COMPARISON OF ACTUAL ENGINE CYCLE WITH
THEORETICAL FUEL-AIR CYCLE.

by

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Submitted in Partial Fulfillment of the
Requirements for the Degree of
Master of Science
(i.e., Bachelor of Science)

and

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Submitted in Partial Fulfillment of the
Requirements for the Degree of
Bachelor of Science

from the
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1950

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ACKNOWLEDGEMENTS

The authors wish to express their sincere appreciation to Professor P. M. Ku who first suggested this problem, and subsequently gave his invaluable advice and supervision throughout the work of this thesis.

The authors also wish to express their gratitude to Professor W. A. Leary, Professor A. R. Rogowski, Mr. J. C. Livengood, Mr. D. H. Tsai, and the staff at the Sloan Laboratory, especially Mr. D. Doremus, for their willing and ever-available advice and help, without which this thesis would not have been possible.
Dear Sir:

In partial fulfillment of the requirements for the degrees of Master of Science and Bachelor of Science at the Massachusetts Institute of Technology, we are pleased to submit herewith a thesis entitled COMPARISON OF ACTUAL ENGINE CYCLE WITH THEORETICAL FUEL-AIR CYCLE.

Yours very truly,

Jacob George Bartas

Jules John Van Deun
TABLE OF CONTENTS

Frontpiece
Abstract
Equipment and Procedure
Discussion of Results
Sources of Error
Tabulated Results
Graphical Results
Suggestions for Future Research
Appendix

- Table of Symbols
- Sample Calculations
- Experimental Data
- P-V Diagrams
- Indicator Diagrams
- Theoretical Fuel-Air Graphs
- Indicator Schematics
- Engine Schematic
- Photographs of Equipment
- Bibliography
ABSTRACT

The object of the thesis was to compare the actual engine cycle characteristics with those of the best available theoretical cycle by using the equivalent cycle as a means of interpreting the latter, and p-v diagrams to measure the actual performance.

The actual efficiency of the engine does not change with varying inlet temperature, inlet pressure, and exhaust pressure. For these conditions the value of the ratio of the actual efficiency to the equivalent efficiency is .82 to .84.

The rate of increase of the equivalent cycle efficiency compared with that of the actual cycle efficiency was found to be greater at higher compression ratios.

When the ratio of the efficiencies was plotted against the fuel-air ratio, the values were constant for each of the compression ratios used, varying from .88 at a compression ratio of 4 to .76 at a compression ratio of 10.

It was found that the rate of increase of the imep/Pi with increasing compression ratio is less for the actual cycle than for the equivalent cycle. With regard to the other variables, fuel-air ratio, and inlet and exhaust pressure, the trends of both the actual and the equivalent cycle were identical.

The comparison of the equivalent cycle characteristics with the theoretical constant volume fuel-air cycle characteristics shows that for the majority of the parameters the
trends are identical, but the range of the respective values differ to an appreciable extent.

In the case of volumetric efficiency, it was found that it increased for the equivalent cycle rather than decreasing as it does for the theoretical cycle.
EQUIPMENT AND PROCEDURE

The engine used was a $3\frac{1}{4} \times 4\frac{1}{2}$ variable compression ratio Cooperative Fuel Research engine, usually referred to as a C. F. R. engine. The compression ratio was varied by changing clearance volume, a calibrated micrometer being used to insure accuracy.

A strobotac was used in conjunction with a tachometer to maintain a constant speed of 1200 RPM.

Engine power was absorbed by a balanced direct current dynamometer with an M. I. T. hydraulic load scale giving brake load readings in inches of mercury.

Oil and jacket temperatures were kept at 150° F and 180° F, respectively, by injecting steam and cold water, in varying amounts, into the oil heat exchanger and the engine jacket. An oil pump kept the oil pressure at the desired level of 60 psi.

To eliminate the effects of atmospheric pressure changes from day to day a throttle valve was employed to stabilize exhaust pressure at 31.5 inches of mercury during all "sea level" runs. A vacuum pump controlled the exhaust pressure at all "altitude" runs. For the same reason, the inlet pressure was throttled to 28.5 inches of mercury.

The mass rate of flow of air was measured by using a 0.515 inch A. S. M. E. square edged orifice in accordance with the standard procedure as given in bibliography.
reference #2. The equation used was:

$$M_{ac} = C \sqrt{\frac{P_o \cdot \Delta P}{T_o}} \cdot \frac{K}{K_m} \cdot \frac{Y}{Y_m}$$

where:

$$C = 0.0194 \text{ lb/sec} \sqrt{\frac{M_a \cdot \text{H}_2\text{O}}{\text{R}}}$$

a constant derived to minimize computing time during operation. The value of "C" contains all constants in the general flow equation in addition to such dimensional corrections as were necessary to make possible the calculation of "Ma" directly from instrument readings. The corrections for the expansion factor, "Y", and the flow coefficient, "K", were accomplished by using graphs of $Y/Y_m$ vs. $p/p_i$ and $K/K_m$ vs. $K_{au}$ as was suggested in the above-mentioned reference.

