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The idealized limited pressure compression ignition cycle: an empirical treatment of its response to varying operating parameters

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THE IDEALIZED LIMITED - PRESSURE

COMPRESSION - IGNITION CYCLE

AN EMPIRICAL TREATMENT OF
ITS RESPONSE TO VARYING
OPERATING PARAMETERS

by

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SYNOPSIS.

A conventional analysis of the limited-pressure compression-ignition cycle results in theoretical equations that express cycle performance as a function of the thermodynamic properties of the working fluid at salient points in the cycle. Thus the assessment of cycle performance is based on a complete cycle computation carried out with the aid of thermodynamic charts. This approach does not produce speedy and accurate results because

- a) the procedure involves laborious trial-and-error methods,
- and b) the charts available do not cover the whole range of mixture strengths at which an internal combustion engine following this cycle may operate.

It is shown that existing thermodynamic charts may be used to develop empirical equations for cycle efficiency, work output, and mean effective pressure, for a range of mixture strengths and, in general, with improved accuracy for mixture strengths requiring interpolation between the charts.

An investigation of other theoretical quantities of interest to the designer, such as maximum cycle temperature and exhaust temperature, is included.

The data are presented in the form of nomographs enabling theoretical performance to be estimated for naturally aspirated diesel engines over a wide range of operating conditions.

Graphs prepared from the nomographs show trends similar to those obtained from tests on a single cylinder four-stroke compression-ignition engine.

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SECTION 1.

INTRODUCTION.

For a theoretical analysis of what takes place in an internal combustion engine certain assumptions must be made concerning the processes constituting the engine cycle and the thermodynamic properties of the working substance. If the assumptions underlying such an analysis represent an approachable ideal, the theoretical performance of the engine may be used as a convenient and meaningful standard of reference.

In the present investigation the limited-pressure, compression-ignition cycle was selected as the ideal representation of the actual "solid-injection" type diesel engine cycle, whilst the properties of the working fluid were obtained from thermodynamic charts.¹⁾ These charts offer a highly accurate account of the physical conditions and properties of the medium in its various states.

The mixed cycle involves the following basic thermodynamic processes :

1. isenthalpic mixing of the burned products of combustion from the previous cycle with the fresh air inducted

2. adiabatic compression of the mixture
3. a combustion process occurring partly instantaneously at constant volume and partly progressively at constant pressure
4. isentropic expansion of the burned products to a point at which the exhaust valve opens and some of the gases expand further in expelling the remainder.

In the mixed cycle some assumption has to be made as to the proportion of heat added at constant volume and at constant pressure. In the actual engine it is often necessary to arrange the injection timing and the injection characteristics to limit the peak pressure to an acceptable value. A realistic approach requires that a similar restriction is placed on the maximum pressure of the ideal mixed cycle. The resulting arbitrary division of the available total heat into a portion added to the working substance at constant volume and a portion added at constant pressure provides a fuel-air cycle that is a reasonably accurate guide for calculations on the behaviour of high compression engines with injected fuel.

The use of thermodynamic charts offers the most satisfactory approach to the problem of the cyclical properties of the working

substance. In their most convenient form they consist of :

1. a modified air chart prepared by Hottel, Williams, and Satterfield ¹),
2. a number of burned mixture charts prepared by the same authors ¹), and an additional one prepared by Ryan ²),

Since the process of combustion changes the properties of the working substance appreciably, it is necessary to consider separately the thermodynamic properties before and after combustion. Variations in the specific heats as a function of the instantaneous temperatures are allowed for in all charts. The combustion charts also take into consideration dissociation effects at the high combustion temperatures and re-association effects during the expansion stroke.

The charts allow complete assessment of the cyclical properties of the working fluid. The essential condition for the computation of the cycle performance is that these properties be determined for all transition points between the processes throughout the cycle. The complete solution of the cycle is a slow and laborious process, which has seriously limited the use of theoretical cycle data ³).

Since the actual number of operating conditions that influence the cycle performance is relatively small, it is possible to

replace the cumbersome thermodynamic cycle solutions by empirical expressions for those cycle performance quantities that are of interest. At the same time this approach offers an opportunity to eliminate, to some extent, the limitations imposed by the thermodynamic charts which are necessarily confined to certain values of cycle operating parameters, or to a certain range of values.

LIST OF SYMBOLS*

- E - total internal energy (sensible plus chemical),
BTU/pound-mole
- e - total internal energy (sensible plus chemical),
BTU/lb. of air constituents
- F - ratio of fuel used to that required theoretically
for complete combustion.
- f - weight fraction of residual gases in equivalent
burned mixture at the beginning of compression

* The symbols and definitions conform in general to the nomenclature used in the thermodynamic charts ^{1, 2}).

- H - total enthalpy (sensible plus chemical), BTU/pound-mole
- h - total enthalpy (sensible plus chemical), BTU/lb. of air constituents
- MEP - mean effective pressure, lb./in².
- p - pressure, lb./in². abs.
- Q - heat, BTU/lb. of fuel ; or a parameter expressing cycle performance
- r - compression ratio
- S - entropy, BTU/pound-mole. °R.
- s - entropy, BTU/lb. of air constituents. °R.
- T - temperature, degrees Rankine
- V - volume, ft³/pound-mole.
- v - volume, ft³/lb. of air constituents
- W - work output, BTU/lb. of fresh air
- y - chart conversion factor, pound-mole of mixture per lb. of air constituents, plus appropriate fuel
- η - efficiency

Suffixes :

- a - actual
- c - combustion
- e - exhaust
- fr - fresh air

g - gross value
i - intake
n - net value
p - pumping
r - ratio
s - sensible portion of thermodynamic function only,
i.e. chemical portion excluded
th - theoretical
1-6 - cycle transition points

SECTION 2.

THERMODYNAMIC CHARTS

The thermodynamic charts used in the present analysis have been discussed frequently in the literature ^{1,4,5,6}), and a full description will not be included here. However, a brief discussion of the particular suitability of the charts to compression-ignition cycle computation may be of some value. Two types of charts are applicable to this work.

1. A modified air chart which permits the complete calculation of a cycle in which the maximum temperature reached after combustion is less than 2520°R .

Unfortunately, this limiting temperature, above which chemical dissociation is considered to affect the properties of the working fluid, restricts the use of this chart to fuel-air mixtures below $F = 0.20^*$) for the most favourable combination of engine operating conditions, and normally to values below $F = 0.15$. Clearly, the output of an engine operating at such a low mixture strength will be considerably

* F is the symbol used to indicate the ratio of the fuel used to that required theoretically for complete combustion. This fuel fraction is a measure of mixture strength, or fuel-air ratio. The terms "fuel fraction" and "mixture strength" are used in this report.

below the rated value. With due allowance to physical reality it can be said that the modified air chart by itself is of little value in compression-ignition cycle analysis.

2. Burned mixture charts are employed to analyze the cycle processes taking place after combustion when the peak cycle temperature exceeds 2520°R . Since dissociation increases the number of molecules taking part in the cycle, a variation that involves a change in the value of the specific thermodynamic properties, the burned mixture charts are constructed on a mass basis. The unit mass used is one pound of air constituents, plus appropriate fuel. This mass basis necessitates the use of separate charts for each value of mixture strength (F)^{*}).

Mixture strengths used in compression-ignition engine operation vary from $F = 0.3$ to $F = 0.8$. For this range two burned mixture charts are available, i.e. for $F = 0.5$ and $F = 0.8$. Hence, the specific thermodynamic properties of the working fluid after combustion can be evaluated with accuracy only for those cycles for which the adopted value of the mixture strength is either $F = 0.5$ or $F = 0.8$. When the mixture strength does not correspond to those values, it has been recommended practice⁴), to use the burned

* The various material bases of the thermodynamic charts, as well as the terminology used in describing the conventional units employed in cycle analysis are discussed in some detail in Appendix 7 , page 151.

mixture chart of the nearest F - value, under the assumption that the specific properties will not change materially with small variations in the mixture strength. It is shown in a later section that relatively large errors in cycle performance data, notably in values of the efficiency, will occur if this practice is adopted.

SECTION 3.

THE LIMITED - PRESSURE COMPRESSION-IGNITION ENGINE.

In considering the relative cycle performance of an IC engine, some standard of comparison is required that constitutes a theoretical limit of performance. In the present analysis the limited-pressure cycle was selected as this "approachable ideal". For convenience the cycle is referred to as the theoretical cycle, the limiting cycle, or sometimes the mixed cycle.

3-1 Characteristics.

In a high-compression engine it is often necessary to arrange the injection timing and the injection characteristics to limit the peak pressure to an acceptable value. A realistic approach requires that a similar restriction be placed on the maximum pressure of the idealized cycle. On the other hand efficiency and operational considerations require at least part of the combustion process to take place at constant volume ⁷). As a result the available heat of combustion of the fuel is divided into a portion added to the working fluid at constant volume, and a remaining portion added at constant pressure. Since combustion characteristics vary with the design of fuel injection equipment and combustion chambers, as well as with adjustments made to each individual engine, an arbitrary division is made of the total available heat into these two quantities. The

FIG. 1 - P-V DIAGRAM FOR THE LIMITED-PRESSURE CYCLE.

resulting cycle is sometimes referred to as the mixed cycle; it is considered to be a reasonably accurate guide to the indicated performance of the majority of engines using injected fuel.

Further assumptions underlying the ideal mixed cycle are :-

1. There is no heat exchange to and from the working fluid.
2. There is no chemical change in either the fuel or the air before combustion.
3. Chemical equilibrium occurs at all points of the cycle.
4. Valve openings and closings take place instantaneously at specified points in the cycle.

3-2 Thermodynamic Analysis.

The sequence of processes occurring in the mixed cycle is represented in the $P - V$ diagram shown in Figure 1. In this diagram at point (1) of the cycle the pressure (p_1) is equal to the intake pressure (p_i). In process 1-2 the working fluid is compressed adiabatically to a pressure (p_2) below the maximum cycle pressure (p_3). In the combustion process 2-3 heat is added to the working fluid in both the constant volume process 2-2' and the constant pressure process 2'-3. Isentropic expansion takes place between points (3) and (4). The exhaust valve opens instantaneously at bottom dead centre. A major portion of the

products of combustion passes out of the cylinder, expanding adiabatically from pressure (p_4), to exhaust pressure (p_e), or (p_5). This process is irreversible; however, the portion of the working fluid remaining in the cylinder at point (5) is assumed to have undergone an isentropic expansion doing work in pushing the expelled portion into the exhaust manifold. The pressure and entropy at point (5) are then equal to the pressure and entropy at point (4'), i.e. for this portion of the working fluid the expansion can be considered to continue isentropically from point (4) to point (4'). Since two defining co-ordinates are the same for the points (5) and (4'), the state of the working fluid is identical at these points, although the mass has changed considerably.

During the exhaust stroke, constant-pressure process 5-6', all the remaining products of combustion are forced out of the cylinder with exception of a small residual quantity in the clearance volume. Again the properties per unit mass of working fluid at point (6') correspond to those at point (4').

As mentioned earlier, in cycles of this type the maximum cycle pressure is limited to an arbitrary value. The intake and exhaust processes are assumed to take place at constant pressures, the relative magnitudes depending on the local barometric pressure, the event of supercharging, inlet and exhaust duct losses, and the

influence of back-pressure of possible exhaust-driven ancillary gear.

In the naturally aspirated diesel engine discussed here the cylinder pressure at the end of the exhaust stroke, point (6') on the P-V diagram, will be slightly above atmospheric pressure, due to the friction losses of the flow through exhaust valve and exhaust manifold. Similarly, allowing for the pressure loss caused by the intake manifold and intake valve, the pressure in the cylinder during the intake stroke will be less than atmospheric.

In the idealized mixed cycle the inlet valve opens instantaneously at point (6). Prior to this, the residual gas in the clearance space is assumed to expand isentropically to intake pressure, process 6'-6. The final process making up the cycle, 6-1 is considered to be a constant-pressure induction process in which the induced fresh air charge mixes adiabatically with the residual gases.

3-3 Formal Solution of the Mixed Cycle.

Using the above model of the diesel engine cycle, the formal cycle solution involves the determination of the fluid properties at the various transition points between the idealized processes that make up the cycle.

Although the general procedure in solving the cycle is discussed extensively elsewhere ^{1,4,5,6}), it was found that some simplifications could be introduced in the series of computations compiled for this

report, and for that reason the procedure adopted is discussed in some detail.

The specific thermodynamic properties of the working fluid in the cycle are affected by the residual gases from the preceding cycle. Hence, when embarking on a cycle calculation, it is necessary to make assumptions concerning the state of the working fluid at a particular transition point, or station, between two cycle processes.

Conventionally, this station is chosen at the beginning of the compression stroke, i.e. point (1) in the P - V diagram of Figure 1. At this starting point of the cycle calculation, it is sufficient to assume the temperature (T_1) of the cylinder gases. A check on the assumed value is not possible until the entire cycle solution is completed, the number of trial solutions required to attain close agreement between computed and assumed values depending on the accuracy of the original assumption.

At point (1) of the cycle compression starts of one pound-mole of air plus an unknown weight fraction (f) of impurged residual products of combustion from the preceding cycle. The fraction (f) has small values at the high compression ratios used for the diesel engine; it is of negligible influence in this first part of the cycle, and no initial assumption as to its value need be made.

The ratio (F) of fuel used to that required for stoichiometrically complete combustion is zero at point (1) of the cycle, and the quantity of mixture present is referred to as one pound-mole of air constituents.

For the assumed value of temperature (T_1) and the given intake pressure (p_1), or (p_1), point (1) can be located on the modified air chart, and the values per pound-mole of air constituents of the other thermodynamic properties can be read off the chart. Due allowance must be made for the effect of pressure and temperature on enthalpy, internal energy and entropy. A number of auxiliary plots on the modified air chart facilitate the computation of the corrected values of these properties.

In process 1-2 the working fluid is compressed isentropically to a volume $V_2 = V_1/r$, where (r) is the volumetric compression ratio. This enables point (2) to be located on the modified air chart by the two state values for volume and entropy.

The combustion process 2 - 3 requires special consideration. To determine the specific thermodynamic properties of the working fluid at point (2') certain assumptions concerning the rate of injection and the rate of burning of the fuel must be made. However, the state of the working fluid at this intermediate point bears no relationship to the cycle performance, and for the solution of the cycle only the conditions at point (3) are relevant. By combining processes 2 - 2' and 2' - 3 the enthalpy (H_3) at point (3) can be evaluated, the net combustion heat added being assessed from the fuel-air ratio and the temperature of the liquid fuel.

In general, for mixture strengths exceeding $F = 0.20$, the combustion end temperature (T_3) will be outside the range of temperatures covered in the modified air chart. Hence, starting from point (3) in the cycle, a burned mixture chart must be used to determine the properties of the gases at the remaining transition points. In addition to the requirement that the cycle mixture strength (F) should correspond to the F -value of the burned mixture chart, the enthalpy (H_3) must be expressed in the chart material base. Accordingly, it is necessary to convert the material base from one pound-mole of air constituents to one pound of air constituents, plus appropriate fuel, by means of a chart conversion factor (y). An auxiliary plot on the modified air chart gives the

value of (y) as a function of the mixture strength (F) and the weight fraction of residual gases (f) .

Process 3-4 is an isentropic expansion, the volume at point (4) of the cycle being equal to $(y.V_1)$. As pointed out in the previous section, the specific properties of the mixture at point (5) correspond to those at point (4'). Each of the remaining points in the cycle is located on the burned mixture chart from a constant entropy line (s_3) intersecting the constant pressure line for either the exhaust pressure (p_e) or the intake pressure (p_i) . During the exhaust stroke, that is constant pressure process 5-6', additional working fluid is forced out of the cylinder. Again the properties of a unit mass of material at point (6') correspond to those at point (4'). At top dead centre the exhaust valve closes instantaneously, and the gases remaining in the clearing space are assumed to expand isentropically during process 6'-6. At the latter point the residual gas reaches intake pressure (p_i) , the inlet valve opens instantaneously, and intake of fresh air takes place during the remainder of the suction stroke.

At the end of the constant pressure mixing process 6-1 the working fluid consists of a mixture of residual gases and fresh air. Since the properties of these constituents are known, both the residual gas fraction (f) and the temperature (T_1) may be computed.

The unpurged gas fraction (f) can be found from the relation

$$f = \frac{v_{6'}}{v_{4'}} , \quad \text{where} \quad v_{6'} = \frac{v_4}{r} , \quad \text{since}$$

the state of the working fluid is the same at points (4'), (5), and (6').

The sensible enthalpy of the mixture per pound of air constituents at point (1) is found by calculating the separate values of the sensible enthalpies of the weight fraction (f) of residual gases and the weight fraction ($1 - f$) of fresh air. After conversion to a material base of one pound-mole the corresponding temperature (T_1) may be read from the modified air chart. The computed value of (T_1) should be reasonably close to the estimated temperature at the beginning of compression. Experience has shown that if the trial value differs from the computed value by more than 4°R , the cycle must be re-computed, using the newly found value of (T_1). As a rule the first trial value is estimated to within 10°R of the computed value by an experienced analyzer. In that case the second trial computation will produce the required accuracy.

3-4 Theoretical Equations for Cycle Performance.

When the specific thermodynamic properties of the working fluid at the cycle transition points have been determined, the fundamental quantities of cycle performance may be expressed in terms of these properties.

Referring to the P - V diagram in Figure 1, the gross work output (W_g) of the cycle, i.e. excluding pumping work, is equal to the work of expansion from point (2') to point (4) minus the work of compression from point (1) to point (2). This may be written as

$$W_g = p_3(V_3 - V_2) \frac{144}{778} + (E_3 - E_4) - (E_2 - E_1)$$

BTU/lb. -mole of mixture(1)

or

$$W_g = H_3 - E_4 - p_3 V_2 \frac{144}{778} - (E_2 - E_1)$$

BTU/lb. -mole of mixture(2)

Conversion of the material base of Equation (2) to the material base of the burned mixture charts is accomplished by means of the chart conversion factor (y) which is expressed in pound-moles of mixture per pound of air constituents, plus appropriate fuel. A variation in this conversion factor is experienced between cycle transition points (2)

and (3), since combustion changes the composition of the working fluid. However, this variation was found to be of negligible influence on cycle performance, and an average value of $y = 0.035$ may be used in the computations.

