1960

An analysis of the otto cycle taking into consideration the effect of varying operating parameters

S. .E. Bonamy
Wollongong University College

Follow this and additional works at: <https://ro.uow.edu.au/theses>

University of Wollongong

Copyright Warning

You may print or download ONE copy of this document for the purpose of your own research or study. The University does not authorise you to copy, communicate or otherwise make available electronically to any other person any copyright material contained on this site.

You are reminded of the following: This work is copyright. Apart from any use permitted under the Copyright Act 1968, no part of this work may be reproduced by any process, nor may any other exclusive right be exercised, without the permission of the author. Copyright owners are entitled to take legal action against persons who infringe their copyright. A reproduction of material that is protected by copyright may be a copyright infringement. A court may impose penalties and award damages in relation to offences and infringements relating to copyright material.

Higher penalties may apply, and higher damages may be awarded, for offences and infringements involving the conversion of material into digital or electronic form.

Unless otherwise indicated, the views expressed in this thesis are those of the author and do not necessarily represent the views of the University of Wollongong.

Recommended Citation

Bonamy, S. .E., An analysis of the otto cycle taking into consideration the effect of varying operating parameters, Doctor of Philosophy thesis, School of Mechanical Engineering, University of Wollongong, 1960. <https://ro.uow.edu.au/theses/2064>

AN ANALYSIS OF THE OTTO CYCLE TAKING INTO
CONSIDERATION THE EFFECT OF VARYING
OPERATING PARAMETERS

BY

S. E. BONAMY, B.E. (Syd.), M.Sc. (Birm.), A.S.T.C.,
A.M.I.E. (Aust.), A.M.I.Mech.E.

Thesis Submitted for the Degree of

DOCTOR OF PHILOSOPHY

School of Mechanical Engineering,
WUC University of New South Wales.

1960.

013432

SYNOPSIS

The present report concerns a study of the Otto cycle with the aid of thermodynamic charts. The charts used take into consideration factors such as residual gases from the previous cycle, variations in specific heats, dissociation and reassociation.

A large number of cycles is analysed for a range of values of operating parameters such as manifold pressure ratio, fuel-air ratio, compression ratio and temperature of incoming fresh charge. From a graphical analysis of the results, equations are developed which enable the temperature at the beginning of compression and the weight fraction of residual gases from the previous cycle, to be estimated accurately for the range of operating parameters used. These quantities are usually assumed when analysing a cycle from the charts, necessitating a lengthy trial and error process. Equations are developed also for calculating the mean effective pressure and thermal efficiency of a cycle, irrespective of the values of the operating parameters. These equations are considered of great value to the research worker or designer of internal combustion engines, since they supersede the lengthy graphical process of calculating engine performance from thermodynamic charts.

A new approach is developed for the analysis of cycles from charts in cases where the intake pressure is greater than the exhaust pressure or when the engine is supercharged.

An examination is made of the effects of variations in operating parameters on the performance of the Otto cycle and on the magnitude of the maximum cycle pressures and temperatures. The values obtained are compared graphically with those of other workers.

Experiments are conducted to examine the effects of increased back pressure on engine performance for a range of intake pressures. The experimental values obtained for mean effective pressure and thermal efficiency are compared with those calculated from the theoretical work.

ACKNOWLEDGMENTS

The writer wishes to express his sincere thanks to Professor A. H. Willis, Dean of the Faculty of Engineering, for the opportunity of conducting this research in his School and for his continued interest in the subject.

The invaluable assistance given by Mr. R. W. Upfold, in the first series of computations and in the examination of the effects of back pressure on engine performance is gratefully acknowledged.

Thanks are due also to Mr. R. Kinnell, Technical Officer, for his assistance in the preparation of the graphs.

LIST OF SYMBOLS

C	A factor allowing for molecular contraction due to combustion
E	Total Internal Energy (sensible plus chemical), B.T.U./lb. mol.
E^0	Total Internal Energy at zero pressure for perfect gas conditions, B.T.U./lb. mol.
e	Total Internal Energy (sensible plus chemical), B.T.U./lb. 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; equals weight fraction of fuel in the burned state at the beginning of compression
G_1	Gas Constant = 1545.4 ft.lb./mol. $^{\circ}$ R.
H	Total Enthalpy (sensible plus chemical), B.T.U./lb. mol.
h	Total Enthalpy (sensible plus chemical), B.T.U./lb. of air constituents
P	Pressure, lb./in. 2 abs.
q_v	Residual heat of combustion of burned mixtures cooled below 2880 $^{\circ}$ R., B.T.U./lb. of air constituents
r	Compression ratio
S	Entropy, B.T.U./lb. mol. $^{\circ}$ R.
s	Entropy, B.T.U./lb. air constits. $^{\circ}$ R.
T	Temperature, degrees Rankine

- V Volume, c.ft./lb. mol.
- v Volume, c.ft./lb. of air constituents
- X Manifold pressure factor = $\frac{P_b - 1.10}{P_i}$
- y Chart conversion factor, lb. mols. of mixture per lb. of air constituents
- z Mols. of fresh charge entering in portion of a cycle for which exhaust pressure is less than intake pressure
- Y Ratio of specific heats

Suffixes

- b Back or exhaust manifold
- fr. Fresh charge
- i Intake
- S Sensible portion of thermodynamic function only, that is, chemical portion is excluded

	<u>C O N T E N T S</u>	Page
	SYNOPSIS	i
	ACKNOWLEDGMENTS	iii
	LIST OF SYMBOLS	iv
SECTION 1.	INTRODUCTION	1
SECTION 2.	THERMODYNAMIC CHARTS	6
	(i) Modified Air Chart	6
	(ii) Burned Mixture Charts	7
SECTION 3.	SOLUTION OF THE OTTO CYCLE	9
	(i) Exhaust Pressure greater than Intake Pressure (Throttled Cycle)	9
	(ii) Exhaust Pressure less than Intake Pressure	11
	(iii) Work Output and Efficiency	16
SECTION 4.	RANGE OF CYCLES ANALYSED	20
	(i) Influence of Manifold Pressure Ratio	20
	(ii) Influence of Compression Ratio, Fuel-Air Ratio and Fresh Charge Temperature	20
SECTION 5.	WEIGHT FRACTION OF RESIDUAL GASES	22
SECTION 6.	TEMPERATURE OF CHARGE AT BEGINNING OF COMPRESSION	26
SECTION 7.	WORK OUTPUT PER POUND OF FRESH CHARGE	31
SECTION 8.	SUMMARY OF EQUATIONS	33
SECTION 9.	EFFECTS OF CHANGING OPERATING VARIABLES ON ENGINE PERFORMANCE	37
	(i) Intake and Exhaust Pressures	37

	<u>C O N T E N T S</u> (Cont'd)	Page
SECTION 9 (Cont'd)	(ii) Fresh Charge Temperature	39
	(iii) Compression Ratio	40
	(iv) Fuel-Air Ratio	42
	(v) Comparative Results	44
SECTION 10	EFFECTS OF CHANGING OPERATING VARIABLES ON MAXIMUM CYCLE TEMPERATURES AND PRESSURES	48
	(i) Intake and Exhaust Pressures	48
	(ii) Fresh Charge Temperature	50
	(iii) Compression Ratio	52
	(iv) Fuel-Air Ratio	53
SECTION 11	EXPERIMENTAL INVESTIGATION ON THE EFFECTS OF BACK PRESSURE ON ENGINE PERFORMANCE	55
	(i) Description of Apparatus	56
	(ii) Experimental Procedure	58
	(iii) Experimental Results	59
SECTION 12	COMPARISON BETWEEN EXPERIMENTAL AND THEORETICAL RESULTS	60
SECTION 13	CONCLUSIONS	64
	REFERENCES	72
SECTION 14	APPENDICES	74
	1. Exhaust Pressure greater than Intake Pressure - Sample Calculation	75
	2. Exhaust Pressure less than Intake Pressure - Sample Calculation	79
	3. Results of Series 1 of Computations from Thermodynamic Charts, Tables 1 to 10	82

C O N T E N T S (Cont'd)

Page

SECTION 14
(Cont'd)

4. Results of Series 2 of Computations from Thermodynamic Charts, Tables 11 to 22	90
5. Calculated Values of Output and Thermal Efficiency from Equations summarized in Section 8, Table 23	99
6. Results of Morse Test, Table 24	100
7. Results of Increased Back Pressure Tests, Tables 25 to 30	101

1. INTRODUCTION

An approximate theoretical analysis of the Otto cycle is often made on the basis of air as the working substance. The air is assumed to behave as a "perfect gas" and losses in the induction and exhaust systems are considered negligible. Thermodynamic properties at various points in the cycle are calculated using "perfect gas" laws and heat supply is assumed to be from an external source. Air Standard Efficiency (A.S.E.) which is a function only of compression ratio, r and specific heat ratio, γ is based on such an analysis. Although this method is useful in representing a simple mathematical limitation of the actual cycle, it does not give a true theoretical estimate of the performance to be expected.

Early attempts to improve the air cycle analysis introduced allowances to be made for specific heat variations with temperature¹. On this basis an intrinsic energy versus temperature chart, with pressure and volume scales superimposed, was prepared for gas engine mixtures², and diagrams relating mean effective pressure, maximum cycle pressure and efficiency were constructed³. However, it was not until the characteristics of the actual working medium were examined that a satisfactory theoretical analysis was made. The working substance varies in a petrol engine in that during the compression stroke it comprises a mixture of fuel-vapour, air and residual gases from the previous cycle whereas during the expansion stroke it is mainly burnt gases. In addition to

specific heat variations, consideration must be given to "dissociation" effects at the high combustion temperatures and "reassociation" effects during the early part of the expansion stroke. Tizard and Pye⁴ were among the first to investigate the effects of dissociation on maximum cycle temperatures but their method, involving the use of dissociation constants and volumetric heats, was one of trial and error which proved rather cumbersome.

A temperature-internal energy chart was introduced⁵ which, by the choice of suitable scales, could be used to estimate the work done by, and the thermal efficiency of, internal combustion engines; specific heat variations, dissociation and fuel-air ratios were taken into account, but the effects of residual gas, found to be most important in the present work, were neglected in the analyses of cycles.

The introduction of thermodynamic charts, (see refs. 6 to 11) giving properties of products of combustion of various fuels with air, provided a new and simplified technique for the solution of internal combustion engine cycles. The charts found to be the most satisfactory for the solution of Otto cycles and used exclusively in the analysis presented here are those at first introduced by Hershey, Eberhardt and Hottel¹⁰ and later expanded and reproduced for the National Advisory Committee for Aeronautics by Hottel, Williams and Satterfield¹¹. These latter charts have been found to be extremely accurate and take into account factors such as

variable specific heats, chemical dissociation, residual gases from the previous cycle and fuel-air ratios.

The idealized Otto cycle is made up of the following four basic steps:-

- (a) Induction: Isoenthalpic mixing of residual gases from the previous cycle with a fresh charge of fuel vapour and air.
- (b) Compression: Isentropic compression of the mixture.
- (c) Combustion: A constant volume and therefore a constant internal energy process.
- (d) Expansion and Exhaust: Isentropic expansion of the burned products to a point at which the exhaust valve opens, followed by further expansion of portion of the gas in expelling the remainder.

The solution of a throttled Otto cycle, using thermodynamic chart is, generally, one of trial and error; for a chosen fresh charge temperature, intake and exhaust pressure, fuel-air ratio and compression ratio, a weight fraction of residual gases and a temperature of charge at the beginning of compression are assumed and the cycle is worked through to obtain new values of these quantities. These new values are checked with the assumed values and, if found unsatisfactory, the complete cycle is repeated using the new values. Depending on the accuracy of the initial assumptions, it may be necessary to work through the cycle as many as four times before attaining the desired accuracy.

Hottel and his co-workers¹¹ have given empirical equations from which the weight fraction of residual gases and temperature of charge at the beginning of compression may be approximated; these involve an assumption of the residual gas temperature and do not take into account variations in fuel-air ratio. Their equations have been found unreliable and are not used in the present investigation.

From an analysis in the present work of a large number of Otto cycles for a range of operating parameters, such as intake and exhaust pressures, fresh charge temperatures, fuel-air ratios and compression ratios, new equations are developed for estimating weight fraction of residual gases and temperature of fresh charge at the beginning of compression in terms of these parameters. A combination of these equations with others, developed for work output per pound of fresh charge in terms of the operating parameters, enables simple relationships to be obtained for mean effective pressure and thermal efficiency in terms of compression ratio, fuel-air ratio, fresh charge temperature and manifold pressures. These equations may be used to supersede the long-handed method of calculating engine cycles from the thermodynamic charts and may be used for analysing either throttled or supercharged Otto cycles. An alternative method to that used by Hottel and his co-workers¹¹ is developed for the analysis, from the charts, of supercharged cycles or cycles in which the intake pressure is greater than the exhaust pressure; this offers a

considerable saving in time.

The influence of operating parameters on thermal efficiency and mean effective pressure is examined using the results from the thermodynamic charts¹¹ and the equations developed.

Results from experimental work on the effects of increased back pressure on the performance of a multi-cylinder engine at various intake pressures are found to reveal close ~~agreement~~ ^{agreement} with those obtained theoretically.

2. THERMODYNAMIC CHARTS

The charts prepared by Hottel, Williams and Satterfield¹¹ include: (i) A Modified Air Chart which is used for compression processes and (ii) Burned Mixture Charts, for a range of fuel-air ratios, which are used for expansion processes.

(i) Modified Air Chart

The Modified Air Chart is essentially a temperature versus entropy chart for air, modified to give thermodynamic properties not only of air but also of fuel-air mixtures for a wide range of mixture strengths and containing various quantities of residual gases. It is plotted on the basis of one pound-mole of mixture since the pressure, volume and temperature relationships per pound-mole for the various mixture strengths are substantially the same. The internal energy and enthalpy values given are those for the sensible portion of the thermodynamic function only (i.e. the energy of combustion is excluded) since the chart covers a range of mixtures differing in chemical energy content. The fuel chosen as the basis of construction of the chart for the various fuel-air mixtures is Octene (C_8H_{16}) since its molecular weight is close to that of fuels used in petrol and diesel engines and in gas turbines. The error involved in using this fuel instead of the actual fuel is negligible since the fuel vapour represents only a small portion of the total volume of mixture. No restriction is placed on the energy of combustion of the actual fuel used since, as mentioned above,

only the sensible portions of enthalpy and internal energy are plotted.

The temperature scale for the chart ranges from 360°R. to 2520°R. , the base temperature of 360°R. corresponding to zero enthalpy, and the upper temperature of 2520°R. being that below which the effects of chemical dissociation are considered negligible. Supplementary plots are provided on the chart giving mixture correction factors to be applied to enthalpy, internal energy, entropy and volume for the weight fraction of residual gases present and the fuel-air ratio of the mixture.

(ii) Burned Mixture Charts.

These charts, plotted for various fuel-air ratios, are internal energy versus entropy charts plotted on the basis of one pound of dry air constituents plus the corresponding fuel. The six charts which are of interest here give properties of the products of combustion of Octene with air for ratios of fuel used to that required for complete combustion of 0.8, 0.9, 1.0, 1.1, 1.2 and 1.5. A mass basis is used in favour of a mole basis since dissociation causes a change in the number of moles per unit mass at high temperatures.

Lines of constant volume, constant pressure, constant enthalpy and constant temperature are superimposed on the charts; the internal energy and enthalpy values include the chemical energy obtainable from the combination of the dissociated components at the corresponding high temperatures.

The Burned Mixture Charts were not plotted at the same time as the Modified Air Chart and different temperature bases have been used; the Burned Mixture Charts have been plotted on a temperature base of 560°R. , corresponding to a sensible internal energy value of zero, and due allowance must be made for this when converting from one chart to the other.

3. SOLUTION OF THE OTTO CYCLE

(i) Exhaust Pressure greater than Intake Pressure (Throttled Cycle).

In general, the procedure adopted in solving a throttled Otto cycle with the aid of the charts is, briefly, as follows:

P-V. Diagram for the Otto Cycle. Exhaust Pressure greater than Intake Pressure.

Referring to the P-V diagram, Figure 1, a temperature T_1 and weight fraction, f of residual gases in the equivalent burned mixture at the beginning of compression are assumed. For the chosen value of intake pressure P_1 and temperature T_1 , a point is located on the Modified Air Chart and the corresponding thermodynamic properties read off. Properties at point 2 are then found by following an isentropic compression process on the chart to volume $V_2 = V_1/r$, where r = compression ratio, allowance being made for mixture

strength and presence of residual gases. The internal energy e_3^* , per pound of air constituents is found by adding the energy of combustion to the sensible energy E_{s2} allowance being made for a chart conversion factor, y . Since volume v_3 , that is, $V_2 y_2$ per pound of air constituents, and e_3 are known, point 3 can be located on the appropriate Burned Mixture Chart. Isentropic expansion to v_4 , that is, $v_3 r$ gives the properties at point 4.

The gas remaining in the cylinder at point 5 is assumed to have undergone an isentropic expansion within the cylinder, from condition 4, doing work on the expelled gas during the process. Its condition corresponds to that at point 4'. The exhaust process 5-6 is considered a constant pressure adiabatic one, so that the properties per pound of air constituents at point 6 are the same as at 4'. Properties at point 4' are found from the chart by continuing the isentropic expansion process to exhaust pressure. The residual gases remaining in the cylinder at the completion of the exhaust stroke are assumed to expand isentropically to intake pressure, process 6-6'. Properties at point 6', therefore, can be read from the chart.

The remainder of the suction stroke, 6'-1, is considered

* Upper and lower case letters apply to the Modified Air Chart and Burned Mixture Charts respectively.

also, as a constant pressure adiabatic one, so that addition of the enthalpy of the weight fraction f of residual gases to the enthalpy of the weight fraction $(1 - f)$ of fresh charge gives the enthalpy per pound of air constituents at point 1. Application of a chart conversion factor allows the enthalpy per mol at point 1 to be calculated, and the corresponding temperature T_1 is read from the Modified Air Chart for the appropriate mixture strength. Also, since $v_4/v_{4'}$ gives the fraction of burned mixture remaining in the cylinder at 5, the fraction f of residual gases is given by $v_4/rv_{4'}$.