To eliminate the possibility of leakage in the air induction system developing during operation, the following procedure was used. Before taking any data, the entire air induction system was critically examined for leaks. When all possible leakage was eliminated, the time for the inlet pressure to drop from 12 inches of mercury to 10 inches of mercury was measured, with the entrance end of the system and the inlet and exhaust valves closed. The result obtained was indicative of the rate of leakage through the valve guides. At frequent intervals during the taking of data, this procedure was repeated and the result compared with the initial results to detect any development of leaks.

The fuel rate of flow was measured by a modified Fischer and Porter rotometer. The standard float was removed and a heavier and longer float was substituted
which had no oscillating tendencies and allowed the fuel flow desired in the range of operations. The rotometer was calibrated by means of a wet test gas meter.

It was decided before the thesis was started to use a gaseous fuel (butane) to facilitate good mixing and to maintain close control over the fuel-air ratio. Endeavoring to lower the cost, the rotometer was first calibrated using compressed air. Then using an equation derived by Fischer and Porter, which showed that the mass rate of flow was proportional to the square root of the density, a theoretical curve for butane was plotted. Check runs with butane showed that the actual and theoretical curves failed to coincide. Further calibration using carbon dioxide followed the same pattern. All runs were made at 25 inches of mercury and temperatures of 71°F to 74°F before the rotometer.

After extensive research in literature and considerable discussion with the thesis advisor it was decided that differences in viscosity could not be neglected. Fischer and Porter's manual suggested the employment of a special float which minimizes viscosity effects in cases where they are not negligible. Since none of the special floats were available, and there is doubt that oscillation would be damped out due to the extreme lightness of the float, an attempt was made to find an empirical formula. After much time had been wasted in pursuit of this idea the project was abandoned and the rotometer was calibrated with butane.
It is of interest to note that low rates of flow appeared to be directly proportional to the viscosity of the various gases. However, at large rates of flow the discrepancies were beyond the limits of experimental error.

The gaseous fuel was obtained from a cylinder containing 96% pure butane. It was decided to keep the pressure before the rotometer at 25 inches of mercury (gage) to ensure adequate fuel feeding during supercharging operations. Since the vapor pressure of butane at room temperature is approximately 40 psi, 25 inches gage appeared to be sufficient. However, at large rates of flow it was discovered that the butane vaporized so rapidly and the requirements of the heating of vaporization were so high as to lower the temperature of the liquid to the point where the vapor pressure could not deliver the gas at 25 inches of mercury.

Therefore, the top of an empty oil drum was removed, inlet steam and water pipes were added, a drain pipe was attached, and the cylinder of butane was immersed in this makeshift heat exchanger. By keeping the water in the drum at about 120°F, gas at 25 inches pressure could be delivered at the rotometer under the most severe operating conditions with relative ease.

Initially, the calibration curve was represented by a graph of pounds of fuel per second vs. rotometer reading. Since the ratio $F/F_{cc}$ represented a controlled variable, another calibration chart representing rotometer reading vs. rate of flow of air for the various values of $F/F_{cc}$
was constructed to facilitate more rapid adjustment of the fuel-air ratio.

Once the butane and air had been measured they were mixed in the vaporizing tank. Again, the inlet pressure was measured by means of a mercury manometer and the temperature kept at the required level by injecting steam and water in desired amounts into the vaporizing tank jacket. When the inlet conditions required very elevated temperatures the air was preheated by means of an electric coil placed around the air pipe before the vaporizing tank.

After all inlet conditions had been stabilized at desired points, the best power spark advance was determined by two different methods. First, the spark was varied until the optimum brake load was obtained. This reading was checked by setting the spark at various readings until a position was found which resulted in the maximum RPM. The dynamometer was so adjusted that the maximum speed was 1200 RPM when it was finally attained. These adjustments were conveniently made by setting a trimmer near the spark control allowing one operator to handle both duties with excellent coordination.

All indicator cards were taken on the M. I. T. high speed indicator, a schematic diagram of which is shown in the appendix.

The indicator cards were then transferred to p-v diagrams by using the p-v table.