Therefore, the change in material base results in

$$W_g = h_3 - e_4 - y p_3 v_2 \frac{144}{778} - y(E_2 - E_1) \quad \text{BTU/lb. of}$$

air constituents, plus appropriate fuel(3)

A further conversion to a base of one pound of original fresh air gives

$$W_g = \frac{h_3 - e_4 - y p_3 v_2 \frac{144}{778} - y(E_2 - E_1)}{(1 - f)} \quad \text{BTU/lb. of}$$

fresh air(4)

In the ideal cycle both the intake and the exhaust process are assumed to take place at constant pressure enabling the pumping loop to be considered rectangular. This results in the convenient approximation for pumping work

$$W_p = \left[\frac{(p_e - p_i)(v_1 - v_6)}{(1 - f)} \right] \cdot \frac{144}{778} \quad \text{BTU/lb. of fresh air(5)}$$

Since $v_6 = v_1/r$, pumping work may be expressed as

$$W_p = \left[\frac{(p_e - p_i)v_1}{(1 - f)} \right] \cdot \left[\frac{r - 1}{r} \right] \cdot \frac{144}{778} \quad \text{BTU/lb. of fresh air(6)}$$

The net work output is

$$W_n = (W_g - W_p) \quad \text{BTU/lb. of fresh air(7)}$$

Substituting Equations (4) and (5) in Equation (7) gives

$$W_n = \left[\frac{h_3 - e_4 - y p_3 v_2 \frac{144}{778} - y(E_2 - E_1)}{(1 - f)} \right] - \left[\frac{(p_e - p_i)v_1}{(1 - f)} \right] \cdot \left[\frac{r - 1}{r} \right] \cdot \frac{144}{778} \quad \text{BTU/lb. of fresh air(8)}$$

Using a basis of one cubic foot of swept volume, the weight of fresh air inducted per cycle is

$$\left[\frac{(1 - f)}{(v_1 - v_6)} \right] \text{ lb. of fresh air}$$

or

$$\left[\frac{(1-f)}{v_1} \right] \cdot \left[\frac{r}{r-1} \right] \text{ lb. of fresh air } \dots\dots\dots(9)$$

Then, the gross work output per cycle per cubic foot of swept volume

$$W_g = \left[\frac{h_3 - e_4 - y p_3 v_2 \frac{144}{778} - y(E_2 - E_1)}{v_1} \right] \cdot \left[\frac{r}{r-1} \right]$$

BTU/ft³. of swept volume \dots\dots\dots(10)

and pumping work per cycle

$$W_p = (p_e - p_i) \frac{144}{778} \text{ BTU/ft.}^3 \text{ of swept volume} \dots\dots\dots(11)$$

The expression for gross mean effective pressure of the cycle is

$$MEP_g = \left[\frac{h_3 - e_4 - y p_3 v_2 \frac{144}{778} - y(E_2 - E_1)}{v_1} \right] \cdot \left[\frac{r}{r-1} \right] \frac{778}{144}$$

lb./in². \dots\dots\dots(12)

and since, for the rectangular pumping loop, the pumping mean effective pressure

$$MEP_p = (p_e - p_i) \text{ lb./in}^2. \dots\dots\dots(13)$$

whilst

$$\text{MEP}_n = (\text{MEP}_g - \text{MEP}_p) \quad \text{lb./in}^2. \quad \dots\dots\dots(14)$$

Then the net mean effective pressure is

$$\text{MEP}_n = \left[\frac{h_3 - e_4 - y p_3 v_2 \frac{144}{778} - y(E_2 - E_1)}{v_1} \right] \cdot \left[\frac{r}{r-1} \right] \cdot \frac{778}{144} - (p_e - p_i) \quad \text{lb./in}^2. \quad \dots\dots\dots(15)$$

Conventionally, the cycle efficiency (η) is the ratio of the net work output and the heat of combustion of the fuel at a standard temperature. On the modified air chart values of the heat of combustion of octene are quoted which are typical for fuels used for compression-ignition cycles ¹). For octene injected in liquid form at 70°F the net calorific value (Q_c) is 19,028 BTU per pound. Hence, the heat available per pound of fresh air is

$$(0.06775 \times 19,028) \quad \text{BTU} \quad \dots\dots\dots(16)$$

On the basis of one pound of fresh air inducted, the cycle efficiency is

$$\eta = \frac{W_n}{0.06775 \times 19,028} \quad \dots\dots\dots(17)$$

Substituting Equation (8) in Equation (17) the cycle efficiency may be expressed as

$$\eta = \left[\frac{h_3 - e_4 - \gamma p_3 V_2 \frac{144}{778} - \gamma (E_2 - E_1) - (p_e - p_i) v_1 \frac{r-1}{r} \frac{144}{778}}{(1 - f) \times 0.06775 F \times 19,028} \right] \dots\dots\dots(18)$$

From the above equations a relationship may be derived that is of interest in the development of the nomographs. Combining Equations (6), (8) and (15) it can be shown that the net mean effective pressure may be expressed thus

$$\text{MEP}_n = \frac{W_n}{W_p} (p_e - p_i) \text{ lb./in}^2. \dots\dots\dots(19)$$

where the pumping work varies from $W_p = 2.65$ to 2.71 BTU/lb. of fresh air over the range of operating conditions adopted for the present analysis. The selected values for the intake and exhaust pressures were $p_i = 14 \text{ lb./in}^2. \text{ abs.}$ and $p_e = 15 \text{ lb./in}^2. \text{ abs.}$ Hence, $(p_e - p_i) = 1.0 \text{ lb./in}^2. \text{ abs.}$ It follows that Equation (19) may be written as

$$\text{MEP}_n = \frac{W_n}{2.68} \text{ lb./in}^2. \dots\dots\dots(20)$$

In this expression the average value of the pumping work was used involving an error of $\pm 1.1\%$, which is an acceptable value.

SECTION 4.

OPERATING CONDITIONS OF THE MIXED CYCLE.

An examination of the theoretical Equations (1) - (20) reveals that the computation of cycle performance involves the complete thermodynamic solution of the cycle; i.e. for a given set of cycle operating conditions the properties of the working fluid must be computed at every step in the limited-pressure cycle. It was shown in Section 3-3 that this involves the determination of the temperature of the working fluid at the beginning of compression by iteration over the complete cycle.

Since the number of operating conditions that influence cycle performance is relatively small, it seems desirable as well as feasible to replace the theoretical equations by empirical relationships expressing cycle performance directly in its governing parameters.

It can be seen from a review of the formal cycle solution that the performance of the idealized cycle is influenced by six operating conditions or independent variables, namely the inlet pressure (p_i), the temperature of fresh inlet air (T_{fr}), the compression ratio (r), the mixture strength (F), the maximum cycle pressure (p_{max}), and the exhaust pressure (p_e). Consequently, an investigation is warranted of the functional relationship.

$$Q = \psi(p_i, T_{fr}, r, F, p_{max}, p_e) \dots\dots\dots(21)$$

where the dependent variable (Q) represents any quantity used in expressing cycle performance. An improved procedure is unlikely to result if the relative effects were to be examined of each independent variable on the cyclical thermodynamic properties of the working fluid used in the theoretical equations. However, the relationship expressed in Equation (21) may be analyzed in a more direct manner, if resort is made to using the data obtained from a large number of formal cycle solutions. By letting the operating conditions of the cycles vary over the whole of their normal range, their relative influence on cycle performance may be studied. Thus empirical relationships between the performance and the above independent variables can be readily established. When presented in the form of nomographs, such equations are a convenient guide to cycle performance under a variety of operating conditions.

SECTION 5.

RANGE OF CYCLES ANALYZED.

For the purpose of analyzing a selected range of compression-ignition cycles, with due regard to variations in design and type of service, the following three operating conditions were chosen as independent variables :-

1. Compression ratio (r), ranging from 14 to 22, in increments of 2.
2. Mixture strength (F), ranging from 0.15 to 1.0. In general, increments were chosen to correspond to the burned mixture charts available. The significance of performance data for intermediate F - values is discussed in the next section.
3. Maximum cycle pressure ($p_{\max}$) of 800, 900, 1000 and 1200 lb./in². abs.

The selected values are representative of a broad cross-section of diesel engines of the naturally aspirated type.

The remaining three operating conditions were maintained constant at the following arbitrary values :-

Intake pressure $p_i = 14 \text{ lb./in}^2. \text{ abs.}$

Temperature of fresh intake air $T_{fr} = 540^\circ\text{R.}$

Exhaust pressure $p_e = 15 \text{ lb./in}^2. \text{ abs.}$

The last stipulation implies that the objective of the present work is narrowed down to establishing an empirical equation to replace the functional relationship.

$$Q = \psi' (r, F, p_{\max}) \dots\dots\dots(22)$$

That is, empirical relationships are to be developed for theoretical cycle performance as a function of three independent variables, namely compression ratio, mixture strength, and maximum cycle pressure.

Although the relative influence of changes in the intake pressure, the temperature of the fresh intake air, and the exhaust pressure on cycle performance are not included here, some preliminary work has been carried out to investigate these additional aspects of cycle behaviour.

SECTION 6.

PROCEDURE AND ACCURACY

On the basis of ranges and increments of the independent parameters outlined in the previous chapter, a total of 80 cycles was computed in Series I of Cycle Analyses. All values of the cyclical thermodynamic properties of the working fluid are tabulated in Appendix 1, Tables 3 - 18, pages 75 - 90. Cycle performance figures and further cycle data of interest were determined using the theoretical equations derived in Section 3-4. The data are listed in Tables 26-41, pages 99 - 114.

As mentioned in the previous section, to establish trends in cycle behaviour due to variations in the ambient atmospheric conditions, or due to the event of supercharging, a second series of 17 cycles was computed. This Series II of Cycle Analyses is considered to be in the nature of a preliminary check. A brief discussion of the findings is given in Appendix 2 , page 122. The cyclical thermodynamic properties, and cycle performance data resulting from this series are presented in Appendix 1, Tables 19-21, pages 91 - 93, and Tables 42-44, pages 115 - 117, respectively.

To verify certain trends in cycle performance and to supplement results established in Series I, a number of 7 additional cycle

FIG. 2 - TYPICAL VARIATION OF CYCLE EFFICIENCY WITH FUEL FRACTION.

FIG. 3 - TYPICAL VARIATION OF THEORETICAL NET WORK OUTPUT WITH FUEL FRACTION

computations were carried out at both intermediate points and points outside the adopted ranges. The cyclical thermodynamic properties and the cycle performance data resulting from this Series III of Cycle Analyses are presented in Appendix 1, Tables 22-23, pages 94 - 95. and Tables 45-46, pages 118 - 119 , respectively.

Finally, a total of 10 cycle computations was completed for values of the cycle mixture strength (F), for which no corresponding burned mixture charts are available. In general, for the purpose of the analysis an existing burned mixture chart of the nearest F -value was used in each case. The resulting data in this Series IV of Cycle Analyses are discussed in a later part of this Section. They are tabulated in Appendix 1, Tables 24-25, pages 96 - 97 , and Tables 47-48, pages 120 - 121 .

The tabulated data served as the basis for the study of the relative influence of each operating condition, or independent variable, on cycle performance. The effect of variations in a certain operating condition was examined by plotting a particular performance quantity against such variable, other variables being maintained constant. Figures 2 - 4 were prepared as examples of this procedure. In Figures 2 and 3 the variations are shown of cycle efficiency (η) and net work output (W_n) as dependent variables

with (F), the fuel fraction or mixture strength, as the independent variable, for a given set of operating conditions. It is emphasised that in these two particular plots only the performance data corresponding to the F -values of 0.15, 0.20, 0.50, 0.80, 0.90 and 1.00 are considered to be accurate; i.e. data obtained in Series I and III of Cycle Analyses. Precise data are obtained only from those calculations in which the cycle solutions are carried out either with the aid of the modified air chart, as in the case for cycle mixture strengths $F = 0.15$ and $F = 0.20$, or with the aid of a modified air chart in conjunction with an available burned mixture chart constructed for an F -value corresponding to the cycle mixture strength, that is for the values $F = 0.50, 0.80, 0.90$, and 1.00. In addition to these precise data, a number of intermediate points are shown on the plots representing performance data established using existing burned mixture charts of an F -value nearest to the cycle mixture strength, Series IV of Cycle Analyses. From the distribution of the points, it is evident that this latter practice results in performance values that are in error, particularly in the case of the efficiency. It is considered that the smooth curves drawn through the points evaluated with the aid of burned mixture charts of the correct F -value give improved values of cycle performance at intermediate mixture strengths. Thus, for these points, a

FIG. 4 - TYPICAL VARIATION OF CYCLE EFFICIENCY WITH COMPRESSION RATIO FOR SELECTED VALUES OF FUEL FRACTION.

higher degree of accuracy is attained from empirical equations based on the mean curve than is possible by individual cycle solutions from the available charts.

A typical example of other plots used in the correlation of performance data is given in Figure 4. The smooth curves drawn through the plotted values of the cycle efficiency (η) illustrate the interdependence of the efficiency and the compression ratio (r) for a number of selected values of the mixture strength (F). By employing well-known curve-fitting techniques⁸, an empirical equation was developed to express this relationship, bearing in mind that its final form should be adapted to nomographic representation. Empirical equation (25) listed in Table 1, page 36, fits this particular set of curves. It expresses the relationship between the variables to the degree of accuracy given in the table.

The exact procedure followed in the development of an empirical relationship from a large number of cycle data is described and illustrated by the example presented in Appendix 3, page 125.

SECTION 7.

EMPIRICAL EQUATIONS.

7-1 Empirical Equations for Cycle Performance.

It is shown in Equation (17) that the net work output (W_n) of the cycle may be readily expressed in the cycle efficiency (η) and the mixture strength (F). Also, from Equation (20), the mean effective pressure (MEP_n) of the cycle is directly proportional to (W_n). As a result, only one empirical equation to express cycle performance need be developed. In the correlation of cycle data the efficiency (η) was singled out as the dependent variable, i.e. Equation (21) may be expressed explicitly as

$$\eta = \psi' (r, F, p_{\max}) \quad \dots\dots\dots(23)$$

Three empirical expressions for (η) were established over the range of the maximum cycle pressure; they are listed in Table 1, Equations (24) - (26). Their reliability may be judged by comparing the calculated values of (η) with the observed values, i.e. the values resulting from the cycle solutions. The deviations were computed as percentages of the observed values. Both the maximum and the average deviations are listed disregarding their algebraic signs. It is emphasized that only those deviations are

included that correspond to a "true" observed value, i.e. a value of (η) obtained from a cycle calculation in which a burned mixture chart of the appropriate F-value was used. Thus the listed deviations reflect in a realistic manner the degree of accuracy achieved in fitting empirical equations to the cycle data curves.

TABLE 1

EMPIRICAL EQUATIONS FOR THE CYCLE EFFICIENCY (η)

Maximum cycle pressure ($p_{\max}$) lb./in ² .abs.	Empirical equation for the cycle efficiency (η)	Equation No.	Deviations, % of observed value	
			Maximum	Average
800	$-0.000,200r^2 + (-0.001,65F + 0.007,65)(r+91.4) - 0.044$	(24)	1.0	0.6
1000	$-0.000,250r^2 + (-0.002,50F + 0.011,50)(r+50.8) - 0.044$	(25)	1.0	0.3
1200	$-0.000,250r^2 + (-0.002,20F + 0.012,28)(r+56.2) - 0.156$	(26)	1.2	0.5
900	-	-	1.2	0.5

7-2 Empirical Equations for Cycle Temperatures.

In the empirical procedure described for the analysis of limited-pressure cycles the formal thermodynamic cycle solution is by-passed completely. No consideration is given to the cyclical thermodynamic properties of the working fluid, since these do not enter into the empirical equations for cycle performance. However, a knowledge of certain values of these properties may be valuable from the design point of view. Notably the maximum cycle temperature and the exhaust temperature are quantities of interest to the designer of limited-pressure diesel engines. For that reason the previous analysis of the idealized limited-pressure cycle was extended to establish relationships between these quantities and mixture strength, compression ratio, and maximum cycle pressure for fixed values of intake and exhaust pressure and intake temperature.

Considering the maximum cycle temperature ($T_{\max}$), or (T_3), the following functional relationship may be stated :-

$$T_{\max} = \phi (r, F, p_{\max}) \dots\dots\dots (27)$$

Also, referring to the P - V diagram in Figure 1, the exhaust temperature (T_e) is equal to $T_5 = T_4'$; since the process 3 - 4' is a reversible adiabatic, the ratio (T_r) of maximum cycle

temperature to exhaust temperature may be expressed thus :-

$$T_r = \frac{T_{\max}}{T_e} = \left(\frac{p_{\max}}{p_e} \right)^{\frac{n-1}{n}} \dots\dots\dots(28)$$

The index of expansion (n) is dependent on the value of the mixture strength (F), since this quality controls the composition of the expanding working fluid.

Therefore, by taking (p_e) as constant,

$$T_e = \phi' (F, T_{\max}, p_{\max}) \dots\dots\dots(29)$$

Alternatively, in analogy to Equation (27),

$$T_e = \phi'' (r, F, p_{\max}) \dots\dots\dots(30)$$

Actual values of the maximum cycle temperature ($T_{\max}$) as determined in the formal cycle solutions are listed in Appendix 1, Tables 26 - 41, pages 99 - 114 .

These tabulated data were used in the development of an exact empirical expression to replace the functional relationship of Equation (27). Using the curve-fitting technique described in Appendix 3 , page 125, a simple and accurate expression for ($T_{\max}$) was established, Equation (31), Table 2, page 40 .

Comparing the values of $(T_{\max})$ listed in the tables of computed cycle data with the values calculated by means of Equation (31), the average deviation appears to be 0.3%, and the maximum deviation 1.4%, which is considered satisfactory.

A similar procedure applied to the exhaust temperature (T_e) produced Equations (32) and (33), Table 2, page 40, to replace the functional relationships of Equations (29) and (30), respectively. The deviations for these expressions are listed in Table 2.

TABLE 2

EMPIRICAL EQUATIONS FOR MAXIMUM TEMPERATURE AND EXHAUST TEMPERATURE.

Temperature	Empirical Equation	Equation No.	Deviations % of observed value	
			Maximum	Average
$T_{\max}$	$1875 + (0.410 p_{\max}) + (1.03r^2 - 45.3r) + (4082F - 958F^3)$	(31)	1.4	0.3
$T_r = T_{\max}/T_e$	$(-0.000394F + 0.00080) (p_{\max} + 1325) + 1.21$	(32)	0.9	0.4
T_e	$846 + (2013 - 0.290 p_{\max}) F + (0.673r^2 - 27.8r)$	(33)	1.5	0.5

SECTION 8

NOMOGRAPHIC PRESENTATION OF RESULTS

The complexity of the final empirical relationships for cycle performance quantities and cycle temperatures of interest, as laid down in Equations (24) - (26), and (31) - (33), suggests the final presentation of the data in the form of alignment charts. The application of well-known chart construction techniques⁸⁾ results in the nomographs of Figures 5 - 9. Since these nomographs are based on cycle computation procedures involving the use of thermodynamic charts, they may be used to supersede such procedures and charts.

In contrast to the conventional procedure of estimating ideal cycle performance by individual cycle constructions for each given set of operating conditions the nomographic chart completely eliminates the necessity to solve the thermodynamic cycle. As a graphical aid it has some distinct advantages when applied to the analysis under consideration. The efficiency, as well as other performance figures of the ideal cycle (or "limiting fuel-air cycle"), can be determined in a fraction of the time involved in the trial-and-error solution of the cycle. If the problem involves the assessment of a trend in cycle performance variation in response to a change in

FIG. 5 - NOMOGRAPH FOR THEORETICAL CYCLE EFFICIENCY AS A FUNCTION OF MAXIMUM CYCLE PRESSURE, COMPRESSION RATIO, AND FUEL FRACTION.

one or more of the operating conditions, the chart offers an immediate answer. Finally, the chart is well-adapted to interpolation. As discussed in Section 6, this feature is of particular importance when considering mixture strength, where the number of correct cycle computations over the practical range of diesel engine operation is limited by the availability of burned mixture charts of the appropriate F-values.

