QUASI EQUILIBRIUM FUEL-AIR HEAT BALANCED CYCLE ANALYSIS (U)

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United States Naval Academy
Annapolis, Maryland

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compared. Performance parameters for both cycles were obtained at equal compression ratios, fuel-air ratios, fuel type, and engine rpm. Results show that for overall stoichiometric heat addition the Heat Balanced cycle can produce greater indicated engine power, higher indicated thermal efficiency and lower indicated specific fuel consumption than the corresponding Otto cycle. Further analysis indicates that the optimum heat balancing conditions occur for constant volume heat addition with rich mixture composition followed by constant pressure heat addition with lean composition.
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I. ABSTRACT

The quasi equilibrium thermodynamic model of the Naval Academy Heat Balanced Engine (NAHBE) has been modified to include the influence of fuel-air chemistry on predicted indicated engine performance. Heat addition to the Air Standard Heat Balanced Cycle was expressed in terms of an appropriate fuel-air ratio and heating value for a standard fuel. Indicated parameters including mean effective pressure, peak pressure, specific fuel consumption and thermal efficiency for compatible Otto and Heat Balanced cycles were calculated and compared. Performance parameters for both cycles were obtained at equal compression ratios, fuel-air ratios, fuel type, and engine rpm. Results show that for overall stoichiometric heat addition the Heat Balanced cycle can produce greater indicated engine power, higher indicated thermal efficiency and lower indicated specific fuel consumption than the corresponding Otto cycle. Further analysis indicates that the optimum heat balancing conditions occur for constant volume heat addition with rich mixture composition followed by constant pressure heat addition with lean composition.
II. INTRODUCTION

The experimental performance of a variety of actual heat balanced internal combustion engines has been shown to depend significantly on the fuel-air characteristics of the particular applications.\textsuperscript{1,2,3} Any I.C. engine performance depends upon, among other parameters, the air-fuel ratio; fuel type, i.e., gasoline, alcohol, diesel fuel oil; fuel heating value; and octane or cetane index. The thermodynamic basis for predicting the influence of these factors on standard cycles such as the Otto, Diesel or Dual cycles, is covered in a variety of I.C. engine texts.\textsuperscript{4,5,6,7} Spark and/or compression ignition engine behavior can be predicted using an air standard cycle or a fuel-air cycle analysis. In the air standard model the charge in the engine is treated as being air alone and heat addition and rejection are used to represent the combustion and exhaust processes. In the fuel-air analysis, the thermochemistry of the engine fuel-air compression and combustion processes, as well as the expansion and exhaust processes of the combustion products are treated in detail. This method allows a particular engine to be considered analytically in such a way as to produce more realistic indicated engine performance predictions than results based on air standard cycle calculations. In this study the preliminary rationale for a fuel-air heat balanced cycle analysis will be developed.
III. HEAT BALANCED CYCLE: PRELIMINARY FUEL-AIR ANALYSIS

A thermodynamic model for the unique heat balanced engine concept has been reported in earlier studies. Figure 1 and Table 1 summarize the particular features relevant to the theoretical heat balanced cycle analysis. In this original heat engine it is assumed that the total mass can be separated into two nonmixing regions; the total heat addition consists of separate constant volume and constant pressure processes, and all thermodynamic processes are quasi-equilibrium in nature. Based on calculations for this air standard cycle it was shown that the quasi-equilibrium heat balanced cycle (QEHBC) for optimum configuration can achieve a greater thermal efficiency and lower peak pressure than the comparable Otto cycle.

These results were obtained from a parametric study of the governing equations for the ideal air standard Otto, Diesel, Dual and Heat Balanced cycles. All cycles were compared on the basis of congruent geometries, equal total masses, and identical total heat additions. At an 8:1 compression ratio two additional heat balanced cycle configurations were investigated. In the first study the mass in the two regions was unchanged while the percentage of the total heat added to these separate masses was varied. In the second treatment the heat addition to the two regions was held constant and the fraction of the total mass in the two regions was changed. Based on these calculations it was shown that for a given compression ratio, total mass, and total heat addition QEHBC performance can be changed dramatically by simply varying the heat and mass balancing ratios.
a) compression process