The new values of temperature T_1 and weight fraction of residual gases f are checked with the assumed values and, if found unsatisfactory, the complete cycle is repeated using the new values.

A sample calculation of a throttled Otto cycle is given in Appendix 1.

(ii) Exhaust Pressure less than Intake Pressure.

In cases where the exhaust pressure is less than the intake pressure, the Otto cycle may be considered analogous to that of a supercharged engine. Unfortunately, the method described by Hottel, Williams and Satterfield¹¹ for the solution of such a cycle is extremely laborious and might take an inexperienced investigator as long as twelve hours to solve one cycle. A brief description of their method is as follows:

P-V. Diagram for the Otto Cycle, Exhaust Pressure less than Intake Pressure.

Referring to the P-V diagram, Figure 2, the temperature of the residual gases, T_6 is assumed and the corresponding sensible internal energy E_{S6} and volume V_6 per mol are read from the Modified Air Chart. Step 6-X is considered as a constant volume mixing process of one mol of residual gases with z mols of fresh charge, so that the sensible internal energy per mol of resultant mixture at X is given by:

$$E_{SX} = \frac{zH_{Sfr.} + E_{S6}}{z + 1} \dots\dots\dots(1)$$

where $H_{Sfr.}$ = sensible enthalpy of fresh charge per mol.

Also, the volume per mol of mixture is given by:

$$V_X = \frac{V_6}{z + 1} \dots\dots\dots(2)$$

and the weight fraction of residual gases in the equivalent

burned mixture at X by

$$f' = \frac{1}{1 + Cz} \dots\dots\dots(3)$$

where C is a factor allowing for molecular contraction due to combustion, which can be determined from a supplementary plot on the Modified Air Chart.

A trial and error process is used to find a value of z that gives agreement between the values of E_{SX} and V_X as calculated from equations (1) and (2) and those read from the Modified Air Chart for the appropriate f', Equation (3). The corresponding sensible enthalpy H_{SX} is read from the chart.

The sensible enthalpy, H_{S1} , per mol of mixture, at the beginning of compression, is calculated by considering the mixing at constant pressure, during step X-1, of z_1 mols of fresh charge with $(1 + z)$ mols of mixture at condition X. Hence $(1 + z + z_1)H_{S1} = (1 + z)H_{SX} + z_1 H_{Sfr}$. $\dots\dots\dots(4)$

Also, the volume V_1 , per mol, and the weight fraction of residual gases f at 1, are obtained from:

$$V_1 = \frac{rV_6}{1 + z + z_1} \dots\dots\dots(5)$$

$$\text{and } f = \frac{1}{1 + C(z + z_1)} \dots\dots\dots(6)$$

A further trial and error process, involving a selection of z_1 to satisfy H_{S1} and V_1 from equations (4) and (5) and values read from the Modified Air Chart for the appropriate f, reveals thermodynamic properties at the beginning of compression. The remainder of the cycle is then worked through in the

manner described for the throttled case. The temperature $T_6 = T_4$, is read from the Burned Mixture Chart and the weight fraction $f = \frac{v_4}{rv_{4'}}$, calculated. These values are checked with the temperature T_6 , initially assumed, and the value of f from equation (6) respectively, and if found unsatisfactory the complete process is repeated.

Alternative Method.

A simplified, yet very accurate, method that may be used in place of the one described above, is as follows:

Referring again to Figure 2, the initial assumptions are the temperature T_1 , and the weight fraction f of residual gases in the equivalent burned mixture at the beginning of compression. Thermodynamic properties at points 2, 3, 4 and 4' are then obtained from the Modified Air and Burned Mixture Charts in the usual manner. An immediate check of $f = \frac{v_4}{rv_{4'}}$ is then possible, before proceeding with the remainder of the cycle.

FIGURE 3

The Alternative Charging Process. Exhaust Pressure less than Intake Pressure.

The residual gas at 6 is assumed compressed isentropically to condition 6', Figure 3, within the cylinder by portion of the incoming charge, a volume $(V_X - V_{6'})$ being required for the process. Since the properties per pound of air constituents at 6 are considered as being the same as at 4', Figure 2, the properties at 6' can be found from the Burned Mixture Chart.

An additional volume $(V_1 - V_X)$ is taken into the cylinder during the charging stroke, making the total volume of fresh charge taken in per cycle, $(V_1 - V_{6'})$.

Considering step 6' - 1 a constant pressure adiabatic mixing process, the sensible enthalpy h_{S1} , per pound of air constituents is calculated from

$$h_{S1} = fh_{S6'} + (1 - f)y_{fr}.H_{S_{fr.}} \dots\dots\dots(7)$$

where

$h_{S6'}$ = sensible enthalpy of residual gases at 6', per pound of air constituents.

$H_{S_{fr.}}$ = sensible enthalpy of fresh charge per mol.

$y_{fr.}$ = fresh charge conversion factor, mols per pound of air constituents.

Also, the sensible enthalpy per mol of mixture at 1

$$H_{S1} = \frac{h_{S1}}{y_1} \dots\dots\dots(8)$$

where y_1 = conversion factor for mixture at 1, mols per pound of air constituents.

Temperature T_1 , corresponding to H_{S1} on the Modified Air Chart can now be read and checked with the assumed value. The cycle calculations are then repeated using the new value of T_1 until the desired accuracy is attained.

It has been found, in solving a number of cycles by the two methods described, that the latter offers a saving in time of up to fifty per cent. This saving is realised because the method eliminates the tedious trial and error process of finding the mols of fresh charge entering the cylinder during both the constant volume and constant pressure mixing processes. Also a check can be made of the weight fraction f of residual gases early in the computations.

When applying this method to a supercharged cycle, the mols of fresh charge added, per mol of residual gases, for computation of supercharger work can be found from:

$$\text{mols of fresh charge} = \frac{1-f}{f} \times \frac{1}{C} \quad \dots\dots\dots(9)$$

An example, making use of this alternative method for a supercharged cycle has been given by the author¹².

A sample calculation for a cycle in which the exhaust pressure is less than intake pressure, using this alternative method, is given in Appendix 2. The results obtained using this method have been found identical with those obtained from the method suggested by Hottel and his co-workers¹¹.

(iii) Work Output and Efficiency.

Instead of calculating the work output per cycle per pound of air constituents as suggested by Hottel and his

co-workers¹¹, it has been found more convenient, for comparison, to calculate the work output per cycle per unit swept volume and hence find the indicated mean effective pressure.

Referring to Figures 1 and 2, work done, excluding

$$\text{pumping work} = \frac{(e_3 - e_4) - (e_{S2} - e_{S1})}{1 - f}$$

$$\text{B.T.U./lb. of fresh charge} \dots\dots\dots(10)$$

where e = internal energy per pound of air constituents at appropriate points, and the suffix S indicates that the chemical energy is excluded.

The pumping work can be estimated with negligible error by assuming the pumping loop, in all cases, to be rectangular:

$$\begin{aligned} \text{Pumping work} &= \frac{(P_b - P_i)(v_1 - v_6)}{1 - f} \times \frac{144}{778} \text{ B.T.U./lb. of fresh charge} \\ &= \frac{(P_b - P_i)(r - 1)v_1}{(1 - f)r} \times \frac{144}{778} \text{ B.T.U./lb. of fresh charge} \\ &\dots\dots\dots(11) \end{aligned}$$

where v = volume, c.ft./lb. of air constituents

P_b = exhaust back pressure, lb./in.² abs.

P_i = intake pressure, lb./in.² abs.

$$\begin{aligned} \therefore \text{Net work done} &= \frac{(e_3 - e_4) - (e_{S2} - e_{S1})}{1 - f} \\ &\quad - \frac{(P_b - P_i)(r - 1)v_1}{(1 - f)r} \times \frac{144}{778} \text{ B.T.U./lb. of} \\ &\quad \text{fresh charge} \dots\dots\dots(12) \end{aligned}$$

Heat available per pound of fresh charge

= weight of fuel used x calorific value

$$= 0.06775 \times F \times 19040 \text{ B.T.U.} \dots\dots\dots(13)$$

where F = fraction of theoretical fuel

Since, for a constant F the heat available per pound of fresh charge is constant,

Thermal Efficiency = constant x net work done per pound of fresh charge $\dots\dots\dots(14)$

The volume of fresh charge entering per cycle

= volume ratio x swept volume $\dots\dots\dots(15)$

$$\text{Volume ratio} = \frac{V_1 - V_6'}{V_1 - V_6} \quad \text{Figures 1 and 3}$$

$$= \frac{r - \frac{V_6'}{V_6}}{r - 1} = \frac{r - \left(\frac{P_b}{P_i}\right)^{\frac{1}{n}}}{r - 1}$$

where n = index of expansion or compression

Specific weight of fresh charge at the beginning of compression

$$= \frac{1 - f}{v_1} \text{ lbs./ft.}^3$$

∴ Weight of fresh charge entering engine per cycle

$$= \frac{r - \left(\frac{P_b}{P_i}\right)^{\frac{1}{n}}}{r - 1} \times \frac{1 - f}{v_1} \text{ lbs./ft.}^3 \text{ of swept volume} \dots\dots\dots(16)$$

Hence net work done per cycle

$$= \left\{ \frac{r - \left(\frac{P_b}{P_i}\right)^{\frac{1}{n}}}{(r - 1)v_1} \right\} \left\{ (e_3 - e_4) - (e_{s2} - e_{s1}) - (P_b - P_i) \left(\frac{r - 1}{r} \right) v_1 \times \frac{144}{778} \right\}$$

$$\text{B.T.U./ft.}^3 \text{ of swept volume} \dots\dots\dots(17)$$

and indicated mean effective pressure

$$= \left\{ \frac{r - \left(\frac{P_b}{P_i} \right)^{\frac{1}{n}}}{(r - 1)v_1} \right\} \left\{ \left[(e_3 - e_4) - (e_{s2} - e_{s1}) \right] \times \frac{778}{144} - (P_b - P_i) \left(\frac{r - 1}{r} \right) v_1 \right\}$$

lbs./in.²(18)

4. RANGE OF CYCLES ANALYSED

Series 1. Influence of Manifold Pressure Ratio

The first series of calculations were conducted to examine, theoretically, the effects of back pressure and intake pressure on the performance of an engine operating on the Otto cycle,¹³ and to obtain relationships between weight fraction of residual gases, temperature at beginning of compression, and manifold pressures. For this series, the compression ratio was chosen as 6.5 to 1 and the fuel-air ratio as 1.2 times that theoretically required for complete combustion. The temperature of the incoming fresh charge was chosen as 600°R. Back pressures ranging from 6 lb./in.²abs. to 20 lb./in.²abs. in 2 pound increments were chosen for intake pressures of 6, 8, 10, 12 and 14 lb./in.²abs., making a total of 40 cycles analysed.

The results of this series of computations are given, Appendix 3, in Tables 1 to 5, and the deductions therefrom in Tables 6 to 10.

Series 2. Influence of Compression Ratio, Fuel-Air Ratio and Fresh Charge Temperature

The second series of calculations were conducted to examine the effects of compression ratio, fuel-air ratio and fresh charge temperature on cycle performance and to obtain relationships between these quantities, the weight fraction of residual gases, and the temperature at the beginning of compression. For each of the extreme limits of manifold

pressure ratio, $\frac{P_b}{P_i} = \frac{20}{6}$ and $\frac{P_b}{P_i} = \frac{6}{14}$, of the cycles

considered in Series 1, cycles were calculated for the following cases:

- (1) Temperature of fresh charge, $T_{fr.}$ varied from $500^{\circ}R.$ to $700^{\circ}R.$ in 50 degree increments at $F = 1.2$ and $r = 6.5$.
- (2) Compression ratio, r varied from 5.5 to 9.5 in increments of 1 at $F = 1.2$ and $T_{fr.} = 600^{\circ}R.$
- (3) Ratio of fuel to air to that theoretically required for complete combustion, F of 0.8, 0.9, 1.0, 1.1, 1.2 and 1.5 at $r = 6.5$ and $T_{fr.} = 600^{\circ}R.$ These values of F were chosen to correspond with the Burned Mixture Charts available.

The results of this series of computations are given, Appendix 4, in Tables 11 to 16 and the deductions therefrom in Tables 17 to 22.

FIGURE 4

Variation of Weight Fraction of residual gases with Manifold Pressures.

FIGURE 5

Weight Fraction, f versus manifold pressure factor, X for a constant fresh charge temperature, compression ratio and fuel-air ratio.

5. WEIGHT FRACTION OF RESIDUAL GASES

In Figure 4 the weight fraction f of residual gases, from Series 1 of the computations, is shown plotted against back pressure for the various intake pressures. Since these curves, when produced, intersect at a common point and the horizontal intercepts between them are equal,

$$f - f_x = m \frac{P_b - P_{bx}}{P_i} \dots\dots\dots(19)$$

where f_x the value of f at the point of intersection of the curves

P_{bx} , the value of P_b at the point of intersection of the curves = 1.10

m = a constant.

Equation (19) is verified by a plot of f versus $\frac{P_b - 1.10}{P_i}$, Figure 5, which reveals a linear relationship as follows

$$f = mX + a \dots\dots\dots(20)$$

$$\text{where } X = \frac{P_b - 1.10}{P_i}$$

$$m = 0.0397$$

$$a = 0.0053$$

It may be assumed, therefore, that linear relationships exist between f and X irrespective of manifold pressure ratio, for various values of compression, ratio, fresh charge temperature and fuel-air ratio.

Supplementary plots of f versus X , Figure 6, from

Weight Fraction, f versus manifold pressure factor, X for various fresh charge temperatures, compression ratios and fuel-air ratios.

FIGURE 6

FIGURE 7

$\frac{\partial f}{\partial x}$ versus Fresh Charge Temperature at a constant compression ratio and fuel-air ratio.

Series 2 of the computations, for the range of compression ratios, fresh charge temperatures and fuel-air ratios used, show that all curves have a common intercept when X equals zero, but have different slopes. Some intermediate points on these curves were checked to verify the linear relationships.

In order to analyse the effects of these variables on f, it is necessary, therefore, to investigate their effect on the variation of the slope, $\frac{\partial f}{\partial X}$ where

$$\frac{\partial f}{\partial X} = \frac{\frac{f_1 - f_2}{P_{b1} - 1.10} - \frac{P_{b2} - 1.10}{P_{i1}}}{P_{i2}}$$

where

suffix 1 refers to an intake pressure of 6 lb./in.² abs. and an exhaust pressure of 20 lb./in.² abs.

suffix 2 refers to an intake pressure of 14 lb./in.² abs. and an exhaust pressure of 6 lb./in.² abs.

$$\therefore \frac{\partial f}{\partial X} = 0.3571 (f_1 - f_2)$$

In Figure 7 a plot of $\frac{\partial f}{\partial X}$ versus $T_{fr.}$ at $F = 1.2$ and $r = 6.5$ gives

$$\frac{\partial f}{\partial X} = m_1 T_{fr.} + b \quad \dots\dots\dots(21)$$

$$\text{where } m_1 = 0.0000340$$

$$b = 0.0193$$

Hence Equation (20) may be written as

FIGURE 8

$\frac{df}{dx}$ versus Compression Ratio, r at a constant fresh charge temperature and fuel-air ratio.

FIGURE 9

$\frac{df}{dx}$ versus Fuel-Air Ratio, F at a constant fresh charge temperature and compression ratio.

$$f = (m_1 T_{fr.} + b) X + a \quad \dots\dots\dots(22)$$

In Figure 8, $\frac{\partial f}{\partial X}$ is plotted against r , from which a logarithmic plot reveals $\frac{\partial f}{\partial X}$ inversely proportional to $r^{1.24}$. The variation with r from the value obtained for $r = 6.5$ may be shown in Equation (22) by substituting $\frac{c}{r^{1.24}}$ in it, where $c = 6.5^{1.24} = 10.19$

$$\text{Hence } f = \frac{c(m_1 T_{fr.} + b) X}{r^{1.24}} + a \quad \dots\dots\dots(23)$$

In Figure 9, $\frac{\partial f}{\partial X}$ is plotted against F and the following relationships are obtained:

When $F = 0.8$ to 1.0

$$\frac{\partial f}{\partial X} = d - m_2 F$$

$$\text{where } d = 0.0578$$

$$m_2 = 0.0170$$

When $F = 1.0$ to 1.5

$$\frac{\partial f}{\partial X} = g + \frac{(F - k)^2}{j}$$

$$\text{where } g = 0.0397$$

$$k = 1.25$$

$$j = 58.4$$

The latter relationship is based on the assumption that the corresponding portion of the curve is parabolic.

Equation (23) is modified to take into account variations of $\frac{\partial f}{\partial X}$ with F from the value for $F = 1.2$, i.e. when $\frac{\partial f}{\partial X} = m = g$ as follows:

When $F = 1.0$ to 1.5

$$f = \frac{1}{m} \left\{ g + \frac{(F - k)^2}{j} \right\} \left\{ \frac{m_1 T_{fr.} + b}{r^{1.24}} \right\} c X + a \dots (24)$$

$$\text{i.e. } f = \left\{ 1 + \frac{(F - k)^2}{A} \right\} \left\{ \frac{m_3 T_{fr.} + B}{r^{1.24}} \right\} X + a \dots (25)$$

$$\text{where } A = mj = 2.318$$

$$B = bc = 0.1967$$

$$m_3 = cm_1 = 0.000346$$

When $F = 0.8$ to 1.0

$$f = \frac{1}{m} \left\{ d - m_2 F \right\} \left\{ \frac{m_1 T_{fr.} + b}{r^{1.24}} \right\} c X + a \dots (26)$$

$$\text{i.e. } f = \left\{ D - GF \right\} \left\{ \frac{m_3 T_{fr.} + B}{r^{1.24}} \right\} X + a \dots (27)$$

$$\text{where } D = \frac{d}{m} = 1.456$$

$$G = \frac{m_2}{m} = 0.428$$

FIGURE 10

Variation of Temperature, T_1 , with Manifold Pressures.

