Effects of specific heat ratio on the power output and efficiency characteristics for an irreversible dual cycle

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Abstract: In the present study, the performance of an air standard Dual cycle is analyzed using finite-time thermodynamics. The variations in power output and thermal efficiency with compression ratio, and the relations between the power output and the thermal efficiency of the cycle are presented. The results show that if compression ratio is less than certain value, the increase of specific heat ratio makes the power output and the thermal efficiency bigger. In contrast, if compression ratio exceeds certain value, the increase of specific heat ratio makes the work output and the thermal efficiency less. The results also show that the maximum power output and the maximum thermal efficiency increase while the compression ratio at the maximum power output point, the working range of the cycle and the compression ratio at maximum thermal efficiency point decrease with increasing specific heat ratio. It is noteworthy that the results obtained in the present study are of significance for providing guidance with respect to the performance evaluation of practical internal combustion engines. [Journal of American Science 2010;6(2):181-184]. (ISSN: 1545-1003).

Key words: Irreversible, Optimization, Dual cycle, performance

1. Introduction

Optimization studies for air-standard reciprocating cycles, i.e., Otto, Diesel and dual cycles, with rate-dependent loss mechanisms have appeared as early as in the 1980s physics literature. In the fundamental analysis of modern Diesel engines, the dual cycle is commonly employed as it includes the heat-addition processes both at constant volume and at constant pressure. Landsberg and Leff (1989) found that these important reversible thermodynamic heat engine cycles can be regarded as special cases of a more universal generalized cycle without any loss. Vecchiarelli et al. (1997) indicated that the hypothetical modification of gas turbine engines to include two heat additions (rather than one) may result in some efficiency improvement as compared with conventional engines. Chen et al. (1998) derived the relations between net work output and efficiency of the Diesel cycles. The relation between net work output and the efficiency as well as the maximum net-work output and the corresponding efficiency for internal-combustion Dual cycles are derived in this paper. Aragon-Gonzalez et al. (2000) derived the maximum irreversible work and efficiency of the Otto cycle by considering the irreversible adiabatic processes with the compression and expansion efficiencies. Ghatak and Chakraborty (2007) analyzed the effect of variable specific heats and heat transfer loss on the performance of the dual cycle when variable specific heats of working fluid are linear functions of its temperature. Zhang et al. (2007) built a generalized endoreversible steady flow thermodynamic cycle consisting of two constant-thermal-capacity heating branches, a constant thermal capacity cooling branch, and two adiabatic branches with consideration of heat resistance loss. The characteristics of the power output, efficiency, and exergy based ecological function were derived. Ge et al. (2008a; 2008b; 2009) analyzed the performance of an air standard Otto, Diesel and dual cycles. In the irreversible cycle model, the non-linear relation between the specific heat of the working fluid and its temperature, the friction loss computed according to the mean velocity of the piston, the internal irreversibility described by using the compression and expansion efficiencies, and the heat transfer loss are considered. Ebrahimi (2009a) studied the effects of stroke length on the performances of the Diesel cycle.

As can be seen in the relevant literature, the investigation of the effect of specific heat ratio on performance of dual cycle with considering heat transfer loss and friction loss of the piston does not appear to have been published. Therefore, the objective of this study is to examine the effect of specific heat ratio on performance of air standard dual cycle.

2. Thermodynamic analysis

The pressure-volume $(P-V)$ diagram of an irreversible dual heat engine is shown in Fig. 1. The compression ($1\rightarrow 2$) process ignition is isentropic; the heat additions are an isobaric process ($2\rightarrow 3$) and an isentropic process ($3\rightarrow 4$); the expansion process ($4\rightarrow 5$) is isentropic; and the heat rejection ($5\rightarrow 1$) is an isobaric process. The net cyclic work output per unit mass of working fluid without considering the lost power due to friction is:

$$W = c_v\left(T_3 - T_2\right) + c_p\left(T_4 - T_3\right) - c_v\left(T_5 - T_1\right) = \frac{R}{\gamma - 1}\left(T_3 - T_2 + \gamma\left(T_4 - T_3\right) - T_5 + T_1\right) \tag{1}$$

Where c_v is the constant volume specific heat, c_p is the constant pressure specific heat, γ is the specific hear ratio and R is the gas constant.