b) first heat addition process

c) second heat addition process

d) expansion process

e) heat rejection process

f) equilibrium and quasi-equilibrium cycles

Figure 1 QEHBC THERMODYNAMIC P-V DIAGRAM
### TABLE I. QEHBC THERMODYNAMIC CYCLE*

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<td>constant volume</td>
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*To attain the QEHBC cycle, regions I and II must remain separated throughout the cycle and all processes are quasi-equilibrium in nature. If the two regions mix and attain equilibrium the cycle produces the classical combined or dual cycle.
For any heat balanced cycle the total mass and heat added to the thermodynamic cycle can be expressed in terms of the two regions as

\[ M_t = M_I + M_{II} \]  \hspace{1cm} (1)

where

\[ M_t = \text{total mass, lbm} \]
\[ M_I = \text{mass in region I, lbm} \]
\[ M_{II} = \text{mass in region II, lbm} \]

and

\[ Q_t = Q_I + Q_{II} \]  \hspace{1cm} (2)

with

\[ Q_t = \text{total heat addition, Btu} \]
\[ Q_I = \text{heat addition to region I, Btu} \]
\[ Q_{II} = \text{heat addition to region II, Btu} \]

Two balancing ratios, the mass balancing ratio, \( \beta_M \), and the heat balancing ratio, \( \beta_H \), have been previously defined\(^8\) for the heat balanced cycle and are given by the relations

\[ \beta_M = \frac{M_I}{M_{II}} \]  \hspace{1cm} (3)

and

\[ \beta_H = \frac{Q_{II}}{Q_I} \]  \hspace{1cm} (4)

Using Equations (3) and (4) the mass in regions I and II can be expressed in terms of \( \beta_M \) and the total mass, \( M_t \), as
Air standard cycle analysis can be modified to approximate the behavior of an actual engine by expressing the thermodynamic cycle heat addition in terms of the actual engine fuel-air requirements. Thus the total heat addition is calculated in terms of the fuel heating value, engine fuel-air ratio, and the fuel-air charge induced into the engine. The total mass in the engine ideally equals the conditions at the end of the intake stroke at BDC, equal to

\[ M_t = \frac{P_1 V_1}{R T_1} \]  \hspace{1cm} (7)

where

\begin{align*}
P_1 &= \text{inlet pressure, lbf/ft}^2 \\
V &= \text{volume at BDC, ft}^3 \\
R_1 &= \text{gas constant for air} \\
&= 53.34 \text{ ft lbf/lbm R} \\
T_1 &= \text{inlet temperature, } ^\circ R
\end{align*}

The volume at BDC consists of the clearance volume, \( V_c \), and the displacement volume, \( V_D \), with the total volume given as

\[ V_1 = V_c + V_D \]  \hspace{1cm} (8)

The compression ratio, \( r_V \), a ratio of the volume of BDC to the volume at TDC, can be written as

\[ r_V = \frac{V_{BDC}}{V_{TDC}} = \frac{V_c + V_D}{V_c} \]  \hspace{1cm} (9)

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The total volume in terms of the compression ratio and displacement volume is then

\[ V_1 = V_D \left( \frac{r_v}{r_v - 1} \right) \]  

(10)

The total volume at BDC of an actual I.C. engine cannot be completely filled with fuel and air, i.e., the induced charge. This is due to the presence of residual gases remaining from the exhaust stroke. Taylor suggests that the residuals and charge at BDC have approximately the same density and the displacement volume can be considered to be filled with reactants and the clearance volume to be filled with residual gases. The total mass of air in the air standard cycle that should be used to determine the required heat addition is thus expressed as

\[ M_c = M_t \left( \frac{V_D}{V_1} \right) = M_t \left( \frac{r_v - 1}{r_v} \right) \]  

(11)

where

- \( M_c \) = mass of air associated with combustion, lbm
- \( M_t \) = total mass of air, lbm
- \( r_v \) = compression ratio

The appropriate heat addition to an air standard cycle in terms of the engine fuel-air characteristics is given as

\[ Q_t = \left[ \frac{HV \times M_c}{AF_t} \right] \]  

(12)

where

- \( Q_t \) = total heat addition, Btu
- \( HV \) = fuel heating value, Btu/lbm fuel
\[ \text{AF}_t = \text{air-fuel ratio, lbm air/lbm fuel} \]
\[ M_c = \text{mass of air, lbm} \]