6. TEMPERATURE OF CHARGE AT BEGINNING OF COMPRESSION

A plot of temperature of charge at the beginning of compression T_1 , versus back pressure for the various intake pressures, Figure 10, reveals a relationship between temperature T_1 and manifold pressure ratio similar to that obtained for f in Equation (19). The reason for this is that the curves, when produced, intersect at a common point and the horizontal intercepts between them are equal. The relationship would be of the form:

$$T_1 - C_1 = C_2 \frac{P_b - C_3}{P_i} \dots\dots\dots(28)$$

where C_1 , C_2 , C_3 are constants.

Equations similar in form to Equations (25) and (27), but relating temperature T_1 with back pressure, intake pressure, compression ratio, fresh charge temperature and fuel-air ratio could then be found. However, less cumbersome equations can be obtained by relating T_1 and f , since, as shown below, a linear relationship exists between these quantities irrespective of manifold pressure ratio, $\frac{P_b}{P_i}$.

For the adiabatic constant pressure mixing process, 6'-1, Figures 1 and 3, in which a weight fraction f pounds of residual gases mixes with a weight fraction $(1 - f)$ pounds of fresh charge, the sensible enthalpy h_{S1} , per pound of air constituents, at point 1 is calculated from

$$h_{S1} = f h_{S6'} + (1 - f)y_{fr}.H_{S_{fr.}} \dots\dots\dots(29)$$

FIGURE II

Relationship between Temperature T_1 and Sensible Enthalpy H_{s1} at the beginning of compression.

where $h_{S6'}$ = sensible enthalpy of residual gases at 6',
per pound of air constituents.

$H_{S_{fr.}}$ = sensible enthalpy of fresh charge per mol.

$y_{fr.}$ = fresh charge conversion factor, mols per
pound of air constituents.

Also, the sensible enthalpy per mol of mixture at 1

$$H_{S1} = \frac{h_{S1}}{y_1} \dots\dots\dots(30)$$

where y_1 = conversion factor for mixture at 1, mols per
pound of air constituents.

Combining equations (29) and (30) and rearranging gives

$$h_{S1} = (h_{S6'} - y_{fr.} H_{S_{fr.}})f + y_{fr.} H_{S_{fr.}} \dots\dots\dots(31)$$

A plot of T_1 versus H_{S1} , Figure 11, computed from Hottel, Williams and Satterfield's Modified Air Chart¹¹ for a fuel-air ratio F , of 1.2 times the chemically correct value, reveals with an accuracy of ± 1 per cent for the range considered that

$$T_1 = 0.127 H_{S1} + 370 \dots\dots\dots(32)$$

Therefore, from Equations (30), (31) and (32),

$$T_1 = 0.127 \left\{ \frac{h_{S6'}}{y_1} - \frac{y_{fr.} H_{S_{fr.}}}{y_1} \right\} f + \left\{ 0.127 \frac{y_{fr.} H_{S_{fr.}}}{y_1} + 370 \right\} \dots\dots\dots(33)$$

In Equation (33), $y_{fr.}$ and $H_{S_{fr.}}$ are constants which depend on the fresh charge temperature and the fuel-air ratio.

Also, the maximum variation in $\frac{h_{S6'}}{y_1}$ and y_1 for the cycles

FIGURE 12

Variation of temperature T_1 , with Weight Fraction, f , for a range of intake temperatures and manifold pressures

considered was found to be ± 2 per cent and, ± 0.5 per cent respectively. Hence it may be assumed with reasonable accuracy that a linear relationship exists between T_1 and f , irrespective of the manifold pressure ratio.

This is verified from a plot of T_1 versus f , Figure 12, for manifold pressure ratios varying from $\frac{6}{14}$ to $\frac{20}{6}$ at a fresh charge temperature of 600°R. , a compression ratio of 6.5 and a fuel-air ratio of 1.2 times that required for complete combustion. The relationship is:

$$T_1 = m_4 f + \ell \quad \dots\dots\dots(34)$$

$$\text{where } \ell = 598$$

$$m_4 = 2016$$

It may be assumed also, that linear relationships exist between T_1 and f , irrespective of manifold pressure ratio, for the various values of compression ratio, fresh charge temperature and fuel-air ratio used. Supplementary plots on Figure 12, using the extreme values of manifold pressure ratio, at $r = 6.5$, $F = 1.2$ and $T_{fr.} = 500, 550, 650$ and 700°R. give the following results for m_4 and ℓ in Equation (34):

T_1	m_4	ℓ
500°R	2016	500
550°R	2016	548
650°R	2022	651
700°R	2010	697

FIGURE 13

Variation of Temperature, T , with Weight Fraction, f , at various compression ratios and fuel-air ratios.

FIGURE 14

$\frac{dT}{df}$ versus Compression Ratio, r at a constant fresh charge temperature and fuel-air ratio.

FIGURE 15

$\frac{dT}{dF}$ versus Fuel-Air Ratio, F at a constant fresh charge temperature and compression ratio

It may be assumed, therefore, with sufficient accuracy that

$$T_1 = m_4 f + T_{fr}. \quad \dots\dots\dots(35)$$

$$\text{where } m_4 = 2016$$

The constant must be equal to T_{fr} . in Equation (34) since T_1 must equal T_{fr} . when there are no residual gases present.

Plots of T_1 versus f , Figure 13, for the various values of compression ratio and fuel-air ratio at a fresh charge temperature of 600°R . show that all curves have a common intercept of $T_1 = 600^\circ\text{R}$. when $f = 0$. Intermediate points on the curves were checked to verify the linear relationships. The variation of slope $\frac{\partial T_1}{\partial f}$ of the curves with compression ratio and fuel-air ratio is shown in Figures 14 and 15 respectively.

From Figure 14,

$$\frac{\partial T_1}{\partial f} = n - m_5 r$$

$$\text{where } n = 2492$$

$$m_5 = 73.6$$

Therefore Equation (35) may be written as

$$T_1 = (n - m_5 r) f + T_{fr}. \quad \dots\dots\dots(36)$$

From Figure 15,

$$\text{when } F = 0.8 \text{ to } 1.0 \quad \frac{\partial T_1}{\partial f} = qF$$

$$\text{where } q = 2330$$

and when $F = 1.0 \text{ to } 1.5$, a logarithmic plot gives

$$\frac{\partial T_1}{\partial f} = \frac{q}{F^{0.8}}$$

Equation (36) is modified to take into account variations of

$\frac{\partial T_1}{\partial f}$ with F from the value for F = 1.2, that is when $\frac{\partial T_1}{\partial f} = m_4$, as follows:

When F = 1.0 to 1.5

$$T_1 = \frac{q}{F^{0.8}} \frac{(n - m_5 r)}{m_4} f + T_{fr}. \quad \dots\dots\dots(37)$$

$$\text{i.e. } T_1 = \frac{(H - m_6 r)}{F^{0.8}} f + T_{fr}. \quad \dots\dots\dots(38)$$

$$\text{where } H = \frac{qn}{m_4} = 2883$$

$$m_6 = \frac{qm_5}{m_4} = 85.1$$

When F = 0.8 to 1.0

$$T_1 = qF \frac{(n - m_5 r)}{m_4} f + T_{fr}. \quad \dots\dots\dots(39)$$

$$\text{i.e. } T_1 = (H - m_6 r) F f + T_{fr}. \quad \dots\dots\dots(40)$$

FIGURE 16

Work Output (excluding pump work) versus Compression Ratio, r at a constant fresh charge temp. and fuel-air ratio.

FIGURE 17

Work Output (excluding pump work) versus fresh charge temperature at a constant compression ratio and Fuel-Air Ratio.

7. WORK OUTPUT PER POUND OF FRESH CHARGE

The work output W , per pound of fresh charge, exclusive of pumping work, has been found to be independent of the manifold pressure ratio, $\frac{P_b}{P_i}$, Tables 6 to 10, Appendix 3. Such a relationship is not to be expected, however, with variations in compression ratio, fuel-air ratio and temperature of fresh charge. An analysis of the effects of these quantities is as follows;

In Figure 16, Work Output W , is plotted against compression ratio r , at a constant fresh charge temperature $T_{fr.}$ of $600^\circ R$. and fuel-air ratio F , of 1.2 times that required for complete combustion. A logarithmic plot from this curve gives

$$W = t r^{0.4} \dots\dots\dots(41)$$

$$\text{where } t = 239$$

Also, a plot of W versus $T_{fr.}$, Figure 17, at a constant r of 6.5 and F of 1.2 reveals

$$W = u - m_7 T_{fr.} \dots\dots\dots(42)$$

$$\text{where } u = 531$$

$$m_7 = 0.04$$

When $T_{fr.} = 600^\circ R$., $W = 507$ B.T.U. per pound of fresh charge, so that Equations (41) and (42) may be combined to read

$$W = v r^{0.4} (u - m_7 T_{fr.}) \dots\dots\dots(43)$$

$$\text{where } v = \frac{t}{507} = 0.472$$

$$\text{Hence } W = (J - m_8 T_{fr.}) r^{0.4} \dots\dots\dots(44)$$

FIGURE 18

Work Output (excluding pumping work) versus Fuel-Air Ratio at a constant fresh charge temperature and compression ratio.

$$\text{where } J = v u = 251$$

$$m_8 = v m_7 = 0.0189$$

The variation of W with F , Figure 18, is negligible within the range $F = 1.0$ to 1.5 , but for values between 0.8 and 1.0

$$W = m_9 F + w \quad \dots\dots\dots(45)$$

$$\text{where } w = 87$$

$$m_9 = 420$$

Since $W = 507$ when $F = 1.0$, substitution of Equation (45) in Equation (44) gives, for the range $F = 0.8$ to 1.0

$$W = (J - m_8 T_{fr.}) r^{0.4} \frac{(m_9 F + w)}{507} \quad \dots\dots\dots(46)$$

$$\text{i.e. } W = (J - m_8 T_{fr.}) r^{0.4} (m_{10} F + x) \quad \dots\dots\dots(47)$$

$$\text{where } x = \frac{w}{507} = 0.172$$

$$m_{10} = \frac{m_9}{507} = 0.828$$

8. SUMMARY OF EQUATIONS

From the results of an extremely careful analysis of 66 Otto cycles, made with the aid of thermodynamic charts, for a selected range of intake pressures, exhaust back pressures, fresh charge temperatures, fuel-air ratios and compression ratios, the following equations have been established:

For the range $F = 0.8$ to 1.0

$$f = \left\{ D - G F \right\} \left\{ \frac{m_3 T_{fr.} + B}{r^{1.24}} \right\} X + a \dots\dots\dots(27)$$

$$T_1 = (H - m_6 r) F f + T_{fr.} \dots\dots\dots(40)$$

$$W = (J - m_8 T_{fr.}) (m_{10} F + x) r^{0.4} \dots\dots\dots(47)$$

For the range $F = 1.0$ to 1.5

$$f = \left\{ 1 + \frac{(F - k)^2}{A} \right\} \left\{ \frac{m_3 T_{fr.} + B}{r^{1.24}} \right\} X + a \dots\dots\dots(25)$$

$$T_1 = \frac{(H - m_6 r) f}{F^{0.8}} + T_{fr.} \dots\dots\dots(38)$$

$$W = (J - m_8 T_{fr.}) r^{0.4} \dots\dots\dots(44)$$

where

F = weight fraction of theoretical fuel

f = weight fraction of residual gases

r = compression ratio

P_b = exhaust back pressure, lb./in.²abs.

P_i = intake pressure, lb./in.²abs.

$T_{fr.}$ = fresh charge temperature, °R.

T_1 = temperature at the beginning of compression, °R.

W = work output exclusive of pumping work, B.T.U. per pound of fresh charge.

$$X = \text{manifold pressure factor} = \frac{P_b - 1.10}{P_i}$$

$$A = 2.318 \quad a = 0.0053$$

$$B = 0.1967 \quad k = 1.25$$

$$D = 1.456 \quad x = 0.172$$

$$m_3 = 0.000346$$

$$G = 0.428 \quad m_6 = 85.1$$

$$H = 2883 \quad m_8 = 0.0189$$

$$J = 251 \quad m_{10} = 0.828$$

Also, since $(e_3 - e_4) - (e_{s2} - e_{s1}) = W(1 - f)$ from Equation (10), Equations (12), (18) and (14) may be modified to read as follows:

$$\text{Net work done} = W - 0.185 \frac{(P_b - P_i)v_1(r - 1)}{r(1 - f)} \quad \text{B.T.U. per} \\ \text{pound of fresh charge} \dots\dots\dots(48)$$

Indicated mean effective pressure

$$= \left\{ \frac{r - \left(\frac{P_b}{P_i}\right)^{\frac{1}{n}}}{v_1(r - 1)} \right\} \left\{ 5.40 W(1 - f) - (P_b - P_i)v_1 \frac{(r - 1)}{r} \right\} \text{lb./in.}^2 \\ \dots\dots\dots(49)$$

$$\text{Thermal efficiency} = \frac{W - 0.185(P_b - P_i)v_1 \frac{(r - 1)}{(1 - f)r}}{1290F} \dots\dots\dots(50)$$

Volume V_1 as read from the Modified Air Chart is essentially the volume of one mol of air. This volume is multiplied by a conversion factor y_1 , which takes into account fuel-air ratio and weight fraction of residual gases, to obtain the volume v_1 per pound of air constituents. Over the temperature range considered for T_1 , Appendices 3 and 4, it may be assumed with accuracy that air behaves as a perfect gas.

FIGURE 19

Relationship between chart conversion factor, y , Fuel-Air Ratio, F and Weight Fraction of residual gases, f

$$\text{Hence } \frac{144 P_1 V_1}{T_1} = G_1$$

where $G_1 = 1545.4 \text{ ft.lb./mol.}^{\circ\text{R.}}$

$$\therefore v_1 = V_1 y_1 = \frac{10.73 T_1 y_1}{P_1} \dots\dots\dots(51)$$

Volumes read from the Charts, tabulated in Appendices 3 and 4, are found to agree with values obtained from Equation (51) with a maximum error of 1 per cent.

Equations (48), (49) and (50), therefore, may be modified to read as follows:

$$\begin{aligned} \text{Net work done} = W - 1.985 T_1 y_1 \frac{\left(\frac{P_b}{P_i} - 1\right)(r - 1)}{r(1 - f)} \quad \text{B.T.U. per} \\ \text{pound of fresh charge} \quad \dots\dots(52) \end{aligned}$$

Indicated mean effective pressure

$$= \left\{ \frac{r - \left(\frac{P_b}{P_i}\right)^{\frac{1}{n}}}{r - 1} \right\} \left\{ \frac{0.503 W(1 - f)P_i}{T_1 y_1} - (P_b - P_i) \frac{r - 1}{r} \right\} \text{ lb./in.}^2 \dots\dots(53)$$

$$\begin{aligned} \text{Thermal efficiency} = \frac{1}{1290 F} \left\{ W - 1.985 T_1 y_1 \frac{\left(\frac{P_b}{P_i} - 1\right)(r - 1)}{r(1 - f)} \right\} \\ \dots\dots(54) \end{aligned}$$

The index n , of compression or expansion of the residual gases may be taken as 1.315 which is an average value found from the charts for the range of cycles considered. Curves relating y_1 and F for various values of f , reproduced from a supplementary plot on Hottel, Williams and Satterfield's Modified Air Chart, are given in Figure 19.

The equations summarized may be used to estimate accurately, the theoretical performance of an engine operating on the Otto cycle irrespective of intake pressure, exhaust pressure, fresh charge temperature, compression ratio and fuel-air ratio. They may be found of particular value when analysing the effects of changing certain operating variables on engine performance, since they take into account factors such as specific heat variations, dissociation and residual gases from the previous cycle. They may be found of use, also, in the design of internal combustion engines operating on the Otto cycle.

FIGURE 20

Variation of Mean Effective Pressure with Manifold Pressures.

9. EFFECTS OF CHANGING OPERATING VARIABLES ON ENGINE PERFORMANCE

The effects on engine performance of changing operating variables such as intake pressure, exhaust pressure, compression ratio, fresh charge temperature and fuel-air ratio can be examined, theoretically, by using the results tabulated in Appendices 3 and 4 or the equations summarized in Section 8.

(i) Intake and Exhaust Pressures.

In Figure 20, mean effective pressure is plotted against back pressure for various intake manifold pressures, under the following conditions:

- (a) Constant fresh charge temperature, $T_{fr.} = 600^{\circ}\text{R.}$
- (b) Constant compression ratio, $r = 6.5$
- (c) Constant fuel-air ratio to that required for complete combustion, $F = 1.2.$

Small increases in back pressure are seen to have a very serious effect on the power output of an engine. For example, at an intake pressure of $14 \text{ lb./in.}^2 \text{ abs.}$, an overall change of back pressure from 6 to $20 \text{ lb./in.}^2 \text{ abs.}$ reduces the mean effective pressure by approximately 30 per cent, that is, more than 2 per cent per pound increase in back pressure. The effect becomes far more serious at low intake manifold pressures: at $6 \text{ lb./in.}^2 \text{ abs.}$ intake pressure, a reduction in mean effective pressure of 63 per cent is realized for the same range of back pressure, that is, approximately 4 per cent per pound increase in back

FIGURE 21

Variation of Thermal Efficiency with Manifold Pressures.

pressure.

The reduction in mean effective pressure with increased back pressure is caused, mainly, by the corresponding increased quantity of high temperature residual gases, Figure 4. These gases increase the temperature and reduce the density of the charge at the beginning of compression and, therefore, restrict the weight of fresh charge admitted per cycle.

The variation of thermal efficiency with back pressure, under the same conditions as listed above, for various intake manifold pressures is shown in Figure 21. At an intake pressure of 14 lb./in.²abs., a reduction in efficiency from 34.3 per cent to 31.5 per cent is realized for a change in back pressure from 6 lb./in.²abs. to 20 lb./in.²abs. The reduction is much greater at lower intake pressures; at 6 lb./in.²abs. the same increase in back pressure reduces the thermal efficiency from 33.1 per cent to 23.7 per cent.