The areas of the various p-v diagrams were then determined by using a planimeter.
DISCUSSION OF RESULTS

The object of the thesis was to compare the actual engine cycle characteristics with those of the best available theoretical cycle by using the equivalent cycle as a means of interpreting the latter, and p-v diagrams to measure the actual performance.

It was expected that the actual cycle would differ from the equivalent cycle since many important factors such as:

- direct heat losses
- exhaust heat losses
- combustion time losses
- leakage

are neglected in the definition of the equivalent cycle. Other differences result from the manner in which the fuel-air charts were constructed, variances of specific heats with increasing temperatures, and the fact that although chemical equilibrium is approached at low fuel-air ratios, it is never actually realized. The first of these factors effects the equivalent cycle, while the latter two are noticed in the actual cycle.

It was realized that unavoidable discrepancies would occur due to experimental errors in the calibration of the rotometer and in the air measurements, and also due to such factors as poor distribution.
EFFICIENCY

It was found that varying inlet temperature, inlet pressure, and exhaust pressure has negligible effect on both the actual and equivalent cycle efficiencies. Curves of both actual and equivalent cycle efficiencies plotted against these particular variables were straight lines. Therefore, the ratios of actual efficiency to the equivalent efficiency was a constant within experimental error, where the limits of this ratio were .82 and .84 for both cases.

Both actual and equivalent efficiencies increased with increasing compression ratio. It was noted that the rate of change of the actual efficiency was the smaller of the two. It should be remembered that heat losses are not taken into consideration in computing the equivalent efficiencies, while in actual operation they are of considerable importance. The heat loss effects are greater at the higher compression ratios due to the higher values of maximum cycle temperature which cause higher average temperature differences between engine parts and the surroundings. It will be recalled that the greatest percentage of the heat losses occur in the vicinity of top-dead-center which coincides with the region of maximum cycle temperatures, and also that the higher pressures accompanying greater compression ratios result in greater leakage rates. It follows from this reasoning that the ratio of actual efficiency to equivalent efficiency should drop off at higher compression ratios. The results show that this parameter
decreases from 0.87 at a compression ratio of 4, to 0.77 at a compression ratio of 10.

When this ratio of the efficiencies was plotted against the fuel-air ratio, there was a slight tendency for it to increase with increasing fuel-air ratio. The rate of change of this ratio is extremely small, and consequently it is believed that the shape of the curve is a result of a combination of experimental variations and irregularities in the Hottel charts used for high fuel-air ratio calculations.

Naturally, as was expected, the range of the actual efficiency was lower than that for equivalent efficiency. Assumption of constant volume burning, constant volume exhaust, and neglect of heat losses and leakage in the calculation of the equivalent cycle, make the equivalent efficiency considerably greater.

I.M.E.P.

The work per cycle is the controlling factor with regard to imep. The combustion time losses and exhaust time losses, which were neglected for the equivalent cycle, have a predominant effect on the work per cycle. Consideration of these quantities results in a lower efficiency for the actual cycle, and therefore lower work per cycle. The final result is a lower value of imep for the actual cycle than for the equivalent cycle at the same compression ratio.

The rate of increase of the $\text{imep}/P_i$ with increasing compression ratio is less for the actual cycle than for the equivalent cycle. This follows as a consequence of the
manner in which the ratio of the actual efficiency to the theoretical efficiency varied with changing compression ratio.

The curve of the equivalent imep vs. fuel-air ratio was very similar to that of the actual cycle. Again it was noticed that at the highest fuel-air ratio there was a slight tendency for the actual curve to deviate from the equivalent curve. The trend here follows the same pattern as that of actual and equivalent efficiencies for the same reasons.

The variation of imep/P_i with changing inlet and exhaust pressure, and inlet temperature for the actual and the equivalent cycles showed identical trends. The absolute values of the imep/P_i for the two cases follow the same pattern as those for the case discussed above.

**EQUIVALENT CYCLE CHARACTERISTICS**

For the majority of the equivalent cycle parameters, the trends were identical to those of the theoretical constant volume fuel-air cycle characteristics calculated by Prof. P. M. Ku within experimental error. The few exceptions are mentioned below.

When T_3/T_1 was plotted against F/F_{cc} for the equivalent cycle calculations, the curve had a tendency to fall off at high fuel-air ratios which is contrary to the curves previously plotted by Prof. P. M. Ku for the theoretical constant volume fuel-air cycle characteristics. This discrepancy is attributed to the inaccuracy of the Hottel chart used for runs at F/F_{cc} of 1.429.