Substituting Equation (11) into Equation (12) yields the proper expression for the total heat addition for the air standard cycle in terms of engine air-fuel requirements as

\[ Q_t = \left[ \frac{HV \times M_t}{\text{AF}_t} \right] \left[ 1 - \frac{1}{r_v} \right] \]  

(13)

Since the total heat addition in the heat balanced cycle consists of portions added separately to regions I and II expressions for these two heat additions can also be written as

\[ Q_I = \left[ \frac{HV \times M_I}{\text{AF}_I} \right] \left[ 1 - \frac{1}{r_v} \right] \]  

(14)

and

\[ Q_{II} = \left[ \frac{HV \times M_{II}}{\text{AF}_{II}} \right] \left[ 1 - \frac{1}{r_v} \right] \]  

(15)

Substituting Equations (13), (14), and (15) into Equation (2), relating \( Q_t \) to \( Q_I \) and \( Q_{II} \) yields

\[ \left[ \frac{HV \times M_t}{\text{AF}_t} \right] = \left[ \frac{HV \times M_I}{\text{AF}_I} \right] + \left[ \frac{HV \times M_{II}}{\text{AF}_{II}} \right] \]  

(16)

Since the same fuel is associated with regions I and II, Equation (16) reduces to

\[ \frac{M_t}{\text{AF}_t} = \frac{M_I}{\text{AF}_I} + \frac{M_{II}}{\text{AF}_{II}} \]  

(17)
Substituting Equations (5) and (6) for $M_I$ and $M_{II}$, respectively gives

$$\frac{M_t}{AF_t} = \left[ \frac{M_t}{AF_I} \right] + \left[ \frac{1}{1 + \beta_M} \right] \frac{M_t}{AF_{II}}$$

$$(1 + \beta_M) = \beta_M \left[ \frac{AF_t}{AF_I} \right] + 1 \left[ \frac{AF_t}{AF_{II}} \right]$$

$$\beta_M \left[ 1 - \frac{AF_t}{AF_I} \right] = \left[ \frac{AF_t}{AF_{II}} - 1 \right]$$

or

$$\beta_M = \left[ \frac{(AF_t/AF_{II}) - 1}{1 - (AF_t/AF_I)} \right]$$

Equation (18) indicates that the mass balancing ratio, $\beta_M$, for the heat balanced cycle should be a function of the total overall air-fuel ratio, $AF_t$, and the air-fuel ratios in regions I and II, $AF_I$ and $AF_{II}$, respectively.

The heat balancing ratio, $\beta_H$, given by Equation (4) can now be rewritten using Equations (14) and (15) giving

$$\beta_H = \frac{Q_{II}}{Q_I} = \left[ \frac{H x M_{II}}{AF_{II}} \right] \left[ \frac{AF_I}{HV x M_I} \right]$$

$$\beta_H = \frac{1}{\beta_M} \left[ \frac{AF_I}{AF_{II}} \right]$$

Substituting Equation (18) for $\beta_M$ in Equation (19) gives the result

$$\beta_H = \left[ \frac{AF_I}{AF_{II}} \right] \left[ \frac{1 - (AF_t/AF_I)}{(AF_t/AF_{II}) - 1} \right]$$

$$\beta_H = \frac{AF_I - AF_t}{AF_t - AF_{II}} = \frac{(AF_I/AF_t) - 1}{1 - (AF_{II}/AF_t)}$$ (20)
IV. PARAMETRIC ANALYSIS: HEAT BALANCED CYCLE HEAT ADDITION

Equation (20) shows that the heat balancing ratio, \( B_H \), for the heat balanced cycle should also be a function of the overall air fuel ratio, \( AF_t \), and the air-fuel ratios in the two regions I and II, \( AF_I \) and \( AF_{II} \).

An inspection of Equations (18) and (20) reveals how the heat balanced cycle might produce the wide variation in performance suggested by the previous cycle studies even for the case of a specified compression ratio and fixed total mass and heat addition. Equation (18) shows that for a particular overall stoichiometry, \( AF_t \), the air-fuel ratios in regions I and II, \( AF_I \) and \( AF_{II} \), can be varied in such a way that the mass balancing ratio, \( B_M \), will remain constant. These changes, however, would result in a variation in the heat balancing ratio, \( B_H \). Similarly, Equation (20) suggests that the air-fuel variations can be made such that the heat balancing ratio, \( B_H \), will remain constant with changes found in the corresponding mass balancing ratio, \( B_M \). Equations (18) and (20) further indicate that the two balancing ratios, \( B_M \) and \( B_H \), are not totally independent but relate to each other through the air fuel ratios in the separated regions I and II.