The main reason for the reduction in thermal efficiency with increased back pressure is as follows:

Work output, per pound of fresh charge, exclusive of pumping work, is found to be independent of manifold pressure ratio $\frac{P_b}{P_i}$, Tables 6 to 10, Appendix 3. Since thermal efficiency is directly proportional to net work output per pound of fresh charge, it must, therefore, depend on the pumping work per pound of fresh charge. At manifold pressure ratios greater than unity, pumping work is negative and at

FIGURE 22

Variation of Mean Effective Pressure & Thermal Efficiency with Fresh Charge Temperature

manifold pressure ratios less than unity, pumping work is positive. The net work output and the thermal efficiency decrease, therefore, as back pressure increases. The rate of decrease is greater at lower intake pressures owing to the greater changes in pumping work.

(ii) Fresh Charge Temperature.

In Figure 22, mean effective pressure is plotted against fresh charge temperature for each of the extreme values of manifold pressure ratio used in the computations. The compression ratio is chosen as 6.5 to 1 and the ratio of fuel to air as 1.2 times that required for complete combustion.

Mean effective pressure is seen to decrease almost linearly as fresh charge temperature increases for the two manifold pressure ratios used. This, no doubt, is due mainly to the reduction in weight of fresh charge admitted per cycle caused by its reduced density at the higher temperatures. The percentage reduction in mean effective pressure from the value obtained for a fresh charge temperature of 500°R . is plotted also on Figure 22. This curve reveals with close approximation, that a linear percentage reduction in mean effective pressure occurs with increased fresh charge temperature, irrespective of manifold pressure ratio. The reduction is approximately 1.5 per cent per 10 degrees Fahrenheit increase.

A variation of fresh charge temperature from 500°R .

to 700°R . is found to have negligible effect on thermal efficiency at a low manifold pressure ratio, $\frac{P_b}{P_i} = \frac{6}{14}$, Table 18, Appendix 4. The reason for this is that the slight reduction, with increased temperature, of work output per pound of fresh charge, exclusive of pumping work, is compensated for by a corresponding slight increase in pumping work which, in this case, is additive. Hence the thermal efficiency, being directly proportional to net work output per pound of fresh charge, remains constant.

At a high value of manifold pressure ratio, however, thermal efficiency is found to decrease steadily as fresh charge temperature increases. At a manifold pressure ratio, $\frac{P_b}{P_i} = \frac{20}{6}$, thermal efficiency decreases from 25.2 per cent at a fresh charge temperature of 500°R . to 22.0 per cent at 700°R ., Figure 22. The reasons for this are as follows:

The work output per pound of fresh charge, exclusive of pumping work, decreases steadily with fresh charge temperature by almost the same amount as for the low manifold pressure ratio, Table 17, Appendix 4. The pumping work, however, increases at a much greater rate since, at low intake pressures, the volume per pound of fresh charge increases more rapidly with temperature. Further, since the pumping work is subtracted in this case to obtain net work, a reduction in thermal efficiency must result.

(iii) Compression Ratio

In Figure 23, mean effective pressure and thermal

FIGURE 23

Variation of Mean Effective Pressure and Thermal Efficiency with Compression Ratio.

efficiency are plotted against compression ratio for manifold pressure ratios of $\frac{20}{6}$ and $\frac{6}{14}$. The corresponding fresh charge temperature is 600°R . and the fuel-air ratio is 1.2 times that required for complete combustion.

Thermal efficiency increases steadily with compression ratio, the rate of increase being slightly greater at the higher manifold pressure ratio. Increases in thermal efficiency from 31.8 per cent to 39.2 per cent and from 20.4 per cent to 29.8 per cent, at manifold pressure ratios $\frac{P_b}{P_i}$, of $\frac{6}{14}$ and $\frac{20}{6}$ respectively, are realized for an increase in compression ratio from 5.5 to 9.5. The increase in work output per pound of fresh charge, exclusive of pumping work, with compression ratio is almost identical for each manifold pressure ratio, Tables 19 and 20, Appendix 4, and is responsible, mainly, for the increase in thermal efficiency. The pumping work remains almost constant at the lower manifold pressure ratio, and decreases steadily as compression ratio increases at the higher manifold pressure ratio. Since the pumping work is negative in the latter case, the corresponding rate of increase in net work output and, therefore, thermal efficiency is greater.

The Air Standard Efficiency curve, added for comparison, illustrates clearly the error involved in using the air standard cycle.

Mean effective pressure is seen to increase steadily with compression ratio from 164 lb./in.²abs. to 201 lb./in.²abs.

FIGURE 24

Variation of Mean Effective Pressure and Thermal Efficiency with Fuel-Air Ratio.

and from 15.5 lb./in.² abs. to 38.6 lb./in.² abs., at the lower and higher manifold pressure ratios respectively, for an increase in compression ratio from 5.5 to 9.5. This increase is caused by the increased weight fraction of fresh charge added per cycle. It has been shown from Figure 8 that the weight fraction of residual gases remaining in the cylinder at the completion of the exhaust stroke, is inversely proportional to $r^{1.24}$ and, since the presence of these high temperature gases increases the temperature of the charge at the beginning of compression, the weight of fresh charge entering the cylinder per cycle must be reduced accordingly.

(iv) Fuel-Air Ratio

In Figure 24, mean effective pressure and thermal efficiency are plotted against fuel-air ratio F , for manifold pressure ratios $\frac{P_b}{P_i}$, of $\frac{20}{6}$ and $\frac{6}{14}$. The corresponding fresh charge temperature is 600°R. and the compression ratio is 6.5 to 1. Values corresponding to $F = 1.3$ and 1.4 are calculated using the equations summarized in Section 8, there being no burned mixture charts available for this range.

The mean effective pressure curves show that maximum power output is to be expected when using a rich mixture, the peak of the curve being reached at a fuel-air ratio F of 1.1 times the chemically correct value when $\frac{P_b}{P_i} = \frac{6}{14}$, and 1.3 times the chemically correct value when

$$\frac{P_b}{P_i} = \frac{20}{6}.$$

Thermal efficiency increases almost linearly with decreasing mixture strength for values of F between 1.5 and 1.0, the rate of increase being greater at the lower manifold pressure ratio. The efficiency tends to decrease, however, as F decreases from 1.0 to 0.8 and in the case when $\frac{P_b}{P_i} = \frac{20}{6}$ a peak efficiency of 27.2 per cent is reached at $F = 0.9$. The peak efficiency would be reached at a somewhat lower value of F for lower manifold pressure ratios.

The reasons for the curve shapes of Figure 23 are as follows:

The work output per pound of fresh charge, exclusive of pumping work, remains almost constant at rich mixture strengths, Figure 18, but decreases steadily as the mixture becomes weaker. The pumping work, on the other hand, at first increases and then decreases slightly with decreasing mixture strength when $\frac{P_b}{P_i} = \frac{20}{6}$, but remains almost constant when $\frac{P_b}{P_i} = \frac{6}{14}$, Tables 21 and 22, Appendix 4. The slope of the efficiency curves must be greater, therefore, with decreasing mixture strength at the higher manifold pressure ratio because, in this case, the pumping work is positive. The decreasing slope of the efficiency curves with weaker mixtures is caused by the corresponding reduction in work per pound of fresh charge as illustrated in Figure 18.

(v) Comparative Results.

Unfortunately, not many theoretical curves are available for comparison with those developed from the equations summarized in Section 8 and the Tables in Appendices 3 and 4.

Goodenough and Baker¹⁴ calculated a number of Otto cycles for various fuel-air ratios and compression ratios, based on data available on dissociation equilibrium constants and specific heats. Although the data used by these authors have been subject to much revision, the trends shown by their curves are still found to be valid.

Hershey, Eberhardt and Hottel¹⁰ calculated a number of Otto cycles based on thermodynamic charts similar to those used in the present work and presented graphs of mean effective pressure and thermal efficiency versus percentage of theoretical air for a range of compression ratios. Separate charts were used for the compression and expansion processes: the "compression" charts were plotted for thermodynamic properties of an air-octane mixture containing a fixed quantity of residual gases prior to combustion, and the "expansion" charts were plotted for the properties of the equilibrium mixture after combustion. These "expansion" or "burned mixture" charts were, later, plotted with greater accuracy and the various "compression" charts were replaced by the Modified Air Chart as used in the present work.

In Figure 25, mean effective pressure and thermal efficiency are plotted against percentage of theoretical air

FIGURE 25

Variation of Mean Effective Pressure and Thermal Efficiency with percentage of theoretical air.

($\frac{1}{F}$ per cent) for a compression ratio of 9 to 1. The intake and exhaust pressures have been chosen as 14.7 lbs./in.²abs., making the pumping work zero, and the fresh charge temperature has been chosen as 520°R. These curves have been established from the equations summarized in Section 8, the results of the calculations being given, Appendix 5, in Table 23.

Curves established by Goodenough and Baker¹⁴ and by Hershey and his co-workers¹⁰ have been plotted also in Figure 25. Intake and exhaust pressures of 14.7 lb./in.²abs. have been used in deriving these curves and, in the former case, the intake fresh charge temperature was chosen as 520°R.. Unfortunately Hershey and his co-workers have not stated the fresh charge temperature used, but from an examination of their work, it appears to have been 520°R.. Goodenough and Baker have not developed curves for a compression ratio of 9, but from an examination of their curve plotted for a compression ratio of 8, it is evident that values close to those obtained by Hershey and his co-workers would be expected at a 9 to 1 compression ratio.

The mean effective pressure curves of Figure 25 all show the same general trend, that is, at first a slight increase followed by a slight, but then more rapid, decrease with increasing percentage of theoretical air. In all cases a maximum mean effective pressure is reached when using approximately 91 per cent of the theoretical air. This corresponds to a fuel-air ratio F , of 1.1 times that

required for complete combustion. The values of mean effective pressure obtained by the other workers, however, are 10 to 15 per cent greater than those from the equations developed in the present work. The reasons for this are as follows:-

(a) Goodenough and Baker's curves were based on data, which as mentioned earlier, have been subject to much revision. Further, they chose Benzene (C_6H_6) as the fuel used and did not consider the effects of residual gases from the previous cycle.

(b) Hershey, Eberhardt and Hottel's¹⁰ curves, on the other hand, were based on charts which have been superseded by those used in the present work. Their separate "compression" charts which were plotted on the basis of a fixed quantity (one per cent) of residual gases, have been replaced by a Modified Air Chart which is suitable for any quantity of residual gases. The weight fraction of residual gases varies between 2.80 and 3.07 per cent in the present case causing the large differences in mean effective pressure.

Thermal efficiency is found to increase with increasing percentage of theoretical air, the curves showing extremely close agreement in all three cases. This is to be expected for the following reasons:

Thermal efficiency is a function of net work output per pound of fresh charge, fuel-air ratio and calorific value of the fuel, Equation (14). The work output per pound of fresh charge, exclusive of pumping work, has been found almost constant for a wide range of mixture strengths and manifold

pressure ratios, Figure 18. Further, since the pumping work is zero and there is little variation in the calorific value of the different fuels used, thermal efficiency, in this case, must depend mainly on the fuel-air ratio.

10. EFFECTS OF CHANGING OPERATING VARIABLES ON MAXIMUM CYCLE TEMPERATURES AND PRESSURES.

The effects of changing operating variables, such as intake and exhaust pressures, fresh charge temperature, compression ratio and fuel air ratio, on the temperature and pressure at any point in the Otto cycle, Figures 1 and 2, could be investigated using the results tabulated in Appendices 3 and 4. The temperatures and pressures considered of most importance to the designer and research worker, however, are the maximum values so that the effects of operating variables on these quantities only are studied here. Equations similar in form to those relating weight fraction of residual gases f , with operating variables, Equations (25) and (27) might be investigated for maximum cycle temperature and pressure, but since these quantities are not of as much importance, it is considered sufficient to examine their variations graphically. Maximum cycle pressures and temperatures for other operating conditions may be estimated with reasonable accuracy from the graphs following.

(i) Intake and Exhaust Pressures.

In Figure 26, maximum cycle temperature T_3 , and pressure P_3 , are plotted against exhaust back pressure for various intake manifold pressures under the following conditions:

- (a) Constant fresh charge temperature, $T_{fr.} = 600^\circ R.$
- (b) Constant compression ratio, $r = 6.5$
- (c) Constant fuel-air ratio to that required for

FIGURE 26

Variation of Maximum Cycle Temperature and Pressure with Manifold Pressures.

complete combustion, $F = 1.2$.

The maximum cycle temperature is seen to decrease linearly with increasing back pressure for all values of intake pressure, the rate of decrease being greater at the lower intake pressures. This is caused by the increase in weight fraction of residual gases present with increased back pressures, the increase being more predominant at the lower intake pressures as shown in Figure 4. The presence of residual gases increases the temperature of the charge at the beginning of compression and, therefore, at the beginning of combustion for a fixed compression ratio, but the corresponding reduced weight of fresh charge available for combustion predominates, causing a reduction in maximum cycle temperature.

The maximum cycle temperature varies from 5140°R. to 5097°R. at an intake pressure of $14 \text{ lb./in.}^2 \text{ abs.}$, and from 5045°R. to 4950°R. at an intake pressure of $6 \text{ lb./in.}^2 \text{ abs.}$, for an increase in back pressure from $6 \text{ lb./in.}^2 \text{ abs.}$ to $20 \text{ lb./in.}^2 \text{ abs.}$. This represents a decrease of 3°F. and of 7.5°F. per pound increase in back pressure, for intake pressures of $14 \text{ lb./in.}^2 \text{ abs.}$ and $6 \text{ lb./in.}^2 \text{ abs.}$ respectively.

The curves relating maximum cycle pressure and back pressure for the various intake pressures appear as a series of straight lines which are almost parallel. The steady decrease in maximum pressure with increased back pressure is to be expected since there is a corresponding decrease in

FIGURE 27

Variation of Maximum Cycle Temperature and Pressure with Fresh Charge Temperature

maximum temperature. The maximum cycle pressure is reduced by an average value of 6 lb./in.^2 per pound increase in back pressure for all values of intake pressure used. A change in intake pressure, however, is seen to have a far more serious effect on maximum pressure: an increase in intake pressure from $6 \text{ lb./in.}^2 \text{ abs.}$ to $14 \text{ lb./in.}^2 \text{ abs.}$ increases the maximum pressure linearly by approximately 600 lb./in.^2 for all values of back pressure, that is, by 75 lbs./in.^2 per pound increase in intake pressure. This increase is caused, partly by the increase in pressure at the end of compression when compressing from a higher intake pressure, but mainly by the reduced weight fraction of residual gases which allow an increase in the weight of fresh charge available for combustion. The weight fraction of residual gases has been found inversely proportional to the intake pressure, Equation (19).

(ii) Fresh Charge Temperature.

In Figure 27, maximum cycle temperature and pressure are plotted against fresh charge temperature for each of the extreme values of manifold pressure ratio used in the computations. The compression ratio is chosen as 6.5 to 1 and the ratio of fuel to air as 1.2 times that required for complete combustion.

Maximum cycle temperature is seen to increase linearly with fresh charge temperature, the rate of increase being slightly greater at the lower manifold pressure ratio. The

maximum temperature changes from 5073°R. to 5200°R. , and from 4905°R. to 5000°R. , at the lower and higher manifold pressure ratios respectively, for a change in fresh charge temperature from 500°R. to 700°R. . This represents an average increase in maximum temperature of 5.6°F. per 10°F. rise in fresh charge temperature, which is equivalent to a maximum temperature rise of approximately one half the fresh charge temperature rise, irrespective of manifold pressure ratio. It might be expected, on first sight, that the maximum temperature should increase by at least an amount equal to the increase in fresh charge temperature, but the presence of residual gases, once again, offsets the increase. It has been shown, Equations (25) and (27), that the weight fraction of residual gases increases with fresh charge temperature. This is due to the increased specific volume of the fresh charge at the higher temperatures, which reduces the weight fraction of fresh charge admitted per cycle.

Maximum cycle pressure decreases linearly as fresh charge temperature increases, the reduction being greater at the lower manifold pressure ratio. The maximum pressure changes from $940 \text{ lb./in.}^2\text{abs.}$ to $698 \text{ lb./in.}^2\text{abs.}$, and from $280 \text{ lb./in.}^2\text{abs.}$ to $215 \text{ lb./in.}^2\text{abs.}$ at the lower and higher manifold pressure ratios respectively, for a change in fresh charge temperature from 500°R. to 700°R. . This represents a reduction of 25.5 per cent and of 23.2 per cent for a 200°F. increase in fresh charge temperature. Hence the

FIGURE 28

Variation of maximum Cycle Temperature and Pressure with Compression Ratio.

maximum cycle pressure is reduced by approximately 0.12 per cent per degree Fahrenheit increase in fresh charge temperature, irrespective of manifold pressure ratio. This strange phenomena of reduced maximum cycle pressure with increased fresh charge temperature, in spite of the corresponding increase in maximum cycle temperature, is caused by the increase in specific volume at intake which increases the specific volume and, therefore, reduces the pressure at the end of combustion.

(iii) Compression Ratio.

In Figure 28, maximum cycle temperature and pressure are plotted against compression ratio for the two extreme values of manifold pressure ratio. The fresh charge temperature is chosen as 600°R. and the fuel-air ratio as 1.2 times that required for complete combustion. Similar curves produced by Goodenough and Baker ¹⁴ have been added for comparison.

Maximum cycle temperature increases with increased compression ratio, the rate of increase being slightly greater at the higher manifold pressure ratio. An increase from 5090°R. to 5237°R. , and from 4875°R. to 5092°R. is realized, at the lower and higher manifold pressure ratios respectively, for an increase in compression ratio from 5.5 to 9.5. The corresponding increases in maximum pressure are from 680 lb./in.²abs. to 1230 lb./in.²abs., and from 190 lb./in.²abs. to 420 lb./in.²abs. respectively. The increase, with compression ratio, of both maximum cycle temperature

FIGURE 29

Variation of Maximum Cycle Temperature and Pressure with Compression Ratio.

and pressure are caused, partly by the increased compression pressure and temperature, and partly by the expulsion of more residual gases which increases the weight fraction of fresh charge available for combustion.

