Effects of irreversible different parameters on performance of air standard dual-cycle

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Abstract: An irreversible *air standard dual cycle* model is proposed in this paper. The performance of an *air-standard dual cycle* with heat transfer loss and variable specific heats of working fluid is analyzed by using *finite-time thermodynamics*. The objective of this study is to analyze the effects of heat loss characterized by a percentage of the fuel's energy, *friction* and variable specific heats of working fluid on the performance of an *air standard dual cycle* with a restriction of maximum cycle temperature. The relations between the *power output* and the compression ratio, and between the thermal efficiency and the compression ratio of the cycle are derived. Moreover, the effects of heat transfer and global losses lumped in a *friction* like term on the performance of the cycle are shown by detailed numerical examples. [Journal of American Science 2011;7(3):608-613]. (ISSN: 1545-1003).

Keywords: *Air Standard, Finite-time thermodynamics, Dual cycle, Power Output, friction*.

1. Introduction

A series of achievements have been made since finite-time thermodynamics was used to analyze and optimize real heat engines. Several author shave applied finite-time thermodynamics (Andresen and Bejan and Chen et al.) to the analysis and optimization of the internal-combustion engine. Mozurkewich and Berry and Hoffman et al. used mathematical techniques from optimal-control theory to determine the optimal motion of the piston in Otto and Diesel engines. Aizenbud and Band and Chen et al. determined the optimal motion of a piston, fitted to a cylinder containing a gas, pumped with a given heating rate and coupled to a heat bath for a finite time. The heat addition process for an air standard cycle has been widely described as subtraction of an arbitrary heat loss parameter times the average temperature of the heat addition period from the fuel's chemical energy. Orlov and Berry obtained the power and efficiency limits for internal-combustion engines. Angulo-Brown et al. and Chen et al. optimized the powers of the Otto and Diesel engines with friction loss during finite times. Klein studied the effect of heat transfer through a cylinder wall on the work output of the Otto and Diesel cycles. Chen et al. derived the relations between net power output and the efficiency for both the Diesel and Otto cycles, with considerations of heat transfer through the cylinder wall. Blank and Wu and Lin et al. considered the effect of heat transfer through a cylinder wall on the work output of the dual cycle. On the basis of these investigations, the relations governing the net power output and the efficiency for the dual cycle with considerations of heat transfer loss and friction-like terms loss during a finite time are derived in this paper. heat transfer between the working fluid and the environment through the cylinder wall is considered and characterized by a percentage of the fuel's energy; friction loss of the piston in all the processes of the cycle on the performance is taken into account. Furthermore, we consider the variable specific heats of the working fluid that is significant in practical cycle analysis. The results obtained in the study may offer good guidance for design and operation of the dual cycle engine.

2. Thermodynamic analysis

An ideal air-standard dual cycle is shown in Fig. 1. The compression process ignition is isentropic $1 \rightarrow 2$; the combustion is modelled by a reversible constant volume process $2 \rightarrow 3$ and a constant-pressure process $3 \rightarrow 4$; the expansion process $4 \rightarrow 5$ is isentropic; and the heat rejection is a reversible constant-volume process $5 \rightarrow 1$. As is usual in FTT-heat engine models, we suppose instantaneous adiabats. For the isochoric branches ($2 \rightarrow 3$ and $5 \rightarrow 1$) and the isobaric branch ($3 \rightarrow 4$) in Fig. 1, we propose that heating from states 2 to 4 and cooling from states 5 to 1 proceed according to constant temperatures; i.e.

$$\frac{dT}{dt} = \frac{1}{K_1} \quad (for \quad 2 \rightarrow 3, \ 3 \rightarrow 4)$$
$$\frac{dT}{dt} = \frac{1}{K_2} \quad (for \quad 5 \rightarrow 1) \ , \qquad (1)$$