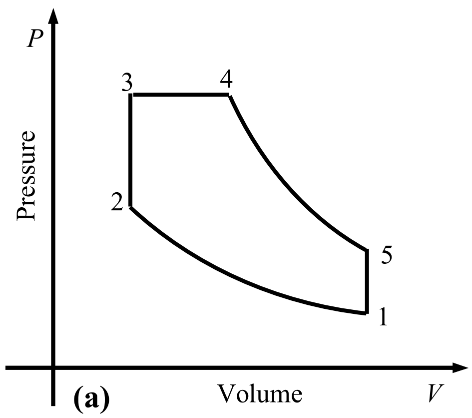

Figure 1. $P-V$ diagram of a dual cycle

The compression ratio r_c, the pressure ratio α and the cut-off ratio β are defined as:

$$r_c = \frac{V_1}{V_2} = \left(\frac{T_2}{T_1}\right)^{\frac{1}{\gamma-1}} \tag{2}$$

$$\alpha = \frac{p_3}{p_2} = \frac{T_3}{T_2} \tag{3}$$

and

$$\beta = \frac{V_4}{V_3} = \frac{T_4}{T_3} \tag{4}$$

The total heat added to the working fluid during process $2 \rightarrow 3$ and $3 \rightarrow 4$ is

$$Q_{in} = c_v\left(T_3 - T_2\right) + c_p\left(T_4 - T_3\right) = \frac{R}{\gamma - 1}\left[T_3 - T_2 + \gamma\left(T_4 - T_3\right)\right] \tag{5}$$

On the other hand, the heat added to the working fluid by combustion per unit mass of the working fluid is given in the following expression linear expression (Ebrahimi, 2009b):

$$Q_{in} = A - B\left(T_2 + T_4\right) \tag{6}$$

where A and B are two constants related to heat transfer and combustion.

In addition, process $1 \rightarrow 2$ is isentropic, hence

Since from the adiabatic processes we have

$$T_2 = T_1 r_c^{\gamma-1} \tag{7}$$

Substituting equation (7) into equation (3) yields

$$T_3 = \alpha T_1 r_c^{\gamma-1} \tag{8}$$

Combining equations (15) (8), we get

$$T_4 = \frac{\left(\gamma-1\right)\left(A - T_1 B r_c^{\gamma-1}\right) - R T_1 \alpha\left(1 - \gamma - \alpha^{-1}\right) r_c^{\gamma-1}}{R\gamma + B} \tag{9}$$

By combining the results obtained from Eqs. (8) and (9) into Eq. (4), gives

$$\beta = \frac{\left(\gamma-1\right)\left(A - T_1 B r_c^{\gamma-1}\right) - R T_1 \alpha\left(1 - \gamma - \alpha^{-1}\right) r_c^{\gamma-1}}{\left(R\gamma + B\right) T_1 \alpha r_c^{\gamma-1}} \tag{10}$$

Combining equations (3), (7) and (9), and bearing in mind that process $4 \rightarrow 5$ is isentropic, we get:

$$T_5 = T_1 \alpha \left[\frac{\left(\gamma-1\right)\left(A - T_1 B r_c^{\gamma-1}\right) - R T_1 \alpha\left(1 - \gamma - \alpha^{-1}\right) r_c^{\gamma-1}}{\left(R\gamma + B\right) T_1 \alpha r_c^{\gamma-1}}\right]^{\gamma-1} \tag{11}$$

Since W is work output per unit mass of working fluid of the cycle, the total power output of the cycle is

$$P_{out} = \dot{m}_t W - P_{lost} \tag{12}$$

where $\dot{m}_t$ is average mass flow rate of the working fluid in the cycle, P_{lost} is the lost power due to friction $= b\left(r_c - 1\right)^2$ and b is constant.

