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Effects of cut-off ratio on performance of an irreversible Dual cycle

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Abstract:
This study is aimed at investigating the effects of cut-off ratio on the Dual cycle performance with considerations of heat transfer loss, variable specific heat ratio and friction irreversible losses. By using finite time thermodynamics theory, the characteristic curves of the power output versus compression ratio and the power output versus thermal efficiency are obtained. The results shows that if compression ratio is less than certain value, the power output decreases with increasing cut-off ratio, while if compression ratio exceeds certain value, the power output first decreases and then starts to increase with increasing cut-off ratio. With further increase in compression ratio, the increase of cut-off ratio results in increasing the power output. Effect of cut-off ratio on the variation of the thermal efficiency with compression ratio is similar to that for the power output. The results also shows that the working range of the cycle, the maximum power output and the optimal power output corresponding to maximum thermal efficiency first increase and then start to decline as the cut-off ratio increases. The results obtained in this work can help us to understand how the cycle performance is influenced by the variation of the cut-off ratio, and they should be considered in practical cycle analysis. [Journal of American Science 2009:5(3) 83-90] (ISSN: 1545-1003)

Key word: Finite time thermodynamics; Performance analysis; Dual cycle; cut-off ratio

1. Introduction

Since finite-time thermodynamics [Andresen et al. 1984; Bejan 1996; Aragon-Gonzalez et al. 2006] is a powerful tool for the performance analysis and optimization of real internal combustion engine cycle, much work has been performed for the performance analysis and optimization of finite time processes and finite size devices [Aizenbud and Band, 1984; Sieniutycz and Shiner, 1994; Aragon-Gonzalez et al., 2000; Lin and Hou, 2008]. Sahin et al. (2002) made a comparative performance analysis of an endoreversible Dual cycle under the maximum ecological function and maximum power conditions. The optimal performances and design parameters such as compression ratio, cut-off ratio and thermal conductance allocation ratio which maximize the ecological objective function are investigated. The results are compared with those of the maximum-power performance criterion. Chen et al. (2004) determined the characteristics of power and efficiency for Dual cycle with heat transfers and friction losses. It is found that there are optimal values of the cut-off ratio at which the power output and efficiency attain their maxima. The effects of cut-off ratio on performance of an irreversible Dual cycle were presented by Parlak [Parlak et al., 2004; Parlak, 2005]. Ust et al. (2005) performed an ecological performance analysis for an irreversible Dual cycle by employing the new thermo-ecological criterion as the objective function. They compared the effects of cut-off ratio on performance of the cycle. Al-Sarkhi et al. (2006) investigated the effects of friction, temperature-dependent specific-heat of the working fluid and cut-off ratio on the performances of the Diesel-cycle. Optimum value of the cut-off ratio of the diesel cycle is derived analytically and compared to the results of an experimental study of the low heat rejection engine by Parlak et al. (2008).

On the basis of these research works, the aim of this paper is to model an air standard Dual cycle with considerations the variable specific heat ratio during a finite time, furthermore study the effects of cut-off ratio on its characteristics of power and efficiency.

2. Thermodynamic analysis

An air standard Dual cycle model is shown in Fig. 1. The compression ($1 \to 2$) process ignition is isentropic; the heat additions are an isobaric process ($2 \to 3$) and an isentropic process ($3 \to 4$); the expansion process ($4 \to 5$) is isentropic; and the heat rejection ($5 \to 1$) is an isobaric process.

