Performance optimization of a Diesel cycle with specific heat ratio

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Abstract: The performance of an air standard Diesel cycle is analyzed by using finite-time thermodynamics. The relations between the power output and the compression ratio, between the thermal efficiency and the compression ratio, and the optimal relation between power output and the efficiency of the Diesel cycle are derived by detailed numerical examples. Moreover, the effects of specific heat ratio of the working fluid on the irreversible cycle performance are analyzed. The results show that there are significant effects of the specific heat of the working fluid on the performance of the Diesel cycle. The conclusions obtained in this investigation are in full agreement with those of published studies for other cycles and may be used when considering the designs of actual Diesel engines. [Journal of American Science 2010;6(1):157-161]. (ISSN: 1545-1003).

Key words: finite-time thermodynamics; Diesel cycle; internal irreversibility; performance optimization

1. Introduction

In the last two decades, by using finite time thermodynamics theory, many optimization studies based on various performance criteria have been carried out for endoreversible and irreversible heat engine models. (Sieniutycz and Salamon, 1990; Chen et al., 2008; Ge et al., 2008a). Mozurkewich and Berry (1982) and Hoffman et al. (1985) used mathematical techniques, developed in optimal-control theory, to reveal the optimal motions of the pistons in Otto and Diesel cycle engines, respectively. Orlov and Berry (1993) deduced the power and efficiency upper limits for internal-combustion engines. Blank and Wu (1993) examined the effect of combustion on the work or power optimised Otto, Diesel and Dual cycles. They derived the maximum work or power and the corresponding efficiency bounds. Lin et al. (1999) derived the relations between the net power and the efficiency for the Dual cycle with due consideration of the heat-transfer losses. Akash (2001) investigated the effect of heat transfer on thermal performance of an air-standard Diesel cycle. Chen et al. (2002) modeled the behaviors of Diesel cycle, with friction losses, over a finite period. Wang et al. (2002) modeled the dual cycle with a friction like term loss and studied the effect of the friction like term loss on cycle performance. Chen et al. (2003, 2004) determined the characteristics of power and efficiency for Otto and Dual cycles with heat transfer and friction losses. Ge et al. (2005) derived the performance characteristics of the diesel cycle with heat transfer and friction like term losses when the maximum temperature of the cycle was not fixed. Parlak and Sahin [2006] defined the internal irreversibility by using entropy production, and analyzed the effect of the internal irreversibility on the performance of irreversible reciprocating heat-engine cycle. Al-Sarkhi et al. (2006) found that friction and the temperature-dependent specific heat of the working fluid of a Diesel engine had significant influences on its power output and efficiency. Ge et al. (2007) studied the effects of variable specific heats of the working fluid on the performances of the Diesel cycle. 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 the temperature dependent specific heat ratio of the working fluid on the performance of the Dual cycle.

As can be seen in the relevant literature, the investigation of the effect of specific heat ratio on performance of Diesel cycle does not appear to have been published. Therefore, the objective of this study is to examine the effect of engine speed on performance of air standard Diesel cycle.

2. An air standard Diesel cycle model

An air-standard diesel cycle model is shown in Fig. 1. Process $1 \to 2s$ is a reversible adiabatic compression, while process $1 \to 2$ is an irreversible adiabatic process that takes into account the internal irreversibility in the real compression process. The heat addition is an isobaric process $2 \to 3$. Process $3 \to 4s$ is a reversible adiabatic expansion, while $3 \to 4$ is an irreversible adiabatic process that takes into account the internal irreversibility in the real expansion process. The heat rejection is an isochoric process $4 \to 1$.

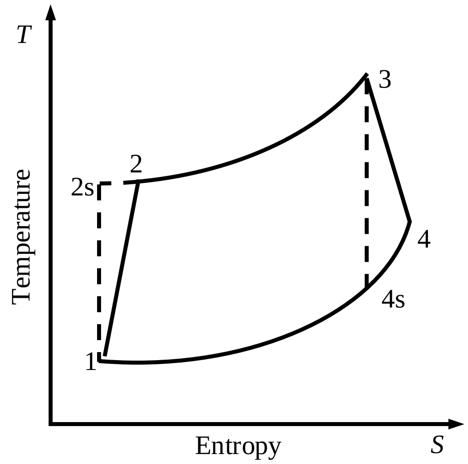

Figure 1. $T-S$ diagram for the air standard Diesel cycle

For the heat addition and heat rejection ($2 \to 3$ and $4 \to 1$ stages, respectively), it is assumed that heating occurs from state 2 to state 3 and cooling ensues from state 4 to state 1 and proceed according to isothermal rates, as shown in Eqs. (1) and (2) (Chen et al., 2002; Al-Sarkhi et al., 2006):