Substituting equations (7)-(9) and (11) into equation (12) yields

$$\begin{aligned} P_{out} &= \frac{\dot{m}_t R}{\gamma-1}\left(T_3 - T_2 + \gamma\left(T_4 - T_3\right) - T_5 + T_1\right) - b\left(r_c - 1\right)^2 = \frac{\dot{m}_t R}{\gamma-1}\Big\{\alpha T_1 r_c^{\gamma-1} - T_1 r_c^{\gamma-1} + \\ &\gamma\left[\frac{\left(\gamma-1\right)\left(A - T_1 B r_c^{\gamma-1}\right) - R T_1 \alpha\left(1 - \gamma - \alpha^{-1}\right) r_c^{\gamma-1}}{R\gamma + B} - \alpha T_1 r_c^{\gamma-1}\right] - T_1 \alpha \left[\frac{\left(\gamma-1\right)\left(A - T_1 B r_c^{\gamma-1}\right) - R T_1 \alpha\left(1 - \gamma - \alpha^{-1}\right) r_c^{\gamma-1}}{\left(R\gamma + B\right) T_1 \alpha r_c^{\gamma-1}}\right]^{\gamma-1} + \\ &+ T_1\Big\} - b\left(r_c - 1\right)^2 \end{aligned} \tag{13}$$

The thermal efficiency of the Dual cycle is

$$\eta = \frac{p_{out}}{\dot{m}_t Q_{in}} \tag{14}$$

Equations (13) and (14) determine the relations between the power output, thermal efficiency and compression ratio. The relation between power output and thermal efficiency to the maximum power output and the corresponding efficiency may be obtained using numerical calculations.

3. Results and discussion

The effects of γ on the performance of the dual cycle with $\dot{m}_t = 0.0156\,kg/s$, $A = 4280\,kJ/kg$, $b_1 = 15\,kW$, $B = 0.5\ kJ.kg^{-1}K^{-1}$, $r_c = 1 \rightarrow 40$ $\gamma = 1.3 \rightarrow 1.4$ and $T_1 = 350\,K$ (Heywood, 1988; Chen et al. 2004; Ghatak and Chakraborty, 2007; Ge et al., 2009; Ebrahimi, 2009c and 2009d) are shown in figures 2-4. Using the above constants and range of parameters, the power output versus compression ratio characteristic and the power output versus efficiency characteristic with varying the mean piston speed can be plotted. Numerical examples are shown as follows.

Figures 2–4 show the effects of the variable specific heat ratio on the cycle performance with heat resistance and friction losses. From these figures, it can be found that the specific heat ratio plays an important role on the power output and the thermal efficiency. They reflect the performance characteristics of an irreversible Diesel cycle engine. It can be seen from these figures that if compression ratio is less than certain value, the increase of specific heat ratio makes the power output and the thermal efficiency bigger. In contrast, if compression ratio exceeds certain value, the increase of specific heat ratio makes the work output and the thermal efficiency less. Therefore, it can be resulted that the effect of specific heat ratio on the performance of the cycle is related to compression ratio.

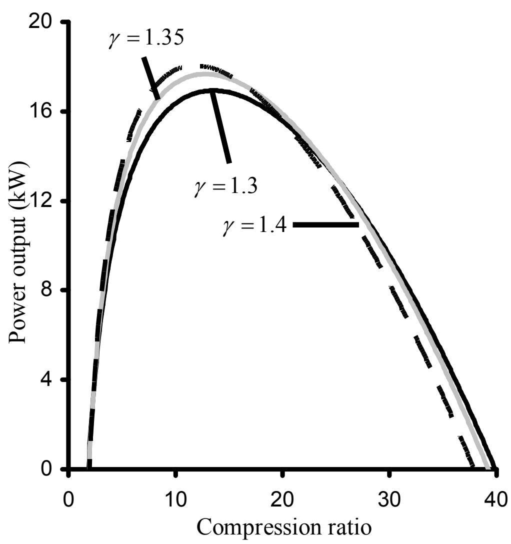

Figure 2. Effect of specific heat ratio on the variation of the power output with compression ratio

It can be also seen that the power output versus compression ratio characteristic and the power output versus efficiency characteristic are parabolic-like curves. They show that the maximum power output point and the maximum efficiency point are very adjacent. With increasing specific heat ratio, the maximum power output and the maximum thermal efficiency increase while the compression ratio at the maximum power output point, the working range of the cycle and the compression ratio at maximum thermal efficiency point decrease. It should be noted that the increase of the value of maximum power output with increasing specific heat ratio is due to the increase in the ratio of the heat added to the heat rejected. Numerical calculation shows that when specific heat ratio increases by about 7.7%, the maximum power output and the maximum thermal efficiency increase by about 6.2% and 22.8%, respectively. Furthermore, the compression ratio at the maximum power output point, the working range of the cycle and the compression ratio at maximum thermal efficiency point decrease by about 14.5%, 5.9% and 5.3%, respectively, as the specific heat ratio increases by about 7.7%.