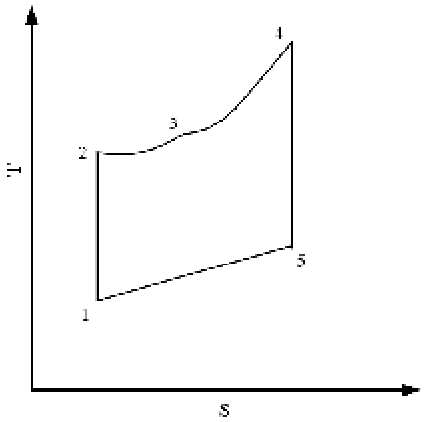

Figure 1. T–s diagram for a dual cycle

where T is the absolute temperature and t is the time. k_1 and k_2 are constants. Integrating Eq. (1) gives

$$t_1 = K_1(T_3 - T_2)\ ,\ t_2 = K_2(T_4 - T_1)\ , \tag{2}$$

where t_1 and t_2 are the heating and cooling times, respectively. Then, the cycle period is

$$t = t_1 + t_2 = K_1(T_3 - T_2) + K_2(T_4 - T_1). \tag{3}$$

The work output is

$$\begin{aligned} W &= C_v(T_3 - T_2) + C_p(T_4 - T_3) \\ &\qquad - C_v(T_5 - T_1), \end{aligned} \tag{4}$$

where C_v is the heat capacity at constant volume, and C_p is the heat capacity at constant pressure, i.e., the product of mass flow rate and the specific heat. Thus, the power output is

$$\begin{aligned} P_1 &= \frac{W}{t} \\ &= \frac{C_v(T_3 - T_2) + C_p(T_4 - T_3) - C_v(T_5 - T_1)}{K_1(T_4 - T_2) + K_2(T_5 - T_1)}. \end{aligned} \tag{5}$$

Defining the compression ratio, g , the pressure ratio, g_p and the cut-off ratio, g_c as follows:

$$g = V_1/V_2 = (T_2/T_1)^{1/(k-1)}\ , \tag{6}$$

$$g_p = P_3/P_2 = T_3/T_2\ , \tag{7}$$

and

$$g_c = V_4/V_3 = T_4/T_3\ , \tag{8}$$

we have

$$T_2 = T_4/(g_p g_c)\ , \tag{9}$$

$$T_3 = T_4/g_c\ , \tag{10}$$

$$T_3 = T_4 g_p g_c{}^k, \tag{11}$$

$$g = [T_4/(T_1 g_p g_c)]^{1/(k-1)}. \tag{12}$$

Substituting Eqs. (6)–(11) into Eq. (5) yields

$$\begin{aligned} P_1 &= \frac{W}{t} \\ &= \{C_v\{T_1[1 - g^{k-1} - (k-1)g_p g^{k-1}] \\ &\quad + kT_4 - T_1^{1-k} T_4^k g_p^{1-k} g^{k(1-k)}\}\} \\ &\quad /\{K_1(T_4 - T_1 g^{k-1}) \\ &\quad + K_2[T_1^{1-k} T_4^k g^{k(1-k)} - T_1]\} \end{aligned} \tag{13}$$

where k is the ratio (C_p/C_v) of specific heats. In special cases, the dual cycle becomes the Diesel cycle or the Otto cycle, i.e. when $g_p = 1$, $g_c = 1$.
The total heat added to the working-fluid during processes $2 \to 3$ and $3 \to 4$ is

$$\begin{aligned} Q_A &= Q_{23} + Q_{34} \\ &= C_v(T_3 - T_2) + C_p(T_4 - T_3) \\ &= C_v[kT_4 + (1 - k - g_p^{-1})g_p g^{k-1} T_1] \end{aligned} \tag{14}$$

For an ideal dual cycle, there are no irreversible losses.

$$\begin{aligned} Q_A &= a - b(T_2 + T_4) \\ &= a - b(T_1 g^{k-1} + T_4), \end{aligned} \tag{15}$$

where a and b are two constants related to the combustion and heat transfer.
Combining Eqs. (14) and (15) yields

$$\begin{aligned} T_4 = \{a - [bg^{k-1} \\ + C_v(1 - k - g_p^{-1})g_p g^{k-1}]T_1\}/(C_p + b). \end{aligned} \tag{16}$$

