhaust gas.

Flame movement and velocity. It was previously thought that after the passage of the spark, turbulence broke up the nucleus flame into small parts which were carried into various parts of the combustion chamber thus virtually setting up large numbers of ignition centres. Flame Photographs\textsuperscript{16} show, however, that after the passage of the spark there is a short interval before the flame appears (ignition lag), the flame then spreads outwards, slowly at first, then more rapidly. The entire flame front is extremely turbulent, turbulence in the remainder of the combustion chamber is of a similar nature but on a smaller scale.

Pressure development. From pressure - time diagrams taken from engine cylinders, it is possible to make a strict distinction between two phases of pressure development; a delay period without an appreciable pressure rise and a combustion time proper. The pressure rise is caused by

the flame front moving across the combustion space, the heat released by combustion causes the unburned portion of the charge to expand and so compress the burned and unburned portions isentropically, the increase in pressure being directly proportional to the percentage of the charge that has been consumed.\(^{17}\)

**Temperature variations in the products of combustion.** Following the pressure variation in the cycle, the first part of the charge burns at constant pressure with a large temperature rise, then undergoes an adiabatic compression. The last portion of the charge is compressed adiabatically and then burns at constant pressure. The temperature rise during the adiabatic compression is greater for the first part of the charge and the resulting temperatures are different, even although the amount of heat added during combustion at constant pressure is essentially the same for both.

The temperature of the charge decreases progressively from the first to the last part of the charge to be burned, the temperature of the gas in the vicinity of the spark plug is therefore higher than that of any other portion

\(^{17}\) C.D. Miller, "The Roles of Detonation Waves and Autoignition in Spark Ignition Engine Knock as Shown by Photographs Taken at 40,000 and 200,000 Frames per Second," S.A.E. Quarterly Transactions, Volume 1, No. 1 (1947)
of the charge.

**Factors affecting flame velocity.** By far the most important factor affecting flame speed is turbulence. The combustion reaction can be expressed in the following manner:

\[
\text{Mass rate of burning} = \text{(Instantaneous area of the flame front)} \times \text{(Velocity of advance)} \times \text{(Density of the mixture)}
\]

The effect of turbulence is to make the flame front ragged and so increase its area. The flame is propagated in much the same way as in stagnant mixtures, the flame tending to follow the shape of the retaining walls. Movements occur in the unburned charge due to general swirl and local turbulence.

It should be noted at this stage the above discussion applies to combustion in the second stage of pressure development. Ricardo\(^{18}\) regards the two stages as quite distinct, "... one the growth and development of a self propagating nucleus of flame, the other as the spread of that flame throughout the combustion chamber." Taking this concept as a working definition for the present, the second stage is purely thermal in character, the flame speed being relatively unaffected by changed in fuel-air ratio, temperature, exhaust gas dilution and other factors.

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Increased turbulence is limited by the increase in direct heat losses to the cylinder walls, for smooth running and optimum efficiency a pressure rise of 30-35 lb./sq.in. per degree of crank angle rotation is indicated.\textsuperscript{19} As the degree of turbulence is proportional to the engine speed, the time required for the second stage of combustion will occupy approximately the same amount of crankshaft rotation at all crankshaft speeds.

The delay period. The delay period is chemical in origin and an analogy can be drawn here with experimental results obtained with combustion bombs (Induction period). Broeze\textsuperscript{20}, showed that the delay period, for a given condition of the engine, is dependent on the nature of the fuel, the mixture strength, temperature and density of the charge.

Results (Figure 6) show that the combustion time is only affected slightly by changes in mixture strength, but is dependent on the type and degree of turbulence.

The delay period, however, is seen to be a minimum when the mixture is about 20\% rich and increases with deviations from this value. The delay period is also

\textsuperscript{19} Ricardo, op.cit. p.13

Figure 5.

Variation of Combustion Time and Delay Period with Fuel-Air Ratio and Turbulence.

(David)

1. Unshrouded Inlet Valve (low turbulence)
2. Shrouded " " (Swirl)
3. Castellated " " (high turbulence)
influenced by the type and degree of turbulence, swirl being more effective than turbulence in reducing ignition delay.

The explanation may be that the decreases in the delay period due to swirl is due to the decreased boundary layer thickness. Broeze suggests,

In the boundary layer along the combustion chamber wall the combustion progress velocity is at first slow until it is caught up by the main body of the charge, in which combustion the combustion chamber wall combustion progress velocity is increased by its state of movement.21

This would explain the efficiency of long reach plugs in igniting lean mixtures as observed by Taub.22

These results can also be called upon to account for the cyclic variations observed in indicator diagrams taken from spark-ignition engines. The variations take place mainly in the initial delay period at low compression ratios or reduced loads.23 This may be due to differences in the fuel-air ratio, especially in the case of carburetted mixtures, or residual gas dilution, as it is noticed that the variations are greater when the sparking plug is not well scavenged.

21 Ibid., p. 465.
VI. THE PROCESS OF IGNITION

A detailed study of the theory of coil ignition is given in Appendix B. A summary of the general characteristics of the spark discharge is first given, together with the definitions of the terms used. The discussion is confined here to the main factors of the spark discharge that determine ignition.

Characteristics of the spark discharge. A summary of the properties of the spark discharge is as follows:

(1) The spark consists of two components, the first being the capacity component which ionizes the gap and renders it conductive. The capacity component is the discharge of the energy stored in the capacitance of the system. The inductive component which follows consists of the energy stored in the magnetic part of the circuit.

(2) The capacity component is of very short duration and carries a high peak current, the inductive component lasts considerably longer carrying a much smaller current, as the energy can only be released slowly.

(3) Every spark gap has a certain time lag, there being a small interval for ionization to appear, then for the process to develop until the space between the electrodes is completely ionized to a stage that produces a spark discharge.

(4) For a given set of gap conditions the time
lag is constant, and the voltage will rise to a value 
$E_1$ dependent on the rate of voltage rise at the gap. 
Under a very slow application of voltage, the spark 
discharges at a voltage $E$. The ratio $\frac{E_1}{E}$ is called 
the "impulse ratio."

Factors affecting sparking potential. The factors 
affecting the sparking voltage are numerous and complex, 
apart from the electrical characteristics of the circuit, 
any factors influencing the ionization of the gases between 
the electrodes will alter the voltage required for the 
spark.

Among the factors affecting sparking potential are 
(a) gap dimensions (b) shape and disposition of the 
electrodes (c) material of the electrodes (d) nature of 
the gas mixture (e) pressure and temperature of the gas 
(f) electrode temperature (g) wave form of the applied 
e.m.f.

Experiments by Patterson and Campbell\textsuperscript{24} showed that 
the effect of substituting a mixture of petrol-air vapour 
for pure air was to reduce the sparking voltage slightly

\textsuperscript{24} G.C. Paterson and N. Campbell, "Some 
Characteristics of the Spark Discharge and its Effect on 
Ignition Explosive Mixtures," Proc. Physical Society, 
Volume 31, Part 4, p. 168. (June, 1919)
The effect of increasing the electrode temperature is to decrease the sparking potential, it is due to this effect that the voltage of the ordinary ignition system is kept within bounds, a 50% reduction has been observed with a rise in electrode temperature of 600°C.\textsuperscript{25} This is probably due to the fact that the hot electrode is surrounded by a layer of heated gas at a density below that of the remaining charge. The effect of turbulence would be to increase the sparking voltage by removing this heated layer. This is observed when sparking voltages are determined in an air blast.\textsuperscript{26}

For a given pair of electrodes, the effect of increasing the gap width is to increase the sparking potential in almost direct proportion. A similar relationship is observed with increase of pressure. In applying these results to the calculation of sparking voltage to an engine running under operating conditions, however, the cumulative effect of all the variables is very difficult to determine, for example, an increase in gas pressure should increase the sparking potential, but the temperature of the plug may be raised at the same time.


\textsuperscript{26} Ibid. p. 441
**Effect of heat energy of the spark.** Considering the process of ignition from a purely thermal viewpoint, from the earlier discussion it is seen that the heat must be supplied as quickly as possible (Figure 1) to minimise the heat loss during the critical period of initiating the combustion. The energy requirements for ignition, shown graphically in Figure 2, indicate that a large increase in spark energy is required to extend the mixture range at any given pressure. It should be noted that in this case it is the capacity component that will cause ignition, any augmentation of the spark energy cannot be obtained by altering the magnetic characteristics of the circuit, that is by making the spark "fatter."

Experimental results have produced evidence to support this view, showing that the capacity spark is the normal form of spark discharge and that the energy required for ignition decreases as the spark potential increases.27

The electrostatic energy in the capacity component is \( \frac{1}{2} CV^2 \), where \( C \) = capacitance of the circuit and \( V \) = sparking voltage. The only way to increase this energy is to increase the breakdown voltage of the gap by increasing the gap length, or by increasing the capacity of the circuit. The former method is to be preferred, as

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27 Paterson and Campbell, *op.cit.*, p. 171
Minimum Spark Energies Required to Ignite
a Petrol-Air Mixture (Paterson & Campbell)

\[ P = \frac{\text{Wt of petrol injected into Combustion Chamber}}{\text{Wt of air which would occupy the chamber at the same temperature and pressure}} \]

![Graph showing the relationship between sparking voltage and spark energy for different values of \( P \).]
Variation of Minimum Spark Energy with Mixture Strength at Different Sparking Potentials (Paterson & Campbell)

\[ P = \frac{\text{Wt of petrol injected into the Combustion Chamber}}{\text{Wt of air which would occupy chamber at the same temperature and pressure}} \]

**FIGURE 7.**

Minimum Spark Energy (Joules x 10^-5) vs Value of p
increasing the gap length also decreases the cooling effect of the electrodes. The limiting factor is the maximum voltage that the ignition system can supply.

**Ignition properties of the components of the spark discharge.** Although a purely thermal theory can give useful results, as has been stated previously, ignition cannot be fully understood from a simple consideration of this kind. Later experiments with coil ignition showed that the capacity component was not necessarily responsible for bringing about ignition. It was found that the igniting power of the spark decreases with suppression of the inductive component. 28

A further series of experiments, carried out with a view to determining the igniting power of the components of the coil discharge for the operating conditions of an internal combustion engine, showed that the performance of the engine was unaffected by changes in the duration of the discharge. It was shown, by means of cathode ray oscillograph analysis, that the only portion of the discharge required for ignition, is the short initial portion of the inductive component. The duration of the

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discharge can be reduced to about one-tenth of its normal value without affecting its igniting properties. 29

Summary of the factors affecting ignition in the internal combustion engine. As it can be seen from the above discussion, the problem of ignition is still obscure and not fully understood.

Probably the rate of dissipation of the spark energy is the most important factor in the process of ignition, the sparking voltage, spark energy in the capacity component and the rate of voltage rise having lesser degrees of influence. The problem is further complicated by the wide range of operating conditions the ignition system is called upon to handle. The inductance component may be required to ignite mixtures under extreme conditions of imperfect carburettion and more especially, under cold starting conditions requiring high heat energy output of long duration as the mixture may be in the form of droplets which must be vapourised before normal ignition will take place.

Considering the problem in the light of the chain reaction theory, during the early stages of ignition and propagation, a surface to volume ratio between the burned

and unburned mixture favourable to reduced heat transfer will be desirable. (See Appendix).

VII. DUAL IGNITION

In this system, used in aircraft engines, the ignition systems are completely independent. Two sparks occur in the combustion chamber and the explosion is propagated from two sources instead of one.

Experimental results with dual ignition. Ricardo carried out a series of carefully conducted tests for the purpose of showing the influence of the sparking plug position on detonation. The engine used for the tests was a single cylinder "E.35" variable compression engine provided with four radial plugs, any two of which could be fired simultaneously. The plug positions and their relation to that of the valves is shown in Figure 8. The results of the tests are given in Table 1 and show that the ignition advance required and the highest useful compression ratio (H.U.C.R.) are both dependent very greatly upon the length of flame path from the sparking plug to the farthest point reached by the flame. The relative positions of the active sparking plugs and that of the various valves are shown to exert a secondary influence

30 Ricardo and Glyde, op.cit., p. 56.
FIGURE 8.

RELATIVE POSITIONS OF VALVES AND SPARKING PLUGS
FOR EXPERIMENTS WITH DUAL IGNITION

(Ricardo and Clyde)
TABLE I.