$$dT/dt = 1/K_1 \quad (\text{for } 2 \to 3) \tag{1}$$

and

$$dT/dt = 1/K_2 \quad (\text{for } 4 \to 1) \tag{2}$$

where T is the absolute temperature and t is the time; K_1 and K_2 are constants. By integrating Eqs. (1) and (2), we obtain:

$$t_1 = K_1\left(T_3 - T_2\right) \tag{3}$$

and

$$t_2 = K_2\left(T_4 - T_1\right) \tag{4}$$

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

$$\tau = t_1 + t_2 = K_1\left(T_3 - T_2\right) + K_2\left(T_4 - T_1\right) \tag{5}$$

The total reversible power output is

$$P_{rev} = \frac{W_{out}}{\tau} = \frac{Mc_p\left(T_3 - T_2\right) - Mc_v\left(T_4 - T_1\right)}{K_1\left(T_3 - T_2\right) + K_2\left(T_4 - T_1\right)} \tag{6}$$

where W_{out} is the total reversible work, c_v is the constant volume specific heat, c_p is the constant pressure specific heat and M is the molar number of the working fluid.

The relation between the specific heat at constant pressure and the specific heat at constant volume is:

$$c_p - c_v = R \tag{7}$$

where R is the molar gas constant of the working fluid.

The compression ratio is defined as:

$$r_c = V_1/V_2 \tag{8}$$

For the processes $1 \to 2s$ and $3 \to 4s$, we have

$$T_{2s} = T_1 r_c^{\gamma-1} \tag{9}$$

and

$$T_{4s} = T_1^{\gamma-1} T_3^{\gamma} r_c^{(1-\gamma)\gamma} \tag{10}$$

Where γ is the ratio of specific heats, $\gamma = c_p/c_v$.

For the two reversible adiabatic processes $1 \to 2s$ and $3 \to 4s$, the compression and expansion efficiencies can be defined as (Ge et al., 2008a; Ebrahimi, 2009c; Lin and Hou, 2008):

$$\eta_c = \left(T_{2s} - T_1\right)/\left(T_2 - T_1\right) \tag{11}$$

and

$$\eta_e = \left(T_4 - T_3\right)/\left(T_{4s} - T_3\right) \tag{12}$$

where:

By substituting Eqs. (7)–(12) into Eq. (6), The total reversible power output becomes:

$$P_{rev} = \frac{\begin{aligned}&MR\Big[T_1\left(\eta_c - \gamma r_c^{\gamma-1} + \gamma\right) - \eta_c\gamma + \\ &T_3\left(\eta_c\eta_e - \eta_c + \eta_c\gamma\right) - \eta_c\eta_e T_1^{\gamma-1} T_3^{\gamma} r_c^{(1-\gamma)\gamma}\Big]\end{aligned}}{\begin{aligned}&(\gamma-1)\Big[T_1\left(K_1 - K_1 r_c^{\gamma-1} - K_2\eta_c\right) + \\ &T_3\left(K_1\eta_c + K_2\eta_c - K_2\eta_c\eta_e\right) - K_1\eta_c + \\ &K_2\eta_c\eta_e T_1^{\gamma-1} T_3^{\gamma} r_c^{(1-\gamma)\gamma}\Big]\end{aligned}} \tag{13}$$

Taking into account the friction loss of the piston and assuming a dissipation term represented by a friction force that is a linear function of the piston velocity gives (Chen et al., 2006; Ge et al., 2007; Ebrahimi, 2009a)

$$f_\mu = -\mu S_p = -\mu\, dx/dt \tag{14}$$

where μ is the coefficient of friction, which takes into account the global losses, x is the piston's displacement and S_p is the piston's velocity. Therefore, the lost power due to friction is

$$\begin{aligned}&P_{los} = dW_{los}/dt = \\ &-\mu\left(dx/dt\right)\left(dx/dt\right) = -\mu\left(S_p\right)^2\end{aligned} \tag{15}$$