8-1 Nomograph for Cycle Efficiency.

The nomograph for cycle efficiency, Figure 5, may be used to estimate - with at least the same inherent degree of accuracy as a cycle calculation performed with the aid of thermodynamic charts - the cycle efficiency for any reasonable combination of values of compression ratio, fuel fraction, and maximum cycle pressure.

It will be noted that the chart features a total of four constant pressure lines in the range 800 to 1200 lb./in². abs. The curves for 800, 1000 and 1200 lb./in². abs. were constructed using Equations (24), (25) and (26) respectively. The 900 lb./in². abs. curve was established by interpolation methods, the errors listed in Table 1 being computed from the analysis of an additional series of cycles for this peak pressure.

In using the nomograph it should be realized that certain combinations of operating conditions are incompatible with the adoption of the mixed cycle, since they cause the P - V diagram to be distorted from the standard shape of Figure 1. As a result two restrictions are to be placed on the applicability of the performance chart to the ranges of operating conditions originally specified.

The first restriction involves those cycles in which due to a high compression ratio the compression end pressure exceeds the maximum pressure adopted for the cycle. Since the occurrence of this irregularity is independent of the mixture strength, the ranges of unsuitable compression ratios were omitted from the constant maximum pressure curves on the diagram. The limiting values of (r) may be estimated for all values of the maximum cycle pressure from the following empirical relationship :

$$r \leq \frac{P_{\max}}{60} + 5.0 \quad \dots\dots\dots(34)$$

The second restriction applies to those cycles in which the amount of injected fuel is insufficient for the maximum cycle pressure to be attained. This involves the combination of a low value of the compression ratio and mixture strength, together with a high

FIG. 6 - NOMOGRAPH FOR THEORETICAL NET WORK OUTPUT
AS A FUNCTION OF CYCLE EFFICIENCY AND FUEL FRACTION.

FIG. 7

FIG. 7 - NOMOGRAPH FOR THEORETICAL MEAN EFFECTIVE PRESSURE AS A FUNCTION OF CYCLE EFFICIENCY AND FUEL FRACTION.

specified maximum cycle pressure. Distortion of the $P - V$ diagram occurs only in the range of F - values between 0.15 and 0.20. In all cases the minimum value of (r) above which no distortion is experienced may be estimated from the following empirical relationship.

$$r \geq \left[\frac{P_{\max} + 1470}{1615F^{0.5}} \right]^2 \dots\dots\dots(35)$$

8-2 Nomograph for Cycle Output.

Once the value of the cycle efficiency (η) is determined from Figure 5, Equation (17) may be used to compute the net work output (W_n) . Alternatively, the value of (W_n) can be read from a second nomograph, Figure 6, which presents graphically the relationship between the three variables of the equation.

8-3 Nomograph for Mean Effective Pressure.

While the value of the mean effective pressure could be obtained from Figure 6 by the inclusion of a second scale on the W_n -axis having a modulus of $(1/2.68)$ times the modulus of the W_n -scale, Figure 7 has been drawn separately to avoid congestion. Again, in the latter nomograph, the value of (η) from Figure 5 is used to

FIG. 8 - NOMOGRAPH FOR MAXIMUM CYCLE TEMPERATURE AS A FUNCTION OF COMPRESSION RATIO, MAXIMUM CYCLE PRESSURE, AND FUEL FRACTION.

FIG. 9 - NOMOGRAPH FOR EXHAUST TEMPERATURE AS A FUNCTION OF MAXIMUM CYCLE PRESSURE, FUEL FRACTION, AND MAXIMUM CYCLE TEMPERATURE.

determine the value of the mean effective pressure (MEP_n).

8 -4 Nomographs for Maximum Cycle Temperature and Exhaust Temperature.

To enable the designer to estimate the limiting values of the peak cylinder temperature and the exhaust temperature the nomographs of Figures 8 and 9 have been provided. The construction of the charts is based on Equations (31) and (32), Table 2, respectively, i.e. they present graphically the magnitudes of the cycle temperatures of interest for any reasonable combination of the three cycle operating conditions, compression ratio, fuel fraction and maximum cycle pressure. Combinations of values of these parameters that cause distortion of the $P - V$ diagram are discussed in Section 8 - 1 on the Nomograph for Cycle Efficiency. Clearly, combinations of that type cannot be expected to give accurate temperature values; they are not to be considered when using the nomographs.

8 - 5 Example of the Use of the Nomographs.

As an example of the use of the charts an analysis is carried out on a 4-stroke Diesel engine, the indicator diagram of which is known to be sufficiently close to the $P - V$ diagram of Figure 1. The compression ratio is 16 : 1, and the maximum cylinder pressure is

limited to 1000 p.s.i.a. Fresh intake air reaches the cylinder at a temperature of 540°R and a pressure of 14 lb./in^2 . abs. The exhaust pressure is 15 lb./in^2 . abs. The fuel-air ratio at maximum power output is 70 per cent of the value theoretically required for complete combustion. The fuel is injected in liquid form at a temperature of 70°F , and its characteristics correspond to those of normal octene. The performance data of interest are the limiting values of the cycle efficiency, the net work output, and the net mean effective pressure at maximum engine output. In addition to these data the maximum cycle temperature and the exhaust temperature are to be estimated.

It will be noted that the intake and exhaust conditions, as well as the fuel characteristics, conform to the values adopted in the preparation of the performance charts.

Solution:

Firstly refer to the nomograph in Figure 5. The value $r = 16$ on the line for $p_{\text{max}} = 1000 \text{ p.s.i.a.}$ is joined with the value $F = 0.7$ on the fuel fraction scale by means of a straight edge. The straight line intersects the η -scale at the point $\eta = 0.543$. This is the value of the cycle efficiency, which then is used to determine the remaining cycle performance quantities. Referring to

the nomographs in Figures 6 and 7, the newly found value of (η) is located on the efficiency scale and joined with the value $F = 0.7$ on the fuel fraction scale. The straight lines, when produced, intersect the third scale at a value for the net work output of $W_n = 487$ BTU/lb. of fresh air, and a value for the net mean effective pressure of $MEP_n = 183$ lb./in².

On Figure (8) maximum cycle temperature, $T_{max.}$, is determined by first joining $r = 16$ and $p_{max.} = 1,000$ p.s.i.a. and then joining the point of intersection of this line on the α -scale with $F = 0.7$ to read on the intersection of maximum cycle temperature scale, $T_{max.} = 4360^\circ R$.

Exhaust temperature T_e is obtained from Figure (9) by producing the line joining $p_{max.} = 1,000$ p.s.i.a. and $F = 0.7$ to intersect the $\left(\frac{T_{max.}}{T_e}\right)$ -axis at 2.43 and then joining this point to $T_{max.} = 4360^\circ R$ to give on the exhaust temperature axis, $T_e = 1790^\circ R$.

Thus it is seen that, by using the nomographs, an examination of the effects of the various operating parameters on ideal engine performance and ideal cycle temperature is easily made. Such effects are useful for making comparisons with and predicting trends in actual engines as is shown in the next section.

SECTION 9

EXPERIMENTAL INVESTIGATION

A series of experiments was conducted with a view of examining the effects of variations in the ~~maximum~~ cycle pressure and fuel-air ratio on thermal efficiency, indicated mean effective pressure, and mean exhaust temperature.

9-1 Description of Apparatus.

The research engine used was a Ricardo B6/S single-cylinder, four-stroke, variable-compression engine of 3.000 inches bore and 4.375 inches stroke. The normal speed range of the unit is 1250-3000 r.p.m.

Although originally designed for general research into spark-ignition problems, the engine may be converted readily to compression-ignition operation by changing the cylinder head and by replacing the magneto by a fuel injection pump. In the compression-ignition version the engine features a Ricardo Comet MK. II Compression Swirl combustion chamber, as employed in many commercial diesel engines. Variable compression is achieved by raising or lowering the cylinder relative to the crankshaft. The movement of the cylinder barrel is measured by means of a micrometer, and the compression ratio corresponding to a given micrometer reading read off a calibration

curve supplied with the unit. The compression ratio, as well as the timing of the injection-point can be varied whilst the engine is running.

Engine output is measured by an electric dynamometer of the swinging field type. Simple switchgear enables the dynamometer to be operated as a motor either to start the engine, or to motor it for the determination of friction and pumping losses.

Special external heat exchangers are incorporated in the engine design to enable controlled cooling of lubricating oil and circulating water. In addition to these an internal electric heater is fitted in the sump, allowing the oil temperature to be controlled at any required value. These provisions ensure identical engine operating conditions during normal running and motoring.

Fuel flow rate was determined by measuring the time for the engine to consume 50 cc. of fuel in a calibrated pipette.

Air consumption was measured by means of a B.S. 1042 orifice plate inserted in the main inlet to a large expansion chamber, which acted as a smoothing capacity. This allowed the orifice to function under steady flow conditions. It has been shown ⁹⁾ that when an air box of at least 1550 cylinder volumes is utilized on a single cylinder engine, there is no appreciable pressure fluctuation

upstream from the tank regardless of the frequency of the oscillations. Since the actual volume of the tank represented approximately 1600 cylinder volumes, no correction was required for possible changes in the discharge coefficients of the metering orifice. The differential pressure across the orifice plate was determined by means of an Askania micromanometer.

The fuel-air ratio was regulated by varying the effective stroke of the Bosch fuel pump.

Two possible errors in the accurate assessment of the fuel-air ratio of the test engine need be mentioned. Firstly, the valve timing of the engine allows for the inlet valve to open 8° before TDC and the exhaust valve to close 7° after TDC. The resulting valve overlap of 15° , while enabling effective replacement of the cylinder charge, may tend to cause a short circuiting of portion of the induced fresh air into the exhaust manifold, thus seriously impairing the accurate measurement of air consumption. Secondly, when the engine operates close to the smoke limit, there is a possibility of unburned fuel escaping into the exhaust.

Concerning the influence of valve overlap, it was found by the designer of the engine ¹⁰⁾ that overlaps of this magnitude do not have any measurable effect on air consumption. Also, to ensure that

all injected fuel was indeed burned during the expansion stroke, a range of fuel-air ratios was selected that allowed the engine to operate with a clear exhaust at all times.

Thus the mixture strength determined from the metered fuel and air flows is the same as the actual mixture strength in terms of the air retained in the cylinder and the fuel burned in it.

Exhaust gas appearance near the smoke limit was judged by visual examination; a small branch pipe and gate valve are fitted to the exhaust pipe, and by opening the gate valve momentarily the exhaust gas may be viewed against a white tile background.

All temperatures were measured by mercury-in-glass thermometers. To determine the exhaust gas temperature a special well was provided in the exhaust pipe.

Maximum cylinder pressures were recorded by means of a Dobbie-McInnes "Farnboro" electric indicator.

In addition to a tachometer driven from the dynamometer shaft, a hand-type Smith's tachometer was used for measuring engine speed.

Views of the apparatus are given in Plates 1 and 2,

9-2 Calibrations.

Compression ratio. The Ricardo engine is of the variable compression

PLATE I

PLATE 2

FIGURE 10 - RICARDO E6/5 RESEARCH ENGINE. COMPRESSION RATIO ADJUSTMENT FOR ENGINE FITTED WITH COMPRESSION-IGNITION CYLINDER HEAD WITH COMET MK. II COMBUSTION CHAMBER.

type, a feature mainly provided for research into spark-ignition problems. The designer of the engine has emphasized that the performance of any compression-ignition engine depends to a large extent on the form and proportions of the combustion chamber.¹⁰⁾ In this respect the limitations of the unit for research on compression-ignition cycle behaviour are recognized and accepted. It is believed, however, that the engine can be utilized to advantage for the purpose of the present study. The shape of the combustion chamber is not taken into consideration in the analysis of the theoretical cycle, and hence, this parameter is not to be altered during the experimental investigations. Unfortunately, this also implies that the compression ratio must be kept at a fixed value, thus limiting the scope of the experimental work.

The constant value of the compression ratio adopted for the tests was $r = 20$. Since for reasons given above the manufacturer did not supply a calibration curve for the compression ratio adjustment of the compression-ignition version of the engine, this calibration had to be carried out in the laboratory in order to determine the correct cylinder barrel setting corresponding to the above value. The method used was by direct measurement of the clearance volume in the compression-ignition head. The resulting calibration curve is presented in Figure 10.

Maximum cylinder pressure. The indicator springs of the "Farnboro" electric indicator were calibrated by producing constant pressure lines at 100 lb./in². intervals on the paper secured to the drum. The usual spring-rate curve was plotted from the parallel lines recorded.

The maximum cylinder pressure was considered to be an independent variable in the present work, its value ranging from 800 - 1,200 lb./in². abs. In a preliminary series of test runs the influence of the injection timing on the maximum cylinder pressure was investigated. An injection point of 37° before TDC was found to give the lowest maximum cylinder pressure compatible with reasonable overall performance over the speed range. At this setting the combustion delay is at a minimum, since the delay period occurs near TDC when the cylinder pressure and temperature are high. Advancement of the injection timing will cause a longer delay period, during which a relatively large quantity of fuel is injected. This results in a high maximum pressure, since the combustion of a large portion of the injected fuel will occur at constant volume.

Although there was a wide scatter of maximum cylinder pressures for any given combination of injection timing and engine load, average readings indicated a maximum cylinder pressure of 800 lb./in².abs.

FIGURE II

RICARDO E6/S RESEARCH ENGINE

APPROXIMATE MIXTURE STRENGTH ADJUSTMENT

for an injection timing of 37° before TDC, increasing to 1200 lb./in². abs. at 47° before TDC. It may be assumed that maximum cylinder pressure varies approximately linearly with injection advance¹¹); engine loading appeared to be of negligible influence in this respect.

Fuel-air ratio. A number of preliminary test runs was conducted to establish the relationship between the setting of the control screw on the fuel pump rack and the fuel-air ratio at a given engine speed. Although an approximate correlation was found, Figure 11, fuel consumption appeared to be erratic - particularly at the low fuel-air ratios - and in the actual performance tests the positive displacement method of fuel metering was resorted to. However, the diagram is useful in obtaining approximate settings, thus streamlining the procedure of arriving at the correct F-value for a certain test.

For a range of metered air consumption values, a second, more accurate, chart was constructed as an aid in the correct adjustment of mixture strength. This chart, Figure 12, depicts the time required to consume 50 cc. of fuel for a range of fuel fraction values $F = 0.16 - 1.20$. The range of air consumption values on the chart corresponds to the normal operating range of the Ricardo test engine.

The specification for the distillate fuel oil used during the tests was as follows :

Distillate Fuel Oil

B.S. Specification 2869-1957 Class A

Specific Gravity S.G. = 0.818

Stoichiometric fuel-air ratio F_c = 0.0667

Calorific values, high Q_c = 18,530 BTU/lb.

low Q_c = 18,330 BTU/lb.

The derivation of the relationship between the metered fuel and air consumptions and mixture strength is given in Appendix 4.2, page 132 .

Finally, a graph was constructed of the flow of air at standard conditions versus the differential pressure across the metering orifice, Figure 13. This chart, together with two additional plots for the appropriate multipliers for ambient air temperature and pressure, provided a convenient means of determining the air consumption of the engine. The orifice plate calculation underlying the chart is given in Appendix 4.3, page 133 .

9-3 Experimental Procedure.

A fixed compression ratio of 20 was selected as being a typical

value for four-stroke, small bore, pre-chamber, compression-ignition engines.

The criterion in the selection of engine speed is the corresponding piston speed and its ensuing influence on turbulence. As can be seen in Figure 11, high values of these parameters increase the quantity of fuel that can be burned efficiently and, hence, increase the maximum engine output. Since turbulence is a parameter not considered in the idealized cycle analysis, its influence must be minimized during the experiments by keeping the engine speed constant. Current practice entails mean piston speeds for four-stroke compression-ignition engines well over 1500 feet per minute. Accordingly, a speed of 2000 R.P.M. was selected as being representative of modern high-speed engines.

Referring once more to the mixture strength adjustment chart, Figure 11, the selection of an engine speed of 2000 R.P.M. enables the experiments to be carried out over an approximate range of $F = 0.35 - 0.75$. Below this range the engine is operating at very low load and engine behaviour tends to become erratic; beyond the top of the range occasional traces of smoke begin to appear in the exhaust gas. In this regard the definition for maximum useful power output of compression-ignition engines is by no means clear ¹²). However, the value of (F) , the mixture strength, rarely exceeds

$F = 0.7$. In most cases operation above $F = 0.6$ is considered to be overload operation, and best economy usually occurs near $F = 0.5$ ⁴).

Five series of test runs were conducted over the range of fuel-air ratios. By careful adjustment of the fuel pump control and by allowing ample time for engine conditions to stabilize after a load change the actual F - values used ranged from 0.35-0.75 in increments of 0.10.

Control of the maximum cylinder pressure by means of adjusting the injection timing proved to be rather inaccurate. However, in each series of tests a number of five different maximum pressures were obtained, that were suitably spaced over the range $p_{\max.} = 800 - 1200 \text{ lb./in}^2 \text{ abs.}$

The brake mean effective pressure was determined by means of the swinging-arm type dynamometer. Since for comparison of the experimental results with the theoretical data the indicated mean effective pressure is of interest, a knowledge of the mechanical efficiency of the engine was essential. The accurate measurement of mechanical efficiency for a high-speed engine is not a simple matter ^{10, 12}). Many methods have been used, of which the motoring of the engine directly after a running test is of reasonable accuracy. The latter procedure was adopted in the present

experimental work, great care being taken to maintain constant cooling water and lubricating oil temperatures. In spite of this the values obtained for friction mean effective pressure showed a fairly wide scatter, and - recognizing the inherent inaccuracy of the method - a mean value was adopted in the calculations.

As expected, the measurement of the exhaust temperature by means of the mercury-in-glass thermometer inserted in the exhaust duct proved to be very sensitive to the position of the instrument in the duct. Whilst great care was taken to maintain the same depth of immersion for each measurement, the readings are indicative only of certain trends in exhaust temperature variations. It is realized that measurement of mean exhaust temperature by this means leaves much to be desired ¹³), although it is only changes of temperature with operating parameters that are of interest here.

Due to lack of suitable instrumentation no attempt was made to determine the maximum cycle temperature.

All experimental results of interest were compiled in Appendix 5, Table 52-56, page 139 - 143.

The calculated values of engine performance quantities are listed in Table 57-61, page 144 - 148 . The derivation of the fundamental

FIG. 14 - VARIATION OF MEAN EFFECTIVE PRESSURE WITH MAXIMUM CYCLE PRESSURE FOR SELECTED VALUES OF FUEL FRACTION.

FIG. 15 - MEAN EFFECTIVE PRESSURE VERSUS FUEL FRACTION
FOR A MAXIMUM CYCLE PRESSURE OF
1000 LB./IN² ABS.

FIG. 16 - VARIATION OF THERMAL EFFICIENCY WITH MAXIMUM CYCLE PRESSURE FOR SELECTED VALUES OF FUEL FRACTION.

FIG. 17 - THERMAL EFFICIENCY VERSUS FUEL FRACTION FOR A MAXIMUM CYCLE PRESSURE OF 1000 LB./IN² ABS.