The curves reproduced from Goodenough and Baker's work are plotted on the basis of intake and exhaust pressures each equal to 14.7 lb./in.² abs. and a theoretically correct air-fuel ratio. The fuel used was Benzene (C₆H₆) instead of Octene (C₈H₁₆) and the temperature of the incoming fresh charge was not given. However, the curves show the same general trend as those of the present work. The large differences in maximum cycle temperature would be accounted for mainly by the difference in fuel-air ratio.

(iv) Fuel-Air Ratio.

In Figure 29, maximum cycle temperature and pressure are plotted against fuel-air ratio for the extreme values of manifold pressure ratio used in the computations. The compression ratio is chosen as 6.5 and the fresh charge temperature as 600°R. Curves by Goodenough and Baker, based also on a compression ratio of 6.5, but on intake and exhaust pressures each of 14.7 lb./in.² abs. and Benzene as a fuel instead of Octene, are added for comparison. These curves show the same general trend as those of the present work, the peak values being reached at almost the same fuel-air ratio.

As the mixture strength is increased beyond the

chemically correct value more of the uncombined Oxygen is used tending to release more heat during combustion and increase the maximum temperature. Opposing this, however, is the tendency to burn more of the ~~Carbon~~ to Carbon Monoxide instead of Carbon Dioxide and, therefore, reduce the heat liberated. Hence, as shown in Figure 29, the maximum cycle temperature at first increases and then decreases with increasing mixture strength, a peak value being reached at a fuel-air ratio of 1.1 times that required for complete combustion.

The maximum cycle pressure, on the other hand, is seen to reach a peak value at a fuel-air ratio of 1.3 times that required for complete combustion. The reason for this peak being reached at a richer mixture strength than for the maximum temperature is as follows:

The combustion process is considered to take place at constant volume and the pressure, therefore, is proportional to the number of molecules and the temperature. As the mixture strength is increased above the chemically correct value, the formation of more Carbon Monoxide in place of Carbon Dioxide increases the number of molecules in the gas and therefore tends to increase its pressure. This tendency for pressure to increase overcomes, for a time after the maximum temperature is reached, the adverse effect of falling temperature and causes the peak pressure to be reached at a greater mixture strength.

PLATE 1: TEST ENGINE AND DYNAMOMETER

- (1) Hose Connection to Air Tank
- (2) Morse "Shorting" Switch
- (3) External Oil Cooler

PLATE 2: EXHAUST PRESSURE CONTROL UNIT

11. EXPERIMENTAL INVESTIGATION ON THE EFFECTS OF BACK PRESSURE ON ENGINE PERFORMANCE.

When the effects of back pressure on engine performance are to be studied, wave action¹⁵ in the exhaust and intake systems could cause a serious hazard. A pressure pulse, for example, moving along an exhaust pipe would, on striking a restriction that might be placed in the pipe to increase back pressure, be reflected as a pulse of greater amplitude. If this reflected pulse returned to the engine when an exhaust port was open, it would have a serious influence on the scavenging of the cylinder. Similar effects might occur from reflected rarefaction waves introduced from positive pulses reaching an open pipe end, or from reflected pulses in the intake system.

In order to minimize the effects of wave action, a multi-cylinder engine was chosen for the tests. Although a multi-cylinder engine has the disadvantage of a fixed compression ratio and, without extensive modifications, no control of fresh charge temperature entering the cylinders, these effects have been examined by others^{16, 17} and are not considered of practical importance here.

In examining effects of back pressure on engine performance, attempts were made to reduce the back pressure below atmospheric and to measure maximum cylinder pressures. To reduce the back pressure, the engine was exhausted into a large tank, in which water sprays were provided for

PLATE 3: INTAKE AND EXHAUST MANIFOLDS

- (1) Morse "Shorting" Switch
- (2) Throttle Control
- (3) Thermocouples
- (4) Manifold Pressure Tappings
- (5) Pressure Tapping for recording Pressure-Time
diagrams from No. 6 cylinder

cooling, and extracted from it by two centrifugal fans arranged in series. These experiments had to be abandoned however, since insufficient cooling could be provided resulting in overheating of the fans, and there was danger of a serious explosion occurring in the tank. A tapping was placed in one cylinder of the engine for recording pressure-time diagrams on a cathode ray oscilloscope. Unfortunately, the physical construction of the engine necessitated a long connecting pipe from the engine cylinder to the pressure "pick-up". Trapped high temperature residual gases in this pipe caused pre-ignition which made it impossible to obtain a stable diagram and caused erratic running of the engine. This, also, had to be abandoned.

The only tests that proved useful in the present work, therefore, were those conducted for a range of intake pressures, the back pressure in each case being varied above atmospheric. The engine was run at a constant speed for all tests, since speed variations would have influenced the dynamics of the intake and exhaust systems and the resulting "breathing" characteristics of the engine.^{18, 19}

(i) Description of Apparatus.

The test engine was a six cylinder, overhead-valve, four-stroke "Holden" automobile engine of 2171 c.c. capacity (3 ins. bore x 3.125 ins. stroke). The compression ratio was 6.5 to 1 and the engine was rated to develop 60 B.H.P. at 3800 R.P.M.. Ancillary equipment such as a generator,

PLATE 4: CONTROL DESK

- (1) Fuel Metering Unit
- (2) Manometers
- (3) Potentiometer
- (4) Thermocouple Selector Switch
- (5) Air Tank
- (6) Air Metering Orifice

water pump, fan and standard radiator cooling system was left intact for the tests, modifications being the addition of an external oil cooler and provision for additional make-up cooling water. The air cleaner was removed and the air intake connected to a large tank which damped out pulsations prior to metering; atmospheric air was drawn into the tank through a short pipe in which a metering orifice plate, designed in accordance with British Standards²⁰, was placed. Iron-constantan thermocouples were placed in the intake ports leading from the manifold to the various cylinders and pressure tappings were provided to measure the mean intake and exhaust manifold pressures.

The engine was coupled to a "Heenan and Froude" Dynamatic Dynamometer (Mk. 1), fitted with an automatic speed control unit.

To provide the desired range of back pressure the exhaust was connected to an expansion box, 8 ins. diameter x 12 ins. long, after which was placed a 6 ins. diameter butterfly control valve taken from an aircraft turbo-supercharger.

Fuel consumption was measured by the positive displacement method and air consumption by the pressure drop across a standard orifice plate. Intake and exhaust manifold pressures were measured on mercury-in-glass manometers. Thermocouples were connected through a selector switch to a potentiometer for measurement of

FIGURE 30

Mechanical Efficiency versus Brake Mean Effective Pressure at 2500 r.p.m.

intake charge temperatures.

A distributor, converted into a switching unit was provided for "shorting out" spark plugs in turn when conducting a "Morse" test ²¹ for determination of mechanical efficiency.

Views of the apparatus are shown in Plates 1 to 4.

(ii) Experimental Procedure.

In order that the experimental results could be compared with those obtained theoretically it was necessary to obtain them on an indicated horsepower basis. Since no means was available for motoring the engine to determine friction horsepower or for taking reliable indicator diagrams, the mechanical efficiencies over the power range available at the chosen test speed of 2500 R.P.M. were determined using the "Morse" technique.²¹ The results of these tests and calculations are given in Appendix 6, Table 24, and the resulting graph of mechanical efficiency versus brake mean effective pressure is shown in Figure 30. Care was taken to maintain engine cooling water and oil temperatures constant during these tests.

A series of tests with increased back pressure was conducted for each of a range of intake manifold pressures selected to give approximately equal steps over the range available at the test speed of 2500 R.P.M. The procedure was as follows:

After a thorough "warm-up" period, the engine throttle

was adjusted to give a chosen intake manifold pressure. For each of a range of exhaust back pressures, controlled by means of the butterfly valve in the exhaust line, fuel consumption, brake load, pressure drop across air metering orifice, intake and exhaust manifold pressures and intake charge temperatures were recorded. The cooling water outlet temperature and the oil temperature was maintained constant for each test by adjusting the supply of make-up water and the water flow through the oil cooler.

An increase in back pressure tended to reduce slightly the intake pressure for a fixed throttle setting; this effect was compensated for by adjusting the throttle setting before each test.

(iii) Experimental Results.

The results of the increased back pressure tests and calculations are given in Appendix 7, Tables 25 to 30.

The air-fuel ratio varied slightly with different throttle openings but this could not be controlled as it was a function of the "Stromberg BXOV.1" carburettor used. Intake charge temperatures however, remained almost constant for all tests.

In order that more direct comparisons could be made with theoretical results, the thermal efficiencies were calculated assuming that the calorific value of the fuel was the same as used in the theoretical analysis (19040 B.T.U./lb.).

FIGURE 31

Variation of Mean Effective Pressure with Manifold Pressures

12. COMPARISON BETWEEN EXPERIMENTAL AND THEORETICAL RESULTS.

The theoretical indicated mean effective pressures and thermal efficiencies, calculated using the equations summarized in Section 8, are tabulated in Appendix 7, Tables 26 to 30. These theoretical results take into consideration the actual manifold pressures, fresh charge temperatures and air-fuel ratios experienced under test conditions and, therefore, allow a direct comparison to be made with the experimental results.

In Figure 31, indicated mean effective pressure is plotted against back pressure for various intake pressures. The corresponding theoretical curves are seen to be in close agreement with those obtained experimentally, the maximum difference, occurring at the highest intake pressure, being only 12 per cent.

The experimental curves, like the theoretical, are almost straight parallel lines, but the rate of decrease of mean effective pressure with increased back pressure is not as great as predicted theoretically; the reduction in mean effective pressure is approximately 1.5 per cent per pound increase in back pressure at an intake pressure of 13.76 lb./in.² abs. and 3.2 per cent per pound increase at 6.88 lb./in.² abs. compared with corresponding theoretical values of 2.0 per cent and 4.4 per cent. The rate of decrease of mean effective pressure with increased back pressure is independent of intake pressure, the slopes being almost

FIGURE 32

Variation of Mean Effective Pressure with Manifold Pressures.

the same for the theoretical curves and for the experimental curves at all intake pressures; a constant reduction of 1.6 lbs./in.² in experimental mean effective pressure is realised per pound increase in back pressure. As stated earlier, the reduction in mean effective pressure with increased back pressure is caused mainly by the increased quantity of high temperature residual gases trapped in the cylinder when the exhaust valve closes. These gases increase the temperature and reduce the density of the charge at the beginning of compression and, therefore, restrict the weight of fresh charge admitted. In the actual cycle there would be appreciable heat transfer from the residual gases to the cooling water during the suction stroke. The corresponding increase in density of the charge at the beginning of compression would increase the weight of fresh charge admitted per cycle, and therefore, increase the mean effective pressure. Heat transfer through the cylinder walls has been neglected in the theoretical analysis, all processes having been considered as adiabatic.

In Figure 32, mean effective pressure is plotted against intake pressure for constant back pressures of 16 and 22 lb./in.² abs., the curves having been interpolated from those of Figure 31.

Both the experimental and theoretical curves are seen to be in close agreement, the rate of increase in mean effective pressure with intake pressure being greater for

FIGURE 33

Variation of Thermal Efficiency with Manifold Pressures.

the theoretical curves, particularly at higher intake pressures. Friction losses in the intake manifold and across the intake ports would increase with the increased gas flow which occurs at higher intake pressures; this would tend to restrict the weight of fresh charge entering the cylinders and therefore, decrease the mean effective pressure below that predicted theoretically. The two experimental curves and the two theoretical curves appear to have the same slopes for each of the exhaust pressures, indicating that the rate of increase in mean effective pressure with intake pressure is independent of exhaust back pressure. A linear relationship exists between mean effective pressure and intake pressure for the experimental curves, the rate of increase being 11 lb./in.^2 per pound increase in intake pressure.

In Figure 33, thermal efficiency is plotted against back pressure for various values of intake pressure. The corresponding theoretical curves, although somewhat displaced from the experimental, show the same general trend; the reduction in efficiency for the same increase in back pressure is almost identical for curves of equal intake pressure, the reduction, generally, being greater at the lower intake pressures. The slight variations in the experimental air-fuel ratios have caused the curves to be slightly displaced; if the air-fuel ratio had remained constant the efficiency would have increased with intake pressures as was shown in

Figure 21.

The main reason for the efficiencies, as determined experimentally, being lower than the corresponding theoretical values is, no doubt, due to the large heat losses from the working substance to the cooling water during combustion and expansion.

13. CONCLUSIONS.

1. The Thermodynamic Charts prepared by Hottel, Williams and Satterfield¹¹, together with the modifications suggested here, have proved to be of particular value in analysing, theoretically, the Otto cycle under various operating conditions. The accuracy of the charts is shown clearly by the lack of scatter of points calculated from them, as plotted in the accompanying graphs.

For the supercharged cycle or for cycles in which the back pressure is less than the intake pressure, the analysis may be simplified greatly by assuming the residual gases at the end of the exhaust stroke to be compressed isentropically to intake pressure by portion of the incoming charge, as described in Section 3.

2. By an analysis of a number of Otto cycles under various operating conditions, equations may be established for estimating the weight fraction of residual gases and the temperature of the charge at the beginning of compression in terms of the operating variables. From these results, and other equations relating work output per pound of fresh charge with the operating variables, the work done per cycle, the mean effective pressure and the thermal efficiency may be found, irrespective of the values of intake pressure, exhaust pressure, fresh charge temperature, compression ratio and fuel-air ratio.

The equations summarized in Section 8 may be used to

supersede the long-hand graphical method of estimating cycle performance from thermodynamic charts as described in Section 3.

3. Using the results of the computations, Appendices 3 and 4, and the equations summarized in Section 8, an analysis of the effects of the operating variables on engine performance reveals the following:

(i) Intake and Exhaust Pressures.

A marked percentage reduction in mean effective pressure occurs with increased back pressure, particularly at low intake pressures. Reductions of approximately 2 per cent per pound increase in back pressure at 14 lb./in.²abs. intake pressure and 4 per cent per pound increase at 6 lb./in.²abs. intake pressure have been realized.

The thermal efficiency of an engine is reduced with increased back pressure, the effect being more pronounced at low intake pressures. Reductions from 34.3 per cent to 31.5 per cent at 14 lb./in.²abs. intake pressure and from 33.1 per cent to 23.7 per cent at 6 lb./in.²abs. result from a change in back pressure from 6 lb./in.²abs. to 20 lb./in.²abs..

(ii) Fresh Charge Temperature.

Mean effective pressure decreases almost linearly as fresh charge temperature increases, a reduction of 1.5 per cent per 10° F. increase being realized for a temperature range from 500° R. to 700° R.

Variations in fresh charge temperature have negligible

effect on thermal efficiency at low values of manifold pressure ratio, but at high manifold pressure ratios, thermal efficiency decreases steadily as fresh charge temperature increases. At a manifold pressure ratio $\frac{P_b}{P_i}$ of $\frac{20}{6}$, thermal efficiency decreases from 25.2 per cent at a fresh charge temperature of 500°R. to 22.0 per cent at a fresh charge temperature of 700°R.

(iii) Compression Ratio.

Mean effective pressure increases almost linearly from 164 lb./in.²abs. to 201 lb./in.²abs. and from 15.5 lb./in.²abs. to 38.6 lb./in.²abs., at manifold pressure ratios of $\frac{6}{14}$ and $\frac{20}{6}$ respectively, for an increase in compression ratio from 5.5 to 9.5. The corresponding increases in thermal efficiency are from 31.8 per cent to 39.2 per cent and from 20.4 per cent to 29.8 per cent.

(iv) Fuel-Air Ratio.

As the fuel-air ratio is increased from a "lean" value, the mean effective pressure increases rapidly until the chemically correct ratio is reached. The rate of increase then slows down and a peak mean effective pressure is reached at a "rich" mixture strength. Further increases in fuel-air ratio cause the mean effective pressure to decrease slightly. The mixture strength at which the peak mean effective pressure occurs depends on the manifold pressure ratio: at manifold pressure ratios of $\frac{6}{14}$ and $\frac{20}{6}$ respectively, the peak mean effective pressures are reached at fuel-air ratios of 1.1 and

1.3 times that required for complete combustion.

Thermal efficiency, on the other hand, tends to increase slightly at first, and then decrease steadily as fuel-air ratio increases, the rate of decrease being greater at the lower manifold pressure ratios. The mixture strength at which the peak thermal efficiency occurs depends on the manifold pressure ratio. At a pressure ratio of $\frac{20}{6}$, a peak value is reached at a fuel-air ratio of 0.9 times that required for complete combustion. The peak efficiency would be reached at lower fuel-air ratios when using lower manifold pressure ratios.

4. From the results tabulated in Appendices 3 and 4, an analysis of the effects of operating variables on maximum cycle temperatures and pressures reveals the following:

(i) Intake and Exhaust Pressures.

Maximum cycle temperature decreases linearly with increasing back pressure for all values of intake pressure, the rate of decrease being greater at the lower intake pressures: the maximum temperature is reduced by 3°F. and by 7.5°F. per pound increase in back pressure for intake pressures of $14 \text{ lb./in.}^2 \text{ abs.}$ and $6 \text{ lb./in.}^2 \text{ abs.}$ respectively.

Maximum cycle pressure is reduced by 6 lb./in.^2 per pound increase in back pressure, irrespective of intake pressure. Changes in intake pressure, however, have a far more serious effect on maximum cycle pressure, the increase being 75 lb./in.^2 per pound increase in intake pressure.

(ii) Fresh Charge Temperature.

An increase in fresh charge temperature from 500°R. to 700°R. increases the maximum cycle temperature linearly from 5073°R. to 5200°R. , and from 4905°R. to 5000°R. for manifold pressure ratios $\frac{P_b}{P_i}$, of $\frac{6}{14}$ and $\frac{20}{6}$ respectively. This represents an average increase of 5.6°F. per 10°F. increase in fresh charge temperature, that is, an increase in maximum cycle temperature of approximately one half the increase in fresh charge temperature, irrespective of the manifold pressure ratio.