INFLUENCE OF SPARKING PLUG POSITION ON DETONATION
(from Ricardo)

Test conditions:

- Speed: 1500 r.p.m.
- Mixture strength: 15% rich (maximum power setting)
- Jacket temperature: 50°C
- Heat input to the induction system: 1350 watts
- Fuel: "Texas gasoline"
- "E.35" variable compression engine

<table>
<thead>
<tr>
<th>Plug Position</th>
<th>Ignition Timing</th>
<th>H.U.C.R.</th>
<th>B.M.E.P.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>giving max. torque</td>
<td></td>
<td>1 lb./sq.in.</td>
</tr>
<tr>
<td>1 and 3</td>
<td>30° early</td>
<td>5.30</td>
<td>138.0</td>
</tr>
<tr>
<td>2 and 4</td>
<td>30° &quot;</td>
<td>5.25</td>
<td>136.6</td>
</tr>
<tr>
<td>3 and 4</td>
<td>34° &quot;</td>
<td>4.98</td>
<td>134.5</td>
</tr>
<tr>
<td>1 and 2</td>
<td>32° &quot;</td>
<td>4.90</td>
<td>133.8</td>
</tr>
<tr>
<td>1 and 4</td>
<td>32° &quot;</td>
<td>4.80</td>
<td>132.1</td>
</tr>
<tr>
<td>2 and 3</td>
<td>32° &quot;</td>
<td>4.84</td>
<td>132.8</td>
</tr>
<tr>
<td>1 only</td>
<td>39° early</td>
<td>4.95</td>
<td>133.5</td>
</tr>
<tr>
<td>2 only</td>
<td>39° &quot;</td>
<td>4.90</td>
<td>133.1</td>
</tr>
<tr>
<td>3 only</td>
<td>40° &quot;</td>
<td>4.84</td>
<td>132.8</td>
</tr>
<tr>
<td>4 only</td>
<td>42° &quot;</td>
<td>4.85</td>
<td>132.8</td>
</tr>
</tbody>
</table>
Still more recently, experiments to determine the effect of combustion time on knock have been carried out. In this case a single cylinder engine was fitted with 17 spark plugs which fired simultaneously. By selecting the number of plugs firing the combustion time could be varied. In these experiments, a significant decrease in the octane requirements was observed.31

Discussion of the results obtained with dual ignition. The results obtained by Ricardo appear to be rather disappointing. This is perhaps due to the fact that the last portion of the charge to burn is subjected to intensive radiation from the two advancing flame fronts. As it has been shown before, a consideration of the ignition process in the engine on a thermal basis gives very reasonable results. Two ignition sources very close together, on thermal considerations, would theoretically give better performance as this would increase the size of the source. (c.f. p. 12). The investigation follows this line of thought.

CHAPTER  III

DESCRIPTION OF THE APPARATUS

An elaborate test apparatus was out of the question so that where possible, existing equipment in the laboratory was used. The problem of getting two or more sparking points inside the combustion chamber was formidable, as it was unlikely that any New Zealand manufacturer would have sufficient experience to make a successful sparking plug with two electrodes of the same order of size as the standard plug. For this reason, a side-valve engine was preferable to an overhead valve engine, as greater accessibility was available with the former type. The difficulty of cooling forced the adoption of a water-cooled engine.

A four cylinder, water cooled, side-valve engine was made available and was installed. Although a single cylinder engine may have offered some advantages, there were none available that were considered suitable.

I. THE ENGINE TEST BED

General arrangement of the equipment. The engine was an Austin "Big Seven", the specifications of which are given in the Appendix. The engine was mounted on a cast iron bedplate through simple mountings made up out of I beams. The mountings were bolted down through holes tapped in the base of the bedplate, the brake being
similarly mounted directly behind the engine.

The drive from the engine to the brake passed through two universal joints, the engine and brake shafts were required to be parallel but did not need to be exactly in line. The spark-advance indicator was fitted to the front of the brake. The instruments for recording temperature were housed in a test panel directly in front of the engine. The instruments for recording air and fuel consumption were part of the equipment of the Ricardo E6/3 engine and were mounted some distance along the wall at the back of the test bed.

Plates I and II show the layout of the equipment.

Measurement of speed. A revolution counter fitted to the brake gave an approximation to the true speed which was checked finally by a stroboscope. The speed could be determined accurately in steps of 250 R.P.M.

Measurement of power. The power was absorbed through a Froude water brake. The capacity of the brake was 100 H.P., well outside the capacity of the engine. For this reason the spring balance, which recorded the brake torque, was not as sensitive as it could have been, especially when the engine was running under light load. The brake mean effective pressure could be determined to an accuracy of \( +0.3 \text{ lb./sq.in.} \).
PLATE I.

LAYOUT OF TEST EQUIPMENT (NEAR SIDE)
PLATE II.

LAYOUT OF TEST EQUIPMENT (OFF SIDE)
Measurement of fuel consumption. Fuel consumption was measured on a volumetric basis, the weight of fuel being deduced after taking specific gravity readings with a hydrometer. Two bulbs, the upper of 50 cc. capacity and the lower of 100 cc. capacity, supplied fuel by gravity to the carburettor. The time taken for the fuel level to pass graduation marks gave a measure of the rate of fuel flow. A selection of three volumes of fuel was thus available to suit the engine load.

Measurement of air consumption. It was found that the original fuel-air meter fitted was inaccurate shortly after testing began. The alternative was to fit an air meter and to obtain the fuel-air ratio by measuring the air flow and fuel flow separately.

Three methods were possibilities: (1) orifice meter with pulsating tank, (2) viscous flow meter (3) hot wire instruments. Of these, the last two were discarded as both require calibration. This would be difficult as no suitable apparatus was available.

The orifice meter would be relatively easily constructed, the size of the pulsating tank required, however, would be of the order of 40 gall. capacity. ¹ This

was too bulky. The problem was solved by utilising some of the results obtained from silencer research (see Appendix).

An orifice, calibrated previously from steam flow, was fitted to a silencer on the intake side of the carburettor. The silencer was designed to damp out all oscillations in the manifold within the speed range of the engine. A lead to an alcohol filled manometer measured the head across the orifice, the manometer having three slopes for low, medium and high air flow rates.

This apparatus proved satisfactory, little oscillation on the manometer being observed even when running at full throttle at low speed. With correction for ambient air temperature and pressure, it was estimated that the air flow could be measured accurately to ± 1%.

Measurement of spark advance. The spark-advance indicator was a neon tube connected to the sparking plug terminal through a 100,000 ohm resistance. A slot was cut in the indicator disc which was fitted to the brake coupling. On the outer fixed part of the indicator, a scale was set showing the position of top dead centre (T.D.C.) The position of the neon tube could be altered against the scale so that when the neon tube fired opposite the slot, the angle of advance could be read off directly against the scale.
In practice, it was found that the indicating flash would fluctuate anything up to 2°, possibly due to irregularities in the cam profile.

Measurement of temperature. Thermometer elements, working on the Wheatstone bridge principle, were set at (a) cooling water inlet, (b) cooling water outlet and (c) drain plug of the sump to record oil temperature.

These thermometers were checked previously against mercury in glass thermometers. Before each test each was zeroed. Additional thermocouple elements could be placed to record manifold or plug temperature if required.

Measurement of manifold pressure. An aircraft manifold pressure gauge was fitted to a tapping in the manifold just below the "hot spot," reading to 0.1 in. mercury.

Engine cooling. The cooling system used was that used for another test engine in the laboratory and consisted of a 40 gal. water tank connected to a small electrically driven circulating pump by \( \frac{\pi}{4} \) in. piping, hence to the engine, the cooling circuit being completed with a return pipe to the top of the tank. An additional lead from the town supply allowed fresh water to be supplied to the tank when the temperature of the water became too high. An
overflow pipe was fitted to the top of the tank. Control of the incoming supply and the flow rate of the circulating water was provided by two valves on the test panel.

With this arrangement, it was found that while the water in the tank was cold, the flow rate through the engine was low. As the temperature of the water rose, however, the flow rate had to be increased until it reached a stage where it was beyond the capacity of the circulating pump. Fresh water then had to be admitted. Due to the positioning of the inlet pipe, the incoming cold water had a tendency to flow directly to the engine, the flow having to be drastically reduced to keep the outlet temperature constant. In the meantime, the original hot circulating water was being pushed out into the overflow pipe.

As it was desirable to have a small temperature difference across the engine, the method used in later tests was to keep the inlet valve open slightly. This gave the best compromise, the small quantity of water entering kept the temperature of the water in the tank at a reasonable level and only a small amount entered the engine and was mixed with some of the hot water in the tank. In this way the temperature of the inlet water could be held at approximately 40°C.

The uncertainty of control of the inlet water temperature was probably the cause of erratic readings in
early experiments.

Carburettor. It was desirable to have a means of altering the mixture strength while the engine was running. The standard carburettor fitted, a Zenith downdraught type, had no means of adjustment as the jets were fixed. It was suggested that a needle valve be fitted to the main jet but there was so little metal housing the jet that it was thought that the die cast alloy would break away if an attempt was made to drill and tap for the shank of a needle valve. An S.U. carburettor having a means of mixture adjustment, was therefore fitted.

The only type available suitable for the engine capacity, was a side draught type and an elbow had to be made up to adapt it to the manifold. It was found with this arrangement that the value of the "hot spot" was lost and that the charge was probably unevenly distributed to the cylinders. It was also found that the range of needle sizes was not wide enough to allow sufficient alteration to the mixture range, especially at large throttle openings.

An attempt was then made to fit a needle valve to the Zenith carburettor. This was successful, giving a wide range of adjustment, the only limit being set by the size of the main jet with rich mixtures.

Two sets of results, with different type carburettors, have to be considered.
II. SPECIAL EQUIPMENT

Multiple sparking plug. It was thought that a successful special plug with two electrodes could not be made by New Zealand manufacturers so a special boss had to be made up housing the central electrode and insulators of standard plugs. A cross-section of this boss is shown in Figure 9.

The central electrode was required to be of as small a diameter as possible but in a suitable heat range to reduce the danger of pre-ignition. Lodge H-40 plugs were finally chosen. (10mm. plugs of suitable type were not obtainable). The outside diameter of the plug boss also had to be as small as possible as there was not much metal around the plug tappings in the cylinder head. It was proposed to open out these tappings, as a result, the electrodes were pocketed to some extent. The nickel alloy earthing electrodes were set at an angle through drillings through the outside of the boss thread where a spot of silver solder over the end ensured that they did not work loose. The plugs were packed with asbestos which proved satisfactory as a gas seal for the relatively short duration of the tests.

This arrangements gave good results except at full throttle, where pre-ignition was observed. This was probably due to the fact that the insulators could not be clamped down properly on the copper gasket at the base due
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to the interference of the end section of the insulator. This interference was allowed so that the size of the boss could be kept to a minimum.

A view of the head with all the plugs fitted is given in Plate III. Plate IV shows the underside of the head and the arrangement of the electrodes.

**Distributor.** This was required to distribute four high tension sparks per cylinder, the peak potential expected being of the order of 16,000 volts.

The rotor was fitted with four high tension ring tracks, leading to four separate distributing electrodes. These electrodes were placed two either side of a central bakelite spacing disc at an angular distance of 150°. The rotor was fully balanced.

The contact case consisted of two 1/4 in. bakelite discs clamped to house the rotor by suitable spacing studs. (see Figure 10). The sixteen distributing terminals engaged in tappings through the bakelite discs and thus could be adjusted for height to give a gap of approximately 1/32 in. above the terminals on the rotor. The whole case could be fitted and located on the standard distributor casing, the contact breaker being common to all the primary circuits of the coils in operation. Electrical leakage was cut down as much as possible by coating all
MULTIPLE SPARKING PLUG ASSEMBLY

FULL SIZE
PLATE III.

CYLINDER HEAD WITH SPARKING PLUGS IN POSITION
PLATE IV.

UNDERSIDE OF CYLINDER HEAD SHOWING THE
POSITION AND ARRANGEMENT OF THE ELECTRODES
exposed sections with electrical varnish.

The distributor was tested and found satisfactory at speeds up to 4,000 r.p.m. There was a tendency for the spark to arc when running on the outer ring track so that in most tests this was not used. Arcing sometimes occurred across the surface of the rotor, probably due to surface film under the sudden application of voltage. The rotor also had a definite capacitance and it was observed that arcing would sometimes take place to a ring track not in operation.
CROSS-SECTION OF SPECIAL DISTRIBUTOR.
FIGURE 11.

ROTOR ARM ASSEMBLY
CHAPTER IV

TECHNIQUE AND RESULTS OF TESTS WITH STANDARD ENGINE

The purpose of these tests was to determine the performance of the engine and to compare it with the makers rating. The opportunity was taken during the course of these experiments to check the accuracy of the equipment, modifications being made where required.

I. FULL THROTTLE TEST

This test was made to determine the output, fuel consumption, volumetric efficiency and optimum spark advance at full power.

**Technique.** The engine was run under light load for 10-15 minutes to reach steady conditions, during this period the air temperature and pressure were recorded, the manometer zero checked and the specific gravity of the fuel determined by means of a hydrometer.

The throttle was then opened wide. The speed was altered in steps of 250 r.p.m. by means of the brake, the engine being given at least two minutes to settle down between each change of brake load. The spark advance was set to give maximum power by slowly rotating the distributor housing by hand and noting the appearance of the flywheel which was illuminated by the stroboscope. The timing was first set approximately and the brake load altered
until fixed marks on the flywheel appeared stationary, the engine was then running at the speed required. (The Stoboscope being previously set at the required speed). The distributor was then rotated slowly, if the fixed marks rotated in the same direction as the flywheel the speed was increasing. The brake was given a further adjustment to make the marks appear stationary and the process repeated until further alteration of the ignition timing produced no further increase in speed. The distributor was then locked in this position and the spark advance noted from the indicator.

The volume of fuel was chosen so that the engine took approximately two minutes for its consumption, the time taken was recorded by a stopwatch previously checked against an electric clock.