FIG. 18 - EXHAUST TEMPERATURE VERSUS MAXIMUM CYCLE PRESSURE FOR A RANGE OF FUEL FRACTION VALUES.

equations of engine performance is presented in Appendix 4-1, page 130 .

Subsequently, the experimental values of indicated mean effective pressure, thermal efficiency and exhaust temperature were plotted against maximum cycle pressure for a range of fuel-air ratios in Figures 14, 16, and 18. In Figures 15, and 17 curves of indicated mean effective pressure and thermal efficiency are plotted against fuel fraction for a selected value of maximum cycle pressure of 1,000 p.s.i.a.; these curves were computed from Figures 14 and 16 since it was easy to maintain a constant fuel-air ratio while experimenting, but difficult to maintain a constant maximum cycle pressure.

9-4 Comparison between Experimental and Theoretical Results.

In Section 5 fixed values were adopted for the three operating parameters, intake pressure, temperature of fresh intake air, and exhaust pressure, of the theoretical cycles. These values are considered sufficiently close to the corresponding values in the experiments to allow a comparison to be drawn between the theoretical and experimental results.

Adopting the same compression ratio, $r = 20$, as for the test engine, the nomographs of Figures 5 and 7 are used to compute a number of values of the cycle efficiency and theoretical mean

effective pressure; curves are plotted of these performance data against maximum cycle pressure and fuel fraction. Figures 14 to 17 show the theoretical curves as well as the corresponding experimental curves plotted from the data in Appendix 5. Heat transfer to or from the working substance has not been taken into account in the theoretical analysis causing the theoretical curves to be somewhat displaced from the experimental. However, it is interesting to observe the similar trends.

Mean Effective Pressures.

In Figure 14 theoretical mean effective pressure increases steadily with maximum cycle pressure at constant fuel air ratios. This is to be expected since an increase in maximum pressure enables more fuel to be burned at constant volume, leading to a higher efficiency as illustrated in Figure 16. Mean effective pressure is directly proportional to efficiency for a constant heat release, and it also must increase with maximum cycle pressure. Experimental curves of mean effective pressure and efficiency, Figures 14 and 16 show at first similar rates of increase to the theoretical curves, as maximum cycle pressure is increased. They then reach a peak value, depending on the fuel-air ratio and decrease steadily with further increases in maximum cycle pressure. This decrease can be attributed mainly to the increased heat loss associated with constant

volume combustion outweighing the gain in efficiency ideally attainable; the effect is far more predominant at low mixture strength because of the corresponding greater proportion of fuel burnt at constant volume than at constant pressure.

Curves of mean effective pressure and efficiency are plotted in Figures 15 and 17 against fuel fraction from Figures 14 and 16 for a constant maximum cycle pressure of $1000 \text{ lb./in.}^2 \text{ abs.}$ The experimental curve of mean effective pressure, Figure 15, reveals at first almost the same rate of increase with increased fuel fraction, but departs rapidly from the theoretical curve at higher values. As shown by the theoretical curve, mean effective pressure ideally should increase almost linearly with fuel fraction. However, in an actual engine it is likely that some of the fuel is unburnt at the point of "cut-off" (point (3) on theoretical diagram, Figure (1) and that combustion may continue during portion of the expansion stroke and perhaps, in some instances, may not be complete when the exhaust valve opens ¹⁴). This effect would be more likely to occur at the higher mixture strengths, causing a reduction in mean effective pressure.

Efficiencies.

A similar effect to the above is shown on the efficiency curves,

FIG. 19 - THEORETICAL MAXIMUM CYCLE TEMPERATURE VERSUS MAXIMUM CYCLE PRESSURE FOR A RANGE OF FUEL FRACTION VALUES.

Figure 16, since for a constant maximum cycle pressure, thermal efficiency is directly proportional to mean effective pressure and inversely proportional to fuel fraction. Theoretical curves computed from data published by others ¹⁴⁾ are included in Figures 15 and 17 for comparison. It should be noted that for these curves a compression ratio of 15 was used instead of 20 as in the present work. Agreement between the two curves is very close, particularly when the open scale used for efficiency is taken into account.

Temperatures.

In Figure 18 theoretical curves of exhaust temperature versus maximum cycle pressure, computed from the nomographs, Figures 8 and 9, for various fuel-air ratios are compared with experimental curves.

Owing to a greater heat release at constant volume, an increase in maximum cycle pressure at constant fuel-air ratio tends to increase the maximum cycle temperature as shown in Figure 19. This tends to increase the exhaust temperature, but the resulting higher expansion ratio with its associated high temperature drop, predominates, the net effect being a reduction in exhaust temperature. Although the experimental values of exhaust temperature are somewhat lower than the theoretical, due to heat transfer and possibly the means adopted

FIG. 20 - EXHAUST TEMPERATURE VERSUS FUEL FRACTION
FOR A MAXIMUM CYCLE PRESSURE OF 1000 LB./IN² ABS.

FIG. 21 - VARIATION OF THEORETICAL MAXIMUM CYCLE TEMPERATURE VERSUS FUEL FRACTION FOR A MAXIMUM CYCLE PRESSURE OF 1000 LB./IN².ABS.

for temperature measurement, the rate of change appears much the same in both cases and is almost independent of fuel-air ratio.

In Figure 20, exhaust temperature is plotted against fuel fraction from Figure 18, for a constant maximum cycle pressure of 1,000 lb/in² abs. Again the rate of change is comparable between theoretical and experimental curves, higher temperatures being experienced with increased fuel quantities, because of the corresponding higher maximum temperatures, Figure 21, and reduced expansion ratios associated with increased constant pressure combustion.

No suitable equipment was available to assess accurately the maximum cycle temperatures. However, it is reasonable to assume from Figures 14 to 20, that experimental curves would show similar relationships to the theoretical curves of Figure 19 plotted from the nomograph, Figure 8. In Figure 21, a curve of maximum cycle temperature versus fuel fraction, for a maximum cycle pressure of 1,000 lb./in² abs., shows close agreement with values published elsewhere ¹⁴).

FIG. 22 - RATIO OF THERMAL TO THEORETICAL EFFICIENCY VERSUS FUEL FRACTION.

FIG. 23

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FIG. 23 - RATIO OF INDICATED TO THEORETICAL MEAN EFFECTIVE PRESSURE VERSUS FUEL FRACTION.

SECTION 10

PREDICTING PERFORMANCE OF A DIESEL ENGINE.

In order that an estimate of the thermal efficiency of a compression-ignition engine may be made, a graph of the ratio of actual thermal efficiency to theoretical efficiency versus fuel fraction is plotted from Figure 16 in Figure 22. Values for individual points on the graph are given in Appendix 6, Table 62-65, page 149 - 150. Although the graph is based on a single cylinder engine having a compression ratio of 20, a fresh air intake temperature of 540°R , and intake and exhaust manifold pressures of 14 and 15 lb./in²abs., it is reasonable to assume that, together with the nomograph of Figure 5, it would be useful in estimating the thermal efficiency of other compression-ignition engines of comparable design, but controlled by a different set of parameters. Maximum cycle pressure is seen to have a greater influence on the efficiency ratio at the lower fuel-air ratios because of the greater heat loss associated with the increased proportion of constant volume combustion.

A graph of the ratio of actual indicated to theoretical mean effective pressure against fuel fraction is plotted in Figure 23. These curves are plotted from and are of similar shape to those of

Figure 22. Theoretically, the ratio of actual indicated mean effective pressure to the theoretical mean effective pressure should equal the efficiency ratio. However, in the experiments the fuel used differed appreciably from octene, C_8H_{16} , which was the fuel selected for the preparation of the thermodynamic charts. This necessitates the introduction of a factor of proportionality. From Equations (17) and (20) it can be seen that

$$(MEP)_{th} = K_{th} \times F_{th} \times M_{th} \dots\dots\dots(36)$$

where $K_{th} =$ a constant

Likewise, for the experiments

$$(IMEP)_a = K_a \times F_a \times M_a \dots\dots\dots(37)$$

Combining Equations (36) and (37) and introducing the ratios of the parameters, actual to theoretical,

$$(MEP)_r = K_r \times \frac{F_a}{F_{th}} \times \eta_r \dots\dots\dots(38)$$

For $\frac{F_a}{F_{th}} = 1$

$$\text{MEP}_r = K_r \times \eta_r \quad \dots\dots\dots(39)$$

where $K_r = 0.899$, a constant.

The derivation of the factor of proportionality (K_r) is given in Appendix 4.4, page 136 .

Figures 22 and 23 may be used together with the nomographs of Figures 5 and 7 for estimating indicated mean effective pressure. Figure 23, is, in fact, a plot of what is sometimes referred to as "diagram factor" by the engine designer.

Since the curves presented in Figures 22 and 23 derive from the experimental values of the thermal efficiency as plotted in Figure 16, their accuracy depends to a great extent on the reproducibility of the curves drawn through the plotted points in the latter diagram. In drawing these curves, the main consideration was to obtain a fair distribution; i.e., the total positive deviations are to equal the total negative deviations consistent with the curves being smooth, and the change in shape being gradual. As a check on the final result, the above data based on Figure 16 were compared to data obtained in a similar manner from Figure 14. For ease of comparison all results were compiled in Appendix 6, Table 62-65, page 149 . In general there is good agreement between the two sets of values of the mean effective pressure ratios; the

average deviation without regard to sign amounts to 0.8%,
whilst the maximum deviations are +1.9% and -1.6%.

SECTION 11

SUMMARY OF CONCLUSIONS

1. Theoretical performance equations used for the analysis of idealized compression-ignition engines are based on the thermodynamic properties of the working fluid at salient points in the cycle. The determination of such properties with the aid of thermodynamic charts requires a complete solution of the cycle; i.e., the cyclical variation of the state of the working fluid must be computed by iteration over the cycle.
2. The introduction of empirical relationships expressing cycle performance in terms of the operating conditions of the cycle eliminates the need for complete cycle solutions, except in those instances where knowledge of the cyclical properties of the working fluid is essential from the design point of view.
3. The influence of maximum cycle pressure, compression ratio, and mixture strength on cycle performance has been examined by empirical relationships developed from thermodynamic charts. Although some data have been included on the effects of intake pressure, fresh air intake temperature, and

exhaust pressure, an extension of this work would be of interest.

4. In addition to the time savings experienced, the method of empirical relationships improves the accuracy of computing performance data for cycles having a mixture strength that requires interpolation between existing thermodynamic charts.
5. Limiting values of the peak cycle temperature and exhaust temperature are of particular interest to the design engineer. The variation of these temperatures in terms of the engine operating conditions are examined in a similar manner to those of engine performance.
6. Nomographs constructed from the empirical relationships find favour since they allow immediate assessment of a trend in cycle performance variation in response to a change in one or more of the operating conditions.
7. Plots prepared from the nomographs of theoretical cycle efficiency and theoretical mean effective pressure in terms of compression ratio, mixture strength, and maximum cycle pressure, compare favourable with those obtained by others.
8. Comparison of the theoretical curves for cycle efficiency and theoretical mean effective pressure with curves

representing experimental performance data reveals similar trends for corresponding variations in operating conditions.

9. Graphs of the ratio of thermal to theoretical efficiency, and indicated to theoretical mean effective pressure against mixture strength are useful to the designer to make reasonable predictions of performance figures to be expected in practice. These curves are to be used with caution since they apply only to a four-stroke single cylinder compression-ignition engine fitted with a "Comet" pre-chamber combustion chamber and operating at a compression ratio of 20 : 1. Graphs from similar series of tests on a wide variety of diesel engines would be of interest.

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APPENDIX 1

CYCLE COMPUTATIONS

1-1 Cyclical Thermodynamic Properties

A total of four Series of Cycle Analyses were carried out, the details of which are given in Section 6 , page 30 . All data on the cyclical thermodynamic properties of the working fluid are compiled in this Appendix, Tables 3 - 25 , page 75 - 97 . The suffixes used for the symbols in the tables correspond to the numbers of the transition points between the cycle processes, as labelled in the P-V diagram of Figure 1 .

Unless indicated otherwise, the magnitude of the fixed cycle operating conditions used in the cycle solutions are :

Intake pressure	p_1	=	14.0 lb./in ² .abs.
Fresh air temperature	T_{fr}	=	540 °R.
Exhaust pressure	p_e	=	15.0 lb./in ² .abs.

The values of the temperature (T_1) at the beginning of compression, as listed in the tables, are assumed values. Referring to the discussion of cycle solution techniques in Section 3-3 , the computed values of these temperatures must agree closely with the initial estimates. The computed values are included in Tables 26 - 48 in this Appendix.

Table 3 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction

$F = 0.15$

r	14	16	18	20	22
T_1	554	551	550		
S_1	3.10	3.06	3.05		
V_1	424	421	420		
E_1	250	237	230		
V_2	30.3	26.3	23.4		
p_2	544	658	772	$> p_3$	$> p_3$
T_2	1523	1591	1657		
E_2	5351	5732	6109		
H_3	15325	15116	15063		
T_3	2355	2330	2323		
V_3	32.1	31.8	31.6		
S_3	5.99	5.90	5.87		
V_4	421	418	417		
p_4	25.0	24.8	24.6		
T_4	973	959	953		
E_4	2420	2340	2320		
T_5	843	835	830		

Table 4 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.5$

r	14	16	18	20	22
E_1	328	307	288		
V_1	438	434	430		
T_1	570	566	562		
S_1	3.30	3.25	3.20		
E_2	5564	5901	6302		
V_2	31.3	27.1	23.9		
T_2	1562	1620	1690		
p_2	542	652	766	$>p_3$	$>p_3$
e_3	728	721	717		
v_3	1.75	1.74	1.73		
T_3	3670	3646	3633		
s_3	0.2604	0.2580	0.2571		
e_4	294	291	290		
v_4	15.7	15.5	15.4		
T_4	1958	1947	1942		
p_4	47.8	48.0	48.5		
$e_{4'}$	187	184	182		
$v_{4'}$	38.1	38.0	37.8		
$T_{4'}$	1484	1471	1465		
e_6	181	179	177		
v_6	40.0	39.5	39.2		
T_6	1461	1451	1445		

Table 5 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.8$

r	14	16	18	20	22
E_1	343	338	329		
V_1	444	439	435		
T_1	578	572	568		
S_1	3.40	3.33	3.28		
E_2	5678	5993	6380		
V_2	31.8	27.4	24.2		
T_2	1581	1635	1703		
p_2	543	649	770	$> p_3$	$> p_3$
e_3	1053	1043	1041		
v_3	2.23	2.22	2.21		
T_3	4532	4507	4503		
s_3	0.4180	0.4155	0.4147		
e_4	528	523	523		
v_4	16.2	16.0	15.9		
T_4	2812	2792	2789		
p_4	67.8	67.8	68.4		
$e_{4'}$	324	320	319		
$v_{4'}$	52.6	52.3	52.0		
$T_{4'}$	2018	2003	2000		
e_6	316	312	311		
v_6	55.5	55.0	55.0		
T_6	1986	1972	1965		

Table 6 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 1.0$

r	14	16	18	20	22
E_1	395	364	338		
V_1	448	442	439		
T_1	583	577	572		
S_1	3.46	3.38	3.33		
E_2	5737	6086	6405		
V_2	32.0	27.6	24.4		
T_2	1592	1652	1707		
p_2	540	651	770	$> p_3$	$> p_3$
e_3	1281	1271	1267		
v_3	2.47	2.46	2.45		
T_3	4890	4865	4856		
s_3	0.4772	0.4748	0.4738		
e_4	702	697	695		
v_4	16.5	16.3	16.2		
T_4	3422	3324	3319		
p_4	80.0	80.6	80.9		
$e_{4'}$	425	421	419		
$v_{4'}$	62.4	65.4	61.8		
$T_{4'}$	2360	2343	2336		
e_6	416	412	410		
v_6	65.8	62.1	65.4		
T_6	2326	2310	2302		

Table 7 - Maximum cycle pressure $p_3 = 900 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction

$F = 0.15$

r	14	16	18	20	22
T_1		551	550	548	
S_1		3.06	3.05	3.03	
V_1		421	420	419	
E_1		237	230	220	
V_2		26.3	23.4	21.0	$> p_3$
p_2		658	772	900	
T_2		1591	1657	1715	
E_2		5732	6109	6444	
H_3		15598	15493	15420	
T_3		2387	2376	2366	
V_3	$< V_2$	29.0	28.8	28.7	
S_3		5.86	5.83	5.79	
V_4		418	417	416	
p_4		24.5	24.5	24.4	
T_4		951	947	942	
E_4		2302	2280	2255	
T_5		830	825	822	

Table 8 - Maximum cycle pressure $p_3 = 900 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.5$

r	14	16	18	20	22
E_1	328	307	288	278	
V_1	438	434	430	429	
T_1	570	566	562	560	
s_1	3.30	3.25	3.20	3.17	
E_2	5564	5901	6302	6625	
V_2	31.3	27.1	23.9	21.5	
T_2	1562	1620	1683	1745	
p_2	542	652	766	895	$> p_3$
e_3	745	734	730	728	
v_3	1.58	1.56	1.56	1.56	
T_3	3732	3692	3678	3671	
s_3	0.2575	0.2540	0.2528	0.2520	
e_4	288	282	282	281	
v_4	15.7	15.5	15.4	15.4	
T_4	1936	1909	1906	1902	
p_4	47.4	47.0	47.7	47.6	
$e_{4'}$	183	179	178	177	
$v_{4'}$	37.8	37.0	37.0	37.0	
$T_{4'}$	1470	1452	1447	1442	
e_6	178	175	173	172	
v_6	39.5	38.7	38.7	38.6	
T_6	1449	1430	1425	1421	

Table 9 - Maximum cycle pressure $p_3 = 900 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.8$

r	14	16	18	20	22
E_1	343	338	319	303	
V_1	444	439	435	433	
T_1	578	572	568	565	
S_1	3.40	3.33	3.28	3.23	
E_2	5678	5993	6380	6704	
V_2	31.7	27.4	24.2	21.6	
T_2	1581	1635	1703	1758	
p_2	543	649	770	895	$> p_3$
e_3	1070	1058	1053	1051	
v_3	2.00	1.99	1.98	1.98	
T_3	4583	4550	4537	4530	
s_3	0.4140	0.4110	0.4095	0.4089	
e_4	518	511	508	508	
v_4	16.2	16.0	15.9	15.8	
T_4	2768	2745	2737	2733	
p_4	66.7	67.0	67.3	67.2	
$e_{4'}$	318	313	310	310	
$v_{4'}$	52.0	51.5	51.4	51.3	
$T_{4'}$	1995	1973	1963	1960	
e_6	310	305	303	302	
v_6	54.6	54.3	54.1	53.9	
T_6	1964	1942	1933	1930	