Maximum cycle pressure, on the other hand, decreases linearly by approximately 0.12 per cent per degree Fahrenheit increase in fresh charge temperature, irrespective of manifold pressure ratio. Reductions from $940 \text{ lb./in.}^2\text{abs.}$ to $698 \text{ lb./in.}^2\text{abs.}$ and from $280 \text{ lb./in.}^2\text{abs.}$ to $215 \text{ lb./in.}^2\text{abs.}$, at the lower and higher manifold pressure ratios respectively, are realized for an increase in fresh charge temperature from 500°R. to 700°R.

(iii) Compression Ratio.

Maximum cycle temperature and maximum cycle pressure both increase as compression ratio increases. An increase in compression ratio from 5.5 to 9.5 increases the maximum cycle temperature from 5090°R. to 5237°R. and from 4875°R. to 5095°R. for manifold pressure ratios of $\frac{6}{14}$ and $\frac{20}{6}$ respectively. The corresponding increases in maximum pressure are from $680 \text{ lb./in.}^2\text{abs.}$ to $1230 \text{ lb./in.}^2\text{abs.}$, and from $190 \text{ lb./in.}^2\text{abs.}$

to 420 lb./in.² abs.

(iv) Fuel-Air Ratio.

Maximum cycle temperature at first increases and then decreases with increasing fuel-air ratio, a peak value being reached at a fuel-air ratio greater than the chemically correct value. Peak values of maximum cycle temperature of 5175°R. and 4970°R. were obtained at manifold pressure ratios of $\frac{6}{14}$ and $\frac{20}{6}$ respectively, each at a fuel-air ratio of 1.1 times that required for complete combustion.

Maximum cycle pressures also, at first increase and then decrease with increasing fuel-air ratio, the peak values being reached at a fuel-air ratio of approximately 1.3 times the chemically correct value, for each of the manifold pressure ratios. Peak values of maximum cycle pressure of 820 lb./in.² abs. and 254 lb./in.² abs. are obtained at manifold pressure ratios of $\frac{6}{14}$ and $\frac{20}{6}$ respectively.

5. Comparison between Experimental and Theoretical Results.

The equations summarized in Section 8 appear to be extremely reliable for calculating mean effective pressures of an actual engine. When operating over a range of manifold pressure ratios, fuel-air ratios and intake temperatures, the maximum variation experienced between experimental values obtained from a multi-cylinder engine, and the corresponding theoretical values was only 12 per cent.

The rate of reduction in mean effective pressure with increased back pressure at a constant intake pressure is

slightly less in practice than predicted theoretically. This is caused by heat transfer from residual gases to cooling water which allows a greater weight of fresh charge to be admitted per cycle.

The theoretical rate of increase in mean effective pressure with intake pressure, at constant back pressure is, generally, in close agreement with that obtained experimentally. Frictional losses in the intake system, however, tend to reduce the experimental rate of increase below the theoretical at high intake pressures.

From both experimental and theoretical results, the rate of increase in mean effective pressure with intake pressure is independent of back pressure; also, the rate of decrease in mean effective pressure with increased back pressure is independent of intake pressure. It is found by experiment that a constant reduction of 11 lb./in. in mean effective pressure is realised per pound increase in intake pressure, and a constant increase of 1.6 lb./in.² is realised per pound reduction in back pressure; hence the same gain in mean effective pressure would result from an increase of 1 lb./in.² in intake pressure as would from a reduction of approximately 7 lb./in.² in back pressure.

Heat transfer through the cylinder walls during combustion and expansion causes experimental values of thermal efficiency to be lower than the theoretical values. However, the theoretical analysis appears quite useful for

estimating the rate of change in efficiency with back pressure likely to occur in practice.

REFERENCES

1. Keenan J.H.: "Thermodynamics", John Wiley and Sons, New York, 1941.
2. Moreley T.B. "The Limit of Efficiency of Gas Engines" Engineering, Dec. 1914.
3. Walker W.J.: Thermodynamic Cycles in relation to the Design and Future Development of Internal-Combustion Engines, Proc. Inst. Mech. Engrs., 1920, p.1235.
4. Empire Motor Fuels Committee Report, Proc. Inst. Automotive Engineers, Vol. 18, Part 1, 1923.
5. Gilchrist J.M. Chart for the Investigation of Thermodynamic cycles in Internal Combustion Engines and Turbines.
6. Tanaka and Awano, Report 118, March 1935, and Report 128, September 1935, Aeronautical Research Institute, Tokyo Imperial University.
7. Kuhl, Beilage 373, 6 (1935) Forschung auf dem Gebiete des Ingenieurwesens.
8. Egbert and O'Brien, M.I.T. Internal Report, 1938.
9. Weibe R., Schultz J.F. and Porter J.C., Ind. Eng. Chem., 34, 575 (1942).
10. Hershey R.L., Eberhardt J.E. and Hottel H.C.: "Thermodynamic Properties of the Working Fluid in Internal Combustion Engines", S.A.E. Journal, October, 1936.
11. Hottel H.C., Williams G.C., and Satterfield C.N.:

- "Thermodynamic Charts for Combustion Processes", Parts 1 and 2, John Wiley and Sons, New York, 1949.
12. Bonamy S.E., "Solution of Supercharged Engine Cycles", Automobile Engineer, October, 1959.
 13. Bonamy S.E. and Upfold R.W., "Exhaust Back Pressure", Automobile Engineer, December, 1959.
 14. Goodenough G.A. and Baker J.B., "A Thermodynamic Analysis of Internal Combustion Engine Cycles", University of Illinois Eng. Exp. Sta. Bull., 160, 1927.
 15. Bannister F.K. and Mucklow G.F.: "Wave Action Following Sudden Release of Compressed Gas from a Cylinder, Proc. Inst. Mech. Engrs., 1948, Vol. 159, p.269.
 16. Ricardo H.R.: "The High-Speed Internal Combustion Engine", Blackie and Son Ltd.
 17. Taylor C.F. and Taylor E.S.: "The Internal Combustion Engine", International Text Book Co.
 18. Morse P.M., Boden R.H. and Schechter H.: "Acoustic Vibration and Internal Combustion Engine Performance", Journal App. Phys., Vol. 9, No. 1, Jan. 1938.
 19. Hussman A.W. and Pullman W.A.: "Pressure Fluctuations in Multi-Cylinder Manifolds", Pennsylvania State University, Eng. Research Tech. Paper No. 127, Nov.1957.
 20. B.S. Code No. 1042: 1943 - "Flow Measurement"
 21. Judge A.H.: "The Testing of High Speed Internal Combustion Engines", Chapman and Hall.

14. APPENDICES

APPENDIX 1

Exhaust Pressure greater than Intake Pressure.

Sample Calculation

Consider the case when exhaust pressure, $P_b = 16 \text{ lb./in.}^2\text{abs.}$, intake pressure, $P_1 = 10 \text{ lb./in.}^2\text{abs.}$, temperature of fresh charge, $T_{fr.} = 600^\circ\text{R.}$, compression ratio, $r = 6.5$, fuel-air ratio, $F = 1.2$ times that required for complete combustion.

Note: The weight fraction of residual gases, f and the temperature at the beginning of compression T_1 , used here, are the values obtained after working through this cycle three times. The final calculation, only, is reproduced.

Assume weight fraction of residual gases, $f = 0.0622$ and temperature at beginning of compression, $T_1 = 724^\circ\text{R.}$

Refer to Figure 1.

From Modified Air Chart,* when $F = 1.2$, $f = 0.0622$, $T_1 = 724^\circ\text{R.}$, and $P_1 = 10 \text{ lb./in.}^2\text{abs.}$,

$$E_{S1}^0 = 1320 \text{ B.T.U./mol.}$$

Pressure correction factor,

$$\frac{E_{S1} - E_{S1}^0}{P_1} = -0.18$$

$$\therefore E_{S1} = 1318 \text{ B.T.U./mol.}$$

$$y_1 = 0.03549$$

$$e_{S1} = E_{S1} \cdot y_1 = 46.8 \text{ B.T.U./lb. air constits.}$$

$$V_1 = 775 \text{ c.ft./mol.}$$

* Owing to their size, the Modified Air chart and Burned Mixture charts are not reproduced here. They may be examined in Part 2 of Reference 11.

Volume correction factor for mixture and entropy correction factor for pressure are negligible.

Allowing for entropy correction factor for mixture,

$$S_1 = 6.04 \text{ B.T.U./mol. } ^\circ\text{R.}$$

$$V_2 = \frac{775}{6.5} = 119 \text{ c.ft./mol.}$$

Following an isentropic process to Volume V_2 on the Modified Air Chart, point 2 is located. The entropy correction factor for the mixture, however, must be taken into consideration by measuring the entropy, $S_2 = 6.04$, from the appropriate mixture correction line at $F = 1.2$, $f = 0.0622$.

$$P_2 = 119 \text{ lbs./in.}^2\text{abs.}$$

$$T_2 = 1307^\circ\text{R.}$$

$$E_{S2}^0 = 5020 \text{ B.T.U./mol.}$$

Pressure correction factor,

$$\frac{E_{S2} - E_{S2}^0}{P_2} = -0.08$$

$$\therefore E_{S2} = 5010 \text{ B.T.U./mol.}$$

$$e_{S2} = E_{S2} \cdot y_1 = 177.8 \text{ B.T.U./lb. air constits.}$$

$$e_3 = y_1(E_{S2} - E_{S,560}^0) + (1 - f)1299F + f q_v$$

$$E_{S,560}^0 = 390 \text{ B.T.U./mol.}$$

$$q_v = 336 \text{ B.T.U./lb. air constits.}$$

$$\text{Hence } e_3 = 1647 \text{ B.T.U./lb. air constits.}$$

$$v_3 = V_2 \cdot y_1 = 4.22 \text{ c.ft./lb. air constits.}$$

Point 3 is located on Burned Mixture Chart for $F = 1.2$

$$T_3 = 5064^\circ\text{R.}$$

$$P_3 = 501 \text{ lb./in.}^2\text{abs.}$$

$$s_3 = 0.5908 \text{ B.T.U./lb. air constits. } ^\circ\text{R.}$$

$$v_4 = 6.5 v_3 = 67.3 \text{ c.ft./lb. air constits.}$$

Following an isentropic path on the Burned Mixture Chart ($s_3 = s_4$) gives:

$$T_4 = 3308^\circ\text{R.}$$

$$P_4 = 50.5 \text{ lb./in.}^2\text{abs.}$$

$$v_4 = 1042 \text{ c. ft./lb. air constits.}$$

Continuing the isentropic expansion to point 4' ($s_{4'} = s_4$

$$P_{4'} = P_6 = 16 \text{ lb./in.}^2\text{abs.})$$

$$T_{4'} = T_6 = 2600^\circ\text{R.}$$

$$v_{4'} = v_6 = 67.3 \text{ c.ft./lb. air constits.}$$

$$f = \frac{v_4}{r v_{4'}} = \underline{0.0627} \text{ compared with the assumed}$$

value of 0.0622

Continuing the expansion to point 6' ($s_{6'} = s_3$, $P_{6'} =$
10 lb./in.²abs.)

$$T_{6'} = 2342^\circ\text{R.}$$

$$v_{6'} = 98 \text{ c.ft./lb. air constits.}$$

$$e_{6'} = 765 \text{ B.T.U./lb. air constits.}$$

From Modified Air Chart,

$$H_{S6'} = 16030 \text{ B.T.U./mol., when } T_{6'} = 2342^\circ\text{R.}$$

Sensible Enthalpy of charge at point 1,

$$h_{S1} = y_{6'} f H_{S6'} + (1 - f) y_{fr.} H_{S_{fr.}}$$

$$y_{6'} = 0.03550 \text{ lbs. air constits./mol.}$$

$$y_{fr.} = 0.03527 \text{ lbs. air constits./mol.}$$

$$H_{S_{fr.}} = 1810 \text{ B.T.U./mol.}$$

$$\text{Hence } h_{S1} = 98.94 \text{ B.T.U./lb. air constits.}$$

$$H_{S1} = \frac{h_{S1}}{y_1} = 2788 \text{ B.T.U./mol.}$$

Corresponding temperature, T_1 , from Modified Air Chart = 726°R.
compared with assumed value of 724°R.

The calculated values of T_1 and f compare favourably with the assumed values. Negligible change would occur if the cycle was repeated using the new values.

Work Output, excluding Pumping Work

$$= \frac{(1647 - 1042) - (177.8 - 46.8)}{1 - 0.0627}$$

$$= 505.7 \text{ B.T.U./lb. fresh charge}$$

$$\text{Pumping Work} = \frac{(16 - 10)(6.5 - 1)775 \times 0.03549 \times 144}{(1 - 0.0627) 6.5 \times 778}$$

$$= 27.6 \text{ B.T.U./lb. fresh charge.}$$

$$\text{Net Work Output} = 505.7 - 27.6$$

$$= 478.1 \text{ B.T.U./lb. fresh charge}$$

$$\text{Thermal Efficiency} = \frac{478.1}{0.06775 \times 1.2 \times 19040}$$

$$= \underline{30.9 \text{ per cent.}}$$

$$\text{Volume ratio} = \frac{6.5 - \frac{98.0}{67.3}}{6.5 - 1} = 0.917$$

Weight of fresh charge taken in per cycle

$$= 0.917 \times \frac{1 - 0.0627}{775 \times 0.03549} = 0.313 \text{ lbs./c.ft. of swept vol.}$$

$$\begin{aligned} \text{Mean Effective Pressure} &= 0.313 \times 478.1 \times \frac{778}{144} \\ &= \underline{81.0 \text{ lb./in.}^2} \end{aligned}$$

APPENDIX 2

Exhaust Pressure less than Intake Pressure.

Sample Calculation

Consider the case when exhaust pressure, $P_b = 6 \text{ lb./in.}^2 \text{ abs.}$, intake pressure, $P_i = 14 \text{ lb./in.}^2 \text{ abs.}$, temperature of fresh charge, $T_{fr.} = 600^\circ \text{R.}$, compression ratio, $r = 6.5$, fuel-air ratio, $F = 1.2$ times that required for complete combustion.

Note: The weight fraction of residual gases, f and the temperature at the beginning of compression T_1 , used here, are the values obtained after working through this cycle twice. The final calculation, only, is reproduced.

Assume weight fraction of residual gases, $f = 0.0205$ and temperature at the beginning of compression, $T_1 = 640^\circ \text{R.}$ Following a procedure similar to that in Appendix 1, values at the various points in Figures 2 and 3 are found and tabulated below:

Point	T deg.R.	P lb/in. ² abs.	V c.ft./ mol.	v c.ft./ lb.air constit.	E _S B.T.U. /mol.	e _S B.T.U. /lb.air constit	S B.T.U. /mol. deg.R.	s B.T.U./lb. air constit. deg. R.
1	640	14	492		850	30.0	4.41	
2	1177	169	75.7		4150	146	4.41	
3	5138	818		2.67		1666		0.5582
4	3340	80		17.4		1051		0.5582
4'	1900	6		131		648		0.5582
6	1900	6		131		648		0.5582
6'	2300	14		69.2		753		0.5582

$$f = \frac{17.4}{6.5 \times 131} = \underline{0.0204} \text{ compared with the assumed value of } 0.0205.$$

$$y_1 = 0.03530 \text{ lbs. air constits./mol.}$$

$$y_{6'} = 0.03550 \text{ lbs. air constits./mol.}$$

$$y_{fr.} = 0.03527 \text{ lbs. air constits./mol.}$$

$$H_{S_{fr.}} = 1810 \text{ B.T.U./lb. air constits.}$$

$$H_{S_{6'}} = 15670 \text{ B.T.U./mol.}$$

$$\begin{aligned} \text{Hence } h_{S1} &= 0.03550 \times 0.0204 \times 15670 + 0.03527 \times 0.9796 \times 1810 \\ &= 74.9 \text{ B.T.U./lb. air constits.} \end{aligned}$$

$$H_{S1} = \frac{74.9}{0.03530} = 2122 \text{ B.T.U./mol.}$$

Corresponding temperature, T_1 , from Modified Air Chart = 642°R.
compared with the assumed value of 640°R.

The calculated values of T_1 and f compare favourably with the assumed values. Negligible change would occur if the cycle was repeated using the new values.

Work Output, excluding Pumping Work

$$= \frac{(1666 - 1051) - (146 - 30)}{0.9796}$$

$$= 509 \text{ B.T.U./lb. fresh charge.}$$

$$\text{Pumping Work} = \frac{(6 - 14)(6.5 - 1)492 \times 0.03530 \times 144}{0.9796 \times 6.5 \times 778}$$

$$= -22.2 \text{ B.T.U./lb. fresh charge.}$$

$$\text{Net Work Output} = 509 - (-22.2)$$

$$= 531.2 \text{ B.T.U./lb. fresh charge.}$$

$$\text{Thermal Efficiency} = \frac{531.2}{0.06775 \times 1.2 \times 19040}$$

$$= \underline{34.3 \text{ per cent.}}$$

$$\text{Volume Ratio} = \frac{6.5 - \frac{69.2}{131}}{6.5 - 1} = 1.086$$

Weight of fresh charge taken in per cycle

$$= 1.086 \times \frac{0.9796}{17.4} = 0.0612 \text{ lbs./c.ft. of swept vol.}$$

$$\begin{aligned} \text{Mean Effective Pressure} &= 0.0612 \times 531.2 \times \frac{778}{144} \\ &= \underline{175.6 \text{ lb./in.}^2} \end{aligned}$$

APPENDIX 3Results of Series 1 of Computations from Thermodynamic Charts.TABLE 1. Intake Pressure = 14 lb./in.²abs.