The water temperature was held at 72°C ± 2°C. This was about the limit imposed by the cooling system. A short run was carried out on the engine to determine the effect of changes in water temperature, it was found that variations of this magnitude has negligible effect on engine performance.

The readings taken were brake load, manometer, time for fuel consumption and amount of fuel, speed, spark advance, inlet and outlet water temperature, oil temperature and manifold pressure.
Where possible, calculations were carried out during the run and the results plotted so that any obvious error could be detected and the reading checked.

Results. The maximum power output of the engine was 26.2 H.P. at 4,000 r.p.m.

The minimum specific fuel consumption was 0.651 lb./B.H.P.hr. and the maximum volumetric efficiency 72%. The complete engine performance is shown in Figure 12.

II. FUEL CONSUMPTION TESTS

These tests were carried out with an S.U. carburettor.

Technique. A fuel consumption "loop" was obtained by holding the speed and throttle opening constant and varying the mixture strength. The needle size in the carburettor fixed the range of mixture strength possible, the mixture strength being changed by tightening the jet adjusting nut. The technique employed is otherwise similar to that used in the previous test.

Results. A typical set of curves is shown in Figure 13, in this case, Brake Mean Effective Pressure (B.M.E.P.) and specific fuel consumption are plotted against air-fuel ratio. Plotting specific fuel
consumption against B.M.E.P. does not show up the characteristics as well, as the B.M.E.P. could only be determined to an accuracy of \( \pm 0.3 \text{ lb./sq.in.} \) and did not show the minimum portion of the curve (maximum economy) which is of the utmost importance.

Further test results and Figures are given in the Appendix.

The mixture loops obtained are typical, the fuel consumption increase with further richening of the mixture beyond the best power point. Weakening of the mixture beyond the best economy mixture causes a large decrease in B.M.E.P. and a rapid rise in specific fuel consumption. Misfiring and uneven running is also observed in this region.
FIGURE 12.
ENGINE PERFORMANCE STANDARD AT FULL THROTTLE

B.M.E.P.

B.H.P.

S.F.C.

ENGINE SPEED R.P.M.

BRAKE HORSEPOWER
EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

Standard Engine 1500 R.P.M.

AIR-FUEL RATIO

B.M.E.P. lb/sq.in.

Specific Fuel Consumption lb/B.H.P.hr

Misfiring in this region
The fact that a number of engine factors could not be controlled to a high degree of accuracy limited the scope of the experiments. The first series of tests were exploratory, of a comparative nature only.

I. TESTS WITH S.U. CARBURETTER

Technique. The primary circuits of three coils were wired in series, by means of a changeover switch the number of coils in operation could be selected and coupled with the correct supply voltage (6, 12, or 18v.) The current through the primary was therefore the same as with the standard ignition coil. The number of working coils could be altered rapidly while the engine was running without appreciably changing the engine conditions. The condenser used was the standard condenser fitted to the distributor casing of 0.22 mf. capacity.

Consecutive readings were taken with different numbers of plugs in operation so that the effect of changes in engine conditions could be minimised. The technique was otherwise identical with that used in the previous series of experiments.

The plug gaps were set a 0.020" for all tests.
Results. Results show that a considerable extension in the mixture range can be accomplished with two or three plugs in operation. Three plugs extend the mixture range slightly more than two, the increase being not as appreciable as the change from one plug to two.

It was found that with fuel consumption tests at constant throttle and constant speed, when B.M.E.P. and specific fuel consumption are plotted against air-fuel ratio, the curves of power output and fuel consumption are almost coincident with different numbers of plugs until the single plug "breaks away" at a critical mixture strength. (Results are shown in Figure 14)

At rich mixtures, it appears that the number of plugs has no effect on the power output.

The effect on ignition advance was slight, a difference in the required angle of advance being about $2^\circ$ less in the case of two plugs. This figure could only be approximate as it was found in practice that the indicator showed fluctuations in the advance, possibly due to slight changes in engine speed or irregularity of the cam profile. It was necessary, therefore, to take mean readings.

It was observed that with lean mixtures, there was a tendency for the engine to misfire with a subsequent fall in speed. This was far more noticeable at low throttle openings with a single plug in operation, and could only
FIGURE 14.

1500 R.P.M.
Manifold Pressure 15.7 in. Hg.

![Graph showing B.M.E.P. and Specific Fuel Consumption vs. Air-Fuel Ratio at 1500 R.P.M. and 15.7 in. Hg. manifold pressure.](image-url)
FIGURE 15.

FUEL CONSUMPTION LOOPS

2000 R.P.M.

SPECIFIC FUEL CONSUMPTION

lb./B.H.P. hr.

BRAKE MEAN EFFECTIVE PRESSURE

lb./sq.in.
be stopped by richening the mixture.

In operation, it was observed that the elbow connecting the carburettor to the intake manifold became coated with dew. It was apparent that the value of the 'hot spot' was lost and that the mixture was probably not well distributed to the individual cylinders. From tests with the standard engine, with rich mixtures, the performance of the engine was not appreciably affected by the fitting of this elbow (the difference of power output at full throttle being small) the mixture, however, being forced to make a right angle turn, vapourised at the bend instead of at the 'hot spot'.

II. TESTS WITH ZENITH CARBURETTOR

The S.U. carburettor having been replaced with the usual Zenith type, the only change being fitting of a variable main jet, a further series of tests were carried out with a somewhat wider range of mixture strengths.

Technique. To check the result that there was no increase in power with rich mixtures, a test was made at constant throttle opening and mixture strength and letting the speed vary. This was achieved with the assistance of an aircraft type fuel-air meter in the exhaust. The
mixture was adjusted to about 10% richer than that chemically correct and the reading on the fuel-air meter noted. (The fuel-air meter was found from preliminary tests to be hopelessly inaccurate and was therefore discarded for obtaining any quantitative results).

Readings were then taken of power output etc., using a similar technique as in previous experiments, and the speed was then altered in steps of 250 r.p.m. The mixture was then adjusted to give the same reading on the fuel-air meter as previously. The process repeated for the complete speed range. Check calculations were made during the run.

A further test was made to check the effect of ignition advance. In this case, the mixture strength, throttle opening and speed were fixed, the ignition timing being altered. The effect on performance with one and two plugs in operation was noted.

Results. Results show that the extension in available mixture range is similar to that obtained with the S.U. carburettor. An increase in speed showed some signs of increasing the mixture range with better fuel economy. (Figure 17)

No increase in power was noticed at a 10% rich mixture over a wide speed range, the maximum power
Figure 16.

1500 R.P.M.
Manifold Pressure 24.8 in Hg.

B.M.E.P. lb./sq.in.

SPECIFIC FUEL CONSUMPTION lb./B.H.P. hr.

AIR - FUEL RATIO
FIGURE 17.

2500 R.P.M.
Manifold Pressure 17.4 in. Hg.

![Graph showing the relationship between B.M.E.P. and specific fuel consumption across different air-fuel ratios at 2500 R.P.M. with a manifold pressure of 17.4 in. Hg.](attachment:image.png)
FIGURE 18

POWER OUTPUT AT CONSTANT AIR-FUEL RATIO

+ 2 Plugs
* 1 Plug  Mixture 10% Rich

Brake Horsepower

1000 1500 2000 2500

Speed R.P.M.
FIGURE 19

EFFECT OF IGNITION ADVANCE ON PERFORMANCE

1500 R.P.M.

IGNITION ADVANCE (Degrees B.T.D.C.)

B.M.E.P. in lb/sq. in.

SPECIFIC FUEL CONSUMPTION in lb./B.H.P. hr.
Output, which is produced by mixtures approximately 15% rich, is not changed with this type of ignition system. (Figure 18)

Figure 19 shows that the ignition setting is more critical with a single plug, especially with retarded spark. The power output and efficiency falls to roughly the same extent with overadvanced spark.

III. TESTS WITH PRIMARY IGNITION COIL CIRCUITS IN PARALLEL

In these experiments, the effect of changing the primary circuit was investigated, while keeping the natural frequency constant.

Technique. To keep the condenser voltage down to a reasonable level, a different condenser was fitted into the circuit when using two coils. (See Appendix for calculations). The number of coils in operation could be selected by means of a changeover switch. Fuel consumption tests (constant throttle - constant speed - variable mixture) were carried out with a similar technique as before.

Results. The results were rather surprising showing an increase in power of roughly 4 lb./sq.in. brake mean effective pressure with rich mixtures.
FIGURE 20.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

2000 R.P.M.
Manifold Pressure 13.9 in.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

2000 R.P.M.
Manifold Pressure 13.9 in.

B.M.E.P. (lb./sq. in.)

SPECIFIC FUEL CONSUMPTION (lb./B.H.P. hr.)

AIR-FUEL RATIO
FIGURE 21.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

1500 R.P.M.
Manifold Pressure 23.8 in.

AIR-FUEL RATIO

SPECIFIC FUEL CONSUMPTION
lb./B.H.P. hr.

B.M.E.P. 16/sq. in.
With leaning off of the mixture, two plugs sometimes gave poorer results than that with the single plug and the characteristic "fish-hook" is delayed until a much leaner mixture is reached. (Figure 20) At larger throttle openings the power output and fuel consumption curves are almost coincident. (Figure 21) At small throttle openings, however, the results do not appear to show any consistency. Speed and throttle opening both appear to affect the power output, generally the specific fuel consumption deteriorates with two plugs and lean mixtures more than with the single plug.

A possible conclusion is that the single plug is better able to burn lean mixtures, while two plugs, with this circuit configuration, is better able to burn rich mixtures.

**IV. EXPERIMENTS WITH THE PRIMARY COILS IN SERIES**

From the results obtained with two coils in parallel it was necessary to re-examine the early results obtained with the coils in series and to investigate the effects of changes in circuit parameters, if any.

**Technique.** In these series of tests, two coils were connected in series as before, only in this case the standard condenser was removed and a smaller condenser
was fitted to give the same natural circuit frequency as the orthodox single coil. The single coil had its own separate condenser and the two circuits could be selected by means of the changeover switch.

Fuel consumption tests were carried out as previously described.

Results. Results show that generally there is a slight increase in power with rich mixtures, the power curves becoming very close together as the mixture is made leaner. (Figure 22) The improved efficiency with rich mixtures is over a much wider range than with two coils wired in parallel and the range of mixtures that could be burnt without misfiring is extended to some extent.

V. EXPERIMENTS TO DETERMINE THE EFFECTS OF CHANGES IN COIL CIRCUIT PARAMETERS

From the results obtained in the previous experiments, with differences in performance with different primary coil circuit arrangements, it was possible to come to the conclusion that the parameters of the primary circuit (frequency, etc.), were a controlling factor. In order to test this conclusion a further series of tests were made as described below.
EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

1500 R.P.M.
Manifold Pressure 16.2 in.
Technique. In the first test, a single coil was used and a choice of two different sizes of condenser (0.1 and 0.25 mf.) was available, the selection being made by means of a changeover switch. A fuel consumption test was made at constant throttle - constant speed and the mixture varied and the results plotted on Figure 23.

In the second test, two coils were connected with their primaries in parallel and a choice of three different capacitances was available. A similar test was made to that given above, at the same throttle opening and speed.

Results. Results show that in the case of the single coil, there was no difference in performance.

In the case of the coils in parallel, the change in performance was very small, with the 0.05 mf. capacitance, a slight decrease in power was observed, but this could well be within the experimental error of the experiment. (Figure 24)

General observations. With suitable adjustment, it was possible to get the engine to run at very low idling speeds, very much lower than that being obtained with a single plug. Idling speeds of approximately 60 r.p.m. were possible with two plugs.
In one of the early tests with two coils in series, the mixture strength was leaned off as far as the limits of the carburettor would allow. It was found that the engine would still run, although very unevenly, at an air-fuel ratio of 19.2/1 with two plugs in operation.
FIGURE 23.
EFFECT OF CIRCUIT PARAMETERS ON PERFORMANCE

Single Coil

- 2000 R.P.M.

- With 0.25 μF condenser.
- With 0.1 μF condenser.

B.M.E.P. (lb/sq.in.):

Specific Fuel Consumption (lb/B.H.P. hr.):

Air-Fuel Ratio

12 13 14 15 16 17 18
FIGURE 24.

EFFECT OF CIRCUIT PARAMETERS ON PERFORMANCE

Two Coils, primaries in Parallel.
- O With 0.25 μF Condenser
- ▼ " 0.1 μF. "
- + " 0.05 μF "

2000 R.P.M.
CHAPTER VI.

SUMMARY OF RESULTS, RECOMMENDATIONS AND CONCLUSIONS

It must be borne in mind that the results are usually of a comparative nature as the apparatus did not permit a high degree of accuracy. This could be due to four main causes, (a) variations in the water temperature, (b) variations in the atmospheric conditions, (c) variations in the oil temperature, and (d) variations in the fuel.

Any quantitative results should therefore be treated with caution.

I. SUMMARY OF RESULTS

Power output. In some cases an increase in power was noted at rich mixtures with two sparking points. This inferred an increase in power at full throttle, and was estimated at approximately 4%. In other cases, the power output was unchanged dependent on the characteristics of the coil circuit and will be considered later.