Table 10 - Maximum cycle pressure $p_3 = 900 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 1.0$

r	14	16	18	20	22
E_1	395	364	338	319	
V_1	448	442	439	435	
T_1	583	577	572	568	
S_1	3.46	3.38	3.33	3.28	
E_2	5737	6086	6405	6760	
V_2	32.0	27.6	24.4	21.8	
T_2	1592	1652	1707	1767	
p_2	540	651	770	895	$> p_3$
e_3	1298	1287	1280	1278	
v_3	2.21	2.20	2.19	2.19	
T_3	4929	4907	4892	4887	
s_3	0.4725	0.4697	0.4681	0.4676	
e_4	687	681	677	677	
v_4	16.5	16.3	16.2	16.1	
T_4	3290	3273	3261	3260	
p_4	78.8	79.4	79.8	80.4	
$e_{4'}$	417	412	408	407	
$v_{4'}$	61.8	61.3	61.1	61.0	
$T_{4'}$	2328	2309	2298	2294	
e_6	408	402	399	398	
v_6	65.0	64.5	64.3	64.3	
T_6	2295	2275	2264	2261	

Table 11 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.15$

r	14	16	18	20	22
T_1		551	550	548	547
S_1		3.06	3.05	3.03	3.02
V_1		421	420	419	419
E_1		237	230	220	214
V_2		26.3	23.3	21.0	19.1
p_2		658	771	900	1015
T_2		1591	1657	1715	1768
E_2		5732	6114	6444	6759
H_3		16081	15910	15805	15777
T_3		2442	2423	2412	2406
V_3	$< V_2$	26.8	26.6	26.4	26.3
S_3		5.85	5.79	5.75	5.73
V_4		418	417	416	416
p_4		24.4	24.4	24.2	24.2
T_4		951	942	933	932
E_4		2292	2260	2215	2205
T_5		829	822	817	815

Table 12 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.5$

r	14	16	18	20	22
E_1	318	300	288	278	
V_1	432	431	430	429	
T_1	567	564	562	560	
S_1	3.23	3.22	3.20	3.18	
E_2	5461	5897	6302	6625	
V_2	30.9	27.0	23.9	21.5	
T_2	1544	1620	1690	1745	
p_2	543	651	766	895	$> p_3$
e_3	756	748	742	735	
v_3	1.44	1.43	1.42	1.41	
T_3	3770	3742	3719	3700	
s_3	0.2536	0.2512	0.2492	0.2475	
e_4	283	278	275	274	
v_4	15.5	15.5	15.4	15.4	
T_4	1909	1890	1878	1865	
p_4	46.9	46.7	47.0	46.6	
$e_{4'}$	179	176	175	173	
$v_{4'}$	37.0	37.0	36.5	36.3	
$T_{4'}$	1452	1435	1431	1421	
e_6	175	172	170	168	
v_6	38.7	38.6	38.3	38.1	
T_6	1430	1416	1408	1403	

Table 13 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction

$F = 0.8$

r	14	16	18	20	22
E_1	358	338	319	303	
V_1	442	439	435	433	
T_1	576	572	568	565	
S_1	3.38	3.33	3.28	3.23	
E_2	5625	5993	6380	6704	
V_2	31.6	27.4	24.2	21.6	
T_2	1572	1635	1703	1758	
p_2	545	649	770	895	$> p_3$
e_3	1085	1073	1066	1062	
v_3	1.82	1.81	1.80	1.79	
T_3	4627	4594	4575	4564	
s_3	0.4102	0.4071	0.4052	0.4042	
e_4	507	500	496	494	
v_4	16.1	16.0	15.9	15.8	
T_4	2729	2706	2690	2685	
p_4	65.3	65.2	65.5	65.7	
$e_{4'}$	312	307	304	302	
$v_{4'}$	51.2	50.9	50.6	50.6	
$T_{4'}$	1970	1948	1935	1930	
e_6	304	300	297	296	
v_6	54.3	53.7	54.0	52.9	
T_6	1938	1917	1909	1903	

Table 14 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 1.0$

r	14	16	18	20	22
E_1	395	364	338	319	
V_1	448	442	439	435	
T_1	583	577	572	568	
S_1	3.46	3.38	3.33	3.28	
E_2	5737	6086	6405	6760	
V_2	32.0	27.6	24.4	21.8	
T_2	1592	1652	1707	1767	
p_2	540	651	762	895	$> p_3$
e_3	1316	1302	1293	1289	
v_3	2.00	1.99	1.98	1.98	
T_3	4973	4942	4927	4919	
s_3	0.4688	0.4655	0.4635	0.4626	
e_4	674	665	662	661	
v_4	16.5	16.3	16.2	16.1	
T_4	3250	3221	3207	3206	
p_4	77.7	77.5	78.0	78.7	
$e_{4'}$	410	404	400	399	
$v_{4'}$	61.0	60.5	60.2	59.9	
$T_{4'}$	2302	2281	2267	2264	
e_6	401	395	392	390	
v_6	64.4	63.8	63.4	63.0	
T_6	2269	2248	2235	2230	

Table 15 - Maximum cycle pressure $p_3 = 1200 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.15$

r	14	16	18	20	22
T_1				548	547
S_1				3.03	3.02
V_1				419	418
E_1				220	214
V_2				21.0	19.1
p_2				900	1018
T_2				1715	1770
E_2				6444	6764
H_3				16575	16473
T_3				2501	2489
V_3	$< V_2$	$< V_2$	$< V_2$	22.9	22.8
S_3				5.69	5.65
V_4				416	415
p_4				24.0	23.7
T_4				924	915
E_4				2160	2120
T_5				810	805

Table 16 - Maximum cycle pressure $p_3 = 1200 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.5$

r	14	16	18	20	22
E_1	328	307	288	278	268
V_1	438	434	430	429	427
T_1	570	566	562	560	558
S_1	3.30	3.25	3.20	3.18	3.15
E_2	5564	5901	6302	6625	6933
V_2	31.3	27.1	23.9	21.5	19.4
T_2	1562	1620	1690	1745	1798
p_2	542	652	766	895	1015
e_3	793	777	768	762	757
v_3	1.24	1.22	1.21	1.20	1.20
T_3	3904	3845	3824	3789	3775
s_3	0.2522	0.2472	0.2444	0.2425	0.2414
e_4	279	271	266	263	262
v_4	15.7	15.5	15.4	15.4	15.3
T_4	1893	1856	1839	1825	1822
p_4	46.6	46.0	45.3	45.4	46.0
$e_{4'}$	177	172	169	168	166
$v_{4'}$	37.0	36.5	36.0	35.7	35.7
$T_{4'}$	1442	1418	1405	1399	1389
e_6	173	168	165	163	161
v_6	38.6	38.0	36.7	37.4	37.2
T_6	1421	1397	1383	1375	1369

Table 17 - Maximum cycle pressure $p_3 = 1200 \text{ lb./in.}^2 \text{ abs.}$

Fuel fraction $F = 0.8$

r	14	16	18	20	22
E_1	343	338	319	303	296
V_1	444	439	435	433	431
T_1	578	572	568	565	563
S_1	3.40	3.33	3.28	3.23	3.22
E_2	5678	5993	6380	6704	7030
V_2	31.8	27.4	24.2	21.6	19.6
T_2	1581	1635	1703	1758	1814
p_2	543	649	770	895	1016
e_3	1121	1102	1092	1085	1081
v_3	1.55	1.53	1.52	1.52	1.52
T_3	4729	4678	4653	4633	4625
s_3	0.4063	0.4014	0.3988	0.3970	0.3961
e_4	495	485	479	476	475
v_4	16.2	16.0	15.9	15.8	15.7
T_4	2688	2648	2628	2615	2612
p_4	64.2	64.2	64.2	64.5	64.8
$e_{4'}$	306	299	295	292	291
$v_{4'}$	51.0	50.0	49.7	49.3	49.0
$T_{4'}$	1943	1915	1899	1889	1883
e_6	299	292	288	285	284
v_6	53.6	52.6	52.4	52.2	52.0
T_6	1914	1885	1870	1858	1853

Table 18 - Maximum cycle pressure $p_3 = 1200 \text{ lb./in.}^2 \text{ abs.}$

Fuel fraction $F = 1.0$

r	14	16	18	20	22
E_1	395	364	338	319	307
V_1	448	442	439	435	434
T_1	583	577	572	568	566
s_1	3.46	3.38	3.33	3.28	3.25
E_2	5737	6086	6405	6760	7041
V_2	32.0	27.6	24.4	21.8	19.7
T_2	1592	1652	1707	1767	1818
p_2	540	651	770	895	1012
e_3	1351	1332	1320	1313	1308
v_3	1.70	1.68	1.67	1.67	1.67
T_3	5055	5019	4996	4981	4971
s_3	0.4638	0.4593	0.4564	0.4548	0.4536
e_4	658	647	639	636	633
v_4	16.5	16.3	16.2	16.1	16.0
T_4	3193	3158	3132	3119	3109
p_4	76.2	76.3	76.4	76.5	76.5
$e_{4'}$	401	393	387	385	384
$v_{4'}$	63.3	59.3	58.9	58.5	58.4
$T_{4'}$	2271	2240	2224	2212	2206
e_6	392	384	379	377	375
v_6	63.3	62.5	62.1	61.9	61.5
T_6	2238	2209	2189	2179	2164

Table 19 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$
 Fuel fraction $F = 0.5$
 Compression ratio $r = 18$
 Fresh air temperature $T_{fr} = 540^\circ \text{R.}$
 Exhaust pressure $p_e = 15.0 \text{ lb./in}^2 \text{ abs.}$

p_i	11.0	12.5	14.0	15.5	17.0	18.0
E_1	308	298	288	278	272	267
V_1	550	486	430	386	356	335
T_1	566	564	562	560	559	558
s_1	3.73	3.45	3.20	2.97	2.81	2.66
E_2	6331	6293	6302	6251	6228	6220
V_2	30.6	27.0	23.9	21.4	19.8	18.6
T_2	1693	1687	1690	1681	1687	1677
p_2	603	680	766	857	938	990
e_3	778	758	742	728	721	713
v_3	1.47	1.44	1.42	1.40	1.39	1.38
T_3	3851	3781	3719	3671	3647	3621
s_3	0.2608	0.2546	0.2492	0.2448	0.2426	0.2400
e_4	263	269	275	282	288	291
v_4	19.7	17.4	15.4	13.8	12.8	12.0
T_4	1827	1852	1878	1906	1934	1950
p_4	35.3	40.8	47.0	52.7	58.3	62.5
$e_{4'}$	187	181	175	170	168	165
$v_{4'}$	38.2	37.2	36.5	36.0	35.8	35.4
$T_{4'}$	1486	1456	1431	1407	1400	1384
e_6	163	167	170	172	177	178
v_6	48.1	42.5	38.3	35.2	32.7	29.2
T_6	1378	1394	1408	1420	1439	1445

Table 20 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$
 Fuel fraction $F = 0.5$
 Compression ratio $r = 18$
 Intake pressure $p_i = 14.0 \text{ lb./in}^2 \text{ abs.}$
 Exhaust pressure $p_e = 15.0 \text{ lb./in}^2 \text{ abs.}$

T_{fr}	420	460	500	540	580	620
E_1	- 338	- 120	84	288	498	700
V_1	332	370	399	430	463	493
T_1	436	480	521	562	603	644
S_1	1.42	2.10	2.67	3.20	3.69	4.15
E_2	4383	5027	5651	6302	6886	7489
V_2	18.4	20.6	22.2	23.9	25.7	27.4
T_2	1355	1470	1579	1690	1788	1890
p_2	807	782	776	766	758	754
e_3	661	592	716	742	768	793
v_3	1.31	1.35	1.39	1.42	1.46	1.51
T_3	3229	3539	3629	3719	3814	3903
s_3	0.2220	0.2326	0.2410	0.2492	0.2577	0.2650
e_4	259	265	269	275	282	288
v_4	11.9	13.2	14.3	15.4	16.6	17.7
T_4	1808	1831	1853	1878	1905	1930
p_4	58.0	53.5	50.0	47.0	44.1	42.4
$e_{4'}$	147	158	165	175	183	191
$v_{4'}$	33.5	34.7	35.7	36.5	37.8	39.0
$T_{4'}$	1299	1349	1389	1431	1470	1506
e_6	142	152	161	170	178	186
v_6	35.0	36.3	37.3	38.3	39.5	40.7
T_6	1277	1327	1367	1408	1449	1483

Table 21 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2.\text{abs.}$
 Fuel fraction $F = 0.5$
 Compression ratio $r = 18$
 Intake pressure $p_i = 14.0 \text{ lb./in}^2.\text{abs.}$
 Fresh air temperature $T_{fr} = 540^\circ\text{R.}$

P_e	3.0	7.0	11.0	15.0	19.0
E_1	208	241	268	288	307
V_1	417	422	428	430	434
T_1	546	552	558	562	566
S_1	3.00	3.08	3.15	3.20	3.25
E_2	6052	6116	6219	6302	6348
V_2	23.2	23.5	23.8	23.9	24.1
T_2	1647	1658	1675	1690	1697
P_2	773	766	767	766	768
s_3	0.2459	0.2469	0.2482	0.2492	0.2503
v_3	1.40	1.41	1.42	1.42	1.42
T_3	3684	3694	3709	3719	3732
e_3	732	739	739	742	745
e_4	273	273	274	275	276
v_4	14.9	15.1	15.3	15.4	15.5
T_4	1868	1871	1873	1878	1887
p_4	48.2	47.5	47.0	47.0	46.4
$e_{4'}$	73	119	151	175	194
$v_{4'}$	117	64.5	45.9	36.5	31.0
$T_{4'}$	930	1162	1315	1431	1517
e_6	166	168	168	170	170
v_6	37.8	38.0	38.3	38.3	38.5
T_6	1392	1397	1403	1408	1412

Table 22 - Intake pressure $p_i = 14.0 \text{ lb./in}^2 \text{ abs.}$
 Fresh air temperature $T_{fr} = 540^\circ \text{R.}$
 Exhaust pressure $p_e = 15.0 \text{ lb./in}^2 \text{ abs.}$

$p_{\max}$	800	800	1000	1200
F	0.15	0.2	0.1	0.15
r	15	18	18	24
T_1	552	550	548	546
S_1	3.08	3.05	3.03	3.00
V_1	423	421	420	418
E_1	242	230	220	210
V_2	28.2	23.4	23.3	17.4
p_2	600	770	768	1152
T_2	1556	1656	1650	1822
E_2	5534	6109	6072	7067
H_3	15193	16889	14048	16411
T_3	2340	2530	2210	2480
V_3	31.8	34.5	24.3	22.7
S_3	5.93	6.66	4.96	5.62
V_4	419	417	418	414
p_4	24.8	28.3	21.0	24.8
T_4	964	1100	813	913
E_4	2380	3120	1577	2100
T_5	838	922	740	810

Table 23 - Compression ratio

 $r = 18$ Intake pressure $p_i = 14.0 \text{ lb./in}^2 \text{ abs.}$ Fresh air temperature $T_{fr} = 540^\circ \text{R.}$ Exhaust pressure $p_e = 15.0 \text{ lb./in}^2 \text{ abs.}$

$P_{\max}$ F	800	1000	1400
	0.9	0.9	0.5
E_1	328	328	288
V_1	438	438	430
T_1	570	570	562
S_1	3.30	3.30	3.20
E_2	6390	6390	6302
V_2	24.3	24.3	23.9
T_2	1704	1704	1690
p_2	762	762	766
e_3	1152	1178	793
v_3	2.34	1.90	1.68
T_3	4716	4790	3900
s_3	0.4452	0.4354	0.2413
e_4	606	575	260
v_4	16.1	16.1	15.4
T_4	3056	2947	1815
p_4	74.6	71.5	44.8
$e_{4'}$	367	350	166
$v_{4'}$	57.0	55.3	35.7
$T_{4'}$	2162	2098	1390
e_6	358	342	161
v_6	60.0	58.2	37.2
T_6	2131	2070	1369

Table 24 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2 \text{ abs.}$
 Compression ratio $r = 18$

F for cycle F for BMC	0.35 0.5	0.4 0.5	0.6 0.5	0.6 0.8	0.7 0.8
E_1	261	273	300	300	307
V_1	427	427	432	432	434
T_1	557	559	564	564	566
S_1	3.14	3.16	3.23	3.23	3.25
E_2	6209	6242	6309	6309	6348
V_2	23.7	23.7	24.0	24.0	24.1
T_2	1674	1679	1691	1691	1697
p_2	770	773	767	767	768
e_3	563	614	820	827	933
v_3	1.46	1.56	1.93	1.89	2.05
T_3	3055	3248	3992	3856	4189
s_3	0.1993	0.2200	0.2884	0.3521	0.3845
e_4	192	225	352	372	446
v_4	15.1	15.2	15.6	15.6	15.7
T_4	1507	1657	2208	2211	2499
p_4	37.8	41.6	54.3	55.6	62.5
$e_{4'}$	126	145	220	233	275
$v_{4'}$	30.6	33.0	41.6	42.2	47.5
$T_{4'}$	1199	1290	1632	1641	1817
e_6	121	140	213	226	268
v_6	37.5	34.9	44.1	44.8	50.0
T_6	1172	1268	1604	1610	1787

Table 25 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$
 Compression ratio $r = 18$

F for cycle F for BMC	0.4 0.5	0.55 0.5	0.6 0.8	0.65 0.8	0.7 0.8
E_1	273	296	300	303	307
V_1	427	431	432	433	434
T_1	559	563	564	565	566
S_1	3.16	3.22	3.23	3.23	3.25
E_2	6242	6289	6309	6335	6348
V_2	23.7	24.0	24.0	24.0	24.1
T_2	1679	1687	1691	1695	1697
p_2	773	767	767	770	768
e_3	638	793	852	905	958
v_3	1.28	1.51	1.54	1.60	1.67
T_3	3340	3903	3939	4105	4269
s_3	0.2132	0.2650	0.3438	0.3600	0.3757
e_4	214	305	353	388	424
v_4	15.2	15.5	15.6	15.6	15.7
T_4	1606	2006	2137	2273	2414
p_4	40.3	49.5	53.7	56.4	60.0
$e_{4'}$	138	191	222	242	262
$v_{4'}$	32.3	39.0	41.6	43.6	46.3
$T_{4'}$	1258	1506	1591	1678	1764
e_6	134	186	216	236	256
v_6	34.3	40.7	43.7	45.9	48.6
T_6	1237	1483	1568	1652	1736

1-2 Cycle Performance Data and Other Cycle

Quantities of Interest

The performance of the idealized mixed cycle is evaluated by calculating the net work output, the net mean effective pressure, and the cycle efficiency. The values of these quantities are listed in Tables 26 - 48 , page 99 - 121 , for the four Series of Cycle Analyses. Also included are the limiting values of the peak cylinder temperature and the exhaust temperature, as well as the computed values of the residual gas fraction and the temperature at the beginning of compression.

The values of fixed cycle operating conditions used in the cycle solutions, if not included in the table headings, are given on page 74 .