In the case of the volumetric efficiency, two variations
from the above-mentioned, previously calculated results for the theoretical cycle characteristics were noticed. It was found that the volumetric efficiency variations were in accordance with those of changing compression ratio and fuel-air ratio, and changing inlet and exhaust pressures.

In the case where the inlet temperature was increased, the equivalent curve sloped upward, whereas the theoretical curve dropped off to a negligible extent. The primary reason for this behavior is again a consequence of heat transfer. As the inlet temperature is increased, the temperature differential between the engine parts and entering charge decreases. Consequently there is less heat transfer to the entering charge. At the lower inlet temperatures, the first portion of the charge to enter the cylinder expands when coming into contact with the hot engine parts, and hinders the entrance of the subsequent portions of the charge, especially that portion which is in the vicinity of the inlet valve when it closes.

Visual comparison of the equivalent and theoretical curves is often misleading since in many cases it was necessary to deviate from the scale used in the theoretical graphs.
The thesis results indicate that the efforts expended to minimize all possible sources of error were effective. The care taken to insure accuracy in air flow measurements is an example of this.

As was mentioned in the Equipment and Procedure section, the rate of flow of air past the valve guides was measured before any data runs were taken. Subsequent readings were taken and compared with the original ones to detect any leaks which could have developed in the air induction system. When supercharging at high inlet pressures, the mixing tank safety valve was checked periodically for leakage.

Seemingly trivial possibilities for error were avoided whenever possible. For instance: the compression ratio was always adjusted while the engine was cold to assure identical settings at all times, eliminating differences in cylinder size which occur at varying temperatures.

Nevertheless, many errors were unavoidable. Much difficulty was experienced with spark advance since the same brake load reading could often be obtained over a spark range of from 4 to 6 degrees.

As mentioned above, the purity of the butane was 96%. The compositions of the remaining components were unknown, although they were probably quite similar to those of butane. Data obtained operating on a full cylinder or a nearly empty cylinder may be in error, since butane was not the only fuel burned, depending on whether the vapor pressures of the other
components were lower or higher or both.

While inlet pressure was readily stabilized, the exhaust pressure had a tendency to oscillate. Under severe operating conditions the variations were as high as plus or minus .3 inches of mercury from the desired value.

Temperature control by means of steam and cold water was easily and accurately accomplished. Variations in inlet temperature, all jacket and tank temperatures, and oil temperature were kept within plus or minus 2 degrees Fahrenheit with a minimum of effort.

There appeared to be considerable voltage fluctuation in the line during certain hours of the day. During operations on a weekday afternoon the speed may have varied as much as plus or minus 10 rpm over a period of 5 minutes. Running under identical conditions on a Saturday morning produced negligible changes of speed in time interval of thirty minutes or more. This was especially noticed during attempts to set best power spark advance by using the maximum speed method as described in the Equipment and Procedure section.

The computations of actual cycle quantities can only be as accurate as the original indicator diagrams. A relative indication of the accuracy of an indicator diagram can be obtained by noting the amount of scattering on the diagram. It was learned from experience that a contaminated diaphragm in the pressure pickup unit is the major contributor to scattering on the inlet portion of the curve. Therefore, after every 5 hours of operation, including motoring time and warmup time,
the pressure pickup unit was removed from the cylinder, dis-
mantled, and thoroughly cleaned.

A considerable amount of scattering occurred during the combustion stroke on the indicator card due to inhomogeneous burning. Therefore, it was a matter of technique to draw the line representing the mean value of the points on the indicator diagram. Although this effect had little bearing on the area of the p-v diagram, it is responsible for considerable error in the determination of the maximum pressure of the actual cycle.

There is also unavoidable human error in converting indicator cards to p-v diagrams and in measuring the area of said diagrams.

Of possible major interest is the accuracy of the Hottel chart used in equivalent cycle calculations at an $F/\bar{F}_{cc}$ of 1.429. The relation between atmospheric pressure and the adjoining pressure lines is definitely wrong. A deviation is easily noticeable without measurement. This obvious error casts some doubt on the accuracy of the entire chart.