Substituting Eq. (16) into Eqs. (13) and (14) yields

$$\begin{aligned}P_1 = \{&C_v \{T_1[1-g^{k-1}-(k-1)g_p g^{k-1}]\\ &+[k/(C_p+b)]\{a-[b\\ &+C_v(1-k-g_p^{-1})g_p]g^{k-1}T_1\}\}\\ &-C_v\{T_1^{1-k}g_p^{1-k}g^{k(1-k)}\{\{a-[b\\ &+C_v(1-k-g_p^{-1})g_p]g^{k-1}T_1\}/(C_p+b)\}^k\}\}\\ &/\{K_1\{\{a-[bg^{k-1}\\ &+C_v(1-k-g_p^{-1})g_p g^{k-1}]T_1\}\\ &/(C_p+b)-T_1 g^{k-1}\}\\ &+K_2\{T_1^{1-k}g^{1-k}g^{k(1-k)}\{\{a-[bg^{k-1}\\ &+C_v(1-k-g_p^{-1})g_p g^{k-1}]T_1\}\\ &/(C_p+b)-T_1 g^{k-1}\}^k-T_1\}\},\end{aligned} \tag{17}$$

$$\begin{aligned}Q_A = C_v\{&a k + b[(1-k-g_p^{-1})g_p - k]g^{k-1}T_1\}\\ &/(C_p+b).\end{aligned} \tag{18}$$

$$f_m = -mv = -m\frac{dx}{dt}, \tag{19}$$

where m is a coefficient of friction which takes into account the global losses and x is the piston displacement. Then, the lost power is

$$P_m = \frac{dW_m}{dt} = -m\frac{dx}{dt}\frac{dx}{dt} = -mv^2. \tag{20}$$

If we take the piston mean velocity $\bar{v}$ as v,

$$\bar{v} = \frac{x_1-x_2}{\Delta t_{12}} = \frac{x_2(g-1)}{\Delta t_{12}}, \tag{21}$$

where x_2 is the piston position at minimum volume and Δt_{12} is the time spent in the power stroke. Thus, the resulting power output is

$$\begin{aligned}P_1 = \{&C_v \{T_1[1-g^{k-1}-(k-1)g_p g^{k-1}]\\ &+[k/(C_p+b)]\{a-[b+C_v\\ &\times(1-k-g_p^{-1})g_p]g^{k-1}T_1\}\}\\ &-C_v\{T_1^{1-k}g_p^{1-k}g^{k(1-k)}\{\{a-[b+C_v\\ &\times(1-k-g_p^{-1})g_p]g^{k-1}T_1\}/(C_p+b)\}^k\}\}\\ &/\{K_1\{\{a-[bg^{k-1}+C_v\\ &\times(1-k-g_p^{-1})g_p g^{k-1}]T_1\}/(C_p+b)-T_1 g^{k-1}\}\\ &+K_2\{T_1^{1-k}g^{1-k}g^{k(1-k)}\{\{a-[bg^{k-1}+C_v\\ &\times(1-k-g_p^{-1})g_p g^{k-1}]T_1\}\\ &/(C_p+b)-T_1 g^{k-1}\}^k-T_1\}\}-b(g-1)^2,\end{aligned} \tag{22}$$