Table 26 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2 \text{ abs.}$
 Fuel fraction $F = 0.15$

Compression ratio (r)	14	16	18	20	22
Net work output (W_n), BTU/lb.fresh air	118.0	120.3	120.8		
Net mean effective press. (MEP) $_n$, lb/in ² .	43.7	44.6	44.7	$p_2 > p_3$	$p_2 > p_3$
Cycle efficiency (η), per cent.	61.1	62.4	62.6		
Maximum cycle temperature (T_3), °R.	2355	2330	2323		
Exhaust temperature (T_5), °R.	843	835	832		

Table 27 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2 \text{ abs.}$
 Fuel fraction $F = 0.5$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0294	0.0256	0.0227		
Corrected temperature (T_1), °R.	569.9	565.6	562.4		
Net work output (W_n), BTU/lb.fresh air	360.6	361.5	360.3		
Net mean effective press. (MEP _n), lb./in ² .	134.5	135.4	134.9	$p_2 > p_3$	$p_2 > p_3$
Cycle efficiency (η), per cent.	56.0	56.1	55.9		
Maximum cycle temperature (T_3), °R.	3670	3646	3633		
Exhaust temperature (T_5), °R.	1484	1471	1465		

Table 28 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2 \text{ abs.}$
 Fuel fraction $F = 0.8$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0220	0.0192	0.0170		
Corrected temperature (T_1), $^{\circ}\text{R.}$	578.1	572.7	568.7		
Net work output (W_n), BTU/lb.fresh air	514.1	517.4	515.6	$p_2 > p_3$	$p_2 > p_3$
Net mean effective press. (MEP _n), lb./in ² .	190.3	192.9	193.0		
Cycle efficiency (η), per cent.	49.9	50.2	50.0		
Maximum cycle temperature (T_3), $^{\circ}\text{R.}$	4532	4507	4503		
Exhaust temperature (T_5), $^{\circ}\text{R.}$	2018	2003	2000		

Table 29 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2.\text{abs.}$

Fuel fraction $F = 1.0$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0189	0.0156	0.0146		
Corrected temperature (T_1), °R.	582.6	574.6	572.2		
Net work output (W_n), BTU/lb.fresh air	599.7	601.6	602.1	$p_2 > p_3$	$p_2 > p_3$
Net mean effective press. (MEP _n), lb./in ² .	221.2	222.7	223.7		
Cycle efficiency (η), per cent.	46.5	46.7	46.7		
Maximum cycle temperature (T_3), °R.	4890	4865	4856		
Exhaust temperature (T_5), °R.	2360	2343	2336		

Table 30 - Maximum cycle pressure $p_3 = 900 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.15$

Compression ratio (r)	14	16	18	20	22
Net work output (W_n), BTU/lb.fresh air		121.7	122.2	122.7	
Net mean effective press. (MEP_n), lb./in ² .		45.1	45.3	45.5	
Cycle efficiency (η), per cent.		63.0	63.3	63.6	
Maximum cycle temperature (T_3), °R.		2387	2376	2366	
Exhaust temperature (T_5), °R.		830	825	822	

$V_3 < V_2$

$p_2 > p_3$

Table 31 - Maximum cycle pressure $p_3 = 900 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.5$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0296	0.0262	0.0231	0.0208	$p_2 > p_3$
Corrected temperature (T_1), $^{\circ}\text{R.}$	569.6	565.4	562.3	559.8	
Net work output (W_n), BTU/lb.fresh air	366.3	370.5	369.2	368.4	
Net mean effective press. (MEP_n), lb./in^2 .	136.6	138.7	138.2	138.4	
Cycle efficiency (η), per cent.	56.8	57.5	57.3	57.2	
Maximum cycle temperature (T_3), $^{\circ}\text{R.}$	3732	3692	3678	3671	
Exhaust temperature (T_5), $^{\circ}\text{R.}$	1470	1452	1447	1442	

Table 32 - Maximum cycle pressure $p_3 = 900 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.8$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0223	0.0194	0.0172	0.0154	$p_2 > p_3$
Corrected temperature (T_1), $^{\circ}\text{R.}$	577.9	572.5	568.3	565.3	
Net work output (W_n), BTU/lb.fresh air	525.0	530.3	531.0	530.0	
Net mean effective press. (MEP_n), lb./in^2 .	194.3	197.8	198.8	198.4	
Cycle efficiency (η), per cent.	50.9	51.4	51.5	51.4	
Maximum cycle temperature (T_3), $^{\circ}\text{R.}$	4583	4550	4537	4530	
Exhaust temperature (T_5), $^{\circ}\text{R.}$	1995	1973	1963	1960	

Table 33 - Maximum cycle pressure $p_3 = 900 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 1.0$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0191	0.0167	0.0148	0.0132	$p_2 > p_3$
Corrected temperature (T_1), °R.	582.2	576.3	571.7	568.1	
Net work output (W_n), BTU/lb.fresh air	615.6	618.6	620.5	618.8	
Net mean effective press. (MEP _n), lb./in ₂ ²	227.0	229.0	230.6	231.6	
Cycle efficiency (η), per cent.	47.8	48.0	48.1	48.0	
Maximum cycle temperature (T_3), °R.	4929	4907	4892	4887	
Exhaust temperature (T_5), °R.	2328	2309	2298	2294	

Table 34 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.15$

Compression ratio (r)	14	16	18	20	22
Net work output (W_n), BTU/lb.fresh air	$V_3 < V_2$	122.0	122.9	124.1	124.3
Net mean effective press. (MEP_n), lb./in ² .		45.2	45.5	46.0	46.1
Cycle efficiency (η), per cent.		63.0	63.7	64.3	64.4
Maximum cycle temperature (T_3), °R.		2442	2423	2412	2409
Exhaust temperature (T_5), °R.		829	822	817	815

Table 35 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{.abs.}$

Fuel fraction $F = 0.5$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0299	0.0261	0.0234	0.0212	$p_2 > p_3$
Corrected temperature (T_1), °R.	569.3	565.0	562.1	559.7	
Net work output (W_n), BTU/lb.fresh air	371.9	374.3	376.0	376.2	
Net mean effective press.(MEP _n), lb./in ² .	140.8	141.2	140.5	141.3	
Cycle efficiency (η), per cent.	57.7	58.1	58.3	58.4	
Maximum cycle temperature (T_3), °R.	3770	3742	3719	3700	
Exhaust temperature (T_5), °R.	1452	1435	1431	1421	

Table 36 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.8$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0225	0.0197	0.0174	0.0156	$p_2 > p_3$
Corrected temperature (T_1), °R.	577.5	572.2	568.1	565.0	
Net work output (W_n), BTU/lb.fresh air	536.8	541.2	543.3	543.8	
Net mean effective press. (MEP _n), lb./in ² .	199.8	201.1	203.4	203.6	
Cycle efficiency (η), per cent.	52.1	52.5	52.7	52.7	
Maximum cycle temperature (T_3), °R.	4627	4594	4575	4564	
Exhaust temperature (T_5), °R.	1970	1948	1935	1930	

Table 37 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 1.0$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0194	0.0169	0.0150	0.0134	$p_2 > p_3$
Corrected temperature (T_1), °R.	582.2	576.2	571.6	568.1	
Net work output (W_n), BTU/lb.fresh air	628.6	635.0	636.4	635.2	
Net mean effective press. (MEP _n), lb./in ² .	231.8	235.1	236.4	237.8	
Cycle efficiency (η), per cent.	48.8	49.3	49.4	49.3	
Maximum cycle temperature (T_3), °R.	4973	4942	4927	4919	
Exhaust temperature (T_5), °R.	2302	2281	2267	2264	

Table 38 - Maximum cycle pressure $p_3 = 1200 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.15$

Compression ratio (r)	14	16	18	20	22
Net work output (W_n), BTU/lb.fresh air				126.0	127.2
Net mean effective press. (MEP _n), lb./in ² .	$V_3 < V_2$	$V_3 < V_2$	$V_3 < V_2$	46.7	47.2
Cycle efficiency (η), per cent.				65.3	65.9
Maximum cycle temperature (T_3), °R.				2501	2489
Exhaust temperature (T_5), °R.				810	805

Table 39 - Maximum cycle pressure $p_3 = 1200 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.5$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0302	0.0266	0.0238	0.0215	0.0195
Corrected temperature (T_1), $^{\circ}\text{R.}$	569.3	564.8	561.8	559.4	557.2
Net work output (W_n), BTU/lb.fresh air	376.3	382.4	385.3	387.1	387.0
Net mean effective press. (MEP_n), lb./in^2 .	140.3	143.1	144.2	145.5	144.9
Cycle efficiency (η), per cent.	58.4	59.3	59.8	60.1	60.0
Maximum cycle temperature (T_3), $^{\circ}\text{R.}$	3904	3845	3824	3789	3775
Exhaust temperature (T_5), $^{\circ}\text{R.}$	1442	1418	1405	1399	1389

Table 40 - Maximum cycle pressure $p_3 = 1200 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.8$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0227	0.0200	0.0177	0.0160	0.0138
Corrected temperature (T_1), $^{\circ}\text{R.}$	577.2	571.9	567.7	564.6	561.1
Net work output (W_n), BTU/lb.fresh air	548.2	557.2	560.8	562.4	561.8
Net mean effective press. (MEP_n), lb./in^2 .	202.9	207.0	209.9	210.6	210.3
Cycle efficiency (η), per cent.	53.2	54.0	54.4	54.5	54.4
Maximum cycle temperature (T_3), $^{\circ}\text{R.}$	4729	4678	4653	4633	4625
Exhaust temperature (T_5), $^{\circ}\text{R.}$	1943	1915	1899	1889	1883

Table 41 - Maximum cycle pressure $p_3 = 1200 \text{ lb/in}^2 \cdot \text{abs.}$

Fuel fraction $F = 1.0$

Compression ratio (r)	14	16	18	20	22
Residual gas fraction (f)	0.0187	0.0172	0.0153	0.0137	0.0125
Corrected temperature (T_1), $^{\circ}\text{R.}$	579.8	575.9	571.3	567.7	565.0
Net work output (W_n), BTU/lb.fresh air	644.9	653.5	659.4	660.7	662.5
Net mean effective press. (MEP_n), lb./in^2 .	237.8	241.9	245.0	247.4	247.1
Cycle efficiency (η), per cent.	50.0	50.7	51.2	51.3	51.4
Maximum cycle temperature (T_3), $^{\circ}\text{R.}$	5055	5019	4996	4981	4971
Exhaust temperature (T_5), $^{\circ}\text{R.}$	2271	2240	2224	2212	2206

Table 42 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2.\text{abs.}$
 Fuel fraction $F = 0.5$
 Compression ratio $r = 18$
 Fresh air temperature $T_{fr} = 540 \text{ }^\circ\text{R.}$
 Exhaust pressure $p_e = 15.0 \text{ lb./in}^2.\text{abs}$

Intake pressure (p_i), $\text{lb./in}^2.\text{abs.}$	11.0	12.5	14.0	15.5	17.0	18.0
Residual gas fraction (f)	0.0287	0.0260	0.0234	0.0214	0.0198	0.0188
Corrected temperature (T_1), $^\circ\text{R.}$	566.3	564.3	562.1	560.4	559.3	558.5
Net work output (W_n), BTU/lb.fresh air	380.0	378.6	376.0	372.0	368.3	366.0
Net mean effective press. (MEP_n), $\text{lb./in}^2.$	111.1	125.7	140.5	156.1	167.9	177.0
Cycle efficiency (η), per cent.	59.0	58.7	58.3	57.7	57.1	56.8
Maximum cycle temperature (T_3), $^\circ\text{R.}$	3851	3781	3719	3671	3647	3621
Exhaust temperature (T_5), $^\circ\text{R.}$	1486	1456	1431	1407	1400	1384

Table 43 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2.\text{abs.}$

Fuel fraction $F = 0.5$

Compression ratio $r = 18$

Intake pressure $p_i = 14.0 \text{ lb./in}^2.\text{abs.}$

Exhaust pressure $p_e = 15.0 \text{ lb./in}^2.\text{abs.}$

Fresh air temperature (T_{fr}), $^{\circ}\text{R.}$	420	460	500	540	580	620
Residual gas fraction (f)	0.0197	0.0213	0.0222	0.0234	0.0244	0.0251
Corrected temperature (T_1), $^{\circ}\text{R.}$	438.9	479.7	520.7	562.1	603.6	644.8
Net work output (W_n), BTU/lb.fresh air	369.3	371.4	374.6	376.0	376.9	377.9
Net mean effective press.(MEP _n), lb./in ² .	180.0	162.2	151.4	140.5	131.4	123.8
Cycle efficiency (η), per cent.	57.3	57.6	58.1	58.3	58.5	58.6
Maximum cycle temperature (T_3), $^{\circ}\text{R.}$	3229	3539	3629	3719	3814	3903
Exhaust temperature (T_5), $^{\circ}\text{R.}$	1299	1349	1389	1431	1470	1506

Table 44 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$

Fuel fraction $F = 0.5$

Compression ratio $r = 18$

Intake pressure $p_i = 14.0 \text{ lb./in}^2 \text{ abs.}$

Fresh air temperature $T_{fr} = 540^\circ \text{R.}$

Exhaust temperature (p_e), $\text{lb./in}^2 \text{ abs.}$	3.0	7.0	11.0	15.0	19.0
Residual gas fraction (f)	0.0071	0.0130	0.0185	0.0234	0.0279
Corrected temperature (T_1), $^\circ \text{R.}$	545.9	551.6	557.1	562.1	566.7
Net work output (W_n), BTU/lb.fresh air	399.6	393.1	385.0	376.0	366.5
Net mean effective press.(MEP _n), lb./in^2 .	157.6	152.1	145.9	140.5	135.9
Cycle efficiency (η), per cent.	62.0	61.0	59.7	58.3	56.9
Maximum cycle temperature (T_3), $^\circ \text{R.}$	3684	3694	3709	3719	3732
Exhaust temperature (T_5), $^\circ \text{R.}$	930	1162	1315	1431	1517

Table 45 - Intake pressure $p_i = 14.0 \text{ lb./in}^2.\text{abs.}$
 Fresh air temperature $T_{fr} = 540 \text{ }^\circ\text{R.}$
 Exhaust pressure $p_e = 15.0 \text{ lb./in}^2.\text{abs.}$

Maximum cycle pressure (p_3), $\text{lb./in}^2.\text{abs.}$	800	800	1000	1200
Fuel fraction (F)	0.15	0.2	0.1	0.15
Compression ratio (r)	15	18	18	24
Net work output (W_n), BTU/lb.fresh air	119.1	157.7	81.7	127.7
Net mean effective pressure (MEP_n), $\text{lb./in}^2.$	44.1	58.7	29.9	47.3
Cycle efficiency (η), per cent.	61.7	61.2	63.4	66.2
Maximum cycle temperature (T_3), $^\circ\text{R.}$	2340	2530	2210	2480
Exhaust temperature (T_5), $^\circ\text{R.}$	838	922	740	810

Table 46 - Compression ratio	$r =$	18
Intake pressure	$p_i =$	14.0 lb./in ² .abs.
Fresh air temperature	$T_{fr} =$	540 °R.
Exhaust pressure	$p_e =$	15.0 lb./in ² .abs.

Maximum cycle pressure (p_3), lb./in ² .abs.	800	1000	1400
Fuel fraction (F)	0.9	0.9	0.5
Residual gas fraction (f)	0.0157	0.0161	0.0240
Corrected temperature (T_1), °R.	570.4	569.9	561.5
Net work output (W_n), BTU/lb.fresh air	562.4	594.1	391.6
Net mean effective pressure (MEP_n), lb./in ² .	208.9	221.1	145.5
Cycle efficiency (η), per cent.	48.5	51.2	60.8
Maximum cycle temperature (T_3), °R.	4716	4790	3900
Exhaust temperature (T_5), °R.	2162	2098	1390

Table 47 - Maximum cycle pressure $p_3 = 800 \text{ lb./in}^2 \text{ abs.}$
 Compression ratio $r = 18$

Fuel fraction (F) for the cycle	0.35	0.4	0.6	0.6	0.7
Fuel fraction (F) for the BMC	0.5	0.5	0.5	0.8	0.8
Residual gas fraction (f)	0.0275	0.0256	0.0208	0.0205	0.0184
Corrected temperature (T_1), $^{\circ}\text{R.}$	558.1	559.7	564.8	564.6	566.4
Net work output (W_n), BTU/lb.fresh air	263.1	295.2	428.3	408.3	463.0
Net mean effective press.(MEP_n), lb./in^2 .	99.2	111.4	161.0	153.4	173.3
Cycle efficiency (η), per cent.	58.3	57.3	55.4	52.8	51.3
Maximum cycle temperature (T_3), $^{\circ}\text{R.}$	3055	3248	3992	3856	4189
Exhaust temperature (T_5), $^{\circ}\text{R.}$	1199	1290	1632	1641	1817

Table 48 - Maximum cycle pressure $p_3 = 1000 \text{ lb./in}^2 \text{ abs.}$
 Compression ratio $r = 18$

Fuel fraction (F) for the cycle	0.4	0.55	0.6	0.65	0.7
Fuel fraction (F) for the BMC	0.5	0.5	0.8	0.8	0.8
Residual gas fraction (f)	0.0261	0.0220	0.0208	0.0199	0.0189
Corrected temperature (T_1), $^{\circ}\text{R.}$	559.1	562.8	563.8	565.0	566.0
Net work output (W_n), BTU/lb.fresh air	306.8	410.9	427.3	457.7	485.7
Net mean effective press.(MEP _n), lb./in. ² .	115.8	154.3	160.6	171.4	181.7
Cycle efficiency (η), per cent.	59.5	58.0	55.2	54.6	53.8
Maximum cycle temperature (T_3), $^{\circ}\text{R.}$	3340	3903	3939	4105	4269
Exhaust temperature (T_5), $^{\circ}\text{R.}$	1258	1506	1591	1678	1764

APPENDIX 2

ADDITIONAL DATA ON CYCLE BEHAVIOUR

The relative influence of changes in the intake pressure, the temperature of the fresh intake air, and the exhaust pressure on cycle performance are not included in this report. However, some preliminary work has been carried out to investigate these additional aspects of cycle behaviour. Approximate changes in cycle performance figures as a response to variations in either (p_i) , (T_{fr}) , or (p_e) are given in Tables 49 - 51 , page 123-124. The values are based on Series II of Cycle Analyses, the results of which are tabulated in Tables 42 - 44 , page 115 - 117. The trends demonstrated in the tables are similar to those found by others ^{3, 11}).

It should be noted that the reported variations are the result of changes of only one of the three independent variables. Whether or not the changes are additive when two or three of the variables vary simultaneously is a matter requiring further investigation. It is suggested that an extended survey should cover the study of the combined effects of variations in the above three operating conditions over ranges sufficiently wide to include supercharged cycles ; i.e. the objective of such investigation would be to replace the functional relationship

$$Q = \psi (p_i, T_{fr}, p_e) \dots\dots\dots (40)$$

where (Q) represents any cycle performance parameter, by an empirical expression suitable for nomographic solution.

Table 49

**Estimated Variations in Cycle Performance
Due to Changes in Intake Pressure (p_i)**

Intake pressure (p_i) lb./in ² .abs.	Cycle efficiency (η)	Net work output (W_n)	Mean effective pressure (MEP_n)
13	101	101	93
14	100	100	100
15	999	99	107

Table 50

Estimated Variations in Cycle Performance
 Due to Changes in Temperature of Fresh Intake Air (T_{fr})

Fresh air temp. (T_{fr}) ° R.	Cycle efficiency (η)	Net work output (W_n)	Mean effective pressure (MEP_n)
500	100	100	107
540	100	100	100
580	100	100	94

Table 51

Estimated Variations in Cycle Performance
 Due to Changes in Exhaust Pressure (p_e)

Exhaust pressure (p_e) lb./in ² .abs.	Cycle efficiency (η)	Net work output (W_n)	Mean effective pressure (MEP_n)
14	101	101	101
15	100	100	100
16	99	99	99

APPENDIX 3

DEVELOPMENT OF EMPIRICAL EQUATIONS

When the object of an investigation is to study the effect of one or more independent variables on a dependent parameter in a certain process, a number of standard methods are used to correlate the test data and to determine explicit empirical relationships between the variables under consideration ⁸).