Fuel consumption. There was no consistency in the values obtained for specific fuel consumption at different throttle openings probably due to the causes
given above.

A decrease in specific fuel consumption of 12% was observed in one particular case. Usually the improvement was due to the increase in the mixture range which could be burnt without misfiring, and this in turn varied with the coil circuit. The improvement in efficiency was generally less than the value given and in some cases there was no improvement. In the particular case of two coils in parallel, the thermal efficiency was sometimes lower than that with a single plug at weak mixtures. At small throttle openings, two sparking points showed up to better advantage.

**Mixture range.** The extension of the mixture range was apparently dependent on a number of factors, including the coil circuit, the throttle opening and the speed. In order to get some correlation in the results the conditions of the charge at the point of passing the spark was considered.

Knowing the pressure and temperature of the charge in the intake manifold, the pressure at the end of the compression when the spark passes can be calculated knowing the ignition advance and assuming the compression is isentropic. (See Appendix)

The leanest practicable mixture, that is the mixture giving best economy without misfiring occurring,
was plotted against the pressure at the point of ignition. (Figure ). The ideal specific fuel consumption curves for the single plug and two plugs (Primaries in series, 0.1 mf. condenser) are plotted as well.

The results show that different coil circuits give slightly different results and show that leaner mixtures can be used with two sparking points.

In the case of the single plug in the standard engine, the results are almost coincident with those obtained in the multiple sparking plug boss using a single sparking point. It was thought that in this case, the points would be poorly scavenged, housing residual exhaust gas which would affect the performance. It appears however, that this factor is not as important as originally supposed.

**Effect of coil circuit parameters.** For a given sparking gap, changes in the natural frequency of the primary circuit had very little effect within the limits tested. It was obvious, however, that there was some characteristic of the circuit that exerted some influence on the subsequent performance of the engine.

It was not within the scope of the investigation to determine the performance of current and voltage on the discharge, the discussion must be based on the
Variation of Leanest Practicable Mixture with Pressure at the Point of Ignition 1500 R.P.M.

Two Coils in Series: 22 μF Cond.

Two Coils in Parallel: 5 μF Cond.

Single Coil: Standard Engine

Ideal S.F.C. lb./h.p.hr.

Approximate Pressure at the Point of Ignition lb./sq.in.
present theory of the coil discharge. (For a mathematical
treatment of the coil discharge, see Appendix)

As the gap width was kept constant, the breakdown
voltage of the gap should stay constant, and therefore
the energy in the capacity component would also stay
constant, for a given condition of the engine. If
then, the capacity component was the normal form of the
spark discharge as postulated by Paterson and Campbell\(^1\),
there should be no difference in the performance of the
ignition spark. This, however, is inconsistent with
the experimental results.

The maximum voltage attained is also dependent on
the natural frequency of the primary circuit. (see Appendix)
In most of the experiments, the primary frequency was
held constant, so that the impulse ratio was kept at the
same value for a given condition of the engine. The
impulse ratio could not be used, therefore to explain
the variations in performance.

The distributor could have possibly been the
cause of some of the variations noted. In operation,
however, comparable results were obtained to those with
the standard ignition coil and distributor. (Figure\(^\_\)\)

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\(^1\) C.C. Paterson and N. Campbell, "Some
Characteristics of the Spark Discharge and its Effect on
Part 4, p. 168 (June, 1919)
Any attempts to obtain a relationship between the parameters of the coil circuit and the performance of the engine failed, the relationship is probably complex and would require further experimental work with special apparatus.

Combustion time. The only method available of obtaining some indication of the combustion time was by recording the amount of ignition advance required. As it was shown by a special experiment, the ignition timing is not as critical with two plugs as with one. Reading for ignition advance could be further in error on this account.

Differences of only a few degrees were noted, typical results being shown in Table 2. It is possible that this slight difference may be significant as the spark has very little effect once the explosion has been initiated.\(^2\) As the combustion chamber shape was the same in each case, any reduction in the total combustion time would probably be in the initial delay period. To test this conclusion would require accurate pressure-time diagrams.

### TABLE 2.

**REQUIRED IGNITION ADVANCE FOR MAXIMUM POWER WITH DIFFERENT NUMBERS OF SPARKING POINTS IN OPERATION**

**Conditions of the test:**

- **Engine speed:** 2,000 r.p.m.
- **Manifold pressure:** 21.9 in.Hg.
- **Outlet water temperature:** 70°C.
- **Fuel:** "Europa" first grade.

<table>
<thead>
<tr>
<th>Air-Fuel Ratio</th>
<th>Ignition Timing giving maximum torque</th>
<th>One plug</th>
<th>Two plugs</th>
<th>Three plugs</th>
</tr>
</thead>
<tbody>
<tr>
<td>11.9</td>
<td>10° B.T.D.C.</td>
<td>10</td>
<td>10</td>
<td>9</td>
</tr>
<tr>
<td>12.5</td>
<td>16</td>
<td>12</td>
<td>13</td>
<td>13</td>
</tr>
<tr>
<td>13.0</td>
<td>15</td>
<td>13</td>
<td>13</td>
<td>13</td>
</tr>
<tr>
<td>14.0</td>
<td>17</td>
<td>15</td>
<td>15</td>
<td>13</td>
</tr>
<tr>
<td>14.6</td>
<td>17</td>
<td>17</td>
<td>17</td>
<td>17</td>
</tr>
<tr>
<td>15.5</td>
<td>21</td>
<td>22</td>
<td>22</td>
<td>23</td>
</tr>
<tr>
<td>16.6</td>
<td>24</td>
<td>25</td>
<td>25</td>
<td>23</td>
</tr>
<tr>
<td>17.7</td>
<td>27</td>
<td>36</td>
<td>36</td>
<td>33</td>
</tr>
<tr>
<td>19.2</td>
<td>Would not run</td>
<td>36</td>
<td>36</td>
<td>33</td>
</tr>
</tbody>
</table>
Discussion of the results with reference to the theories of ignition. In a study of this nature, it is extremely difficult to relate the results to the theories of ignition. Only the final effects are shown in the power output and fuel consumption figures, little indication is given of the exact ignition process. It is the period immediately following "break" that is of vital importance. The whole process takes place in such a short period of time that it is very difficult to record experimentally exactly what happens. The problem is further complicated by the complex nature of the coil discharge.

The results tend to conflict with the thermal theory which indicates that a number of simultaneous sparks close together would be more effective than a single spark of the same heat content. If this was the case, provided the breakdown voltage of the gap and the frequency of the primary circuit were held constant, the performance figures would not be altered by changing the coil circuit from a series to a parallel arrangement. This, however, is not borne out by experiment.

Consideration of the results in the light of the ionization or chain reaction theories is limited by the fact that the performance of current and voltage in the period of say, on millionth of a second after "break"
is unknown. Cipriani and Middleton state:

Some very high frequency secondary oscillations can be obtained in the portion of the spark discharge immediately following ignition. These oscillations, well up in the megacycle range, can and do, under part throttle conditions, contribute to the flame propagation process.\footnote{3}

Unfortunately, the theory of the discharge of an ignition coil under working conditions gives no indication of any high frequency oscillations in the period following break, the main consideration is the discharge of the magnetic component. In a simple mathematical treatment for the changes in the coil circuit as were made in the experiments, any effect on these oscillations, if they were present, would not be shown.

The results obtained with change of circuit parameter show that quite large changes in primary frequency have little, if any, effect on performance. If ionization were a controlling factor, as was suggested in the combustion of carbonic oxide-air mixtures\footnote{4}, it appears likely that it would show itself in an experiment of this nature.


II. RECOMMENDATIONS

From the experience gained from this series of experiments it is possible to come to some conclusions as a guide to further work along these lines.

Test apparatus. A single cylinder engine would appear to be preferable for obtaining accurate results. The variables on a multi-cylinder engine are hard to control and an accurate picture of the combustion process difficult to obtain. The uncertainty of mixture distribution to the individual cylinders is only one factor. For demonstration purposes, admittedly, an engine of the type used in these experiments is perfectly satisfactory, but for a research project, something better is required.

Ricardo, after years of research experience, states:

In the author's (Ricardo) experience it never pays to employ a commercial or proprietary engine for research. The temptation to do so is, of course, enormous, both on the score of time and cost, but almost invariably it will be found that the engine is unsuitable; either it is not sufficiently robust to withstand the high pressures which the research will involve, or it is not versatile enough to permit of the changes which the research demands, or again, though reliable when used for its intended purpose, and at its designed output, it may prove very unreliable and inconsistent under the conditions imposed by research. 5

This comment appears to be well justified when one considers the difficulties experienced in operation and installation of the engine.

More specifically, it is suggested that the possibilities of getting two sparking points close together in the combustion chamber of the Ricardo E6/3 engine, at present installed in the laboratory, be investigated. In this particular type of engine, the sparking plug tapping is not very accessible and a special plug would probably have to be manufactured. However, this is not beyond the realms of possibility, even with New Zealand manufacturers, and could have a fair chance of success.

Although the experiments did show that a distributor supplying four sparks per cylinder could be successfully constructed, it would be desirable to remove any possibilities of error arising out of the operation of this unit. This is another factor favouring the use of a single cylinder engine.

It is considered that greater use could be made of the type of air meter used in these experiments, that is, fitting a suitable muffler to the intake manifold instead of the more usual pulsation tank. The muffler can be of simple construction, is easy to install, and requires very much less space than the pulsation tank.
It also has the additional advantage over other types of air meter, that when an orifice gauge is used, calibration is not required.

An oil heater is thought necessary as even with a small engine, it takes some time for the oil to reach equilibrium temperature. Small variations in the oil temperature are not troublesome at large throttle openings, but when tests are made at part throttle, as was the case in many experiments, control of this factor becomes more important.

The inlet and outlet water temperatures require accurate control and the system used was not particularly successful. The only feasible solution appears to be to have a heat exchanger in the cooling water circuit. This has the advantage that the cooling water is not being continually changed and therefore does not deposit undesirable salts in the cooling passages.

Scope of further investigation. The first investigation should be a confirmation of the results obtained above but with more rigourously controlled engine conditions.

The effect on combustion time could be determined, especially in the delay period, where the spark characteristics are likely to have their greatest effect.
For this purpose, pressure-time diagrams of the compression and power stroke would be necessary.

There is some evidence to show that a reduction in the required octane rating is possible with a reduced combustion time. This aspect could be studied relatively easily in the E6/S variable compression engine.

A study of the effect of the circuit parameters is certainly indicated. An oscillograph analysis of the discharge may throw some light on the problem, in particular, the value of peak voltage and the time of duration of the discharge. It would also be interesting to study the condition of the charge at the point of passing of the spark. In the Ricardo E6/S engine, the heat supplied to the charge can be determined and from a pressure-time diagram the pressure and temperature can be deduced. As the degree of turbulence has only small influence on the initiation of the explosion, the results obtained with stagnant mixtures may be able to be applied.

III. CONCLUSIONS

From the experimental results it is possible to

come to the following conclusions:

(1) The number of sparking points in operation is not the only controlling factor in determining the performance of the ignition system. The coil circuit configuration has considerable effect, the exact characteristics of which were not revealed by this series of experiments. Results, however, infer that a consideration of the ignition process in the engine on a purely thermal basis is not justified.

(2) With two sparking points close together
   (a) The required ignition advance is reduced slightly.
   (b) The ignition timing is less critical than that with a single sparking gap.
   (c) There is a significant increase in the range of mixtures that can be burnt without misfiring.
   (d) The addition of a third spark gap has a relatively small effect.

(3) The range of mixtures that can be burnt is dependent on the pressure and possibly the temperature at the point of ignition.

Application of the theories of ignition is almost impossible as it is seen that the ignition and combustion process in an internal combustion engine is complicated
by the fact that the initiation of the explosion
and the subsequent propagation of the flame are not
easily separated as distinct processes. If the
process of ignition cannot be considered entirely on a thermal
basis, the question then arises, which property of the
spark discharge determines ignition? This problem requires
further investigation in the light of some of the more
modern theories before an improvement in present ignition
systems can be expected. It is hoped that this investigation
provides some evidence on one of the least understood
factors of engine design.
APPENDIX A.

MATHEMATICAL DEVELOPMENT OF THE CHAIN REACTION THEORY. 1

When a mass element passes through a combustion wave, it gains heat first by conduction from preceding hotter elements and looses heat to succeeding cooler elements.

Correspondingly, the source $h$ of the thermal and chemical energy per unit mass at first increases above the level $h_0$ for the unburned or adiabatically burned gas, then later decreases to the same level. It follows that in a unit area segment of a plane combustion wave the excess energy $\int_{x_u}^{x_b} (h - h_0) dx$ is stored. Where $x_u$ and $x_b$ denote reference points before and behind the wave.

This energy is acquired at the expense of the energy content of the burned gas as the flame grows from a small sphere to its final size.

The energy that is thus lost by the burned gas is insignificant except at the flame origin. Here the excess energy must be furnished by the ignition source.