P_b	20	18	16	14	12	10	8	6
E_{S1}	1260	1220	1161	1072	1020	975	900	850
V_1	548	538	531	520	515	508	500	492
T_1	713	705	695	682	670	663	651	640
S_1	5.15	5.04	4.95	4.86	4.76	4.67	4.58	4.41
E_{S2}	4909	4800	4700	4595	4430	4350	4300	4150
V_2	84.3	82.7	81.7	80.0	79.2	78.1	76.9	75.7
T_2	1288	1275	1260	1243	1220	1208	1200	1177
P_2	162	165	165	167	167	167	167	169
e_3	1651	1653	1654	1657	1659	1661	1667	1666
v_3	2.99	2.93	2.90	2.84	2.80	2.76	2.72	2.67
T_3	5100	5110	5107	5110	5120	5128	5137	5138
s_3	.5640	.5635	.5635	.5624	.5605	.5602	.5593	.5582
P_3	720	730	725	750	770	788	804	818
e_4	1043	1045	1047	1050	1046	1050	1050	1051
v_4	19.42	19.07	18.82	18.43	18.23	17.97	17.67	17.36
T_4	3320	3320	3330	3343	3320	3348	3330	3340
P_4	72.0	72.8	73.5	75.8	77.5	77.5	79.0	80.0
$e_{4'}$	816	799	783	760	737	714	684	648
$v_{4'}$	53.0	57.2	62.2	70.0	78.0	90.0	106	131
$T_{4'}$	2520	2465	2410	2328	2240	2150	2040	1900
$e_{6'}$	765	763	763	760	757	757	755	753
$v_{6'}$	70.0	70.0	70.0	70.0	69.5	69.3	69.0	69.2
$T_{6'}$	2340	2332	2332	2328	2330	2315	2302	2300

TABLE 2. Intake Pressure = 12 lb./in.²abs.

P _b	20	18	16	14	12	10	8	6
E _{S1}	1362	1320	1225	1130	1090	1010	940	860
V ₁	656	648	630	621	608	600	588	578
T ₁	732	722	705	692	683	668	657	644
S ₁	5.76	5.65	5.45	5.32	5.21	5.07	4.92	4.78
E _{S2}	5100	4985	4780	4670	4580	4430	4315	4180
V ₂	100.9	99.7	96.9	95.5	93.5	92.3	90.4	88.9
T ₂	1318	1300	1270	1255	1241	1221	1202	1181
P ₂	141	141	140	142	143	142	143	143
e ₃	1647	1649	1646	1652	1656	1660	1662	1664
v ₃	3.58	3.54	3.44	3.39	3.31	3.27	3.20	3.14
T ₃	5070	5080	5075	5096	5105	5110	5125	5122
s ₃	.5774	.5772	.5740	.5738	.5740	.5728	.5720	.5702
P ₃	598	605	626	632	642	662	670	688
e ₄	1038	1039	1041	1041	1048	1048	1050	1050
v ₄	23.3	23.0	22.3	22.0	21.5	21.2	20.8	20.4
T ₄	3300	3310	3310	3304	3330	3321	3332	3341
P ₄	59.3	59.5	62.5	63.0	64.8	66.1	67.1	68.5
e _{4'}	844	826	803	780	760	734	704	666
v _{4'}	55.0	59.5	64.7	72.0	81.2	93.0	110	137
T _{4'}	2620	2570	2480	2400	2328	2225	2115	1970
e _{6'}	767	767	761	760	760	759	758	754
v _{6'}	82.0	81.5	81.0	81.0	81.2	81.0	80.0	80.0
T _{6'}	2350	2350	2325	2322	2328	2320	2318	2305

TABLE 3. Intake Pressure = 10 lb./in.²abs.

P _b	20	18	16	14	12	10	8	6
E _{S1}	1500	1420	1320	1260	1160	1055	980	920
V ₁	805	795	775	758	741	728	712	700
T ₁	755	740	724	710	695	678	665	653
S ₁	6.37	6.19	6.04	5.89	5.70	5.57	5.37	5.23
E _{S2}	5360	5190	5010	4920	4740	4570	4410	4300
V ₂	124	122	119	117	114	112	110	108
T ₂	1354	1332	1307	1293	1266	1241	1218	1200
P ₂	118	118	119	120	120	120	120	119
e ₃	1643	1645	1647	1651	1654	1657	1660	1664
v ₃	4.40	4.33	4.22	4.14	4.04	3.97	3.88	3.81
T ₃	5053	5055	5064	5072	5078	5090	5093	5108
s ₃	.5930	.5921	.5908	.5892	.5878	.5876	.5864	.5855
P ₃	483	488	501	519	531	538	551	563
e ₄	1038	1040	1042	1042	1044	1048	1050	1053
v ₄	28.6	28.2	27.5	26.9	26.3	25.8	25.2	24.7
T ₄	3292	3304	3308	3310	3315	3325	3336	3352
P ₄	48.0	49.1	50.5	51.8	52.9	54.3	55.6	56.9
e _{4'}	875	857	836	811	786	760	728	689
v _{4'}	57.8	62.1	67.3	74.8	84.0	97.0	114	144
T _{4'}	2740	2675	2600	2505	2418	2322	2204	2061
e _{6'}	769	767	765	762	761	760	757	756
v _{6'}	99.0	98.5	98.0	98.2	97.0	97.0	97.2	96.2
T _{6'}	2355	2352	2342	2338	2322	2322	2315	2310

TABLE 4. Intake Pressure = 8 lb./in.²abs.

P _b	20	18	16	14	12	10	8	6
E _{S1}	1730	1630	1525	1420	1310	1180	1060	960
V ₁	1062	1046	1022	995	964	938	906	882
T ₁	794	777	760	740	720	698	678	664
S ₁	7.18	7.06	6.85	6.64	6.43	6.16	5.97	5.76
E _{S2}	5780	5610	5460	5210	4980	4700	4540	4370
V ₂	163	161	157	153	148	144	139	136
T ₂	1416	1392	1368	1336	1302	1258	1233	1208
P ₂	93.8	93.9	94.8	94.8	94.8	94.8	95.9	97.4
e ₃	1634	1639	1644	1646	1648	1650	1656	1661
v ₃	5.81	5.72	5.59	5.43	5.26	5.11	4.93	4.80
T ₃	5010	5018	5034	5046	5045	5054	5062	5082
s ₃	.6128	.6128	.6118	.6100	.6081	.6058	.6044	.6042
P ₃	361	368	378	391	402	417	432	438
e ₄	1032	1034	1038	1040	1044	1044	1047	1053
v ₄	37.8	37.2	36.4	35.3	34.2	33.2	32.1	31.2
T ₄	3278	3282	3295	3310	3318	3318	3326	3342
P ₄	36.3	36.6	37.0	39.0	40.8	42.0	43.6	44.8
e _{4'}	919	901	875	854	826	793	758	720
v _{4'}	60.8	65.8	71.8	78.8	89.0	102	122	152
T _{4'}	2888	2833	2750	2662	2561	2441	2316	2178
e _{6'}	774	774	772	768	765	762	758	758
v _{6'}	125	124	124	123	123	122	122	122
T _{6'}	2375	2375	2370	2358	2339	2328	2316	2320

TABLE 5. Intake Pressure = 6 lb./in.²abs.

P _b	20	18	16	14	12	10	8	6
E _{S1}	2170	2000	1850	1700	1545	1380	1220	1070
V ₁	1546	1504	1472	1410	1374	1316	1262	1224
T ₁	867	840	813	788	760	733	705	679
S ₁	8.47	8.22	7.96	7.69	7.46	7.16	6.83	6.56
E _{S2}	6550	6280	5950	5690	5430	5120	4780	4530
V ₂	238	231	226	217	211	202	194	188
T ₂	1530	1490	1442	1405	1365	1323	1271	1232
P ₂	69.1	70.3	68.0	70.0	70.2	70.2	70.4	70.6
e ₃	1620	1625	1629	1635	1643	1648	1652	1658
v ₃	8.49	8.25	8.06	7.72	7.51	7.19	6.88	6.67
T ₃	4950	4967	4969	4989	5004	5022	5036	5045
s ₃	.6396	.6381	.6373	.6351	.6347	.6323	.6292	.6286
P ₃	247	254	260	273	280	293	310	318
e ₄	1024	1027	1031	1034	1038	1043	1044	1046
v ₄	55.2	53.7	52.4	50.2	48.8	46.7	44.8	44.1
T ₄	3252	3261	3274	3282	3300	3305	3314	3323
P ₄	24.5	25.4	26.4	27.4	28.4	29.6	30.9	31.3
e _{4'}	984	962	937	907	881	845	805	763
v _{4'}	65.0	70.5	77.2	85.3	96.3	110	130	162
T _{4'}	3112	3040	2954	2850	2750	2630	2488	2332
e _{6'}	783	780	778	774	774	770	764	763
v _{6'}	167	167	166	166	166	164	163	162
T _{6'}	2410	2400	2395	2378	2375	2360	2338	2332

TABLES 6 and 7

Calculated Results from Series 1 of computations, Tables 1 and 2

Back Press. P_b lb./in ² abs.	Corrected f	Enthalpy H_{s1} B.T.U./ mol.	Corrected T_1 deg. R.	Work/lb.fr. chge., excl. Pump Work B.T.U.	Pump Work /lb. fr. charge B.T.U.	Net Work /lb. fr. charge B.T.U.	Thermal Effic. per cent	Volume Ratio	Fr.chge/ unit swept volume lbs.	Mean Effect. Pressure lb./in. ²
TABLE 6. Intake Pressure = 14 lb./in. ² abs.										
20	.0564	2691	714	508	-19.4	489	31.6	0.942	.0458	121.0
18	.0513	2607	705	507	-12.6	494	31.9	0.959	.0477	127.5
16	.0465	2532	694	504	- 6.2	498	32.2	0.977	.0495	133.4
14	.0405	2435	682	502	0	502	32.4	1.000	.0521	141.6
12	.0359	2367	672	510	+ 5.9	516	33.3	1.020	.0540	150.7
10	.0307	2281	660	507	+11.6	519	33.5	1.042	.0562	157.8
8	.0256	2201	651	510	+17.1	527	34.0	1.064	.0587	166.9
6	.0204	2122	642	509	+22.2	531	34.3	1.086	.0612	175.6
TABLE 7. Intake Pressure = 12 lb./in. ² abs.										
20	.0651	2831	733	509	-31	478	30.9	0.911	.0366	94.4
18	.0595	2742	719	510	-23	487	31.5	0.933	.0382	100.5
16	.0532	2631	707	506	-15	491	31.7	0.954	.0404	107.0
14	.0471	2535	693	509	- 7	502	32.4	0.977	.0423	114.5
12	.0408	2441	682	505	0	505	32.6	1.000	.0446	121.6
10	.0352	2353	670	509	+ 7	516	33.3	1.023	.0466	129.7
8	.0292	2257	657	507	+13	520	33.6	1.049	.0490	137.8
6	.0229	2161	644	508	+20	528	34.1	1.076	.0515	147.0

TABLES 8 and 9

Calculated Results from Series 1 of Computations, Tables 3 and 4

Back Press. P_b lb./in. ² abs.	Corrected f	Enthalpy H_{s1} B.T.U./ mol.	Corrected T_1 deg. R.	Work/lb.fr. chge., excl. Pump Work B.T.U.	Pump Work /lb. fr. charge B.T.U.	Net Work /lb. fr. charge B.T.U.	Thermal Effic. per cent	Volume Ratio	Fr.chge/ unit swept volume lbs.	Mean Effect. Pressure lb./in. ²
TABLE 8. Intake Pressure = 10 lb./in. ² abs.										
20	.0761	3006	756	506	-49	458	29.6	0.870	.0281	69.7
18	.0697	2903	740	506	-38	468	30.2	0.893	.0295	74.6
16	.0627	2788	726	506	-28	478	30.9	0.917	.0313	81.0
14	.0553	2671	710	507	-18	489	31.6	0.943	.0331	87.5
12	.0481	2552	694	508	-9	499	32.2	0.972	.0352	95.1
10	.0409	2441	682	505	0	505	32.6	1.000	.0372	101.6
8	.0340	2332	668	506	+8	514	33.2	1.026	.0393	109.1
6	.0265	2216	652	505	+16	521	33.7	1.060	.0418	117.8
TABLE 9. Intake Pressure = 8 lb./in. ² abs.										
20	.0956	3329	795	506	-79	428	27.6	0.809	.0194	44.8
18	.0869	3192	778	508	-64	444	28.7	0.839	.0206	49.4
16	.0779	3046	758	505	-49	456	29.5	0.866	.0219	54.0
14	.0689	2895	739	506	-36	471	30.4	0.898	.0237	60.3
12	.0591	2729	718	503	-22	482	31.1	0.930	.0256	66.7
10	.0500	2585	698	507	-11	496	32.0	0.965	.0276	74.0
8	.0405	2433	680	506	0	506	32.7	1.000	.0299	81.7
6	.0316	2282	664	503	+10	513	33.1	1.036	.0322	89.3

TABLE 10

Calculated Results from Series 1 of Computations, Table 5.

Back Press. P_b lb./in. ² abs.	Corrected f	Enthalpy H_{s1} B.T.U./ mol.	Corrected T_1 deg. R.	Work/lb.fr. chge., excl. Pump Work B.T.U.	Pump Work /lb.fr. charge B.T.U.	Net Work /lb. fr. charge B.T.U.	Thermal Effic. per cent	Volume Ratio	Fr.chge./ unit swept volume lbs.	Mean Effect. Pressure lb./in. ²
TABLE 10. Intake Pressure = 6 lb./in. ² abs.										
20	.1306	3924	867	506	-139	367	23.7	0.715	.0113	22.3
18	.1170	3695	839	504	-114	390	25.2	0.751	.0124	26.0
16	.1044	3490	814	505	- 92	413	26.7	0.791	.0135	30.2
14	.0905	3250	786	505	- 69	436	28.1	0.828	.0150	35.3
12	.0780	3049	760	506	- 50	456	29.5	0.868	.0164	40.5
10	.0651	2837	731	505	- 31	474	30.6	0.912	.0183	46.8
8	.0523	2625	706	508	- 15	493	31.9	0.955	.0202	53.9
6	.0412	2450	681	510	0	510	33.0	1.000	.0217	59.9

APPENDIX 4

Results of Series 2 of Computations from Thermodynamic Charts.

TABLE 11. Fresh Charge Temperature = 500°R to 700°R.
Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{20}{6}$

$T_{fr.}$	500	550	600	650	700
E_{S1}	1500	1790	2170	2550	2880
V_1	1355	1450	1546	1675	1755
T_1	753	804	867	932	983
S_1	7.37	7.86	8.47	9.02	9.46
E_{S2}	5370	5890	6550	7220	7850
V_2	208	223	238	258	270
T_2	1358	1433	1530	1619	1704
P_2	70.8	69.3	69.0	67.8	68.5
e_3	1592	1606	1620	1636	1653
v_3	7.44	7.96	8.49	9.21	9.66
T_3	4907	4930	4950	4976	5005
s_3	.6233	.6320	.6396	.6490	.6560
P_3	280	259	247	227	218
e_4	1005	1016	1024	1036	1045
v_4	48.4	51.8	55.2	59.9	62.8
T_4	3190	3225	3252	3290	3320
P_4	27.4	25.9	24.5	23.0	22.1
$e_{4'}$	945	965	984	1007	1026
$v_{4'}$	62.2	63.7	65.0	67.1	67.8
$T_{4'}$	2960	3050	3112	3197	3260
$e_{6'}$	754	769	783	801	815
$v_{6'}$	160	164	167	172	175
$T_{6'}$	2300	2355	2410	2472	2515

TABLE 12. Manifold Pressure Ratio $\frac{P_b}{P_i} = \frac{6}{14}$
 Fresh Charge Temperature = 500°R to 700°R

$T_{fr.}$	500	550	600	650	700
E_{S1}	260	555	850	1160	1470
V_1	417	452	492	534	573
T_1	537	590	640	696	748
S_1	3.08	3.78	4.41	5.05	5.62
E_{S2}	3050	3600	4150	4710	5250
V_2	64.1	69.5	75.7	82.1	88.1
T_2	1008	1095	1177	1259	1337
P_2	168	172	169	167	164
e_3	1630	1649	1666	1686	1703
v_3	2.27	2.46	2.67	2.90	3.11
T_3	5073	5105	5138	5170	5200
s_3	.5390	.5490	.5582	.5686	.5774
P_3	938	877	818	754	708
e_4	1030	1042	1051	1065	1076
v_4	14.7	16.0	17.4	18.9	20.3
T_4	3070	3305	3340	3386	3426
P_4	93.0	86.8	80.0	75.4	70.5
$e_{4'}$	620	635	648	663	676
$v_{4'}$	125	129	131	136	140
$T_{4'}$	1795	1850	1900	1960	2012
$e_{6'}$	719	736	753	772	789
$v_{6'}$	65.7	67.5	69.2	71.0	72.7
$T_{6'}$	2173	2230	2300	2370	2428

TABLE 13. Compression ratio = 5.5 to 9.5
 Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{20}{6}$

r	5.5	6.5	7.5	8.5	9.5
E_{S1}	2570	2170	1900	1720	1520
V_1	1675	1546	1470	1400	1356
T_1	934	867	822	781	754
S_1	9.14	8.47	8.03	7.63	7.37
E_{S2}	6770	6550	6490	6420	6410
V_2	305	238	196	165	143
T_2	1557	1530	1517	1507	1505
P_2	55.0	69.2	83.0	99.0	113.5
e_3	1591	1620	1640	1657	1670
v_3	10.91	8.49	6.99	5.86	5.08
T_3	4877	4950	5010	5058	5093
s_3	.6630	.6396	.6285	.6182	.6092
P_3	188	247	300	365	422
e_4	1048	1024	1002	982	962
v_4	60.0	55.2	52.4	49.8	48.2
T_4	3360	3252	3005	3110	3040
P_4	23.2	24.5	25.5	26.2	26.5
$e_{4'}$	1018	984	957	932	911
$v_{4'}$	67.7	65.0	63.2	61.6	60.0
$T_{4'}$	3230	3112	3015	2933	2842
$e_{6'}$	810	783	763	745	730
$v_{6'}$	174	167	162	157	154
$T_{6'}$	2500	2410	2335	2270	2210