In general, many of the errors mentioned above had negligible effect on the thesis results.
| Run | $\theta_1$ | $\theta_2$ | $\theta_3$ | $\theta_4$ | $\theta_5$ | $\theta_6$ | $\theta_7$ | $\theta_8$ | $\theta_9$ | $\theta_{10}$ | $\theta_{11}$ | $\theta_{12}$ | $\theta_{13}$ | $\theta_{14}$ | $\theta_{15}$ | $\theta_{16}$ | $\theta_{17}$ | $\theta_{18}$ | $\theta_{19}$ | $\theta_{20}$ | $\theta_{21}$ | $\theta_{22}$ | $\theta_{23}$ | $\theta_{24}$ | $\theta_{25}$ | $\theta_{26}$ | $\theta_{27}$ | $\theta_{28}$ | $\theta_{29}$ | $\theta_{30}$ | $\theta_{31}$ | $\theta_{32}$ | $\theta_{33}$ | $\theta_{34}$ | $\theta_{35}$ | $\theta_{36}$ | $\theta_{37}$ | $\theta_{38}$ | $\theta_{39}$ | $\theta_{40}$ |
|-----|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|
GRAPHICAL RESULTS

SYMBOLS

\( r \) or \( T_{1}^{\circ}R \) as abscissa:

\[ \frac{F}{F_{cc}} \]

\[ \begin{array}{c}
  \circ : 0.8 \\
  \sigma : 0.91 \\
  \omega : 1.0 \\
  \rho : 1.175 \\
  \phi : 1.429 \\
\end{array} \]

\( \frac{F}{F_{cc}} \) as abscissa:

\[ r \]

\[ \begin{array}{c}
  \sigma : 4 \\
  \omega : 6 \\
  \rho : 8 \\
  \phi : 10 \\
\end{array} \]

\( \frac{P_{e}}{P_{1}} \) as abscissa:

\[ \begin{array}{c}
  \sigma : P_{e} \text{ non-standard} \\
  \omega : P_{1} \text{ non-standard} \\
  \rho : T_{1} 550^{\circ}R \\
  \phi : T_{1} 650^{\circ}R \\
\end{array} \]
equation: cubic efficiency vs. \( r/r_c \) to \( r_t/r_t \), $\eta_t$. $T_1 = 550^\circ R$, $650^\circ R$
Equivalent Cycle Imb /i vs. F/Fe, F'/Fe', T/E.
Equivalent-Cylinder, \( \frac{F}{F_c} \) vs. \( \frac{T}{T_c} \), \( T_c \).
Equivalent Cycle T/F vs. F/Fo, F/4, T/Fo, T/8.
Equivalent Cycle $F_c/F$ vs. $F_c$, $P_c/F$, $T_i P_r$.
SUGGESTIONS FOR FUTURE RESEARCH

The authors would like to make several suggestions which they feel may be profitably employed during any future research of a similar nature. These suggestions follow from difficulties encountered as the thesis progressed, and could not be employed since the time element made changes in the experimental setup impossible.

Since the fuel-air ratio was a controlled variable, it was highly desirable to have a system of calibration whereby the fuel-air ratio could be regulated immediately after the calculation of the rate of flow of air. In other words, a system such as the one used during this thesis was found to be very convenient.

Naturally, accuracy should never be sacrificed to promote convenience. Therefore, it is recommended that unless a critical examination of the effects of high pressures in and fluctuating pressures after the rotometer shows no undesirable effects on the initial calibration of the instrument, some other means should be used to meter the fuel flow.

The chances of poor mixing occurring were eliminated to a great extent by the use of the gaseous fuel. This same feature makes possible another step toward more perfect mixing without too much effort. By inserting an orifice similar to the one used to measure the air flow somewhere between the place where the gaseous fuel is injected into the air stream and the inlet to the mixing tank - the purpose being to create a turbulent zone through which the fuel and air must both flow - the
probability of even a slight degree of poor mixing will be eliminated.

Once mixing is uniform, it would be advisable to install two spark plugs for several runs to determine the effect of combustion time on the various cycles.

It was found that backfires occurred more frequently when the fuel used was gaseous than when a liquid fuel was used, particularly when the change was made from motoring to firing conditions. To eliminate this tendency, some device, a distributor for instance, should be used to prevent the ignition spark which occurs when the exhaust valve is open.

As a final suggestion, considerable effort should be exerted in planning the taking and evaluation of data to detect operating errors as soon as possible after each day's operations have been completed. The taking of data should be planned so that if a series of runs taken on the same day are subject to some operating error unique to that day's operation, the error will be easily recognized upon plotting the results, permitting wiser selection of check runs.