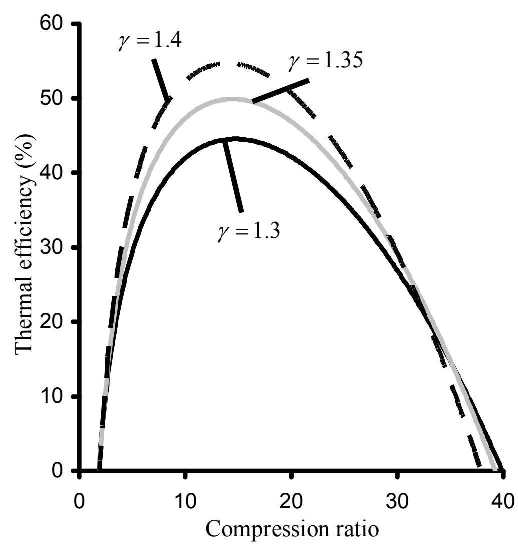

Figure 3. Effect of specific heat ratio on the variation of the thermal efficiency with compression ratio

Figure 4 shows the effects of the specific heat ratio on the power output versus the thermal efficiency characteristic. The power output versus thermal efficiency characteristics exhibit loop shaped curves as is common to almost all real heat engines (Gordon and Huleihil, 1992). From the figure, it is found that the parameter specific heat ratio has a significant influence on the power output versus thermal efficiency characteristic. When specific heat ratio increases, the efficiency at the maximum power output point, as well as the power output at the maximum efficiency point, will also increase. If specific heat ratio increases by about 7.7%, the optimal power output corresponding to maximum efficiency and the optimal thermal efficiency corresponding to maximum power output increase by about 22.8% and 6.6%, respectively.

According to the above analysis, it can be found that the effects of the specific heat of the working fluid on the cycle performance are obvious, and they should be considered in practice cycle analysis to make the cycle model more close to practice.

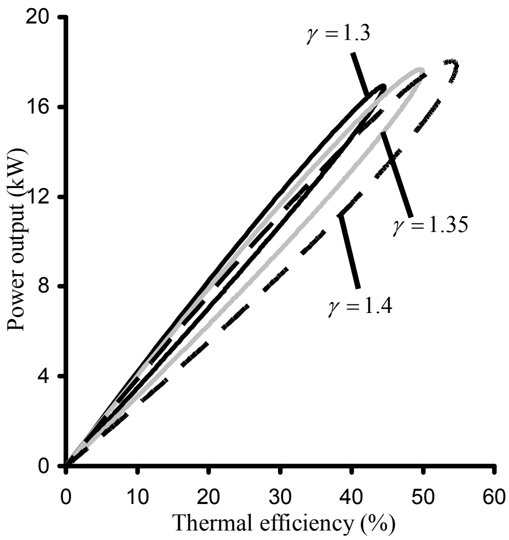

Figure 4. Effect of specific heat ratio on the variation of the power output with thermal efficiency

4-Conclusion

In this paper, the effects of specific heat ratio on the Dual cycle are investigated. The relation between power output, thermal efficiency and compression ratio are derived. These results show that if compression ratio is less than certain value, the increase of specific heat ratio makes the power output and the thermal efficiency bigger. In contrast, if compression ratio exceeds certain value, the increase of specific heat ratio makes the work output and the thermal efficiency less. With increasing specific heat ratio, the maximum power output and the maximum thermal efficiency increase while the compression ratio at the maximum power output point, the working range of the cycle and the compression ratio at maximum thermal efficiency point decrease. The results provide significant guidance for the performance evaluation and improvement of real Dual engines.

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