Consider first a plane combustion wave (Figure). At any point $x$, the sum of the rate of heat gain or loss in a unit volume of a layer $dx$ due to thermal conduction, mass flow and chemical reaction is zero.

\[ \text{i.e.} \quad k \frac{d^2 T}{dx^2} - \rho S C_p \frac{dT}{dx} + q(x) = 0 \]

where $T = \text{temperature}$, $k = \text{heat conductivity}$, $\rho = \text{density}$, $C_p = \text{specific heat}$, $S = \text{velocity of gas flow}$, and $q(x) = \text{rate of heat release per unit volume}$.

Relationship of Temperature and Energy through a Cross-Section of a Combustion Wave [Lewis & von Elbe]

A. Plane Wave

B. Spherical Wave
Now $q(x)$ is initially very small and therefore $T$ increases exponentially until $q(x)$ is appreciable. $\frac{d^2 T}{dx^2}$ then becomes negative as shown in the dotted part of the curve.

A mass element entering the combustion wave first acquires some excess energy by heat conduction from preceding elements and then returns the 'borrowed' energy to succeeding elements. Therefore,

Thermal energy after passage through the wave = Thermal energy + Chemical energy before entering the wave.

Therefore, inside a plane wave segment of unit area carries an excess energy = $\int_0^\infty q(h - h_0) \, dx$

A small initial flame which is formed around a spark derives its energy from the spark itself.

It is the function of the spark to initiate the reaction by producing a high local concentration of heat and chain carriers and to furnish at least as much energy as is necessary to satisfy the excess energy requirement of the smallest flame sphere that is capable of self propagation.

In the calculation $q(x)$ is taken as zero up to $T_1$

Between $T_1$ and $T_b$, $q(x)$ is taken as constant and

$\frac{d^2 T}{dx^2} = 0$

These conditions closely follow the actual conditions.

It should be noted at this stage that no issue arises in a calculation of this kind concerning a 'thermal' or 'kinetic' theory of flame propagation. This issue only arises when an attempt is made to calculate $S_0$ and $(x_3 - x_1)$ from the values of $T_1$ and the rate of chemical reaction.

For a spherical wave $r = r_1$ and $q(x) = 0$
Hence an increase in heat in a unit volume of any shell $4\pi r^2 \, dr$ during time $dt$ is given by

$$k \left[ \frac{\partial^2 T}{\partial r^2} + \frac{2}{r} \frac{\partial T}{\partial r} \right] = C_P \frac{\partial T}{\partial t}$$

Velocity of the combustion wave relative to the gas at temperature $T_b$

$$= S_b = \frac{\partial r}{\partial t}$$

It appears reasonable to assume that the shells expand at such a rate that the mass crossing until area is the same for all shells

i.e. $\rho S = \rho_b S_b = \rho_u S_u = \text{constant}$

thus, during unit time, the mass $\rho_u S_u$ passes through the volume $4\pi r^2$ times unit area of a shell of total volume $4\pi r \, dr$ and receives heat $C_P \rho_u S_u \frac{\partial T}{\partial t}$

$\therefore$ rate of increase per unit volume $= C_P \rho_u S_u \frac{\partial T}{\partial t}$

$$\therefore k \left[ \frac{\partial^2 T}{\partial r^2} + \frac{2}{r} \frac{\partial T}{\partial r} \right] = -C_P \rho_u S_u \frac{\partial T}{\partial r}$$

Which can be integrated to

$$u = u_1 \left( \frac{r}{r_1} \right)^2 e^{-a(r - r_1)}$$

Where $u = \frac{\partial T}{\partial r}$ and $a = \frac{C_P \rho_u S_u}{k}$

and $T - T_0 = \frac{r_1}{u_1} \int \frac{\rho_u S_u}{r} e^{-a(r - r_1)} dr$

$$= \frac{r_1}{u_1} \int e^{-a(r - r_1)} \, dr_1 e^{a r_1} \left[ -E_1(-ar) \right]$$

Considering the plane wave as a special case of the spherical wave when the radius becomes very large

For a plane wave at $x_1$

Heat Balance $= k u_1 = C_P \rho_u S u \int_{x_1}^\infty (T_1 - T_0)$

$$\therefore T_1 - T_0 / u_1 = \frac{k}{C_P \rho_u S_u} = \frac{1}{a_\infty}$$

$a_\infty$ is the value for a very large flame radius.
Now the excess energy \(H\) in the exponential portion of the temperature gradient

\[
[H] = \int_{r_1}^{\infty} 4 \pi r^2 \rho \, (h - h_0) \, dr
\]

and substituting for \((h - h_0)\) we have

\[
[H] = \int_{r_1}^{\infty} 4 \pi r^2 \rho C_p (T - T_u) \, dr
\]

Now substituting the value obtained for \((T - T_u)\) and replacing \(\rho\) and \(C_p\) by mean values

\[
[H]_{r_1} = 4 \pi \bar{\rho} \bar{C}_p |u_1| (r_1/a)^2 \left\{ 1 + a r_1 - a^3 r^2 \int \left[ E_1(-a r) \right] \, dr \right\}
\]

The value of the integral has been calculated

when \(a r_1 = 1\), \(2\), \(3\), \(4\), \(5\)

Integral = 0.514, 0.296, 0.142, 0.058, 0.018

and \(a r_1\) if found by experiment to be generally greater than 2, therefore the integral may be safely neglected

\[
[H]_{r_1} = 4 \pi \bar{\rho} \bar{C}_p |u_1| \left( \frac{r_1}{a} \right)^2 (1 + a r_1) \quad \ldots \ldots \quad (2)
\]

for the second stage \(q(x) = \) constant and \(\frac{\partial^2 T}{\partial t^2} = 0\)

Then

\[
[H]_{r_2} = \int_{r_2}^{r_1} 4 \pi r^2 \bar{\rho} \bar{C}_p (T_1 - T_u) \, dr
\]

\[
= 4/3 \pi r^3 \bar{\rho} \bar{C}_p (T_1 - T_u) \left( r_1^2 - r_2^2 \right) \quad \ldots \ldots \quad (3)
\]

The flame can propagate if \(|u_1|\) is stable, i.e. the rate of heat production in the volume \(4\pi r_2^3\)

\[
q = 4/3 \pi r_2^3 \bar{\rho} \bar{C}_p \left( r_1^2 - r_2^2 \right) |u_1|
\]

and \(q = 0\) from \(r = 0\) to \(r = r_1\)

Thus the flame can propagate when

\[
4/3 \pi r_2^3 q = 4\pi r^2 \bar{\rho} |u_1| \text{ between } r_1 \text{ and } r_2
\]

and

\[
q = \bar{\rho} \bar{C}_p u^8 |u_1|
\]

By combining equations (2) and (3) and making the approximation in writing \((1 + a r_1)\) as \((1 + 1.3 a r_1)\)
\[
H = 4\pi r_1^2 \frac{\rho C_p}{s_\infty} (T_b - T_u) \frac{a^2}{s_\infty} \frac{(1 + 1.3 a r_1)}{1 + a r_1 [1 - (3/4 a r_1)]^{1/2}}
\]

The value of \( r_1 \) is given by the measurable flame quenching distance \( d \), between plane parallel plates.

A factor of \( d/2r_1 \geq 2 \) appears reasonable.

The treatment is only applicable when

1. The dimensions of the flame over which the spark energy is distributed during the time of discharge are smaller than the flame radius \( r_1 \)

2. The time of discharge is small compared with the time required for a substantial growth of the flame radius \( r_1 \).

APPENDIX B.

THEORY OF THE IGNITION COIL

Build up of primary current. Up to the period at "break", since no current flows in the secondary coil, the circuit can be considered simply as a resistance $R$ and an inductance $L_1$ with a condenser $C_1$ in parallel with the inductance. The purpose of the condenser is to limit the voltage set up across the contacts and prevent arcing and subsequent erosion of the contact points. The condenser is mounted as near as possible to the contact points as it is found that this gives best results, the leads to the condenser having a very low resistance or capacitance.

The performance of the circuit is required under transient current conditions. If $v = \text{battery voltage}$ by Kirchoff's Laws, the sum of the transient free voltages and the sum of the steady voltages are both zero.

If $i'$ denotes the current under steady conditions

and $i''$ denotes the current under steady conditions

then current at any time $t$ is $i = i' + i''$ . . . (1)

after the contacts have been closed

$$L_1 \frac{di'}{dt} + Ri' = v \quad \ldots \quad (2)$$

and

$$L_1 \frac{di''}{dt} + Ri'' = 0 \quad \ldots \quad (3)$$
Electrical Schematic of an Induction Coil Ignition System

Circuit Diagram for Two Coils in Parallel
Since in the steady state \( \frac{di}{dt} = 0 \), therefore \( i' = \frac{v}{R} \)

Equation (3) is readily solved by making the substitution \( i'' = I e^{at} \)

Solving (3) and combining with (1) gives the equation for current in the primary 

\[
i = \frac{v}{R} \left( 1 - e^{-\frac{Rt}{L}} \right) \ldots \tag{5}
\]

Voltage across condenser after sudden interruption.

At the moment of "break" the current in the primary is given by (5). The voltage across the condenser can be calculated by energy considerations.

If the resistance in the circuit is small, after instantaneous interruption by the switch, the current flows through the condenser, charging the capacitance to a voltage \( V \).

Stored electrostatic energy \( = \frac{1}{2} CV^2 \)

and this must equal the magnetic energy in the inductance carrying a current \( I \)

i.e. \[ \frac{1}{2} LI^2 = \frac{1}{2} CV^2 \]

\[ V = I (L/C)^{\frac{1}{2}} \ldots \tag{6} \]

Voltage induced in secondary after sudden interruption of the primary current. Up to the moment of breakdown of the spark gap, the primary circuit can be considered quite distinct from the secondary circuit as there is no current flowing in the secondary.
If \( M \) = mutual inductance between the secondary and the primary, then the voltage induced in the secondary before gap breaks down is
\[
E = M \frac{dI}{dt}
\]

On interruption the transient current will flow through the circuit composed of the inductance, resistance and capacitance. Denoting the potential \( e \) across these components in order with \( L, R \), and \( C \),
\[
e''_L + e''_R + e''_C = 0 \quad \ldots \quad (7)
\]
corresponding to
\[
L \frac{dI''}{dt} + R I'' + \frac{1}{C} \int I \, dt = 0 \quad (8)
\]
Differentiating and dividing by \( L \)
\[
\frac{d^2 I''}{dt^2} + \frac{R}{L} \frac{dI''}{dt} + \frac{I''}{LC} = 0 \quad (7A)
\]
Taking as a tentative solution
\[
I'' = I e^{at}
\]
and substituting the derivatives in \((7A)\)
\[
I e^{at} \left[ a^2 + \frac{Ra}{L} + \frac{1}{LC} \right] = 0
\]
Solving the quadratic gives
\[
a = \frac{-R}{2L} \pm \sqrt{\left( \frac{1}{LC} \right) - \frac{R^2}{4L^2}} \quad (8)
\]
The positive sign is taken as it is proponentant in most cases. Writing the square root as \( f\), and introducing the electromagnetic time constant \( T = \frac{L}{R} \)
\[
\text{Since} \quad e^{jf} = \cos ft + j \sin ft
\]
the complete transient is given by
\[
i''_I = e^{-t/2T} (I_1 \cos ft + I_2 \sin ft) = e^{-t/2T} I \cos (ft + b) \quad (8)
\]
Where \( b \) = phase angle, \( I \) is fixed by the initial conditions.

For small ohmic resistance, frequency \( f = \frac{1}{(LC)^{3/2}} \)

Now if the condenser voltage is originally small, the phase \( b \) may be omitted. i.e. \( b = 0 \)

\[ V_{\text{induced}} = M \frac{dI}{dt} = E \]

\[ = MI e^{-t/2T} (-f \sin ft - 1/2T \cos ft) \]

\[ = -MI e^{-t/2T} f^2 + (1/2T)^2 \sin (ft + d) \ldots (9) \]

where \( d \) is given by \( \tan d = \frac{R}{2fL} \)

Now for small ohmic resistance, a similar approximation can be made as in (8), equation (9) reduces to

\[ E = -MI f e^{-t/2T} \sin (ft + d) \ldots \ldots \ldots \ldots (10) \]

Now \( d \) is small, \( E \) will be a maximum when \( ft = \frac{1}{f} \)

\[ E_{\text{max}} = -MI f = \frac{MI}{(L/C)^{3/2}} \ldots \ldots \ldots \ldots (11) \]

Circuit equations for ignition coil under sparking conditions. The previous treatment only applies when there is no current flowing in the secondary coil, or in the period up to "break". Under normal circumstances, when the coil is required to furnish a spark across a short fixed gap, the breakdown potential is smaller than the maximum voltage the coil is capable of delivering. It is the period after "break" that is of interest with the subsequent performance of secondary current and voltage.
A slightly different notation is used as both the primary and the secondary circuits have to be considered. The suffixes 1 and 2 denote the primary and secondary coils respectively.