It would be of considerable interest to plot actual, equivalent, and theoretical fuel-air cycle results on one graph. Dimensionless parameters, such as actual efficiency divided by theoretical efficiency, composed of various combinations of the above-mentioned results should also be plotted. Comparison on this basis would be extremely valuable in determining which type of calculations are required to obtain desired results. Questions such as: "Will the theoretical fuel-air cycle yield an answer of sufficient accuracy for a particular purpose or should additional time be spent in calculating an equivalent cycle?" will be readily answered.
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>Air flow, lbs./intake stroke</td>
</tr>
<tr>
<td>B.L.</td>
<td>Brake load, in. Hg.</td>
</tr>
<tr>
<td>Btu.</td>
<td>778 ft. - lbs.</td>
</tr>
<tr>
<td>e</td>
<td>Actual efficiency</td>
</tr>
<tr>
<td>e_i</td>
<td>Indicated equivalent efficiency</td>
</tr>
<tr>
<td>e_V</td>
<td>Volumetric efficiency</td>
</tr>
<tr>
<td>E</td>
<td>Internal energy, Btu's.</td>
</tr>
<tr>
<td>E_c</td>
<td>Energy of combustion, Btu's.</td>
</tr>
<tr>
<td>E_s</td>
<td>Sensible energy, Btu's.</td>
</tr>
<tr>
<td>f</td>
<td>Fraction of residual gas, °/o.</td>
</tr>
<tr>
<td>°F</td>
<td>Temperature, degrees Fahrenheit</td>
</tr>
<tr>
<td>F</td>
<td>Fuel-air ratio</td>
</tr>
<tr>
<td>F_c</td>
<td>Chemically correct fuel-air ratio, .065</td>
</tr>
<tr>
<td>F_cc</td>
<td>Chemically correct fuel-air ratio, .065</td>
</tr>
<tr>
<td>imep</td>
<td>Indicated mean effective pressure</td>
</tr>
<tr>
<td>J</td>
<td>Joule's constant, 778 ft. - lbs/Btu.</td>
</tr>
<tr>
<td>k</td>
<td>Indicator spring constant, lbs/in.</td>
</tr>
<tr>
<td>K</td>
<td>F/F_cc</td>
</tr>
<tr>
<td>K/K_m</td>
<td>Approach velocity correction factor.</td>
</tr>
<tr>
<td>L</td>
<td>Length of p-v diagram, in.</td>
</tr>
<tr>
<td>M_a</td>
<td>Mass rate flow of air</td>
</tr>
<tr>
<td>M_ac</td>
<td>Mass rate flow of air, corrected</td>
</tr>
<tr>
<td>M_au</td>
<td>Mass rate flow of air, uncorrected</td>
</tr>
<tr>
<td>N</td>
<td>Engine speed, RPM</td>
</tr>
<tr>
<td>psia</td>
<td>Lbs/in.^2, absolute</td>
</tr>
<tr>
<td>p-v</td>
<td>Pressure - volume</td>
</tr>
</tbody>
</table>
\( \Delta p \) : Pressure drop, in. of H\(_2\)O

\( P \) : Pressure, lbs/in.\(^2\)

\( P_e \) : Exhaust Pressure

\( P_i \) : Inlet Pressure

\( P_o \) : Pressure before orifice

\( P_{\text{max}} \) : Maximum actual cycle pressure

\( r \) : Compression ratio

\( R \) : Temperature, degrees Rankine

S.A.b.p. : Best power spark advance

T : Temperature

\( T_e \) : Exhaust temperature

\( T_i \) : Inlet Temperature

V : Cylinder volume

\( V \) : Chart volume

\( V_d \) : Displacement volume

W : Work

\( Y/Y_m \) : Expansion correction factor

All numerical and capital letter subscripts refer to points on the p-v diagram as shown in the appendix.
SAMPLE CALCULATIONS ON RUN 4-4-3

Actual Cycle

Indicated mean effective pressure, actual efficiency, and maximum pressure were calculated by using the equations listed below and results measured from indicator and p-v diagrams. These equations are readily derived and are found in numerous sources including those noted in the bibliography.

Indicated mean effective pressure:

\[
imep = \frac{\text{Work/cycle}}{V_d}
\]

As mentioned above the work can be found by measuring the area of the p-v diagram and multiplying by the length of the diagram and an appropriate spring constant. Determining the imep is analogous. The area is divided by the length to give the average height of the diagram. Multiplying this height by the spring constant gives the indicated mean effective pressure of the cycle.

\[
imep = \frac{\text{Area} \times k}{L} = \frac{5.44 \times 100}{5} = 108.8 \text{ lb/in}^2
\]

Actual efficiency:

\[
e_{\text{i}} = \frac{\text{Work/cycle}}{\text{Heat added/cycle}}
\]

The method of determining work is explained above. The amount of heat added is calculated by multiplying the amount of fuel added by its heating value.
\[
e_i = \frac{\text{imep} \times V_d \times \frac{N}{2}}{K \times F_{cc} \times M_a \times E_c} \times \frac{1}{12 \times 778 \times 60}
\]

where the units of:

- \(12 = \text{in/ft}\)
- \(778 = \text{ft-lb/Btu}\)
- \(60 = \text{sec/min}\)