where

$$b = \frac{mx_2^2}{(\Delta t_{12})^2}.\frac{1}{2} \tag{23}$$

The thermal efficiency of the cycle is

$$\begin{aligned}h = \frac{P}{Q_A} = \{&C_v \{T_1[1-g^{k-1}-(k-1)g_p g^{k-1}]\\ &+[k/(C_p+b)]\{a-[b+C_v\\ &\times(1-k-g_p^{-1})g_p]g^{k-1}T_1\}\}\\ &-C_v\{T_1^{1-k}g_p^{1-k}g^{k(1-k)}\{\{a-[b+C_v\\ &\times(1-k-g_p^{-1})g_p]g^{k-1}T_1\}/(C_p+b)\}^k\}\}\\ &/\{C_p(a-2bg^{k-1}T_1)/(C_p+b)\}\\ &-b(g-1)^2\{K_1\{\{a-[bg^{k-1}+C_v\\ &\times(1-k-g_p^{-1})g_p g^{k-1}]T_1\}/(C_p+b)-T_1 g^{k-1}\}\\ &+K_2\{T_1^{1-k}g^{1-k}g^{k(1-k)}\{\{a-[bg^{k-1}+C_v\\ &\times(1-k-g_p^{-1})g_p g^{k-1}]T_1\}\\ &\quad/(C_p+b)-T_1 g^{k-1}\}^k-T_1\}\}\}\\ &\quad/\{C_p(a-2bg^{k-1}T_1)/(C_p+b)\}.\end{aligned} \tag{24}$$

Eqs. (22) and (24) are the major results of this paper. They determine the relations between the power output P, efficiency h, compression ratio g, and the pressure ratio g_p. The relation between power output and efficiency may be obtained using numerical calculations.

Two limiting cases are manifested from Eqs. (22) and (24). The first case, when $g_p = 1$, describes the relationship of the power and efficiency of the Diesel cycle

$$
\begin{aligned}
P_D = &\{C_v\{T_1(1-k g^{k-1})+[k/(C_p+b)][a \\
&+(C_p-b)g^{k-1}T_1]\}-C_v\{T_1^{1-k}g^{k(1-k)}\{[a \\
&+(C_p-b)g^{k-1}T_1]/(C_p+b)\}^k\}\} \\
&/\{K_1\{[a+(C_p-b)g^{k-1}T_1]/(C_p+b) \\
&-T_1g^{k-1}\}+K_2\{T_1^{1-k}g^{k(1-k)}\{[a+ \\
&(C_p-b)g^{k-1}T_1]/(C_p+b) \\
&-T_1g^{k-1}\}^k-T_1\}\}-b(g-1)^2,
\end{aligned}
\tag{25}
$$

$$
\begin{aligned}
h_D = &\frac{P_D}{\dot{Q}_{DA}} \\
&=\{C_v\{T_1(1-k g^{k-1}) \\
&+[k/(C_p+b)][a+(C_p-b) \\
&\times g^{k-1}T_1]-T_1^{1-k}g^{k(1-k)}\{[a \\
&+(C_p-b)g^{k-1}T_1]/(C_p+b)\}^k\}\} \\
&/\{C_p(a-2bg^{k-1}T_1)/(C_p+b)\} \\
&-\{b(g-1)^2\{K_1\{[a+(C_p-b) \\
&\times g^{k-1}T_1]/(C_p+b)-T_1g^{k-1}\} \\
&+K_2\{T_1^{1-k}g^{k(1-k)}\{[a+(C_p-b) \\
&\times g^{k-1}T_1]/(C_p+b)-T_1g^{k-1}\}^k-T_1\}\}\} \\
&/\{C_p(a-2bg^{k-1}T_1)/(C_p+b)\}
\end{aligned}
\tag{26}
$$

when $b=0$, the result is the same as that of Chen et al.

The second case, when $g_c=1$, describes the relationship of the power and efficiency of the Otto cycle

$$
\begin{aligned}
P_o = &\{\{C_v\{T_1(1-g^{k-1}) \\
&+[1/(C_v+b)][a+(C_v-b)g^{k-1}T_1] \\
&\times(1-g^{k-1})\}\}/\{(K_1+K_2g^{1-k}) \\
&\times\{[a+(C_v-b)g^{k-1}T_1] \\
&/(C_v+b)-g^{k-1}T_1\}\}\}-b(g-1)^2,
\end{aligned}
\tag{27}
$$