Tables II-VI give the results for a four stroke air standard heat balanced cycle. The following conditions were assumed in all calculations:

- fuel = \( C_8H_{18} \) (octane)
- \( D \times L = 4" \times 4" \)
- inlet \( P_1 = 14.7 \) psia
- inlet \( T_1 = 70^\circ F \)
The stoichiometric reaction of $C_8H_{18}$ and air is equal to

$$C_8H_{18} + 12.5[O_2 + 3.76 N_2] \rightarrow 8CO_2 + 9H_2O + 47 N_2$$

where the total air-fuel ratio is given by

$$AF_t = \left[ \frac{(12.5)(4.76)(28.97)}{(1.0)(114)} \right] \frac{\text{lbm air}}{\text{lbm fuel}} = 15.12 \frac{\text{lbm air}}{\text{lbm fuel}}$$

The mass flow rate of air associated with the required engine heat addition calculation is given by the expression

$$M_c = \frac{M_c \times N}{2} = \frac{M_t N}{2} \left[ 1 - \frac{1}{r_v} \right]$$

where

$M_c$ = combustion air mass flow rate $\text{lbm air/min}$

$M_c$ = mass of combustion air, $\text{lbm air/intake stroke}$

$M_t$ = total mass of air, $\text{lbm}$

$N$ = engine revolution, $\text{rev/min}$

$r_v$ = compression ratio

$2$ = revolutions/intake stroke

The mass flow rate of fuel required for the proper heat addition is then equal to

IV-2
\[ M_f = \frac{M_{tN}}{2A_{Ft}} \left[ 1 - \frac{1}{r_v} \right] \]  

(22)

where

\[ M_f \] = fuel mass flow rate, lbm fuel/min

\[ A_{Ft} \] = total overall air-fuel ratio, lbm air/lbm fuel.

The indicated mean effective pressure is equal to the net cycle work divided by the displacement volume, or

\[ \bar{P} = \frac{W_{\text{net}}}{V_D} \]  

(23)

with

\[ \bar{P} \] = indicated mean effective pressure, lbf/ft²

\[ W_{\text{net}} \] = net cycle work, ft lbf

\[ V_D \] = displacement volume, ft³

The indicated power for the cycle can be obtained using the mean effective pressure then as the standard relationship

\[ \dot{W} = \frac{\bar{P}LAC}{33,000} \frac{n}{n} \]  

(24)

where

\[ \dot{W} \] = indicated engine power, hp

\[ \bar{P} \] = mean effective pressure, lbf/ft²

\[ L \] = stroke, ft/stroke

\[ A \] = piston cross-sectional area, ft²

\[ N \] = engine speed, rev/min

\[ c \] = number of cylinders = 1

\[ 33,000 = \text{ft lbf/hp-min} \]

\[ n = \text{number of revolutions/stroke} = 2 \]
The indicated thermal efficiency for the cycle can be calculated from the expression

\[
\eta = \left[ \frac{\dot{W}}{\dot{M}_f HV} \right]
\]

(25)

where

\[
\dot{W} = \text{indicated engine power, Btu/min}
\]
\[
\dot{M}_f = \text{fuel mass flow rate, lbm fuel/min}
\]
\[
HV = \text{fuel heating value, Btu/lbm fuel}
\]

Using Equations (22) and (24) an expression for the indicated specific fuel consumptions can be written as

\[
\text{ISFC} = \frac{\dot{M}_f}{\dot{W}}
\]

(26)

with

\[
\text{ISFC} = \text{indicated specific fuel consumption, lbm/hp-hr}
\]
\[
\dot{M}_f = \text{fuel mass flow rate, lbm fuel/hr}
\]
\[
\dot{W} = \text{indicated engine power, hp}
\]