As is usual in finite time thermodynamic heat engine cycle models, there are two instantaneous adiabatic processes ($1 \to 2$) and ($4 \to 5$). For the isochoric branches ($2 \to 3$ and $5 \to 1$) and the isobaric branch ($3 \to 4$) in Fig. 1, it is assume that heating from state 2 to state 4 and cooling from state 5 to state 1 proceed at to temperature rates, as shown in Eq. (1) [Chen el al., 2004; Qin et al., 2003]:

$$t_1 = K_1\left(T_4 - T_2\right), \quad t_2 = K_2\left(T_5 - T_1\right) \tag{1}$$

Where K_1 and K_2 are constants linked to the

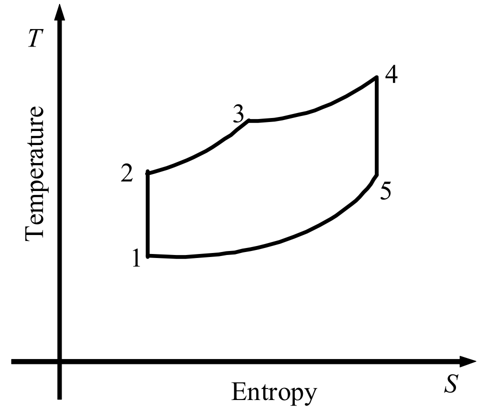

Fig. 1. $T-S$ diagram for the air standard Dual

mean variation rate of the temperature. t_1 and t_2 are the heating and cooling times, respectively. In this way, the cycle's period is given by:

$$\tau = t_1 + t_2 = K_1\left(T_4 - T_2\right) + K_2\left(T_5 - T_1\right) \tag{2}$$

In a real cycle, the specific heat ratio is generally modeled as a linear function of mean charge temperature [Gatowski el al., 1984; Ebrahimi, 2006]. Thus, it can be supposed that the specific heat ratio of the working fluid is function of temperature alone and has the linear forms:

$$\gamma = \gamma_\circ - k_1 T \tag{3}$$

where γ is the specific heat ratio and T is the absolute temperature. $\gamma_\circ$ and k_1 are constants. The heat added to the working fluid, during processes ($2 \to 3$) and ($3 \to 4$) is

$$Q_{in} = M\left(\int_{T_2}^{T_3} c_v dT + \int_{T_3}^{T_4} c_p dT\right) =$$

$$M\int_{T_2}^{T_3}\left(\frac{R}{\gamma_\circ - k_1 T - 1}\right)dT + M\int_{T_3}^{T_4}\left(\frac{\left(\gamma_\circ - k_1 T\right)R}{\gamma_\circ - k_1 T - 1}\right)dT \tag{4}$$

$$= \frac{MR}{k_1}\ln\left(\frac{\gamma_\circ - k_1 T_2 - 1}{\gamma_\circ - k_1 T_4 - 1}\right) + MR\left(T_4 - T_3\right)$$

where M is the molar number of the working fluid which is function of engine speed. R and c_p are molar gas constant and molar specific heat at constant pressure for the working fluid, respectively.

The heat rejected by the working fluid during the process ($5 \rightarrow 1$) is

$$Q_{out} = M\int_{T_1}^{T_5} c_v dT = M\int_{T_1}^{T_5}\left(\frac{R}{\gamma_\circ - k_1 T - 1}\right)dT = \frac{MR}{k_1}\ln\left(\frac{\gamma_\circ - k_1T_1 - 1}{\gamma_\circ - k_1T_5 - 1}\right) \tag{5}$$

where c_v is the molar specific heat at constant volume for the working fluid.

According to Refs [Ge et al., 2005; Al-Sarkhi et al., 2006], a suitable engineering approximation for the reversible adiabatic process with variable γ can be assumed, i.e. this process can be broken up into infinitesimally small processes, for each of these processes, the adiabatic exponent γ can be regarded as constant. For example, any reversible adiabatic process between states i and j can be regarded as consisting of numerous infinitesimally small processes, for each of which a slightly different value of γ applies. For any of these processes, when infinitesimally small changes in temperature dT and in volume dV of the working fluid takes place, the equation for a reversible adiabatic process with variable γ can be written as follows:

$$TV^{\gamma-1} = \left(T + \mathrm{d}T\right)\left(V + \mathrm{d}V\right)^{\gamma-1} \tag{6}$$

From Eq. (6), we get the following equation

$$T_i\left(\gamma_\circ - k_1T_j - 1\right) = T_j\left(\gamma_\circ - k_1T_i - 1\right)\left(V_j / V_i\right)^{\gamma_\circ - 1} \tag{7}$$