$$
\begin{aligned}
h_o = &\frac{P_o}{\dot{Q}_{OA}}=C_v\{T_1(1-g^{k-1}) \\
&+[1/(C_v+b)][a+(C_v-b) \\
&\times g^{k-1}T_1](1-g^{k-1})\}-b(g-1)^2 \\
&\times(K_1+K_2g^{1-k})\{[a+(C_v-b) \\
&\times g^{k-1}T_1]/(C_v+b)-g^{k-1}T_1\}\} \\
&/\{C_v\{[a+(C_v-b)g^{k-1}T_1] \\
&/(C_v+b)-g^{k-1}T_1\}\},
\end{aligned}
\tag{28}
$$

$$
\begin{aligned}
g_p = &(g_p)_o=T_4/(T_1g^{k-1}) \\
&=[a+(C_v-b)g^{k-1}T_1] \\
&/[(C_v+b)g^{k-1}T_1],
\end{aligned}
\tag{29}
$$

Where $(g_p)_o$ is the pressure ratio for the Otto cycle.

3. Numerical examples

To illustrate the preceding analysis, we have $t=33.33ms$, $C_v=0.7165kJ/K$ and $C_p=1.0031kj/K$. Taking equal heating and cooling times ($t_1=t_2=t/2=16.6ms$), the constant temperature rates K_1 and K_2 are estimated as $K_1=8.128\times10^{-6}s/K$ and $K_2=18.67\times10^{-6}s/K$. We consider $m=7s$ with $s=12.9kg/s$ and s is the friction coefficient of the exhaust and compression strokes. Taking the same values of x_2 and Δt_{12} as in Mozurkewich's paper and following the same procedure to determine the losses for the full cycle, we obtain $b=32.5\,W$. The maximum cycle-temperature ratio is $t=2742/329=8.33$. The typical parameter ranges of the four parameters are $g_p=1-(g_p)_o$, $a=2500-4000\ kJ/kg$, $b=0.3-1.8kJ/[(kg)K]$ and $T_1=300-400\ K$. The effects of g_p on the power output vs. efficiency curve for $a=3500\ kJ/kg$, $b=1\ kJ/[(kg)K]$ and $T_1=350\ K$ are shown in Fig. 2. The $g_p=1$ characteristic curve corresponds to the Diesel-cycle performance. The $g_p=g_{po}$ characteristic curve corresponds to the Otto-cycle performance. There is a maximum-power output point. When $g_p=g_m=16.512$ and $g_p=1.2$, the power output approaches its maximum value at $P_{max}=29.0214\ kW$ and the corresponding efficiency is $h_p=0.5174$. The effect of b on the power output vs. efficiency curve for $a = 3500\ kJ/kg$, $T_1 = 350\ K$, $b = 32.5\ W$ and $g_p = 1.2$ is shown in Fig. 3. The effect of a on the power output vs. efficiency curve for $b = 1\ kJ/[(kg)K]$, $b = 32.5\ W$, $T_1 = 350\ K$ and $g_p = 1.2$ is shown in Fig. 4. The effect of T_1 on the power output vs. efficiency curve for $a = 3500\ kJ/kg$, $b = 1\ kJ/[(kg)K]$, $b = 32.5\ W$ and $g_p = 1.2$ is shown in Fig. 5. The effects of b on the power output vs. efficiency curve for $a = 3500\ kJ/kg$, $b = 1\ kJ/[(kg)K]$, $T_1 = 350\ K$ and $g_p = 1.2$ are shown in Figs. 6.

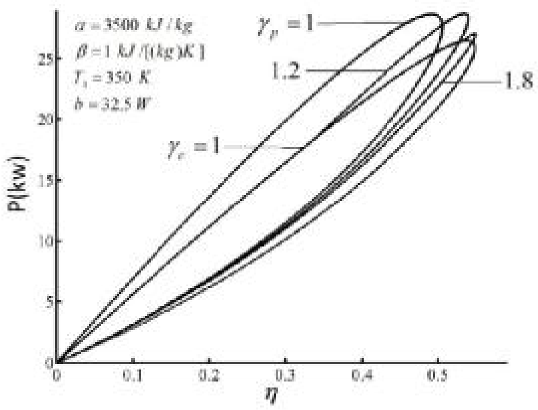

Figure 2. Effect of g_p on the $P-h$ characteristic.