Numerical results obtained using Equations (21)-(26) are found in Tables II-VI and are presented graphically in Figures 2-6. Table II and Figure 2 show the predicted indicated mean effective pressure for various heat and mass balancing ratios. Table II shows that for the particular values of \( \beta_M \) and \( \beta_H \) the QEHBC in all cases exceeded the output of 311.38 psi for the corresponding Otto cycle. Figure 2 also indicates that ideal power output reaches its maximum values for heat and mass balancing ratios less than one.
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*Results obtained for equivalent Otto cycle.
(maximum value obtained)
Figure 2 INDICATED MEP VS BALANCING RATIOS

- Bore x stroke: 4" x 4"
- \( P_1 \) @ BDC: 14.7 psia
- \( T_1 \) @ BDC: 70°F
- Fuel (octane): \( C_8 H_{18} \)
- Total AF ratio: 15.12:1
- Heating value: 20604 Btu/lbm
- Piston speed: 1300 rev/min
- Cylinders: 1
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<th>0.8</th>
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</table>

*Results obtained for equivalent Otto cycle. (maximum value obtained)
Figure 3 THERMAL EFFICIENCY VS BALANCING RATIOS
## TABLE IV. ISFC vs. BALANCING RATIO

(0.21868 lbm fuel/hp-hr)*

<table>
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<tr>
<th>$\beta_H$</th>
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<th>1.0</th>
<th>1.2</th>
<th>1.4</th>
<th>1.6</th>
<th>1.8</th>
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<td>0.18501</td>
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<td>0.18739</td>
<td>0.18836</td>
<td>0.18920</td>
<td>0.18994</td>
<td>0.19059</td>
<td>0.19117</td>
<td>0.19169</td>
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<tr>
<td>0.4</td>
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<td>0.18244</td>
<td>0.18322</td>
<td>0.18392</td>
<td>0.18455</td>
<td>0.18510</td>
<td>0.18559</td>
<td>0.18602</td>
<td>0.18641</td>
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<td>0.18190</td>
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<td>0.18360</td>
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<td>0.18460</td>
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<td>0.18467</td>
<td>0.18471</td>
<td>0.18476</td>
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<td>0.18571</td>
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<td>0.18714</td>
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<td>0.18695</td>
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<td>1.8</td>
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<td>0.18916</td>
<td>0.18882</td>
<td>0.18859</td>
<td>0.18842</td>
<td>0.18829</td>
<td>0.18820</td>
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<td>0.19229</td>
<td>0.19141</td>
<td>0.19081</td>
<td>0.19046</td>
<td>0.19010</td>
<td>0.18987</td>
<td>0.18970</td>
<td>0.18957</td>
<td>0.18946</td>
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</tbody>
</table>

*Results obtained for equivalent Otto Cycle.
(maximum value obtained)
Figure 4 ISFC VS BALANCING RATIOS
Indicated thermal efficiency calculations for the heat balanced cycle are listed in Table III and plotted in Figure 3. Table III and Figure 3 reveal that the thermal efficiency is greater than Otto cycle value of 57.484% for all values of $\beta_M$ and $\beta_H$. A comparison of Figures 2 and 3 further shows that the mean effective pressure as well as thermal efficiency is larger than those for the Otto cycle for heat balanced cycles having $\beta_M$ and $\beta_H$ less than one.

The indicated specific fuel consumption predictions are presented in Table IV and graphically shown in Figure 4. Table IV suggests that the Otto cycle specific fuel consumption of 0.21868 lbm fuel/hp-hr is greater than all heat balanced cycle values. Figure 4 shows that the minimum fuel consumption rates occur at values of $\beta_M$ and $\beta_H$ less than one.

These results of improved performance are in agreement with heat balanced engine predictions reported in earlier studies. The required air fuel ratios for the two separate heat additions are found in Tables V and VI and plotted in Figures 5 and 6. A study of Table V and Figure 5 reveals that the air fuel ratio for region I, i.e., composition associated with constant total volume heat addition, is rich for all values of $\beta_M$ and $\beta_H$ less than one and lean for all values of $\beta_M$ and $\beta_H$ greater than one. Likewise Table VI and Figure 6 suggest that the air fuel ratio for region II, mass associated with constant pressure heat addition, is lean for all values of $\beta_M$ and $\beta_H$ less than one and rich for all values of $\beta_M$ and $\beta_H$ greater than one. In addition, the conditions of
TABLE V. $AF_I$ vs. BALANCING RATIO