Neglecting the secondary resistance, the general coil circuit equations may be written:

\[
\begin{align*}
L_1 \frac{di_1}{dt} + L \frac{di_2}{dt} + R_1 i_1 + e_1 &= 0 \quad \ldots \quad (12) \\
L_2 \frac{di_2}{dt} + L \frac{di_1}{dt} + e_2 &= 0 \quad \ldots \quad (13)
\end{align*}
\]

Where \( L_{12} \) and \( L_{21} \) are the coefficients of mutual of the secondary on the primary and the primary on the secondary respectively, \( t \) is the time after "break", \( e_1 \) and \( e_2 \) are the potential differences across the condensers.

From cathode ray oscillograph analysis\(^1\), of typical ignition coils, it was found that on breaking the primary circuit rises in a manner determined by the coil constants (see above) to the breakdown potential of the gap, the voltage then falls rapidly to a value at which it remains sensibly constant during the remainder of the discharge. The first part of the trace on the oscillograph record is the discharge of the capacity component, the duration of this component that is, the time for the spark potential to fall from the breakdown voltage to a constant value, is

\(^1\) Finch and Mole op. cit. p. 72
of the order of one microsecond. The normal duration of the discharge is of the order of several milliseconds, it appears reasonable to assume, therefore, that for all practical purposes, the sparking potential is constant and independent of the current throughout the life of the discharge.

Under sparking conditions, the current distribution in secondary will then be uniform, and the primary-secondary and secondary-primary mutual inductances are equal, that is,

\[ L_{12} = L_{21} = M \text{ and } e_2 \text{ = constant.} \]

Considering the natural frequency of the transient current \( i_1' \) in the primary

\[ L_1 \frac{di_1'}{dt} + M \frac{di_2}{dt} + R_1 i_1' + \frac{1}{C} \int i \, dt = 0 \quad (14) \]

substituting the value of \( \frac{di_2}{dt} \) from (13) and writing \( n = \frac{M}{L_2} \)

\[ (L_1 - Mn) \frac{di_1'}{dt} - \frac{e_2}{L_2} + R_1 i_1' + \frac{1}{C} \int i \, dt = 0 \quad (14A) \]

Differentiating and dividing by \( (L_1 - Mn) \), since \( e_2 \) is constant

\[ \frac{d^2 i_1'}{dt^2} + \frac{R}{L_1 - Mn} \frac{di_1'}{dt} + \frac{1}{N[(L_1 - Mn)C]} \frac{i_1'}{t} = 0 \quad (15) \]

This equation is of exactly the same form as (7A) and has a similar method of solution

\[ i_1 = i_0 e^{-at} \cos ft \quad . . . . . . \quad (16) \]

Where \( f = \text{natural frequency} = \frac{1}{\sqrt{(L_1 - Mn)C}} \)

and \( a = \text{electromagnetic time constant} = \frac{R}{(L_1 - Mn)} \)

Now from (13) \( \frac{di_2}{dt} = -\frac{e_2}{L_2} - n \frac{di_1}{dt} \)
Integrating \( i_2 = -e^{2t/L_2} - ni_1 + \text{a constant} \)

This can be written in the form

\[
i_2 = n (k - e^{-at} \cos \omega t) - e^{2t/L_2}
\]

When \( t = 0, \ i_2 = 0, \) therefore \( k = 1 \)

and \( i_2 = n (1 - e^{-at} \cos \omega t) - e^{2t/L_2} \ldots . \ldots \) (17)

From (12), substituting the values of \( i_1 \) and \( i_2 \) and their derivatives gives:

\[
- a_{10} e^{-at}(L_1 - Mn) \cos \omega t - f_{10} e^{-at}(L_1 - Mn) \sin \omega t
\]

\[
R_{10} e^{-at} \cos \omega t + e_1 - ne_2 = 0
\]

On inspection, it is seen that the sum of the coefficients of \( \cos \omega t = 0 \)

and \( e_1 = ne_2 + \frac{10}{f C} e^{-at} \sin \omega t \ldots . \ldots \) (18)

Thus, when an ignition coil is furnishing a spark under the normal conditions of practice, the secondary current rises to a maximum value of \( 2i_{10} e^{-a\pi/2f} \) and executes damped oscillations of initial amplitude \( ni_0 \) about a linear axis, the slope of which is given by \( -e^{2t/L_2} \)
Current and Voltage Relationships in an Ignition Coil

Primary Current

Gap Voltage

Breakdown Voltage of Gap

Gap Current

Current rises to a very high value in this interval

Energy is insufficient to support discharge & sparking ceases
APPENDIX B.

CALCULATIONS FOR COIL CIRCUIT CHARACTERISTICS

The inductance, resistance and the capacitance of the coils were determined.

Results. The inductance of the four coils used in the tests at 50 cycles/sec. (corresponding to an engine speed of 1500 r.p.m.) was

<table>
<thead>
<tr>
<th>No.</th>
<th>Coil</th>
<th>Inductance (h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>No.1</td>
<td>0.00464</td>
</tr>
<tr>
<td>2</td>
<td>No.2</td>
<td>0.00491</td>
</tr>
<tr>
<td>3</td>
<td>No.3</td>
<td>0.00465</td>
</tr>
<tr>
<td>4</td>
<td>No.4</td>
<td>0.00450</td>
</tr>
</tbody>
</table>

This only corresponds to one engine speed, the only other frequency available for an accurate determination was 1000 c/s., at this frequency

<table>
<thead>
<tr>
<th>No.</th>
<th>Coil</th>
<th>Inductance (h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>No.1</td>
<td>0.00582</td>
</tr>
<tr>
<td>2</td>
<td>No.2</td>
<td>0.00604</td>
</tr>
<tr>
<td>3</td>
<td>No.3</td>
<td>0.00599</td>
</tr>
<tr>
<td>4</td>
<td>No.4</td>
<td>0.00574</td>
</tr>
</tbody>
</table>

For calculations, the inductance was taken as 0.005 h.

Capacity of condenser = 0.22 mf.
Resistance of primary = 1.3 ohm
Ballast resistance = 0.25 ohm (cold)
                     = 0.6 ohm (hot)
Natural frequency of primary = \((LC)^{-\frac{1}{2}}\)
                           = \((5 \times 10^{-3} \times 0.22 \times 10^{-6})^{-\frac{1}{2}}\)
Maximum current when the ballast resistance is cold
\[ I = \frac{V}{R} = \frac{6}{1.55} = 3.87 \text{ amp.} \]

Maximum current when the ballast resistance is hot
\[ I = \frac{6}{1.9} = 3.17 \text{ amp.} \]

Maximum voltage across the condenser
\[ V = I \sqrt{\frac{L}{C}} = 3.87 \sqrt{\frac{0.005}{0.22 \times 10^{-6}}} = 585 \text{ volts.} \]

The current will rise in the primary circuit after the contacts are closed according to the law
\[ I = \frac{6}{1.55} (1 - e^{-310t}) \]

Two coils in series. In the first arrangement, the standard condenser was used, with a 12v. supply.

The ratio R/L is therefore constant as both the resistance and inductance have been doubled and the current will follow the same law as given above for the single standard coil.

Maximum voltage across condenser
\[ V = I \sqrt{\frac{L}{C}} = 3.87 \sqrt{\frac{2 \times 0.005}{0.22 \times 10^{-6}}} = 825 \text{ volts} \]

Natural frequency of the primary circuit
\[ (LC)^{-\frac{1}{2}} = \sqrt{(2 \times 0.005 \times 0.22 \times 10^{-6})^{-\frac{1}{2}}} = 2.1 \times 10^4 \text{ cycles/sec.} \]
Changing the condenser size to 0.1 mf. by a similar calculation,

Natural frequency = $3.2 \times 10^4$ cycles/sec.

Maximum voltage across the condenser = 1220 volts.

**Three coils in series, standard condenser.** By a similar set of calculations as before, since the voltage resistance and inductance are trebled, the current will follow the same law as with the standard coil.

Natural frequency = $1.7 \times 10^4$ cycles/sec.

Maximum voltage across condenser = 1010 volts.

**Two coils in parallel, 0.5 mf condenser.** In this arrangement, the voltage across each coil is 6v. Lumping the resistance and inductance to form a simple circuit, the resistance and inductance are halved. The current through each coil, however, will be the same as before.

The build up of current will follow the law

$$i = \frac{6}{0.77} (1 - e^{-310t})$$

Natural frequency of the primary circuit

$$= (LC)^{-\frac{1}{2}} = \left(\frac{1}{2} \times 0.005 \times 0.5 \times 10^{-6}\right)^{-\frac{1}{2}}$$

$$= 2.8 \times 10^4 \text{ cycles/sec.}$$

Maximum voltage across the condenser
\[
\frac{L}{6} = I \sqrt{\frac{L}{6}} = 7.74 \times \frac{\frac{1}{2} \times 0.005}{0.5 \times 10^{-6}}
\]

= 550 volts
APPENDIX C.

AIR FLOW METER DESIGN CALCULATIONS.

Some means of damping out oscillations in the inlet manifold was required so that an orifice meter could be fitted to measure the air flow to the engine.

Instead of using the orthodox pulsation tank, it was proposed to employ an exhaust type muffler to reduce the size of the installation. The design was based on results obtained with simple expansion chambers.¹

Design features for the muffler:
(a) must be of reasonable size
(b) must not offer any appreciable impedance to the air flow to affect the volumetric efficiency of the engine in the normal operating range.
(c) must give high attenuation over the operating speed range of the engine.
(d) must be simple and easy to construct.

As a basis for design calculation, assuming a maximum crankshaft speed of 6000 r.p.m., the engine is a four stroke type, four cylinders in line, with a common intake manifold, therefore the intake is subject to two

pulsations per crankshaft revolution.

\[ \text{Maximum frequency} = \frac{6000 \times 2}{60} = 200 \text{ cycles/sec.} \]

Similarly, assuming a minimum speed of 1000 r.p.m. at which air consumption figures are required:

\[ \text{Minimum frequency} = 36 \text{ cycles/sec.} \]

This minimum frequency rules out resonators as theory and experimental results show that this type of muffler has low attenuation at low frequencies.

This factor also virtually rules out simple expansion chambers as a high expansion ratio is required to give an attenuation of over 20 db. This would make the muffler too bulky.

Let \( m = \frac{\text{Area of cross-section of chamber}}{\text{Area of cross-section of exhaust}} \)

Considering a multiple expansion chamber, high attenuation is obtained with two chambers, addition of a third expansion chamber only gives a small increase in the degree of attenuation.

\[ \text{taking as a tentative design a two expansion chamber muffler, there are two types offering:} \]

(1) external connecting tube type

(2) internal connecting tube type.

Now the low frequency pass region, that is, the region at low frequencies where the attenuation falls to a very low value, is lowered by increasing the length of the connecting
tube.

Now with type (1) regions of low attenuation, with a width of 50 cycles/sec. or more, occur between the large loops of the attenuation curves. (Figure 29)

This is objectionable.

The type finally decided, then, is a double chamber expansion type muffler, with an internal connecting tube. This appears to be the best type giving high attenuation in a wide range of frequencies, the pass regions being widely spaced, the attenuation rising rapidly to a high value from the "cut off" frequency. (Figure 29)

It is recommended that the velocity of flow through the connecting tube be less than 0.3 \( c \)

Where \( c = \) velocity of sound \( = 1140 \) ft./sec.

At the maximum engine speed of 6,000 r.p.m., assuming a volumetric efficiency of 70%, engine capacity = 900 cc.

Air consumption \( = \frac{6000}{2} \times 900 \times \frac{70}{100} \times \frac{1}{(2.54)^3} \times \frac{1}{1728} \times \frac{1}{60} \)

\( = 1.12 \) cu.ft./sec.

Diameter of connecting tube \( = \sqrt{\frac{4}{\pi}} \times \sqrt[3]{\frac{144 \times 1.12}{0.3 \times 1140}} \)

\( = 0.77 \) in., say \( \frac{3}{8} \) in.

For this small size connecting tube a large expansion ratio can be used without making the muffler too bulky, a large expansion ratio improves attenuation.
Taking \( m = 64 \)

Diameter of expansion chamber \( = \frac{6}{4} \times (64)^{\frac{1}{2}} \) \( = 6 \) in.

Now the cut off frequency is given by the approximation

\[
 f_{\text{cut off}} = \frac{c}{2\pi} \frac{1}{\sqrt[m]{m l_e l_c + \frac{1}{3}(l_e - l_c)}}
\]

Where \( l_e = \frac{1}{3} \) effective length of the connecting tube

\( l_c = \) length of expansion chamber

From the design curves, taking \( \frac{l_o}{x l_e} = \frac{1}{3} \) and taking as a tentative value \( l_o = 6 \) in.

\[
 f_{\text{cut off}} = \frac{1140}{2\pi} \frac{1}{\sqrt[6]{64 \times l_e^\frac{1}{3} l_o^2 + l_e^2 \cdot \frac{5}{3}}}
\]

\[
 = \frac{1140}{2\pi (96.2)^{\frac{1}{6}}}
\]

\[
 = 36.6 \text{ cycles per second}
\]

The minimum frequency is 36 cycles/sec. so the assumption made for the length of the connecting tube is near enough.