The piston mean velocity is

$$\bar{S}_p = \left(x_1 - x_2\right)/\Delta t_{12} = \left[x_2\left(r_c - 1\right)\right]/\Delta t_{12} \tag{16}$$

Where x_2 is the piston at minimum volume and Δt_{12} is the time spent in the power stroke. Thus, the resulting power output $\left(P_{out}=P_{rev}-P_{los}\right)$ is:

$$P_{out}=\frac{\begin{array}{l}MR\left[T_1\left(\eta_c-\gamma r_c^{\gamma-1}+\gamma\right)+\right.\\ T_1\left(K_1-K_1r_c^{\gamma-1}-K_2\eta_c\right)\\ \left.\eta_c\eta_e T_1^{\gamma-1}T_3^{\gamma}r_c^{(1-\gamma)\gamma}-\eta_c\gamma\right]\end{array}}{\begin{array}{l}(\gamma-1)\left[K_2\eta_c\eta_e T_1^{\gamma-1}T_3^{\gamma}r_c^{(1-\gamma)\gamma}+\right.\\ T_3\left(K_1\eta_c+K_2\eta_c-K_2\eta_c\eta_e\right)-\\ \left.K_1\eta_c+T_1\left(K_1-K_1r_c^{\gamma-1}-K_2\eta_c\right)\right]\end{array}}-b\left(r_c-1\right)^2 \quad (17)$$

Where $b=\mu x_2^2/\left(\Delta t_{12}\right)^2$, and the thermal efficiency of the Diesel cycle engine is expressed by

$$\eta_{th}=P/\left(Q_{in}/\tau\right) \quad (18)$$

where:

$$Q_{in}=Mc_p\left(T_3-T_2\right) \quad (19)$$

Notice that both power and efficiency are convex functions of the compression ratio.

The relations between the power output and the compression ratio, as well as between the efficiency and the compression ratio, can be derived from Eqs. (17) and (18). The maximum power and the corresponding efficiency can be obtained by numerical calculations.

3. Numerical examples and discussions

According to references (Chen et al. 2006; Ghatak and Chakraborty, 2007; Ge et al., 2009; Ebrahimi, 2009b), the following parameters are used: $\eta_e=0.97$, $\eta_c=0.97$, $M=1.57\times10^{-5}\,kmol$, $T_3=2200\,K$, $K_1=8.128\times10^{-6}\,s.K^{-1}$, $\gamma=1.3\rightarrow1.4$, $b_1=15\,kW$ $K_2=18.67\times10^{-6}\,s.K^{-1}$, $r_c=1\rightarrow40$ and $T_1=300\,K$. Using the above constants and range of parameters, the characteristics of $P_{out}-r_c$ and $\eta_{th}-r_c$ can be plotted.

Figures 2–4 show the effects of the specific heat ratio of the working fluid on the power output and the thermal efficiency of the cycle with irreversible friction losses (The dashed lines in the figures denote where the cycle cannot work). 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.

The variations of the power output with respect to the compression ratio and the specific heat ratio are indicated in Figure 2. It can be seen that the power output versus the compression ratio characteristic is parabolic like curve. In other words, there is a maximum power output in the range of compression ratio. With increasing specific heat ratio, the maximum power output and the compression ratio at the maximum power output point increase. Therefore, it can be resulted that the effect of specific heat ratio on the performance of the cycle is related to compression ratio. 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 compression ratio at the maximum power output point increase by about 22.2% and 1.3%, respectively.

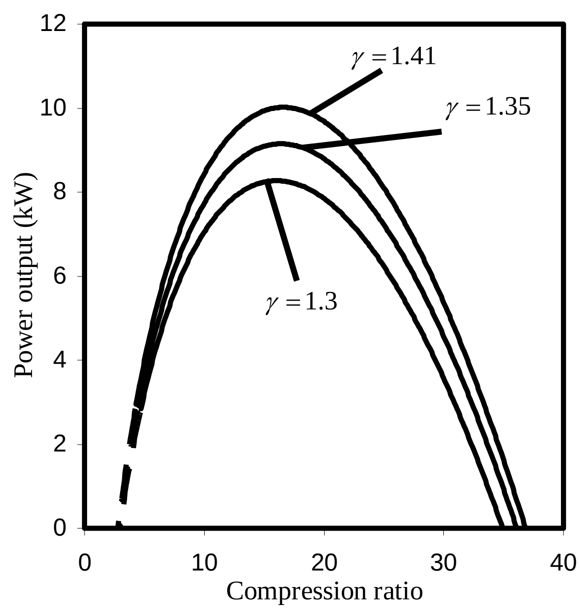

Figure 2. Effect of specific heat ratio on the variation of the output power with compression ratio