(15.12 lbm air/lbm fuel)*

<table>
<thead>
<tr>
<th>$\delta_h$</th>
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<th>0.6</th>
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<th>1.4</th>
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<td>15.876</td>
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</table>

*Results obtained for equivalent Otto Cycle. (total stoichiometric charge)
Figure 5 REGION I AF RATIO VS BALANCING RATIOS
<table>
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<th>$\beta_M$</th>
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<td>8.723</td>
<td>8.1</td>
<td>7.56</td>
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</table>

*Results obtained for equivalent Otto cycle.
(totall stoichimetric charge)*
Figure 6 REGION II AF RATIO VS BALANCING RATIOS
optimum thermodynamic performance, i.e., small balancing ratios cited previously, correspond to a rich constant total volume heat addition followed by a lean constant total pressure heat addition.

The use of an Air Standard cycle to predict the performance of the actual engine precludes the effect of variable specific heats of the charge, residual gases in the clearance volume, and the thermochemistry of the actual fuel-air combustion process. In this sense the previous calculations merely suggest the trends anticipated in the proper design of a heat balanced engine. To further assess the influence of the actual fuel-air properties on $\beta_M$ and $\beta_H$ the total charge of reactants in the displacement volume next were treated as an ideal fuel-air gaseous mixture where

\begin{align}
N_{0t} &= N_{0I} + N_{0II} \\
N_{ft} &= N_{fI} + N_{fII}
\end{align}

(27)

(28)

where

\begin{align}
N_0 &= \text{number of moles of oxidant} \\
N_f &= \text{number of moles of fuel} \\
I,II &= \text{regions I and II}
\end{align}

Allowing the total fuel and air to be divided between regions I and II requires that the total moles still be conserved.

\begin{align}
N_{0I} &= Z_0 N_{0t} \\
N_{0II} &= (1 - Z_0) N_{0t}
\end{align}

(29)

(30)
with

\[ N_{f_I} = Z_f \cdot N_{f_t} \]  
(31)

\[ N_{f_{II}} = (1 - Z_f) \cdot N_{f_t} \]  
(32)

where

\[ Z_0 = \text{mole fraction of total oxidant } 0 < Z_0 < 1 \]
\[ Z_f = \text{mole fraction of total fuel } 0 < Z_f < 1 \]
\[ Z_0 \neq Z_f \]

The total air-fuel ratio of the charge in the displacement volume is given by the expression

\[ AF_t = \frac{(N_0^t)(MW_0)}{(N_{f_t}^t)(MW_f)} \]  
(33)

where

\[ AF_t = \text{total air fuel ratio, lbm air/lbm fuel} \]
\[ N = \text{total number of moles} \]
\[ MW = \text{molecular weight, lbm/lbmole} \]

Likewise the air-fuel ratios in regions I and II are equal to

\[ AF_I = \frac{(N_{0_I})(MW_0)}{(N_{f_I})(MW_f)} \]  
(34)

and

\[ AF_{II} = \frac{(N_{0_{II}})(MW_0)}{(N_{f_{II}})(MW_f)} \]  
(35)
Combining Equations (33) and (34) into Equation (31) and likewise Equations (33), (35) and (32) gives

$$AF_1 = \frac{Z_0}{Z_f} (AF_t)$$  \hspace{1cm} (36)

and

$$AF_{II} = \frac{(1 - Z_0)}{(1 - Z_f)} (AF_t)$$  \hspace{1cm} (37)

The mass balancing ratio \( \beta_M \) can be expressed in terms of the molar fuel-air analysis as

$$\beta_M = \frac{M_I}{M_{II}} = \frac{(N_{O_I} + N_{F_I})(MW_I)}{(N_{O_{II}} + N_{F_{II}})(MW_{II})}$$  \hspace{1cm} (38)

The corresponding molecular weights of regions I and II are equal to

$$MW_I = \begin{bmatrix} N_{O_I} \\ N_{O_I} + N_{F_I} \\ MW_0 & \left[ \frac{N_{F_I}}{N_{O_I} + N_{F_I}} \right] MW_f \end{bmatrix}$$

and

$$MW_{II} = \begin{bmatrix} N_{O_{II}} \\ N_{O_{II}} + N_{F_{II}} \\ MW_0 & \left[ \frac{N_{F_{II}}}{N_{O_{II}} + N_{F_{II}}} \right] MW_f \end{bmatrix}$$  \hspace{1cm} (39)