The attenuation characteristics for this muffler are shown in Figure 29, giving large damping at all frequencies between 36 and 250 cycles/sec.
Muffler Dimensions and Attenuation Curves

Expansion Ratio = 64

To Nozzle

\[ \text{To Muffler} \]

Attenuation Characteristics

<table>
<thead>
<tr>
<th>Frequency cycles/sec.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operating Range of Engine</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Attenuation db.</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
</tr>
<tr>
<td>30</td>
</tr>
<tr>
<td>20</td>
</tr>
<tr>
<td>10</td>
</tr>
<tr>
<td>0</td>
</tr>
</tbody>
</table>

0 100 200 300 400
AIR FLOW CALCULATIONS FOR ORIFICE GAUGE

For this shape orifice (checked previously with steam flow), pressure loss through orifice = \( \frac{1}{8} \rho V^2 \)

Where \( V \) = velocity of flow and \( \rho \) = density of the gas

For the manometer, three slopes are provided for high, medium and low air flows, alcohol being the manometer liquid.

Specific gravity of the alcohol used = 0.82

If \( H \) cm. is the reading on the manometer scale and \( F \) is the slope factor given by the manufacturers

\[ \text{Head across manometer} = \frac{H}{2.54} \times 0.82 \times \frac{1}{12} \text{ ft.} \]

\[ \text{Specific weight of water} = 62.4 \text{ lb./cu.ft.} \]

\[ \text{Pressure across manometer} = 0.027 \times 62.4 \times \frac{H}{F} \text{ lb./sq.ft.} \]

\[ = 1.685 \frac{H}{F} \text{ lb./sq.ft.} \]

For the nozzle, \( P_{\text{air}} = 0.00238 \text{ slugs/cu.ft at } 15^\circ \text{C and 14.7 lb./sq.in.} \)

and coefficient of velocity = 0.98

Velocity of flow in nozzle = \( 0.98 \times \left[ 2 \times 1.685 \times \frac{H}{0.00238} \right]^{\frac{1}{2}} \frac{\text{ft./sec.}}{F} \)

\[ = 36.8 \left( \frac{H}{F} \right)^{\frac{1}{2}} \text{ ft./sec.} \]

Nozzle diameter = 0.618 in.

Area of nozzle = 0.00208 sq.ft.

\[ \text{. Air flow through nozzle} = 36.8 \times 0.00208 \left( \frac{H}{F} \right)^{\frac{1}{2}} \text{ ft} \]

\[ (\text{at } 15^\circ \text{C and 14.7 lb./sq.in.}) = 0.0767 \left( \frac{H}{F} \right)^{\frac{1}{2}} \text{ cu.ft./sec.} \]
125

Mass rate of flow through nozzle = 0.0767 \( \left( \frac{H}{F} \right)^{\frac{1}{3}} \) lb./sec.

and air = 0.00238 \times 32.2 \text{ lb./cu.ft. at } 15^\circ \text{C and } 14.7 \text{ lb./sq.in.}

Mass rate of flow through nozzle = 0.00588 \( \left( \frac{H}{F} \right)^{\frac{1}{3}} \) lb./sec.

Slope factors:

- High slope  \( F = 1.37 \)
- Medium slope  \( F = 3.37 \)
- Low slope  \( F = 8.55 \)

Correction factor for temperature and pressure. If \( M \) is the mass rate of flow, and the suffixes 1 and 2 denote conditions of temperature and pressure, from the above:

\[
\frac{M_1}{M_2} = \left[ \frac{f_1}{f_2} \right]^{\frac{1}{2}} \quad \text{and since } \quad P = fRT
\]

\[
= \left[ \frac{P_1 T_2}{P_2 T_1} \right]^{\frac{1}{3}}
\]

Since the standard conditions were taken as 15\(^\circ\)C. and 14.7 lb./sq.in. (30 in. Mercury)

Correction factor = \[
\left[ \frac{288}{T} \times \frac{P}{30} \right]^{\frac{1}{3}}
\]

Where \( T \) is the absolute temperature and \( P \) is the barometer reading at the time of the test in in. mercury.
APPENDIX E

DETERMINATION OF COMPRESSION RATIO

The bore and stroke of the engine were known from the makers specifications but it was suspected that the engine has been rebored. Two determinations of compression ratio were required, the first with the standard head and the second with the multiple sparking plug bosses in position.

Measurement of clearance volume. The cylinder head was removed, supported on blocks and levelled by means of a spirit level. A pointer set to the face of the head indicated the level of the kerosine which was poured into the combustion volume from a burette.

The thickness of the gasket was determined from a series of micrometer readings, the area of cross section of the combustion space over the gasket was calculated by use of a planimeter.

Measurement of the swept volume. This was determined by two methods. The first was to take a number of measurements of the bore by means of an internal micrometer. The area of cross-section of the bore was thus known, multiplying by the stroke gave the swept volume of the cylinder.

The second method, used to check the first, was to
turn the crankshaft to B.D.C. (determined by setting the moveable plunger of a dial gauge on the piston), and to fill the bore with kerosine, checking the level as for the measurement of clearance volume.

Results.

(1) Standard head.
Volume required to fill compression space (average of 8 readings) = 42.2 cc.
Area of gasket over combustion space (12 readings) = 8.17 sq.in.
Thickness of gasket when compressed = 0.045 in.
Compression volume = (42.2 + 8.17x0.045x2.542) = 48.2 cc.
Average diameter of bore (12 readings) = 2.247 in.
Swept volume = (2.247 x 3.5 x 2.543) = 227.6 cc.
Swept volume by direct measurement (8 readings) = 227.4 cc.
Compression ratio = \[
\frac{227.5 + 48.2}{48.2} = 5.74
\]

(2) Head with Special plug bosses.
Total compression volume = 54.1 cc.
Compression ratio = \[
\frac{227.5 + 54.1}{54.1} = 5.20
\]
APPENDIX E

SPECIFICATIONS OF THE TEST-ENGINE

TYPE: Austin "Big 7", (1938-39 Model)
BORE: 56.77 mm. (2.235 in.)
STROKE: 88.90 mm. (3.500 in.)
CAPACITY: 900 cc.
FIRING ORDER: 1,3,4,2
OIL PRESSURE: 25 lb./sq.in.
CARBURETTOR: Zenith downdraught
SETTING: Choke 23
Main 90
Compensating 50
Slow Running 60
TAPPET CLEARANCES (HOT): Inlet and Exhaust 0.004 in.
VALVE TIMING: Inlet opens T.D.C.
BREAKER GAP: 0.012 in.
PLUG GAP: 0.018-0.020 in.
IGNITION TIMING: On flywheel.
Fully retarded T.D.C.
MAXIMUM POWER OUTPUT: 25 H.P. at 4,000 r.p.m.
APPENDIX F

THERMODYNAMIC CALCULATIONS FOR THE IDEAL CYCLE

The values of indicated specific fuel consumption and indicated brake mean effective pressure were calculated from a consideration of ideal cycles, by means of air tables\(^1\) and thermodynamic charts\(^2\).

Allowance was made for variation in specific heats and exhaust gas dilution.

Outline of the method. The temperature and pressure of the incoming air being known, the isentropic compression was calculated by means of the air charts. To allow for the heating of the charge in the inlet manifold, an approximation to the temperature of the ingoing charge was made from a series of intake manifold temperatures taken from the actual engine (Table 3).

Residual exhaust gas has very little effect on the specific heats or the density of the unburned charge at any given temperature and no effect of the composition of the charge after combustion, for the fuel-air ratios

---


for successive cycles are the same. The displacing effect of the exhaust gas in the induction stroke and the smaller amount of heat released during combustion need only be considered.

The mixing at the end of the induction stroke can be considered as a constant pressure process, the pressure and the enthalphy of the mixture define the conditions at the beginning of the compression stroke. An approximation was made to the entropy, enthalpy and specific volume of the residual exhaust to fix the condition of the charge at the beginning of the compression stroke, carrying the calculations right through with this value, the original assumption was checked, if the assumption was in error another calculation was made with an amended value until the value agreed reasonably well.

The compression ratio fixes the specific volume and the relative volume at the end of the compression stroke and the corresponding temperature, relative pressure and internal energy determined from the air tables. At this point, the quantity of heat released by the combustion of the fuel may be added to give the internal energy at the beginning of the expansion stroke. Both the specific volume and the internal energy are therefore known, the pressure temperature and entropy may be determined
from the Hottel chart. (The Hottel charts are calculated for 80, 90, 100, 110 and 120 per cent of the theoretical fuel.)

The entropy must remain constant during the expansion stroke, the specific volume at the end of the expansion stroke is known, so the conditions can be determined.

The temperature of the residual exhaust can be readily determined as the entropy must remain constant, and the pressure must be equal to that in the exhaust manifold.

A complete calculation is given in Table 4.

Calculations for the condition of the charge at the point of ignition. The equation for piston travel is given by the following:

\[ x = r(1 - \cos a - \frac{1}{2} r \sin^2 \frac{a}{l}) \]

Where

- \( x \) = piston travel
- \( r \) = crank radius
- \( l \) = length of connecting rod
- \( a \) = angle between the crank radius and the line of action of the piston.

Taking \( \frac{r}{l} = 4 \) and considering values of \( a \) up to 40° differences in the value of \( \frac{r}{l} \) of 0.25 will affect the results by less than 0.1%.
Knowing the value of \( x \) for various crank angles, and the compression volume when the piston is at top dead centre, the compression volume at any crank angle can be deduced. The total volume at bottom dead centre is known and thus the compression ratio is known.

Assuming the compression to be isentropic, that is \( pv^n = \text{constant} \), taking a value for \( n \) of 1.3 and knowing the inlet manifold pressure, an approximation to the pressure of the charge can be made.

Sample calculations are given in Table 5.
The temperature of the manifold was recorded by means of a thermocouple at a point just before entry into the cylinder block. The mixture strength was held constant at approximately 10% rich and the speed was held at 2,000 r.p.m. The throttle opening and brake load were varied.

**Conditions of test:**

- **Speed:** 2000 r.p.m.
- **Mixture:** 10% rich
- **Water temperature:** 70°C.

<table>
<thead>
<tr>
<th>Manifold Pressure (in. mercury)</th>
<th>Manifold Temperature (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>14.0</td>
<td>112</td>
</tr>
<tr>
<td>15.8</td>
<td>104</td>
</tr>
<tr>
<td>16.6</td>
<td>102</td>
</tr>
<tr>
<td>17.8</td>
<td>98</td>
</tr>
<tr>
<td>18.8</td>
<td>96</td>
</tr>
<tr>
<td>20.2</td>
<td>93</td>
</tr>
<tr>
<td>21.2</td>
<td>91</td>
</tr>
<tr>
<td>21.8</td>
<td>90</td>
</tr>
<tr>
<td>22.8</td>
<td>88</td>
</tr>
<tr>
<td>23.9</td>
<td>86</td>
</tr>
</tbody>
</table>
TABLE 4.
TABULATED CALCULATIONS FOR A CONSTANT VOLUME CYCLE
FOR A SPARK IGNITION ENGINE

Compression ratio = 5.2, fuel-air ratio = 0.0746
Fuel: octene

<table>
<thead>
<tr>
<th></th>
<th>Residual exhaust gas</th>
<th>Fresh Charge</th>
<th>Start Compression stroke</th>
<th>End Compression stroke</th>
<th>Start Expansion Stroke</th>
<th>End Ex. Stroke</th>
</tr>
</thead>
<tbody>
<tr>
<td>T, °R.</td>
<td>2850</td>
<td>563</td>
<td>802</td>
<td>1498</td>
<td>5090</td>
<td>3740</td>
</tr>
<tr>
<td>P, psia</td>
<td>14.7</td>
<td>8.0</td>
<td>8.0</td>
<td>-</td>
<td>76.2</td>
<td>280</td>
</tr>
<tr>
<td>v, ft³/lb air</td>
<td>83</td>
<td>26.0</td>
<td>37.0</td>
<td>7.26</td>
<td>37.8</td>
<td>83</td>
</tr>
<tr>
<td>V_r, ft²/lb air</td>
<td>-</td>
<td>-</td>
<td>1697</td>
<td>326</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>E, B/lb air</td>
<td>-</td>
<td>-</td>
<td>41.9</td>
<td>212</td>
<td>1548</td>
<td>992</td>
</tr>
<tr>
<td>H, B/lb air</td>
<td>.612</td>
<td>-</td>
<td>-</td>
<td>-.613</td>
<td>.613</td>
<td>.613</td>
</tr>
<tr>
<td>H, B/lb air</td>
<td>848</td>
<td>36.7</td>
<td>96.8</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Weight ratio of residual exhaust gas to total charge

\[ \frac{1}{v_a} = \frac{1}{v_b} + 1 = \frac{4.2}{v_b} + 1 = 0.074 \]

Energy of combustion = 1336 Btu

Work in = 212 - 52 = 160

Work out = 1548 - 992 = 556

Nett E = 396

I.S.F.G. = \[ \frac{0.0746 \times 2545 \times 0.926}{396} \] = 0.444 lb./HP. hr.

I.M.E.P. = \[ \frac{778 \times 396}{144(37.8 - 7.26)} \] = 70.6 lb./sq.in.
IDEAL CYCLES

Compression Ratio - 5:2

Allowance made for residual exhaust gas

\[ F = \frac{\text{Weight of Fuel}}{\text{Chemically Correct wt. of fuel}} \]

Intake Manifold Pressure lb./sq.in.