In the case of a four-variable process including one dependent variable there is usually little likelihood of establishing an accurate empirical expression for such variable or parameter, due to the complexities introduced by interdependence and recurrence, i.e. the influence of one independent variable upon the variation of the dependent parameter with a second independent variable. If this interdependence is either nil or negligible, a rather simple method of correlating four-variable data is available.

To illustrate the technique used in such a situation, an empirical expression is developed here for maximum cylinder temperature as a function of fuel fraction, maximum cycle pressure, and compression ratio. This is followed by an example of a three-

FIG. 24

FIG. 24 - VARIATION OF MAXIMUM CYCLE TEMPERATURE WITH COMPRESSION RATIO FOR SELECTED VALUES OF MAXIMUM CYCLE PRESSURE AND FUEL FRACTION.

variable process, in which the ratio of the maximum cylinder temperature to the exhaust temperature is expressed empirically in the independent parameters, fuel fraction and maximum cycle pressure.

Maximum Cycle Temperature

In Figure 24 maximum cycle temperature ($T_{\max}$) is plotted against compression ratio (r) for the complete range of fuel-air ratios and maximum cycle pressures used. By adopting curve-fitting techniques ⁸) it is found from a plot of the average value of ($T_{\max}$) against (F), for all ($p_{\max}$) and (r) values, that

$$T_{\max, \text{ avg}} = C_1 + C_2 F + C_3 F^3 \quad \dots\dots\dots (41)$$

where C_1 , C_2 , and C_3 are constants.

Also, from Figure 24, it is seen that the variation of ($T_{\max}$) with ($p_{\max}$) is almost independent of (F). By plotting average values of ($T_{\max}$) for all values of (r) against ($p_{\max}$), the following linear relationship is obtained :

$$T_{\max, \text{ avg}} = C_4 p_{\max} + C_5 \quad \dots\dots\dots (42)$$

Because of its independency on (F), Equation (42) may be

substituted for the constant C_1 in Equation (41) to give

$$T_{\max, \text{avg}} = C_4 p_{\max} + C_5 + C_2 F + C_3 F^3 \dots (43)$$

The variation of $(T_{\max})$ with (r) is also found to be independent of (F) . Again by using average values of $(T_{\max})$ over the range of $(p_{\max})$ it is found by curve-fitting that

$$T_{\max, \text{avg}} = C_6 r^2 + C_7 r + C_8 \dots (44)$$

By combining Equations (43) and (44) and making adjustments to constants it can be shown that

$$T_{\max} = C_9 + C_4 p_{\max} + C_6 r^2 + C_7 r + C_2 F + C_3 F^3 \dots (45)$$

Actual values of $(T_{\max})$ determined from cycle analyses are compared with values obtained from Equation (45), and the constants further adjusted to give a satisfactory average deviation as well as acceptable maximum deviations. Substituting the numerical values for the constants in Equation (45), the final expression thus obtained is Equation (31), listed in Table 2, page 40,

$$T_{\max} = 1875 + (0.410 p_{\max}) + (1.03 r^2 - 45.3 r) + (4082 F - 958 F^3) \dots (31)$$

FIG.25- RATIO OF MAXIMUM CYCLE TEMPERATURE TO EXHAUST TEMPERATURE VERSUS MAXIMUM CYCLE PRESSURE.

This equation gives an average deviation - without regard to sign - of $(T_{\max})$ -values of 0.3 %, and maximum deviations of + 1.1 % and - 1.4 %, and is considered satisfactory.

Exhaust Temperature

Exhaust pressure has been considered constant for the series of cycle calculations , so that from Equation (28) a graph of $\frac{T_{\max}}{T_e} = T_r$ against $(p_{\max})$ should reveal a slight curve, the slope depending on the value of $(\frac{n}{n-1})$ and, therefore, on the fuel fraction (F) . Such graphs are plotted for the various values of (F) in Figure 25 . Since the curvatures are slight, the curves are considered as straight lines for the purpose of developing empirical relationships between exhaust temperatures (T_e) and fuel fraction (F) , which will be valid over the selected range of operating conditions. As can be seen from the graphs , no serious error is involved by making this assumption.

It is found that all curves of Figure 25 , when produced , pass through a common point , giving an equation of the form

$$\frac{\frac{T_{\max}}{T_e} + C_{10}}{p_{\max} + C_{11}} = X \quad \dots\dots\dots (46)$$

where the slope of each curve (X) depends on fuel fraction (F) .

On plotting (X) against (F) a linear relationship is revealed , such that

$$X = C_{12} F + C_{13} \quad \text{..... (47)}$$

Inserting Equation (47) in Equation (46) and writing in the values of the various constants the following relationship is obtained :

$$\frac{T_{\max}}{T_e} - 1.21 = (0.000390 F + 0.00080) x (p_{\max} + 1325) \quad \text{..... (32)}$$

Equation (32) , which was listed in Table 2, page 40 , gives an average deviation - without regard to sign - of the (T_r) -values of 0.4 % , and maximum deviations of + 0.8 % and - 0.9 %.

APPENDIX 4

EQUATIONS FOR THE DETERMINATION OF ENGINE PERFORMANCE

4.1 Derivation of Engine Performance Equations

For the series of experiments conducted on the Ricardo research engine the relationships between the conventional performance quantities are derived as follows.

For the four-stroke engine, having a bore of 3.000 inches, a stroke of 4.375 inches, a speed of (N) rpm, and a brake mean effective pressure of (BMEP) lb./in²., when coupled to a dynamometer with a swinging-arm length of 18 inches and a load of (W) lb., the torque relationship is :

$$\frac{\pi}{4} \times 3.000^2 \times (\text{BMEP}) \times \frac{4.375}{12} \times \frac{N}{2} = W \times \frac{18}{12} \times 2\pi N$$

or

$$\text{BMEP} = (7.31 W) \text{ lb./in}^2. \quad \dots\dots\dots (48)$$

For an indicated mean effective pressure of (IMEP) lb./in². the indicated horsepower output of the engine is :

$$\text{IHP} = \frac{\frac{\pi}{4} \times 3.000^2 \times (\text{IMEP}) \times \frac{4.375}{12} \times \frac{N}{2}}{33,000} \text{ hp.}$$

or

$$\text{IHP} = \frac{\text{IMEP} \times N}{25,600} \text{ hp} \quad \dots\dots\dots (49)$$

If the fuel used has a specific gravity of 0.818 , then 50 cc. of fuel burned in (t) seconds corresponds to a fuel consumption of

$$\frac{50 \times 3600 \times 0.818}{t \times 454} = \frac{324.5}{t} \text{ lb./hour}$$

Combining the last two expressions, the indicated specific fuel consumption amounts to

$$\text{i.s.f.c.} = \frac{\frac{324.5}{t}}{\text{IHP}} = \frac{\frac{324.5}{t}}{\frac{\text{IMEP} \times N}{25,600}} \text{ lb./ihp.hr.}$$

or

$$\text{i.s.f.c.} = \frac{8.30 \times 10^6}{t \times \text{IMEP} \times N} \text{ lb./ihp.hr.} \dots\dots\dots (50)$$

The thermal efficiency of the engine is

$$\eta = \frac{550 \times 3600}{\text{i.s.f.c.} \times 778 \times 18,330} = \frac{0.139}{\text{i.s.f.c.}} \dots (51)$$

for a low heat of combustion of the fuel of 18,330 BTU/lb., ref.p.55.

Substituting Equation (50) into Equation (51) there results :

$$\eta = 1.675 \times 10^{-8} \times t \times \text{IMEP} \times N \dots\dots\dots (52)$$

In the conducted experiments the engine speed was kept constant

at $N = 2000$ rpm., hence,

$$\eta = 3.35 \times 10^{-5} \times t \times \text{IMEP} \dots\dots\dots (53)$$

4.2 Equation for the Calculation of Mixture Strength

The metering of the fuel consumption was carried out by the positive displacement method, using a calibrated pipette of 50 cc. and a stopwatch. The time (t) in seconds required for the engine to consume this amount of fuel for an observed air consumption of (L) lb./min. and for a given mixture strength (F) may be derived as follows.

By definition, the stoichiometric ratio (F_c) is the theoretical weight of fuel required for complete combustion of 1 lb. of air. Also, the mixture strength (F) is the fraction of the stoichiometric ratio actually present for combustion.

Hence,

$$F \times F_c = \frac{\text{metered weight rate of fuel flow}}{\text{metered weight rate of air flow}}$$

or

$$F \times F_c = \frac{\frac{50}{t} \times \text{S.G.}_{\text{fuel}} \times \frac{1}{454}}{\frac{L}{60}}$$

The specification of the fuel used in the experiments is given on page 55 . Substituting the relevant values into the last expression, and re-arranging,

$$t = \frac{50 \times 60 \times 0.818}{454 \times 0.0667F \times L} = \frac{81.04}{F \times L} \dots\dots\dots (54)$$

This is the relationship underlying the curves presented in Figure 12 . The chart was used as an aid in the laboratory to adjust the mixture strength of the test engine, as described in Section 9 .

4.3 Orifice Plate Computation

The expression for rate of weight flow of fresh air induced is computed from the following data in accordance with the British Flow Measurement Code B.S. 1042 - 1943 .

Details of metering installation :

Fluid flowing	-	Atmospheric air
Orifice installation	-	Flow from an open space into a pipe
Pressure tapplings, upstream	-	None ; open manometer connection
downstream	-	At distance $D/2$ from downstream face of orifice plate
Diameter of orifice bore	-	$d = 1.134$ in.

Diameter of downstream pipe	-	$D = 4.000 \text{ in.}$
Area ratio	-	$\left(\frac{d}{D}\right)^2 = 0.128$
		This value is less than $\frac{1}{7}$, and hence the area ratio is taken as $m = 0$ (Ref. B.S.1042 - 67)
Differential head measuring device	-	Micro-manometer
Maximum expected differential head	-	$h = 25 \text{ m.m. w.g.}$
Humidity	-	Assume no correction for humidity required, $N = 1.000$
Air temperature	-	60°F.
Air pressure	-	29.92 in. Hg.

Calculation :

Reference
in B.S. 1042

1. Rate weight of flow

$$L = 359.1 \times C Z d^2 E \times \sqrt{h} \times \sqrt{W} \text{ lb./hr.} \quad \text{Eq.(3)}$$

2. For standard conditions at orifice as specified above
the specific weight

$$W = \frac{p}{RT} = \frac{14.70 \times 144}{53.3 \times 520} = 0.07637 \text{ lb./ft}^3.$$

3. For ambient conditions other than standard, i.e. air
temperature $(t_a)^\circ \text{F}$, and barometric pressure $(p_a) \text{ in.Hg.}$

Reference
in B.S.1042

the multiplier is

$$M = \sqrt{\frac{520}{460 + t_a} \cdot \frac{p_a}{29.92}}$$

4. For $m = 0$, $E = 1.000$ D.S. 9
5. For $m = 0$, $C = 0.595$ Fig.19A
6. Assuming $Z = 1.000$, the approximate maximum weight rate of flow, when $h = 1$ in. w.g.,

$$L = 359.1 \times 0.595 \times 1.134^2 \times \sqrt{1} \times \sqrt{0.07637}$$

$$L = 75.8 \text{ lb./hr.}$$

7. For these conditions the Reynolds Number

$$R_d = \frac{L}{15.83 \text{ d } \gamma} \quad \text{Eq. (13)}$$

$$R_d = \frac{75.8}{15.83 \times 1.134 \times 181.5 \times 10^{-6}} = 23,300$$

8. Correction for R_d is 1.008 Fig.19B
 "- "- D is 1.000 Fig.13C
 "- "- ϵ is 0.999 D.S. 11

9. Correct value for $Z = 1.008 \times 1.000 \times 0.999$
 $Z = 1.007$

10. Corrected expression for weight flow per minute,
 taking (h) in m.m.,

Reference
in B.S.1042

$$L = 359.1 \times 0.595 \times 1.007 \times 1.134^2 \times \sqrt{\frac{h}{25.4}} \times \sqrt{0.07637} \times M$$

$$L = 0.2515 M \sqrt{h} \text{ lb./min.}$$

11.	Tolerances : Basic	0.70 %	Fig.19D
	Extra for R_d	0.22 %	Fig.19B
	Extra for D	0.00 %	Fig.19C
	Extra for ℓ	0.30 %	Par. 63
	Overall tolerance	± 0.80 %	Par. 3

12. Summarizing :

Weight rate of flow per minute

$$L = 0.2515 M \sqrt{h} \text{ lb./min. ... (55)}$$

Multiplier for non-standard air conditions

$$M = \sqrt{\frac{520}{460 + t_a} \cdot \frac{p_a}{29.92}} \text{ (56)}$$

4.4 Relationship between Efficiency Ratio and Mean Effective Pressure Ratio

Referring to the derivation of theoretical equations in Section 3-4 , Equations (17) and (20) combined will give the following expression for (theoretical) cycle efficiency :

$$\eta_{th} = \frac{2.68 (\text{MEP})_{th}}{0.06775 F_{th} \times 19,028}$$

In Section 10 this relationship is expressed thus :

$$(\text{MEP})_{th} = K_{th} \times F_{th} \times \eta_{th} \quad \dots\dots\dots(36)$$

where the constant (K_{th}) has the value

$$K_{th} = \frac{0.06775 \times 19,028}{2.68} = 481$$

For the experimental work Equations (53) and (54) were developed in Sections 4-1 and 4-2 , which may be combined to give the expression for (actual) thermal efficiency

$$\eta_a = 3.35 \times 10^{-5} \times \frac{81.04}{F_a \cdot L} \times (\text{IMEP})_a$$

Again this relationship can be written

$$(\text{IMEP})_a = K_a \times F_a \times \eta_a \quad \dots\dots\dots(37)$$

During the tests the average value of the weight flow of air per minute was $L_{avg} = 1.175$ lb./min. Hence, the value of the constant (K_a) in Equation (37) is

$$K_a = \frac{1.175}{3.35 \times 10^{-5} \times 81.04} = 432$$

Dividing Equation (37) by Equation (36) and introducing

$$\text{efficiency ratio } \eta_r = \frac{\eta_a}{\eta_{th}}$$

and

$$\text{mean effective pressure ratio } (\text{MEP})_r = \frac{(\text{IMEP})_a}{(\text{MEP})_{th}}$$

there results :

$$(\text{MEP})_r = \frac{K_a}{K_{th}} \times \frac{F_a}{F_{th}} \times \eta_r \quad \dots\dots\dots(38)$$

Since fuel fractions were equal for theoretical and experimental analyses, $\frac{F_a}{F_{th}} = 1$, while $\frac{K_a}{K_{th}} = K_r = \frac{432}{481} = 0.899$

The final relationship is therefore

$$(\text{MEP})_r = K_r \times \eta_r \quad \dots\dots\dots(39)$$

where $K_r = 0.899$, a constant.

APPENDIX 5

RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 52 - Observed quantities

Item	Quantity	Symbol	Unit	Experimental Results						
1	Test Run No.			82	83	84	85	86	Average	87
2	Compression ratio	r		20	20	20	20	20		20
3	Engine speed	N	rpm	2000	2000	2000	2000	2000		2000
4	Dynamometer load	W	lb.	3.60	3.85	3.75	3.64	3.52		
5	Dynamometer load - motoring	W _m	lb.							6.77
6	Lubrication oil temperature		°F	147	146	145	145	145		143
7	Cooling water temperature, inlet		°F	139	140	140	143	145		145
8	"-", outlet		°F	143	144	144	147	149		143
9	Orifice differential pressure	h	m.m. w.g.	21.78	21.78	21.78	21.78	21.78	21.78	
10	Air temperature at orifice	t _a	°F	62	62	61	61	61	61.5	
11	Barometric pressure	p _a	in. Hg.	30.06	30.06	30.06	30.06	30.06	30.06	
12	Time to consume 50 cc. of fuel	t	sec.	194.4	194.6	195.5	196.2	195.8	195.3	
13	Injection point before TDC.		degrees	37.0	39.5	42.0	44.5	47.0		
14	Maximum cycle pressure	p _{max}	lb./in ² .a.	785	898	984	1070	1120		
15	Exhaust gas appearance			clear	clear	clear	clear	clear		
16	Exhaust gas temperature	t _e	°F	450	428	419	407	403		

Calculated performance data for this series of test runs are listed in Table 57 , page 144 .

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RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 53 - Observed quantities

Item	Quantity	Symbol	Unit	Experimental Results						
1	Test Run No.			91	92	93	94	95	Average	96
2	Compression ratio	r		20	20	20	20	20		20
3	Engine speed	N	rpm	2000	2000	2000	2000	2000		2000
4	Dynamometer load	W	lb.	6.12	6.45	6.47	6.36	6.07		
5	Dynamometer load - motoring	W _m	lb.							6.86
6	Lubrication oil temperature		°F	145	146	147	148	149		148
7	Cooling water temperature, inlet		°F	146	142	142	141	141		141
8	"-", Outlet		°F	150	146	146	145	145		138
9	Orifice differential pressure	h	m.m. w.g.	21.49	21.49	21.49	21.49	21.49	21.49	
10	Air temperature at orifice	t _a	°F	62	62	62	61	61	61.5	
11	Barometric pressure	p _a	in. Hg.	30.20	30.21	30.22	30.23	30.23	30.22	
12	Time to consume 50 cc. of fuel	t	sec.	151.0	152.8	154.1	155.1	158.0	154.2	
13	Injection point before TDC.		degrees	37.0	39.5	42.0	44.5	47.0		
14	Maximum cycle pressure	p _{max}	lb./in ² .a.	804	885	1007	1093	1205		
15	Exhaust gas appearance			clear	clear	clear	clear	clear		
16	Exhaust gas temperature	t _e	°F	565	543	529	512	493		

Calculated performance data for this series of test runs are listed in Table 58 , page 145.

APPENDIX 5

RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 54 - Observed quantities

Item	Quantity	Symbol	Unit	Experimental Results						
1	Test Run No.			101	102	103	104	105	Average	106
2	Compression ratio	r		20	20	20	20	20		20
3	Engine speed	N	rpm	2000	2000	2000	2000	2000		2000
4	Dynamometer load	W	lb.	8.53	8.76	8.84	8.97	8.83		
5	Dynamometer load - motoring	W _m	lb.							6.99
6	Lubrication oil temperature		°F	147	147	148	149	149		145
7	Cooling water temperature, inlet		°F	140	142	143	145	145		144
8	"-", outlet		°F	144	146	147	149	149		141
9	Orifice differential pressure	h	m.m. w.g.	21.49	21.49	21.49	21.49	21.49	21.49	
10	Air temperature at orifice	t _a	°F	64	64	64	63.5	63.5	64	
11	Barometric pressure	p _a	in. Hg.	30.36	30.36	30.36	30.35	30.35	30.36	
12	Time to consume 50 cc. of fuel	t	sec.	122.3	123.0	126.6	126.7	127.4	125.2	
13	Injection point before TDC.		degrees	37.0	39.5	42.0	44.5	47.0		
14	Maximum cycle pressure	p _{max}	lb./in ² .a.	823	941	1046	1111	1178		
15	Exhaust gas appearance			clear	clear	clear	clear	clear		
16	Exhaust gas temperature	t _e	°F	684	666	655	635	630		

Calculated performance data for this series of tests are listed in Table 59 , page 146 .