The compression, r_c , pressure, α and β cut-off ratios are defined as

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

and

$$\alpha = T_3/T_2 \tag{9}$$

and also

$$\beta = V_4/V_3 = T_4/T_3 \tag{10}$$

Therefore, the equations for processes ($1 \rightarrow 2$) and ($4 \rightarrow 5$) are shown, respectively, by the following:

$$T_1\left(\gamma_\circ - k_1T_2 - 1\right)\left(r_c\right)^{\gamma_\circ - 1} = T_2\left(\gamma_\circ - k_1T_1 - 1\right) \tag{11}$$

$$T_4\left(\gamma_\circ - k_1T_5 - 1\right) = T_5\left(\gamma_\circ - k_1T_4 - 1\right)\left(\frac{T_3}{T_4}r_c\right)^{\gamma_\circ - 1} \tag{12}$$

The energy transferred to the working fluid during combustion is given by the following linear relation [Klein, 1991; Zhao and Chen, 2007].

$$Q_{in} = M\left[A - B\left(T_2 + T_4\right)\right] \tag{13}$$

where A and B are two constants related to combustion and heat transfer. The heat loss through the cylinder wall is assumed to be proportional to the average temperature of both the working fluid and the cylinder wall and the wall temperature is constant. From Eq. (13), it can be seen that Q_{in} contained two parts: the first part is the released heat by combustion per second, MA , and the second part is the heat leak loss per second,

$$Q_{leak} = MB\left(T_2 + T_4\right)$$

Taking into account the friction loss of the piston and assuming a dissipation term represented by a friction force which is a linear function of the piston velocity gives [Wang et al., 2002; Ge et al., 2005]

$$f_\mu = -\mu v = -\mu\frac{\mathrm{d}x}{\mathrm{d}t} \tag{14}$$

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

$$P_\mu = \frac{dW_\mu}{dt} = -\mu\left(\frac{\mathrm{d}x}{\mathrm{d}t}\right)^2 = -\mu v^2 \tag{15}$$

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

$$\bar{v} = \frac{x_1 - x_2}{\Delta t_{12}} = \frac{x_2(r_c - 1)}{\Delta t_{12}} \tag{16}$$

where x_2 is the piston's position corresponding to the minimum volume of the trapped gases and Δt_{12} is the time spent in the power stroke. Thus, the power output of the Dual cycle engine can be written as

$$P_{out} = \frac{W}{\tau} - P_{\mu} = \frac{\left[\begin{array}{l} \frac{MR}{k_1}\ln\left(\frac{(\gamma_{\circ}-k_1T_2-1)(\gamma_{\circ}-k_1T_5-1)}{(\gamma_{\circ}-k_1T_4-1)(\gamma_{\circ}-k_1T_1-1)}\right) \\ +MR(T_4-T_3) \end{array}\right]}{K_1(T_4-T_2)+K_2(T_5-T_1)} - b_1(r_c-1)^2 \quad (17)$$

Where

$$b_1 = \frac{\mu x_2^2}{(\Delta t_{12})^2} \quad (18)$$

The thermal efficiency of the Dual cycle engine is expressed by

$$\eta_{th} = \frac{P_{out}}{Q_{in}} = \frac{\left(\begin{array}{l} \frac{MR}{k_1}\ln\left(\frac{(\gamma_{\circ}-k_1T_2-1)(\gamma_{\circ}-k_1T_5-1)}{(\gamma_{\circ}-k_1T_4-1)(\gamma_{\circ}-k_1T_1-1)}\right)+ \\ MR(T_4-T_3)- \\ b_1(r_c-1)^2\left(K_1(T_3-T_2)-K_2(T_4-T_1)\right) \end{array}\right)}{\frac{MR}{k_1}\ln\left(\frac{\gamma_{\circ}-k_1T_2-1}{\gamma_{\circ}-k_1T_4-1}\right)+MR(T_4-T_3)} \quad (19)$$