The following quantities were held constant for all runs:

- \(V_d = 37.33 \text{ in}^3\)
- \(N = 1200 \text{ RPM}\)
- \(E_c = 20,000 \text{ Btu/lb of Butane}\)
- \(F_{cc} = .065\)

Substitution shows that:

\[
e_i = 3.08 \times 10^{-5} \frac{\text{imep}}{K} \times M_a
\]

\[
= 3.08 \times 10^{-5} \frac{108.8}{100} \times .0111 = .3018
\]

**Maximum pressure:**

Maximum pressure is easily found by measuring the overall height of the p-v diagram and multiplying by the spring constant.

\[
P_{\text{max}} = \text{height} \times k = 4.56 \times 100 = 456 \text{ lb/in}^2
\]

Another useful parameter is the amount of air utilized per stroke calculated by dividing the amount of air flow per second by the number of intake strokes per second.

\[
B = \frac{60 \times M_a}{N/2} = \frac{60 \times .0111}{600} = .00111 \text{ lb/ stroke}
\]
SAMPLE CALCULATIONS ON RUN 4-4-3

Desired pressures and temperatures in the various parts of the actual cycle were determined by the equivalent cycle method. Comparison of the theoretical indicator diagram with the actual indicator diagram on this basis eliminates differences due to the fact that the volumetric efficiency of the actual run is not equal to the volumetric efficiency assumed for the ideal process.

The volume scale of the thermodynamic charts is based on one pound of air, including both the air contained in the fresh charge and the remaining air in the residual gas.

According to this definition of air, the weight of "air" taken in per stroke is the weight of air taken in divided by (1-f). The chart volume corresponding to any point on the indicator diagram will therefore be:

\[
\bar{V} = \frac{V(1-f)}{B}
\]

Here, B has been measured, but f must be found by trial and error. The following method was employed:

1. At an appropriate point on the indicator diagram measure the pressure and crank angle. The actual volume occupied by the air is then determined.

\[
V_A = \text{Displacement} \div \text{Clearance} = \frac{0.01815}{0.00432} = 0.2247 \text{ ft}^3
\]
2. Assume a reasonable value of $f$ and calculate the chart volume at this point from the above equation.

Assume $f = .057$

$$V_A = \frac{V_A(1-f)}{B} = \frac{.02247(1-.057)}{.00111} = 19.1 \text{ ft}^3$$

$P_A = 60 \text{ lbs/sq. in.} \quad \text{(from indicator card)}$

3. Determine point (5) at exhaust pressure by use of the "burned" thermodynamic charts.

$$P_5 = P_{\text{exhaust}} = 15.5 \text{ lbs/sq. in.}$$

$$V_5 = 56 \text{ ft}^3$$

4. Find the chart volume at point (2) and check to see if the resulting $f$ coincides with the assumed $f$. If not, repeat the process until a reasonable check is obtained.

$$V_2 = \frac{V_2(1-f)}{B} = \frac{.00432(1-.057)}{.00111} = 3.67 \text{ ft}^3$$

$$f = \frac{V_2}{V_5} = \frac{3.67}{56} = .0655 \quad \text{(doesn't check)}$$

Assume $f = .065$

$$V_A = 18.9 \text{ ft}^3$$

$$V_5 = 56 \text{ ft}^3$$

$$V_2 = 3.64 \text{ ft}^3$$

$$f = .065 \quad \text{(checks with assumed value and will be used from here on)}$$
5. Now locate a point (D) on an appropriate, easily-read portion of the compression stroke.

\[ V_D = \text{Displacement Clearance} = 0.0975 \times 0.00432 = 0.01407 \]

\[ P_D = 25 \text{ lbs/sq. in.} \]

From this point (D) begin construction of a fuel-air cycle by means of the thermodynamic charts.

\[ V_d = \frac{V_d(1-f)}{B} = \frac{0.01407(1-0.065)}{0.00111} = 11.82 \text{ ft}^3 \]

6. Isentropic expansion from (D) to (1) using "unburned mixture" charts.

\[ V_1 = 21.8 \text{ ft}^3 \]

\[ T_1 = 620^\circ R \]

\[ E_{S1} = 21 \text{ Btu} \]

7. Isentropic compression from (D) to (2) using "unburned mixture" charts.

\[ V_2 = 3.64 \text{ ft}^3 \]

\[ P_2 = 116 \text{ psia.} \]

\[ T_2 = 1135^\circ R \]

\[ E_{S2} = 129 \text{ Btu} \]