TABLE 14. Compression Ratio = 5.5 to 9.5
 Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{6}{14}$

r	5.5	6.5	7.5	8.5	9.5
E_{S1}	870	850	810	780	750
V_1	496	492	487	485	482
T_1	646	640	634	630	624
S_1	4.49	4.41	4.32	4.30	4.24
E_{S2}	3800	4150	4380	4660	4860
V_2	90.2	75.7	64.9	57.1	50.7
T_2	1124	1177	1211	1253	1285
P_2	134	169	200	238	275
e_3	1651	1666	1678	1690	1699
v_3	3.19	2.67	2.29	2.02	1.79
T_3	5090	5138	5175	5210	5238
s_3	.5694	.5582	.5490	.5412	.5338
P_3	675	818	955	1094	1233
e_4	1087	1051	1025	1000	978
v_4	17.5	17.4	17.2	17.1	17.0
T_4	3462	3340	3252	3170	3092
P_4	82.8	80.0	79.5	77.8	76.2
$e_{4'}$	665	648	635	623	613
$v_{4'}$	137	131	128	125	123
$T_{4'}$	1967	1900	1845	1808	1765
$e_{6'}$	774	753	737	722	710
$v_{6'}$	70.8	69.2	67.2	66.3	64.9
$T_{6'}$	2375	2300	2240	2185	2140

TABLE 15. Fuel-Air Ratio, $F = 0.8$ to 1.5

$$\text{Manifold Pressure Ratio, } \frac{P_b}{P_i} = \frac{20}{6}$$

F	0.8	0.9	1.0	1.1	1.2	1.5
E_{S1}	2170	2330	2410	2270	2170	2020
V_1	1585	1630	1650	1595	1546	1492
T_1	883	905	915	888	867	832
S_1	8.49	8.68	8.84	8.62	8.47	8.23
E_{S2}	6750	6910	7030	6850	6550	6230
V_2	244	251	254	245	238	230
T_2	1610	1615	1620	1580	1530	1450
P_2	71.3	69.2	69.0	69.3	69.0	68.3
e_3	1112	1236	1365	1492	1620	2010
v_3	8.59	8.85	8.98	8.72	8.49	8.35
T_3	4630	4805	4925	4970	4950	4655
s_3	.5290	.5610	.5910	.6155	.6396	.7092
P_3	214	217	221	236	247	250
e_4	592	677	774	893	1024	1431
v_4	55.9	57.6	58.4	56.7	55.2	54.3
T_4	3040	3296	3547	3405	3252	2868
P_4	21.4	22.5	24.2	24.5	24.5	23.5
$e_{4'}$	580	655	735	853	984	1401
$v_{4'}$	58.8	63.2	68.0	66.6	65.0	61.9
$T_{4'}$	3000	3222	3430	3267	3112	2765
$e_{6'}$	399	456	518	646	783	1213
$v_{6'}$	151	166	178	172	167	158
$T_{6'}$	2320	2504	2695	2544	2410	2110

TABLE 16. Fuel - Air Ratio, $F = 0.8$ to 1.5
 Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{6}{14}$

F	0.8	0.9	1.0	1.1	1.2	1.5
E_{S1}	810	820	830	850	850	845
V_1	492	492	492	493	492	488
T_1	642	642	642	644	640	635
S_1	4.33	4.36	4.38	4.43	4.41	4.43
E_{S2}	4160	4170	4120	4190	4150	4090
V_2	75.7	75.7	75.7	75.8	75.7	75.1
T_2	1212	1207	1188	1189	1177	1143
P_2	173	173	170	169	169	165
e_3	1150	1279	1406	1538	1666	2055
v_3	2.65	2.66	2.66	2.67	2.67	2.67
T_3	4773	4970	5120	5175	5138	4809
s_3	.4517	.4808	.5080	.5348	.5582	.6237
P_3	708	748	780	800	818	810
e_4	615	703	798	922	1051	1458
v_4	17.2	17.3	17.3	17.3	17.4	17.4
T_4	3130	3380	3635	3500	3340	2955
P_4	71.1	77.3	83.5	82.2	80.0	76.3
e_4'	276	316	362	504	648	1091
v_4'	119	129	140	136	131	123
T_4'	1820	1965	2125	2010	1900	1650
e_6'	370	421	475	623	753	1189
v_6'	61.5	66.3	72.2	70.6	69.2	64.8
T_6'	2205	2375	2545	2419	2300	2020

TABLES 17 and 18. Fresh Charge Temperature = 500°R to 700°R

Fresh Chge. Temp. T _{fr} . deg. R.	Corrected f	Enthalpy H _{s1} B.T.U./ mol.	Corrected T ₁ deg. R.	Work/lb.fr. chge., excl. Pump Work B.T.U.	Pump Work /lb. fr. charge B.T.U.	Net Work /lb. fr. charge B.T.U.	Thermal Effic. per cent	Volume Ratio	Fr.chge./ unit swept volume lbs.	Mean Effect. Pressure lb./in. ²
TABLE 17. Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{20}{6}$										
500	.1196	2964	749	510.3	-120.7	389.6	25.2	.7153	.0130	26.4
550	.1249	3436	809	508.0	-129.9	378.1	24.4	.7149	.0121	24.6
600	.1306	3924	867	508.0	-139.4	368.6	23.7	.7150	.0113	22.3
650	.1373	4436	933	504.6	-152.5	352.1	22.6	.7155	.0103	19.5
700	.1424	4870	983	504.0	-160.8	343.2	22.0	.7123	.0097	17.9
TABLE 18. Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{6}{14}$										
500	.0182	1329	537	511.1	+ 18.9	530.0	34.2	1.086	.0723	207.1
550	.0191	1717	588	509.1	+ 20.5	529.6	34.2	1.087	.0668	191.0
600	.0204	2122	642	505.9	+ 22.2	528.1	34.3	1.086	.0612	175.6
650	.0214	2526	695	505.9	+ 23.7	529.6	34.2	1.087	.0564	161.5
700	.0222	2928	743	502.0	+ 26.0	528.0	34.3	1.087	.0525	150.5

TABLES 19 and 20. Compression Ratio = 5.5 to 9.5

Compression Ratio r	Corrected f	Enthalpy H_{S1} B.T.U./ mol.	Corrected T_1 deg. R.	Work/lb.fr. chge., excl. Pump Work B.T.U.	Pump Work /lb. fr. charge B.T.U.	Net work /lb. fr. charge B.T.U.	Thermal Effic. per cent	Volume Ratio	Fr.chge./ unit swept volume lbs.	Mean Effect. Pressure lb./in. ²
TABLE 19. Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{20}{6}$										
5.5	.1612	4483	937	467.6	-151.7	315.9	20.4	.6511	.0091	15.5
6.5	.1306	3924	867	505.8	-139.4	366.4	23.7	.7150	.0113	22.3
7.5	.1106	3519	818	532.8	-132.4	400.4	25.9	.7595	.0129	27.9
8.5	.0951	3222	781	560.6	-125.9	434.7	28.1	.7934	.0144	33.9
9.5	.0847	3021	755	583.9	-122.2	461.7	29.8	.8167	.0155	38.6
TABLE 20. Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{6}{14}$										
5.5	.0234	2184	649	471.1	+ 21.8	492.9	31.8	1.107	.0616	164.1
6.5	.0204	2122	642	509.0	+ 22.2	531.2	34.3	1.086	.0612	175.6
7.5	.0179	2070	632	536.6	+ 22.5	559.1	36.1	1.073	.0613	185.3
8.5	.0161	2039	630	561.9	+ 22.8	584.7	37.8	1.063	.0611	192.9
9.5	.0146	2011	625	584.0	+ 22.8	606.8	39.2	1.056	.0612	200.5

TABLES 21 and 22. Fuel - Air Ratio, $F = 0.8$ to 1.5

Fuel-Air Ratio F	Corrected f	Enthalpy H_{s1} B.T.U./ mol.	Corrected T_1	Work/lb.fr. chge., excl. Pump Work B.T.U.	Pump Work /lb. fr. charge B.T.U.	Net Work /lb. fr. charge B.T.U.	Thermal Effic. per cent	Volume Ratio	Fr.chge./ unit swept volume lbs.	Mean Effect. Pressure lb./in. ²
TABLE 21. Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{20}{6}$										
0.8	.1461	3898	880	420.7	-143.7	277.0	26.8	0.716	.0109	16.4
0.9	.1400	4099	902	461.7	-147.0	314.7	27.1	0.706	.0105	17.9
1.0	.1321	4223	914	508.0	-147.8	350.2	26.7	0.706	.0105	19.5
1.1	.1309	4073	891	502.1	-143.3	358.8	25.3	0.712	.0109	21.1
1.2	.1306	3924	867	505.8	-139.4	366.4	23.7	0.715	.0113	22.3
1.5	.1349	3665	830	506.5	-137.8	368.7	18.3	0.719	.0115	21.9
TABLE 22. Manifold Pressure Ratio, $\frac{P_b}{P_i} = \frac{6}{14}$										
0.8	.0224	2070	640	427.1	+ 22.1	449.2	43.5	1.088	.0618	149.9
0.9	.0206	2097	642	467.5	+ 22.2	489.7	42.2	1.088	.0617	163.2
1.0	.0190	2105	642	503.0	+ 22.1	525.1	40.6	1.088	.0617	174.7
1.1	.0197	2120	642	508.2	+ 22.2	530.4	37.4	1.087	.0614	176.0
1.2	.0204	2122	642	509.0	+ 22.2	531.2	34.3	1.086	.0612	175.6
1.5	.0217	2120	635	501.7	+ 22.3	524.0	26.6	1.086	.0611	172.0

APPENDIX 5

Calculated Values of Output and Thermal Efficiency from
Equations summarized in Section 8.

TABLE 23. $P_b = P_i = 14.7 \text{ lb./in.}^2 \text{abs.}$, $T_{fr.} = 520^\circ \text{R.}$, $r = 9.0$

Per cent of theor. air.	Fuel-Air ratio F	Residual Gas fract. f	Temp. T_1 deg. R.	Net Work /lb. fr. charge B.T.U.	Thermal Effic. per cent	Mean Effect. Press. lb./in. ²
80	1.250	.0280	574	580.4	36.0	200.5
90	1.111	.0286	580	580.4	40.5	203.7
100	1.000	.0291	587	580.4	45.0	201.8
110	0.909	.0300	582	536.7	45.8	188.4
120	0.833	.0307	579	500.3	46.5	176.6

APPENDIX 6

Results of Morse Test at 2500 R.P.M.

$$B.H.P. = \frac{W \times N}{400} = \frac{B.M.E.P. \times S \times N}{33000 \times 12 \times 2}$$

where

W = Dynamometer Balance reading - lbs.

N = Speed - R.P.M.

S = Engine Displacement = 132.5 ins.³

∴ B.M.E.P. = 14.94 W

TABLE 24

Test No.		1	2	3	4	5	6	7	8
Dynamometer Balance Reading - lbs.									
W (6 cyls)		6.91	6.02	4.86	3.88	2.99	2.21	1.73	1.06
Cylinder "Shorted Out"	1	5.49	4.70	3.74	2.90	2.17	1.49	1.00	0.55
	2	5.42	4.77	3.78	2.95	2.18	1.50	1.01	0.56
	3	5.51	4.74	3.72	2.94	2.18	1.50	1.01	0.55
	4	5.50	4.76	3.73	2.93	2.17	1.51	1.00	0.55
	5	5.54	4.76	3.75	2.94	2.19	1.49	1.00	0.56
	6	5.37	4.74	3.73	2.91	2.15	1.49	1.01	0.55
Indicated Load W _I		8.63	7.65	6.71	5.71	4.87	4.29	3.76	3.04
Mech. Eff. W per cent W _I		80.07	78.69	72.43	67.95	61.29	51.50	46.00	34.87
B.M.E.P. lb./in. ²		103.2	89.9	72.6	58.0	44.6	33.0	25.8	15.8

APPENDIX 7Results of Increased Back Pressure Tests.TABLE 25

Test No.	Intake Press. ins. Hg.	Exhaust Press. ins. Hg.	Brake Load lbs.	Time to consume 150 mls. fuel-secs.	Orifice Press drop ins. Colger Oil	Mean Intake Temp. °C
1	15.65	0.39	1.59	67.8	1.67	41
2	15.75	4.17	1.42	66.9	2.13	36
3	15.78	7.28	1.22	69.7	2.04	36
4	15.72	11.50	1.02	70.5	1.92	39
5	15.69	15.38	0.78	71.3	1.83	39
6	11.91	0.67	2.89	52.5	3.80	35
7	11.84	4.05	2.82	52.5	3.77	34
8	11.78	7.62	2.67	53.0	3.70	32
9	12.05	11.97	2.33	55.3	3.40	33
10	11.83	15.55	2.21	55.2	3.37	35
11	12.00	18.70	1.94	57.0	3.15	36
12	8.01	1.10	4.42	44.2	5.93	34
13	7.85	5.23	4.26	44.2	5.93	36
14	7.94	7.50	4.10	45.2	5.71	36
15	7.91	11.98	3.88	45.6	5.60	36
16	7.80	15.76	3.72	46.2	5.52	36
17	7.77	19.18	3.48	46.2	5.32	36

Results of Increased Back Pressure Tests

TABLE 25 (cont'd)

Test No.	Intake Press. ins. Hg.	Exhaust Press. ins. Hg.	Brake Load lbs.	Time to consume 150 mls. fuel-secs.	Orifice Press. drop ins. Colger Oil	Mean Intake Temp. °C
18	3.81	1.67	5.96	33.4	9.03	32
19	3.76	4.53	5.84	33.7	8.92	31
20	3.89	7.65	5.62	33.6	8.64	31
21	3.71	11.96	5.42	33.9	8.45	32
22	3.72	15.88	5.22	34.6	8.25	31
23	3.77	19.90	4.94	35.6	8.03	32
24	1.72	1.97	6.73	33.0	10.60	32
25	1.65	4.95	6.56	33.4	10.40	31
26	1.71	8.48	6.32	33.2	10.18	31
27	1.67	11.51	6.17	33.8	9.85	32
28	1.67	15.75	5.93	34.1	9.74	31
29	1.65	19.73	5.72	34.4	9.46	32

Specific Gravity of Fuel = 0.721

Specific Gravity of Colger Oil = 0.814

Orifice diameter = 1.357 ins.

Orifice Tappings: D and $\frac{D}{2}$

Barometer Reading = 29.77 ins. Hg.

Mean Dry Bulb Temp. = 83°F

Mean Wet Bulb Temp. = 66°F

TABLES 26 to 28Deductions from Experimental Results and correspondingTheoretical Values.

Exhaust Press. lb./in. ² abs.	Fuel Cons. lb./ min.	Air Cons. lb./ min.	Air/ Fuel Ratio	B.M.E.P. lb./in. ²	I.M.E.P. lb./in. ²	Therm. Effic. %	Theor. Values	
							M.E.P. lb./ in. ²	Therm. Effic. %
<u>TABLE 26:</u> Tests 1 to 5; Mean Intake Press. = 6.88 lb./in. ² abs.								
14.78	0.211	2.08	9.85	23.75	54.2	23.89	54.5	25.95
16.63	0.214	2.35	10.95	21.21	52.0	22.62	49.7	25.11
18.15	0.206	2.30	11.15	18.23	48.2	21.84	45.3	24.37
20.22	0.203	2.23	10.95	15.24	46.2	21.17	41.3	23.29
22.12	0.201	2.18	10.82	11.65	40.2	18.63	37.1	22.23
<u>TABLE 27:</u> Tests 6 to 11; Mean Intake Press. = 8.76 lb./in. ² abs.								
14.92	0.273	3.14	11.48	43.18	72.6	24.78	74.1	28.89
16.57	0.273	3.12	11.42	42.13	71.8	24.51	69.0	28.34
18.32	0.270	3.10	11.42	39.89	69.4	23.91	63.9	27.71
20.45	0.259	2.97	11.42	34.81	64.9	23.34	57.9	26.91
22.21	0.260	2.95	11.36	33.02	63.5	22.79	52.7	26.20
23.75	0.251	2.86	11.36	28.98	59.4	22.01	49.5	25.56
<u>TABLE 28:</u> Tests 12 to 17; Mean Intake Press. = 10.73 lb./in. ² abs.								
15.13	0.324	3.92	12.08	66.03	93.3	26.81	99.4	31.36
17.15	0.324	3.92	12.08	63.64	90.9	26.12	92.3	30.79
18.26	0.317	3.85	12.12	61.25	89.0	26.16	88.5	30.46
20.46	0.314	3.81	12.12	57.97	85.5	25.35	81.6	29.78
22.31	0.310	3.79	12.20	55.58	83.5	25.08	76.1	29.17
23.99	0.310	3.71	11.95	51.99	80.5	24.18	71.3	28.59

TABLES 29 and 30

Deductions from Experimental Results and corresponding Theoretical Values

Exhaust Press. lb./in. ²	Fuel Cons. lb./ min.	Air Cons. lb./ min.	Air/ Fuel ratio	B.M.E.P. lb./in. ²	I.M.E.P. lb./in. ²	Therm. Effic. %	Theor. Values	
							M.E.P. lb./ in. ²	Therm. Effic. %

TABLE 29: Tests 18 to 23; Mean Intake Press. = 12.73 lb./in.²abs.

15.40	0.429	4.84	11.25	89.04	114.3	24.82	129.0	29.54
16.81	0.425	4.81	11.27	87.25	112.4	24.63	125.1	29.26
18.34	0.426	4.73	11.08	83.96	109.4	23.90	118.4	28.95
20.45	0.423	4.68	10.95	80.97	106.5	23.47	111.3	28.50
22.37	0.414	4.62	11.13	77.99	104.0	23.39	105.0	28.07
24.34	0.403	4.57	11.32	73.80	100.1	23.17	98.8	27.61

TABLE 30: Tests 24 to 29; Mean Intake Press. = 13.76 lb./in.²abs.

15.55	0.434	5.24	12.05	100.55	125.3	26.89	143.3	31.91
17.01	0.429	5.19	12.08	98.01	122.7	26.65	137.7	31.63
18.74	0.432	5.13	11.88	94.42	119.1	25.71	131.5	31.29
20.23	0.424	5.05	12.17	92.17	117.3	25.77	125.7	30.98
22.30	0.420	5.02	11.92	88.59	113.6	25.19	119.0	30.54
24.26	0.417	4.95	11.85	85.46	110.7	24.76	112.8	30.10