L.S.F.C. lb./I.H.P.Hr.

<table>
<thead>
<tr>
<th>Intake Manifold Pressure</th>
<th>L.S.F.C.</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>0.45</td>
</tr>
<tr>
<td>7</td>
<td>0.45</td>
</tr>
<tr>
<td>8</td>
<td>0.45</td>
</tr>
<tr>
<td>9</td>
<td>0.45</td>
</tr>
<tr>
<td>10</td>
<td>0.45</td>
</tr>
<tr>
<td>11</td>
<td>0.45</td>
</tr>
<tr>
<td>12</td>
<td>0.45</td>
</tr>
</tbody>
</table>

F = 1.0

F = 0.9

F = 1.2

F = 1.1
### APPROXIMATE PRESSURE AT DIFFERENT POINTS OF COMPRESSION STROKE

<table>
<thead>
<tr>
<th>Degrees</th>
<th>Piston Change in B.T.D.C. % Stroke</th>
<th>Compression Volume cc.</th>
<th>Compression Ratio R</th>
<th>Multiplying Factor for Pressure R^1.3</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
<td>48.1</td>
<td>5.74</td>
</tr>
<tr>
<td>5</td>
<td>0.2</td>
<td>0.5</td>
<td>48.6</td>
<td>5.69</td>
</tr>
<tr>
<td>10</td>
<td>1.0</td>
<td>2.3</td>
<td>50.4</td>
<td>5.48</td>
</tr>
<tr>
<td>15</td>
<td>2.1</td>
<td>4.6</td>
<td>52.7</td>
<td>5.24</td>
</tr>
<tr>
<td>20</td>
<td>3.7</td>
<td>8.2</td>
<td>56.3</td>
<td>4.90</td>
</tr>
<tr>
<td>25</td>
<td>5.8</td>
<td>13.2</td>
<td>61.3</td>
<td>4.50</td>
</tr>
<tr>
<td>30</td>
<td>8.3</td>
<td>18.9</td>
<td>67.0</td>
<td>4.11</td>
</tr>
<tr>
<td>35</td>
<td>11.1</td>
<td>25.1</td>
<td>73.2</td>
<td>3.77</td>
</tr>
<tr>
<td>40</td>
<td>14.3</td>
<td>32.5</td>
<td>80.6</td>
<td>3.41</td>
</tr>
</tbody>
</table>
APPENDIX G.

ADDITIONAL PERFORMANCE CURVES OF THE ENGINE

OBTAINED FROM INDIVIDUAL EXPERIMENTS.

Operating conditions:
Outlet water temperature: $72^\circ C. \pm 2^\circ C.$
Ignition timing set for maximum torque
Fuel: "Europa" first grade.
All readings corrected for ambient air conditions.
I. EXPERIMENTS WITH S.U. CARBURETTOR

Coil Circuit: Two and three coils in series with a 12 v. or 18 v. supply. Standard condenser, 0.22 mf. capacitance.
FIGURE 31.

1500 R.P.M.
Manifold Pressure 19.6 in. Hg.

B.M.E.P. lb./sq.in.

SPECIFIC FUEL CONSUMPTION lb./B.H.P. hr.

AIR - FUEL RATIO

-50
-55
-60
-65
-70
FIGURE 32.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

2000 R.P.M.
Manifold Pressure 13.8 in.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

2000 R.P.M.
Manifold Pressure 13.8 in.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

2000 R.P.M.
Manifold Pressure 13.8 in.
2000 R.P.M.
Manifold Pressure 16.0 in. Hg

B.M.E.P. lb./sq.in.

SPECIFIC FUEL CONSUMPTION lb./b.h.p. hr.

AIR-FUEL RATIO
2000 R.P.M.

Manifold Pressure 13.3 in Hg

![Graph showing B.M.E.P. and Specific Fuel Consumption vs. Air-Fuel Ratio at 2000 R.P.M.](image-url)
1500 R.P.M.
Manifold Pressure 18.0 in. Hg.

FIGURE 36.

B.M.E.P.  lb./sq.in.

SPECIFIC FUEL CONSUMPTION  lb./B.H.P. hr.

AIR - FUEL RATIO
1500 R.P.M.

Manifold Pressure 19.7 in. Hg.
II. EXPERIMENTS WITH COILS IN SERIES

Test equipment:
Zenith carburettor with variable main jet.

Coil circuit: Two coils in series with 12 v. supply standard condenser 0.22 mf. capacitance.
FIGURE 38.

2000 R.P.M.
Manifold Pressure 18.0 in. Hg.

![Graph showing specific fuel consumption vs air-fuel ratio at 2000 RPM with manifold pressure of 18.0 in. Hg.](image-url)
FIGURE 39.

1500 R.P.M.
Mannifold Pressure 21.8 in. Hg.

AIR - FUEL RATIO
FIGURE 40.

1500 R.P.M.
Manifold Pressure 15.5 in. Hg.

B.M.E.P. (lb./sq.in.)

SPECIFIC FUEL CONSUMPTION (lb./B.H.P.hr.)

AIR - FUEL RATIO
Fig. 41. Graph showing the relationship between air-fuel ratio and manifold pressure. The graph compares different plug configurations:

- **Additional Range with Three Plugs**
- **Additional Range with Two Plugs**
- **Mixture Range with One Plug**

The chart also indicates a 'Chemically Correct Mixture (Approximately)' line. The x-axis represents manifold pressure in inches of mercury, while the y-axis represents the air-fuel ratio.
III. EXPERIMENTS WITH COILS IN SERIES AND CONSTANT FREQUENCY.

Test equipment. Zenith carburettor with variable main jet.

Coil Circuit: Two coils in series with a 12 v. supply with an 0.1 mf. capacitance.

Single coil, 6 v. supply, 0.2 mf. capacitance.
FIGURE 42.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

1500 R.P.M.
Manifold Pressure 21.0 in.

AIR-FUEL RATIO

SPECIFIC FUEL CONSUMPTION

B.M.E.P., lb/sq.in.
EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

1500 R.P.M.
Manifold Pressure 15-1 in.
EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

1500 R.P.M.
Manifold Pressure 19.7 in.

FIGURE 44.

B.M.E.P. (lb/sq.in.)

SPECIFIC FUEL CONSUMPTION (lb/b.h.p.hr.)

AIR-FUEL RATIO

11 12 13 14 15 16 17
Figure 45.

Effect of Air-Fuel Ratio on Performance

2000 R.P.M.

Manifold Pressure 18.8 in.

B.M.E.P. (lb/sq.in.)

Specific Fuel Consumption (lb/B.H.P. hr.)

Air-Fuel Ratio

12 13 14 15 16 17 18
FIGURE 46.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

1500 R.P.M.
Manifold Pressure 14.5 in.

B.M.E.P. (lb/sq.in.)

SPECIFIC FUEL CONSUMPTION (lb/B.H.P.hr.)

AIR-FUEL RATIO
IV. EXPERIMENTS WITH COILS IN PARALLEL AND CONSTANT FREQUENCY

Test equipment.

Zenith carburettor fitted with a variable main jet.

Coil circuit: Two coils in series with a 6 v. supply with an 0.5 mf. capacitance.

Single coil, 6 v. supply, 0.2 mf. capacitance.
FIGURE 47.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

1500 R.P.M
Manifold Pressure 19.4 in.

B.M.E.P. lb./sq.in.

SPECIFIC FUEL CONSUMPTION lb./B.H.P. hr.

AIR-FUEL RATIO

12 13 14 15 16 17 18
FIGURE 48

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

2000 R.P.M.
Manifold Pressure 19.1 in.

B.M.E.P. lb/sq.in.

SPECIFIC FUEL CONSUMPTION lb./B.H.P.hr.

AIR-FUEL RATIO

65

60

55

50

45

40

35

30

25

20

15

10

5

0
EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

2000 R.P.M.
Manifold Pressure 17.1 in.

(Specific Fuel Consumption vs. Air-Fuel Ratio graph)

B.M.E.P. lb/sq.in.

SPECIFIC FUEL CONSUMPTION lb./B.H.P. hr.

AIR-FUEL RATIO
FIGURE 50.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

1500 R.P.M.
Manifold Pressure - 16.1 in.

AIR-FUEL RATIO
EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

2000 R.P.M.
Manifold Pressure 20.9 in.

AIR-FUEL RATIO vs. B.M.E.P. (lb/sq.in.)

AIR-FUEL RATIO vs. SPECIFIC FUEL CONSUMPTION (lb./B.H.P. Hr.)
VARIATION OF B.M.E.P. WITH INTAKE MANIFOLD PRESSURE

AT MAXIMUM POWER SETTING 1500 R.P.M

MANIFOLD PRESSURE in. Hg

BRAKE MEAN EFFECTIVE PRESSURE lb/sq.in.

AIR-FUEL RATIO

FIGURE 52.
IV. RESULTS OBTAINED WITH THE STANDARD ENGINE.
EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

Standard Engine  2000 R.P.M.

B.M.E.P.  lb./sq.in.

Specific Fuel Consumption  lb./H.P.hr.

AIR-FUEL RATIO
EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

Standard Engine 1500 R.P.M.

Figure 54.
FIGURE 55.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

Standard Engine 2000 R.P.M.

EFFECT OF AIR-FUEL RATIO ON PERFORMANCE

Stardard Engine 2000 R.P.M.

AIR-FUEL RATIO

B.M.E.P. lb./sq.in.

Specific Fuel Consumption lb./B.H.P.hr.
TABLE 6.

VOLUMETRIC EFFICIENCY OF THE STANDARD ENGINE AT FULL THROTTLE.

Test conditions:

Mixture: Maximum power (approximately 15% rich)

Water temperature 70°C.

Ignition advance set to give maximum torque.

Zenith Carburettor.

<table>
<thead>
<tr>
<th>Engine Speed r.p.m.</th>
<th>Swept volume c.f.m.</th>
<th>Air Flow rate c.f.m.</th>
<th>Volumetric Efficiency per cent</th>
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</thead>
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<tr>
<td>1500</td>
<td>24.1</td>
<td>17.5</td>
<td>72.7</td>
</tr>
<tr>
<td>1750</td>
<td>28.1</td>
<td>20.4</td>
<td>72.7</td>
</tr>
<tr>
<td>2250</td>
<td>36.1</td>
<td>25.5</td>
<td>70.8</td>
</tr>
<tr>
<td>2500</td>
<td>40.2</td>
<td>28.0</td>
<td>69.6</td>
</tr>
<tr>
<td>3000</td>
<td>48.2</td>
<td>31.5</td>
<td>65.3</td>
</tr>
<tr>
<td>3500</td>
<td>56.2</td>
<td>34.6</td>
<td>61.6</td>
</tr>
<tr>
<td>4000</td>
<td>64.3</td>
<td>37.1</td>
<td>57.7</td>
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</table>
ENGINE UNDER MOTORING TESTS DURING INITIAL RUNNING-IN PROCESS.
BIBLIOGRAPHY

A. BOOKS


Fraas, Arthur P., Combustion Engines.

Heldt, P.M., High Speed Combustion Engines.
New York: P.M. Heldt. 1946.

Jost, W., Explosion and Combustion Processes in Gases.

Judge, A.W., The testing of High Speed Internal Combustion Engines.

Lewis, B., and Elbe, G. von, Combustion, Flame and Explosions of Gases.


London: Blackie and Son Ltd. 1953.

London: Blackie and Son Ltd. 1950.


Young, A.P., and Griffiths, L., Automotive Electrical Equipment.
London: Iliffe and Sons Ltd. 1950.
B. PERIODICAL ARTICLES


El tgroth, G.V., Electronic Ignition Systems. Electronics 18: 106-112 (April, 1945)


Karpov, V.P., Analysis of Thermal Ignition Processes in Internal Combustion Engines. Engineers Digest 9: 113-115 (April, 1948)


Miller, C.D., The Role of Detonation Waves and Autignition in Spark Ignition Engine Knock as Shown by Photographs Taken at 40,000 and 200,000 Frames per Second. S.A.E. Quarterly Trans. Vol 1, No 1, (1947)


Peter, M.F., Summerville, W.L., and Davis, M., An Investigation into the Effectiveness of Ignition Sparks. N.A.C.A. Technical Report No. 359 (1930)


Sloane, R.E., Ignition of Gaseous Mixtures by Corona Discharge. Phil. Mag. 19: 998-1002 (1936)


Taylor-Jones, E., Morgan, J.D., and Wheeler, R.V., On the Form of Temperature Wave Spreading by Conduction from Point and Spherical Sources; with a Suggested Application to the Problem of Spark Ignition. Phil. Mag. 43: 359-364 (1922)


Viallard, R., Ignition of Explosive Gas Mixtures by Electric Sparks.
Journ. Phys. Chem. 16: 555-564 (1948)

Watson, E.A., Coil Ignition Systems.
Journ. Inst. Electrical Engineers. 70: 363-372 (January, 1932)


Industrial and Engineering Chemistry, 23: 538-542 (May, 1931)

C. BOOKS OF TABLES


Hottel, H.C., Willis, G.C., and Satterfield, C.N.,
Thermodynamic Charts for Combustion Process.