8. Constant volume burning from (2) to (3) using "burned" charts.

\[ E \text{ of comb.} = 1300(1-f) = 1300(1-0.065) = 1215 \text{ Btu} \]

\[ E_3 = E \text{ of comb.} \times E_{S2} = 1215 \times 129 = 1344 \text{ Btu} \]

\[ V_3 = 3.64 \text{ ft}^3 \]
\[ P_3 = 555 \text{ psia.} \]
\[ T_3 = 4945^\circ \text{ R} \]

9. Isentropic expansion from (3) to (4).

\[ V_4 = V_1 = 21.8 \text{ ft}^3 \]
\[ P_4 = 68 \text{ psia.} \]
\[ T_4 = 3550^\circ \text{ R} \]
\[ E_4 = 778 \text{ Btu} \]

10. At this point it is convenient to determine \( T_5 \) and check \( V_5 \) and \( P_5 \).

\[ V_5 = 56 \text{ ft}^3 \]
\[ P_5 = 15.5 \text{ psia.} \]
\[ T_5 = 2110^\circ \text{ R} \]

11. To calculate the results of the fuel-air cycle it is necessary to establish \( E_1 \).

\[ E_1 = E \text{ of comb.} \]
\[ E_{sl} = 1215 \text{ Btu} \]

12. Results of fuel-air cycle.

\[ W_1 = E_1 - E_4 = 1236 - 778 = 458 \text{ Btu} \]
\[ \text{imep} = \frac{W \times J}{V_1-V_2} = \frac{458 \times 778}{144(21.3 - 3.64)} = 136.3 \text{ psia.} \]
\[ e_1 = \frac{W}{E \text{ of comb.}} = \frac{458}{1215} = .377 \]
DATA SHEET

For all runs:

Speed.........................1200 R.P.M.
Oil Temperature..................150°F
Jacket Temperature................180°F
Oil Pressure......................60 p.s.i.

SYMBOLS

\( T_i \) : Inlet temperature.
\( P_i \) : Inlet pressure.
\( P_e \) : Exhaust pressure.
\( r \) : Compression ratio.
\( F \) : Fuel air ratio.
\( F_{cc} \) : Chemically correct fuel air ratio.
\( M_a \) : Mass rate of flow of air.
\( B.L. \) : Brake load.
\( S.A.B.p. \) : Best power spark advance.
\( k \) : Indicator spring constant.
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Run #11

Run #12
Run #15

Run #16
Run #19

Run #20
Run #25

Run #26
Run #36

Run #37
Run #40
The engine inlet pressure was kept constant at 28.7 inches of mercury, except when it was a variable, by means of a throttle valve in order that the fluctuation in the daily barometer reading would not affect our test.

The engine exhaust pressure was maintained between 30.1 to 30.8 inches of mercury except when it was a variable. A vacuum pump was used to control the exhaust pressure when it was a variable between 15 to 45 inches of mercury.
REPRODUCTIONS OF INDICATOR DIAGRAMS
ARE AVAILABLE IN MASTER COPY IN
CENTRAL LIBRARY
Constant-volume Fuel-air Cycle Characteristics

Part I. Variation of \( \text{imep}, \text{imep}, \frac{P_e}{P_i}, \frac{F}{F_c}, \) and \( f \) with \( r, \frac{P_e}{P_i}, \) and \( T_i \).
Constant-volume Fuel-air Cycle Characteristics

Part II Variation of $\frac{T_1}{T_i}$, $\frac{T_2}{T_i}$, $\frac{T_3}{T_i}$, $\frac{T_4}{T_i}$, and $\frac{T_5}{T_i}$ with $r$, $\frac{F}{F_c}$, $\frac{P_e}{P_i}$, and $T_i$
Constant-volume Fuel-air Cycle Characteristics

Part III Variation of $\frac{P_2}{P_1}$, $P_3$, $P_4$, $\frac{T_4}{T_1}$, and $e_v$ with $r$, $\frac{F}{F_c}$, $P_e$, and $T_i$
OUTPUT - 350V D.C.

110V AC

200W
2 mfd.

FG17

100,000 ohm
0.125 mfd.

SPARK COIL

10,000 ohm

RF choke

0.001 mfd

2 Megohm

335 mH

75 mH

3000 ohm

45000 ohm

49000 ohm

Three Pole Double Throw Switch

Make

Break

GROUND

TRIP CIRCUIT - M.I.T INDICATOR
Figure 6.- Construction of pick-up unit of M.I.T. balanced-pressure indicator.
1. Indicator Pickup
2. Knockmeter Pickup
3. Rate of Pressure Pickup
4. Spark Plug
BIBLIOGRAPHY