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RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 55 - Observed quantities

Item	Quantity	Symbol	Unit	Experimental Results						
1	Test Run No.			110	111	112	113	114	Average	115
2	Compression ratio	r		20	20	20	20	20		20
3	Engine speed	N	rpm	2000	2000	2000	2000	2000		2000
4	Dynamometer load	W	lb.	9.68	10.04	10.26	10.31	10.35		
5	Dynamometer load - motoring	W _n	lb.							6.89
6	Lubrication oil temperature		°F	142	145	148	149	149		145
7	Cooling water temperature, inlet		°F	150	149	151	146	146		143
8	"- , outlet		°F	154	153	155	150	150		141
9	Orifice differential pressure	h	mm. Hg.	21.78	21.78	21.78	21.78	21.78	21.78	
10	Air temperature at orifice	t _a	°F	63	64	65	65	65	64.5	
11	Barometric pressure	p _a	in. Hg.	30.36	30.35	30.34	30.32	30.30	30.33	
12	Time to consume 50 cc. of fuel	t	sec.	106.6	108.0	108.1	108.0	108.6	107.8	
13	Injection point before TDC.		degrees	37.0	39.5	42.0	44.5	47.0		
14	Exhaust gas appearance			clear	clear	clear	clear	clear		
15	Maximum cycle pressure	p _{max}	lb./in ² .a.	803	900	993	1103	1188		
16	Exhaust gas temperature	t _e	°F	810	795	771	756	756		

Calculated performance data for this series of test runs are listed in Table 60 , page 147.

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RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 56 - Observed quantities

Item	Quantity	Symbol	Unit	Experimental Results						
1	Test Run No.			122	123	124	125	126	Average	127
2	Compression ratio	r		20	20	20	20	20		20
3	Engine speed	N	rpm	2000	2000	2000	2000	2000		2000
4	Dynamometer load	W	lb.	10.55	10.84	11.30	11.28	11.41		
5	Dynamometer load - motoring	W _m	lb.							6.74
6	Lubrication oil temperature		°F	149	149	152	150	149		148
7	Cooling water temperature, inlet		°F	145	147	146	149	149		148
8	"- , outlet		°F	149	151	150	153	153		146
9	Orifice differential pressure	h	m.m. w.g.	21.78	21.78	21.78	21.78	21.78	21.78	
10	Air temperature at orifice	t _a	°F	66	66	66	66	66	66	
11	Barometric pressure	p _a	in. Hg.	30.29	30.29	30.28	30.27	30.26	30.28	
12	Time to consume 50 cc. of fuel	t	sec.	92.7	92.5	92.1	92.1	91.2	92.1	
13	Injection point before TDC.		degrees	37.0	39.5	42.0	44.5	47.0		
14	Maximum cycle pressure	p _{max}	lb./in ² .a.	796	891	990	1069	1134		
15	Exhaust gas appearance			clear	clear	clear	clear	clear		
16	Exhaust gas temperature	t _e	°F	937	923	903	896	894		

Calculated performance data for this series of test runs are listed in Table 61 , page 148.

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RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 57 - Calculations

Reference	Performance quantity	Symbol	Unit	Experimental Results				
Table 52	Test Run No.			82	83	84	85	86
Eq. (56)	Air flow multiplier $\sqrt{\frac{520}{460 + t_a} \cdot \frac{p_a}{29.92}} =$	M		1.000	1.000	1.000	1.000	1.000
Eq. (55)	Air consumption $(0.2515 M \sqrt{h_{avg}}) =$	L_{avg}	lb./min.	1.175	1.175	1.175	1.175	1.175
Eq. (54)	Fuel fraction $(\frac{81.04}{t_{avg} \cdot L_{avg}}) =$	F		0.353	0.353	0.353	0.353	0.353
Eq. (48)	Brake mean effective pressure (7.31 W) =	BMEP	lb./in ² .	26.3	28.1	27.4	26.6	25.7
Eq. (48A)	Friction mean effective pressure (7.31 W _m) =	FMEP	lb./in ² .	50.0	50.0	50.0	50.0	50.0
Eq. (48B)	Indicated mean effective pressure (BMEP + FMEP) =	IMEP	lb./in ² .	76.3	78.1	77.4	76.6	75.7
Eq. (53)	Thermal efficiency ($3.35 \times 10^{-5} \times t_{avg} \times IMEP$) =	η		0.499	0.511	0.506	0.502	0.495

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RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 58 - Calculations

Reference	Performance quantity	Symbol	Unit	Experimental Results				
Table 53	Test Run No.			91	92	93	94	95
Eq. (56)	Air flow multiplier $\sqrt{\frac{520}{460 + t_a} \cdot \frac{p_a}{29.92}} =$	M		1.007	1.007	1.007	1.007	1.007
Eq. (55)	Air consumption $\left(0.2515 M \sqrt{h_{avg}} \right) =$	L_{avg}	lb./min.	1.174	1.174	1.174	1.174	1.174
Eq. (54)	Fuel fraction $\left(\frac{81.04}{t_{avg} \cdot L_{avg}} \right) =$	F		0.448	0.448	0.448	0.448	0.448
Eq. (48)	Brake mean effective pressure (7.31 W) =	BMEP	lb./in ² .	44.7	47.2	47.3	46.5	44.4
Eq. (48A)	Friction mean effective pressure (7.31 W _m) =	FMEP	lb./in ² .	50.0	50.0	50.0	50.0	50.0
Eq. (48B)	Indicated mean effective pressure (BMEP + FMEP) =	IMEP	lb./in ² .	94.7	97.2	97.3	96.5	94.4
Eq. (53)	Thermal efficiency ($3.35 \times 10^{-5} \times t_{avg} \times \text{IMEP}$) =	η		0.489	0.502	0.503	0.498	0.487

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RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 59 - Calculations

Reference	Performance quantity	Symbol	Unit	Experimental Results				
Table 54	Test Run No.			101	102	103	104	105
Eq. (56)	Air flow multiplier $\sqrt{\frac{520}{460 + t_a} \cdot \frac{p_a}{29.92}} =$	M		1.006	1.006	1.006	1.006	1.006
Eq. (55)	Air consumption $(0.2515 M \sqrt{h_{avg}}) =$	L_{avg}	lb./min.	1.173	1.173	1.173	1.173	1.173
Eq. (54)	Fuel fraction $(\frac{81.04}{t_{avg} \cdot L_{avg}}) =$	F		0.552	0.552	0.552	0.552	0.552
Eq. (48)	Brake mean effective pressure (7.31 W) =	BMEP	lb./in ² .	62.3	64.0	64.6	65.6	64.5
Eq. (48A)	Friction mean effective pressure (7.31 W _m) =	FMEP	lb./in ² .	50.0	50.0	50.0	50.0	50.0
Eq. (48B)	Indicated mean effective pressure (BMEP + FMEP) =	IMEP	lb./in ² .	112.3	114.0	114.6	115.6	114.5
Eq. (53)	Thermal efficiency ($3.35 \times 10^{-5} \times t_{avg} \times \text{IMEP}$) =	η		0.471	0.478	0.481	0.484	0.480

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RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 60 - Calculations

Reference	Performance quantity	Symbol	Unit	Experimental Results				
Table 55	Test Run No.			110	111	112	113	114
Eq. (56)	Air flow multiplier $\sqrt{\frac{520}{460 + t_a} \cdot \frac{p_a}{29.92}} =$	M		1.005	1.005	1.005	1.005	1.005
Eq. (55)	Air consumption $(0.2515 M \sqrt{h_{avg}}) =$	L_{avg}	lb./min.	1.180	1.180	1.180	1.180	1.180
Eq. (54)	Fuel fraction $(\frac{81.04}{t_{avg} \cdot L_{avg}}) =$	F		0.638	0.638	0.638	0.638	0.638
Eq. (48)	Brake mean effective pressure (7.31 W) =	BMEP	lb./in ² .	70.7	73.4	75.0	75.3	75.6
Eq. (48A)	Friction mean effective pressure (7.31 W _m) =	FMEP	lb./in ² .	50.0	50.0	50.0	50.0	50.0
Eq. (48B)	Indicated mean effective pressure (BMEP + FMEP) =	IMEP	lb./in ² .	120.7	123.4	125.0	125.3	125.6
Eq. (53)	Thermal efficiency ($3.35 \times 10^{-5} \times t_{avg} \times \text{IMEP}$) =	η		0.436	0.446	0.452	0.453	0.454

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RICARDO E6/S RESEARCH ENGINE TESTS

EXPERIMENTAL RESULTS

Table 61 - Calculations

Reference	Performance quantity	Symbol	Unit	Experimental Results				
Table 56	Test Run No.			122	123	124	125	126
Eq. (56)	Air flow multiplier $\sqrt{\frac{520}{460 + t_a} \cdot \frac{p_a}{29.92}}$	M		1.000	1.000	1.000	1.000	1.900
Eq. (55)	Air consumption $(0.2515 M \sqrt{h_{avg}})$	L _{avg}	lb./min.	1.174	1.174	1.174	1.174	1.174
Eq. (54)	Fuel fraction $(\frac{81.04}{t_{avg} \cdot L_{avg}})$	F		0.749	0.749	0.749	0.749	0.749
Eq. (48)	Brake mean effective pressure (7.31 W)	BMEP	lb./in ² .	77.1	79.2	82.6	82.4	83.4
Eq. (48A)	Friction mean effective pressure (7.31 W _m)	FMEP	lb./in ² .	50.0	50.0	50.0	50.0	50.0
Eq. (48B)	Indicated mean effective pressure (BMEP + FMEP)	IMEP	lb./in ² .	127.1	129.2	132.6	132.4	133.4
Eq. (53)	Thermal efficiency ($3.35 \times 10^{-5} \times t_{avg} \times \text{IMEP}$)	η		0.392	0.398	0.409	0.408	0.412

APPENDIX 6

COMPARISON OF THEORETICAL AND EXPERIMENTAL PERFORMANCE

Table 62 - Maximum cycle pressure $p_{\max} = 900 \text{ lb./in}^2 \text{ abs.}$
 Compression ratio $r = 20$
 Fuel fraction $F =$ as indicated in the table

fuel fraction (F)	cycle efficiency (η_a) Figure 16	thermal efficiency (η_{th}) Figure 16	efficiency ratio $\eta_r = \frac{\eta_a}{\eta_{th}}$	mean eff. ^{ve} pressure (IMEP) _a Figure 14	indicated mean eff. ^{ve} pressure (MEP) _{th} Figure 14	mean effective pressure ratio $(MEP)_r = \frac{(IMEP)_a}{(MEP)_{th}}$ $(MEP)_r = 0.899 \eta_r$ Equation (39)	
0.35	0.511	0.598	0.855	78.0	100.0	0.780	0.769
0.45	0.502	0.580	0.865	97.0	125.5	0.773	0.778
0.55	0.4755	0.562	0.846	113.5	148.5	0.764	0.761
0.65	0.4445	0.544	0.817	123.3	170.0	0.725	0.735
0.75	0.400	0.526	0.761	130.0	190.0	0.684	0.685

Table 63 - Maximum cycle pressure $p_{\max} = 1000 \text{ lb./in}^2 \text{ abs.}$
 Compression ratio $r = 20$
 Fuel fraction $F =$ as indicated in the table

0.35	0.5075	0.608	0.835	77.8	102.0	0.763	0.751
0.45	0.502	0.590	0.851	97.5	128.0	0.762	0.765
0.55	0.481	0.5725	0.840	115.0	151.5	0.760	0.755
0.65	0.4505	0.555	0.811	125.0	174.0	0.718	0.729
0.75	0.407	0.5375	0.757	132.5	194.5	0.681	0.681

APPENDIX 6

COMPARISON OF THEORETICAL AND EXPERIMENTAL PERFORMANCE

Table 64 - Maximum cycle pressure $p_{\max} = 1100 \text{ lb./in}^2 \text{ abs.}$

Compression ratio $r = 20$

Fuel fraction $F = \text{as indicated in the table}$

fuel fraction (F)	cycle efficiency (η_a) Figure 16	thermal efficiency (η_{th}) Figure 16	efficiency ratio $\eta_r = \frac{\eta_a}{\eta_{th}}$	mean eff. ^{ve} pressure (IMEP) _a Figure 14	indicated mean eff. ^{ve} pressure (MEP) _{th} Figure 14	mean effective pressure ratio	
						$(MEP)_r = \frac{(IMEP)_a}{(MEP)_{th}}$	$(MEP)_r = 0.899 \eta_r$ Equation (39)
0.35	0.4975	0.6155	0.809	76.2	103.5	0.736	0.727
0.45	0.497	0.598	0.831	96.5	129.8	0.743	0.747
0.55	0.4825	0.5805	0.831	115.3	154.0	0.748	0.747
0.65	0.454	0.564	0.805	125.7	177.0	0.711	0.724
0.75	0.411	0.5465	0.752	133.4	197.5	0.675	0.676

Table 65 - Maximum cycle pressure $p_{\max} = 1200 \text{ lb./in}^2 \text{ abs.}$

Compression ratio $r = 20$

Fuel fraction $F = \text{as indicated in the table}$

0.35	0.4815	0.622	0.774	73.3	104.5	0.701	0.696
0.45	0.487	0.604	0.806	94.5	131.0	0.721	0.725
0.55	0.478	0.587	0.815	114.2	155.5	0.734	0.733
0.65	0.453	0.5705	0.794	125.5	179.0	0.701	0.714
0.75	0.4105	0.554	0.741	133.7	200.0	0.669	0.666

APPENDIX 7

MATERIAL BASES COMMONLY USED FOR

THERMODYNAMIC QUANTITIES

In the present analysis it was found that different material bases were used for the various thermodynamic charts and these tended to constitute an appreciable source of confusion.

When formulating expressions for cycle performance due allowance must be made for the appropriate material base of the thermodynamic quantities under consideration. Also, the correct transfer of readings from one chart to another requires a thorough understanding of the implications of a change in material base. This aspect of thermodynamic cycle analysis is discussed below.

The modified air chart is a basic chart for the properties of one pound-mole of pure dry air ¹). In addition to its basic plot supplementary plots are provided to determine the properties of one pound-mole of a mixture of air and fuel, and their products of combustion; that is, for any mixture of air plus burned and/or unburned fuel.

The burned mixture charts, on the other hand, are constructed on a material base of one pound of air constituents, plus a fuel

quantity corresponding to the mixture strength.

The relevant ratios of the various components in the mixture determine the magnitude of its thermodynamic properties.

Consequently, the following ratios must be taken into account :

F - the weight ratio of actual fuel used to
the fuel required for stoichiometrically
complete combustion.

f - the weight ratio of the unpurged combustion
products to the total mixture.

The above definitions give rise to a number of basic
relationships that can be listed as follows :

Fraction of theoretical fuel (F)

1. The burned mixture charts give thermodynamic properties of the products of combustion of a typical fuel $(CH_2)_x$ for selected values of the fuel fraction (F).

The stoichiometrically correct quantity of $(CH_2)_x$ to combine with one pound of dry air is 0.06775 pound. Hence, in all cases where $(CH_2)_x$ is assumed to be the fuel used and where it is available in the fraction (F) of the theoretically correct quantity the relationship can be expressed thus :

1 lb. of dry air combines with $(0.06775F)$ lb. of fuel to give $(1 + 0.06775F)$ lb. of mixture.

2. This latter quantity is the material base of the burned mixture charts. Some confusion seems to derive from the fact that the chart designers have designated this material base as one pound of dry air constituents, plus appropriate fuel.

It should be noted, however, that the term "air constituents" applies to "air" of any composition, i.e. the oxygen originally present may have partially or wholly combined with fuel. Also, the term "appropriate fuel" implies fuel either in the burned or in the unburned state, or both, as in the case of rich mixtures.

3. A convenient procedure is to consider one pound of dry air constituents to be present at the beginning of compression in the cycle, and to use this quantity as a conventional basis for the cycle computation.

Since $(1 + 0.06775F)$ lb. of mixture contains 1 lb. of (original, dry) air constituents, the thermodynamic properties of the working fluid as obtained from the burned mixture charts have the same numerical value when expressed on the basis of 1lb. of air constituents.

Likewise, cycle performance can be expressed in either base without

altering its numerical value. For instance the designers of the thermodynamic charts evaluate the cycle output (W) from chart quantities, and subsequently state its numerical value in BTU/lb. of air constituents.

4. When the occurrence of chemical dissociation necessitates the use of both the modified air chart and the burned mixture charts, a transfer of readings from the one chart to the other is required. This procedure is facilitated by an auxiliary plot of chart conversion factors (y) for the various values of (F) and (f). As a consequence of the foregoing considerations the unit of the conversion factor (y) is pound-moles of mixture per pound of air constituents.

Weight fraction of residual gases (f).

1. Due to incomplete purging in a cyclical operation the mass of gases present in the cylinder at the end of the intake stroke consists of (f) lb. of residual gas plus $(1 - f)$ lb. of fresh charge. That is, $(1 - f)$ lb. of fresh charge is present for each pound of air constituents at the beginning of the compression stroke.

In the case of the Otto cycle the fresh charge is a mixture of fresh air and fuel, whereas in the mixed cycle it comprises fresh air only.

2. Since 1lb. of original, dry air constituents contains only $(1-f)$ lb. of fresh charge, an adjustment is required if cycle performance quantities are to be expressed per lb. of fresh charge. For instance, if the cycle output is (W) BTU/lb. of air constituents, that same output is $\left(\frac{W}{(1-f)}\right)$ BTU/lb. of fresh charge.

Examples.

With the aid of the above relationships the expressions used for cycle performance can now be verified by considering the appropriate units in the equations for, say, gross work output and thermal efficiency.

Gross work output :

$$W_g = \frac{h_3 - e_4 - y p_3 V_2 \frac{144}{778} - y (E_2 - E_1)}{(1 - f)} \dots\dots\dots(4)$$

$$W_g = \frac{\frac{\text{BTU}}{\text{lb. of air constituents}}}{\frac{\text{lb. of fresh air}}{\text{lb. of air constituents}}}$$

$$W_g = \frac{\text{BTU}}{\text{lb. of fresh air}}$$

It can be seen that the unit used for gross work output in Equation (4) conforms to the outcome of the above dimensional

examination.

Thermal efficiency :

$$\eta_{th} = \frac{W_n}{0.06775F \times Q_c} \dots\dots\dots(17)$$

$$\eta_{th} = \frac{\frac{\text{BTU}}{\text{lb. of fresh air}}}{\frac{\text{lb. of fuel}}{\text{lb. of fresh air}} \times \frac{\text{BTU}}{\text{lb. of fuel}}}$$

It can be seen from this dimensionless result that Equation (17) for thermal efficiency is dimensionally correct.