Figure 3 shows the effects of specific heat ratio on thermal efficiency with respect to the compression ratio. It can be seen that the thermal efficiency versus the compression ratio characteristic is parabolic like curve. In other word, the thermal efficiency increase with increasing compression ratio, reach their maximum values and then decrease with further increase in compression ratio. With increasing specific heat ratio, the maximum thermal efficiency increases while the compression ratio at maximum thermal efficiency point remains approximately constant. It should be noted that the increase of the value of maximum thermal efficiency 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 thermal efficiency increases 25.4%.

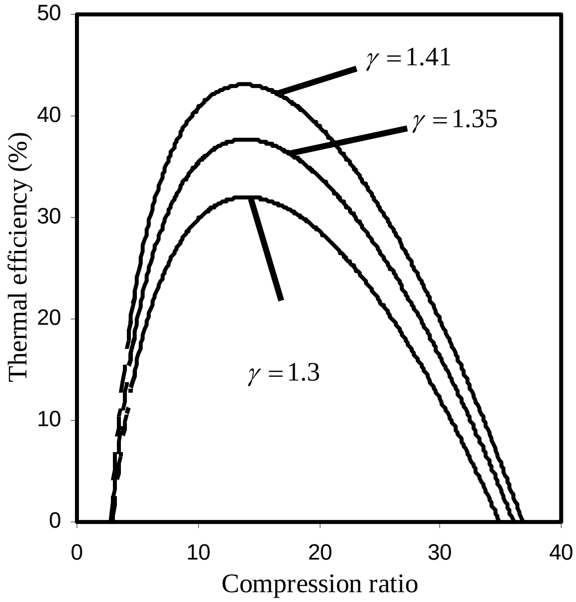

Figure 3. Effect of specific heat ratio on the variation of the thermal efficiency with compression ratio

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 (Chen et al., 1999; 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. They show that the maximum power output point and the maximum efficiency point are very adjacent. 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 19.9% and 43.7%, 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.

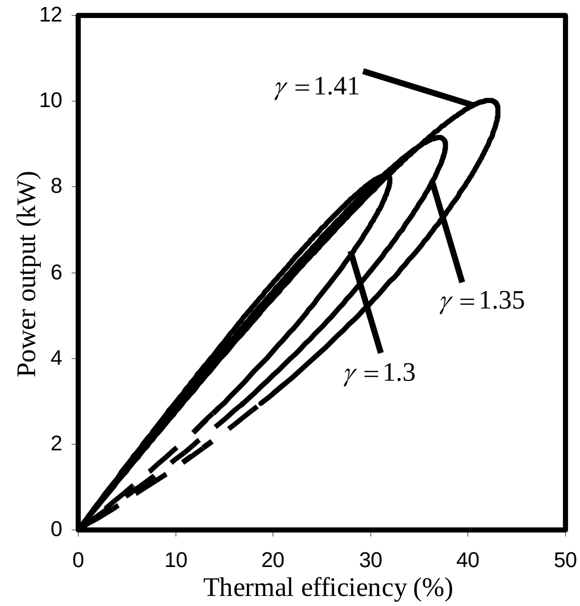

Figure 4. Effect of specific heat ratio on the variation of the output power with thermal efficiency

4-Conclusion

In this paper, an irreversible air standard Dual cycle model which is more close to practice is established. In the model, the friction loss computed according to the mean velocity of the piston and the internal irreversibility described by using the compression and expansion efficiencies are considered. The performance characteristics of the cycle were obtained by numerical examples. The results show that there are significant effects of the specific heat of the working fluid on the performance of the cycle, and this should be considered in practical cycle analysis. The conclusions of this investigation are of importance when considering the designs of actual Diesel engines.

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