Substituting Equations (39) and (40) into (38) yields an expression for \( \beta_M \) in terms of the moles of fuel and air in regions I and II as

$$\beta_M = \frac{N_{O_I} MW_0 + N_{F_I} MW_f}{N_{O_{II}} MW_0 + N_{F_{II}} MW_f}$$  \hspace{1cm} (41)

Using the definition of \( AF_I \) and \( AF_{II} \) found in Equations (34) and (35)

$$\beta_M = \frac{N_{O_I} MW_0 + (N_{O_I} MW_0/AF_I)}{N_{O_{II}} MW_0 + (N_{O_{II}} MW_0/AF_{II})}$$  \hspace{1cm} (42)

IV-18
\[ \beta_M = \frac{N_{O_I}}{N_{O_{II}}} \left[ \frac{1 + \left(\frac{1}{AF_I}\right)}{1 + \left(\frac{1}{AF_{II}}\right)} \right] \]  

Since

\[ N_{O_I} = Z_0 N_{O_t} \]  

and

\[ N_{O_{II}} = (1 - Z_0) N_{O_t} \]

with

\[ AF_I = \frac{Z_0}{Z_f} (AF_{t}) \]

and

\[ AF_{II} = \frac{(1 - Z_0)}{(1 - Z_f)} AF_t \]

then

\[ \beta_M = \left[ \frac{Z_0}{1 - Z_0} \right] \left[ \frac{1 + \left(\frac{Z_f/Z_0}{1 - Z_f/1 - Z_0}\right)}{1 + \left(\frac{1}{AF_t}\right)} \right] \]  

Equation (44) expresses the mass balancing ratio \( \beta_M \) in terms of the overall air-fuel ratio and the mole fractions of fuel and oxidant in region I. An expression for the heat balancing ratio \( \beta_H \) can also be written in terms of the above parameters as

\[ \beta_H = \frac{Q_{II}}{Q_1} = \frac{(N_{f_{II}})(MW_f)(HV)}{(N_{f_I})(MW_f)(HV)} \]

\[ \beta_H = \frac{N_{f_{II}}}{N_{f_I}} = \frac{(1 - Z_f)(N_{f_t})}{Z_f N_{f_t}} = \frac{1 - Z_f}{Z_f} \]
Numerical computations using the molar analysis for a stoichiometric $C_8H_{18}$-air mixture were obtained using Equations (36), (37), (44) and (46). The results are found in Table VII and shown graphically in Figures 7 and 8. Table VI and Figure 7 show that the air fuel ratio for region I, i.e., composition associated with constant total volume heat addition, is rich for all values of $\beta_M$ and $\beta_H$ less than one and lean for all values of $\beta_M$ and $\beta_H$ greater than one. Table VI and Figure 8 further indicate that the air-fuel ratio for region II, i.e., mass associated with constant pressure heat addition, is lean for all values of $\beta_M$ and $\beta_H$ less than one and rich for all values of $\beta_M$ and $\beta_H$ greater than one.

The current results are all related to ideal Otto or spark ignition engines operating at an overall stoichiometric air-fuel ratio. Since real engine performance for the heat balanced engine has shown best power at an air-fuel ratio of about 16 and best thermal efficiency at an air-fuel ratio of about 20, it would be extremely useful to extend the current fuel-air calculations to these overall composition limits. Moreover, since the cycle is applicable to compression ignition as well as spark ignition, the fuel-air approach should also be extended to compression ignition.
TABLE VII. QEHBC AIR FUEL RATIOS vs. BALANCING RATIOS - MOLAR ANALYSIS

\[ \beta_M = 0.6 \rightarrow \]

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<tr>
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<th>(AF_{II})</th>
<th>(\beta_H)</th>
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<td>0.375</td>
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<td>15.12</td>
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</table>

\[ \beta_M = 0.8 \rightarrow \]

<table>
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<th>(Z_f)</th>
<th>(AF_{I})</th>
<th>(AF_{II})</th>
<th>(\beta_H)</th>
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</thead>
<tbody>
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<tr>
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\[ \beta_M = 1.0 \rightarrow \]