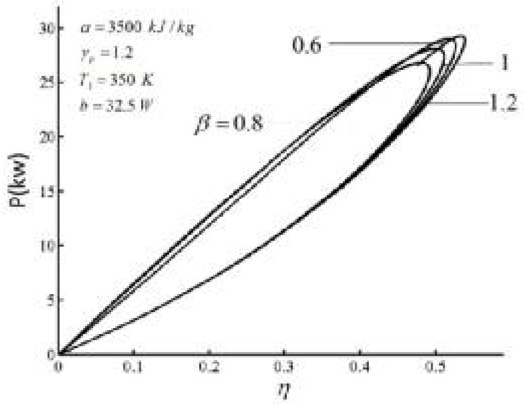

Figure 3. Effect of b on the $P-h$ characteristic.

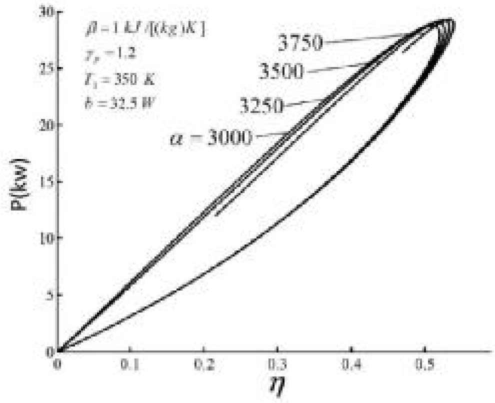

Figure 4. Effect of a on the $P-h$ characteristic.

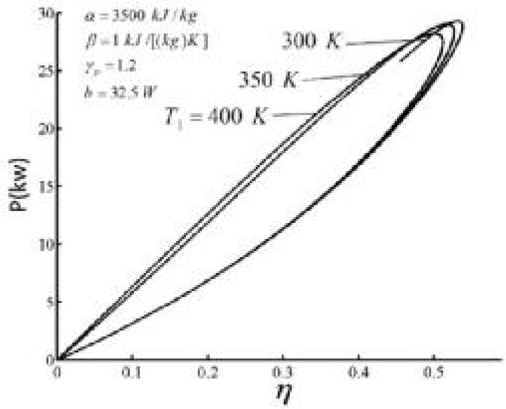

Figure 5. Effect of T_1 on the $P-h$ characteristic.

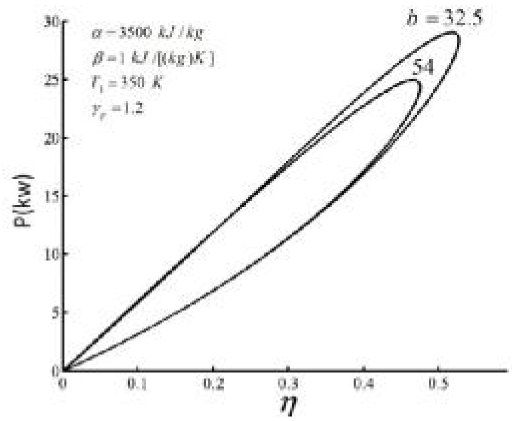

Figure 6. Effect of b on the $P-h$ characteristic.

4. Conclusions

In this paper, the effects of cylinder wall heat-transfer and global losses lumped in a friction-like term on the performance of a dual cycle during a finite time are investigated. The relations between net power output, efficiency, compression-ratio, and the pressure ratio are derived. The maximum power output and the corresponding efficiency and the maximum efficiency and the corresponding power output are also calculated. The results can also be applied to the performance analysis of the Diesel and Otto cycles. The detailed effect analyses are shown by numerical examples. The results can provide significant guidance for the performance evaluation and improvement of real dual engines.

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