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<th>(AF_{I})</th>
<th>(AF_{II})</th>
<th>(\beta_H)</th>
</tr>
</thead>
<tbody>
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<td>0.50</td>
<td>0.50</td>
<td>15.12</td>
<td>15.12</td>
<td>1.000</td>
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</tbody>
</table>

\[ \beta_M = 1.2 \rightarrow \]

<table>
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<th>(AF_{I})</th>
<th>(AF_{II})</th>
<th>(\beta_H)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.53</td>
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<td>10.2854</td>
<td>32.1741</td>
<td>0.2835</td>
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<td>0.54</td>
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<td>13.0028</td>
<td>18.6931</td>
<td>0.5925</td>
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<td>0.5454</td>
<td>15.12</td>
<td>15.12</td>
<td>0.8333</td>
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<tr>
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<tr>
<td>0.56</td>
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<td>2.0719</td>
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<tr>
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<td>4.7362</td>
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</table>

\[ \beta_M = 1.4 \rightarrow \]

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<th>(AF_{I})</th>
<th>(AF_{II})</th>
<th>(\beta_H)</th>
</tr>
</thead>
<tbody>
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<tr>
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<td>0.5833</td>
<td>15.12</td>
<td>15.12</td>
<td>0.7143</td>
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<tr>
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<td>1.0724</td>
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<tr>
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<td>27.3803</td>
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<td>2.0181</td>
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<td>0.61</td>
<td>0.1801</td>
<td>51.2022</td>
<td>7.1924</td>
<td>4.5515</td>
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</table>
bore x stroke 4" x 4"
$P_i @ BDC$ 14.7 psia
$T_i @ BDC$ 70° F
fuel (octane) $C_4 H_{18}$
total AF ratio 15.12:1
heating value 20604 Btu/lbm
piston speed 1300 rev/min
cylinders 1

---

Figure 7 AF — REGION I VS BALANCING RATIOS (MOLAR ANALYSIS)
IV-22
bore x stroke 4" x 4"
P$_1$ @ BDC 14.7 psia
T$_1$ @ BDC 70°F
fuel (octane) C$_8$H$_{18}$
total AF ratio 15:1:1
heating value 20604 Btu/lbm
piston speed 1300 rev/min
cylinders 1

**Figure 8 AF — REGION II VS BALANCING RATIOS (MOLAR ANALYSIS)**

IV-23
V. CONCLUSION AND RECOMMENDATIONS

The quasi-equilibrium air standard heat balanced cycle, QEHBC, has been modified to investigate the influence of the fuel-air thermochemistry on predicted cycle behavior. Based on this study several important conclusions can be stated. All results compare Otto and QEHBC cycles having equal engine geometries, total heat addition, compression ratios, overall stoichiometric air fuel ratios, and engine speed.

1. For stoichiometric heat addition the indicated output of the heat balanced cycle can exceed that of the corresponding Otto cycle.

2. For stoichiometric heat addition the indicated thermal efficiency of the heat balanced cycle can exceed that of the corresponding Otto cycle.

3. For stoichiometric heat addition the indicated specific fuel consumption of the heat balanced cycle is less than that of the corresponding Otto cycle.

4. These results occur for the optimum values of the mass balancing ratio, $\beta_M$, and heat balancing ratio, $\beta_H$, previously developed.

5. The thermodynamic heat balancing ratios, $\beta_M$ and $\beta_H$, are not independent parameters but are both functions of the overall total air fuel ratio and the corresponding air fuel ratios in the two mass balancing regions I and II.

6. For overall stoichiometric proportions the QEHBC air fuel ratios in regions I and II can be varied over a wide range of values and in general are not stoichiometric.
7. The indicated performance of the heat balanced cycle can be dramatically changed by simply varying the air fuel ratios in the two balancing regions I and II.

8. Optimum values for the heat balancing ratios of $\beta_M$ and $\beta_H$ suggest that the air fuel ratio associated with constant total volume heat addition be rich.

9. Optimum values for the heat balancing ratios of $\beta_M$ and $\beta_H$ suggest that the air fuel ratio associated with constant pressure heat addition be lean.
VI. REFERENCES