When the values of r_c and T_1 are given, T_2 can be obtained from Eq. (11). When β is given, substituting Eq. (4) into Eq. (13) yields T_3, and T_4 can be found using Eq. (10). But, when α is given, T_3 can be found from Eq. (9); then, substituting Eq. (4) into Eq. (13) yields T_4. The last unknown is T_5, which can be obtained from Eq. (12). Substituting T_1, T_2, T_3, T_4 and T_5 into Eqs. (17) and (19), respectively, the power output and thermal efficiency of the Dual cycle engine can be obtained. Therefore, the relations between the power output, the thermal efficiency and the compression ratio can be derived.

3. Results and discussion

The following constants and parameter values have been used in this exercise: $T_1 = 285\,K$, $\mu = 0.0129\,kNsm^{-1}$, $M = 1.57\times10^{-5}\,kmol$, $k_1 = 0.00006\,K^{-1}$, $\gamma_{\circ} = 1.41$, $A = 70000\ J.mol^{-1}$, $B = 25\ J.mol^{-1}K^{-1}$, $b_1 = 20\,kW$, $\alpha = 1$ and $\beta = 1-1.8$ [Ghatak and Chakraborty, 2007; Chen et al., 2006; Ge et al., 2005; Mozurkewich and Berry, 1982]. Using the above constants and range of parameters, the characteristic curves of the power output and efficiency, varying with the pressure ratio, and the power output versus efficiency can be plotted. It should be noted that the $\beta = 1$ characteristic curve corresponds to the Otto cycle performance and the $\alpha = 1$ characteristic curve corresponds to the Diesel cycle performance. It is also worth mentioning here that when $T_5 \geq T_4$ and $T_3 > T_4$, the cycle cannot operate normally.

Figs. 2-4 display the influence of the parameter β on the Dual cycle performance with considerations of heat transfer and friction like term losses (The dashed lines in the figures denote where the cycle cannot work normally). From these figures, it can be found that β plays important roles on the power output and the thermal efficiency. It can be seen that the power output versus compression ratio characteristic and the thermal efficiency versus compression ratio characteristic are approximately parabolic like curves. In other words, the power output and the thermal efficiency increase with increasing compression ratio. This is due to the fact that the ratio of the heat added to the working fluid to the heat rejected by the working fluid increases with the increase of the compression ratio. With further increase in compression ratio, the power output and the thermal efficiency decrease, due to the increase of the friction loss and the heat transfer between the working fluid and the cylinder wall. It can be seen that the curves of power output versus thermal efficiency are loop shaped. As can be clearly seen from these figures, the effects of β on the power output and thermal efficiency are related to compression ratio.

The results also show that if compression ratio is less than certain value, the increase of β make the power output less. This can be attributed to the fact that ratio of the heat added to the working fluid to the heat rejected by the working fluid decreases when β increased. While if compression ratio exceeds certain value, the power output first decreases and then starts to increase as the cut-off ratio increases. With further increase in compression ratio, the increase of β results in increasing the power output due to the decrease of the heat transfer between the working fluid and the cylinder wall. The behavior of the thermal efficiency with compression ratio for various cut-off ratios is similar to that for the power output.

Numerical calculation shows that the smallest power output is for the Diesel cycle when $r_c < 17.9$ and is for the Otto cycle when $r_c > 17.9$ and also the largest power output is for the Otto cycle when $r_c < 6.5$ and is for the Dual cycle when $r_c > 6.5$. It can also be concluded from the numerical calculation that the smallest thermal efficiency is for the Diesel cycle when $r_c < 23.6$ and is for the Otto cycle when $r_c > 23.6$ and also the largest thermal efficiency is for the Otto cycle when $r_c < 11.5$ and is for the Dual cycle when $r_c > 11.5$.

Referring to Figs 2-4, it can also be concluded that the working range of the cycle, the maximum power output and the optimal power output corresponding to maximum thermal efficiency increase and then start to decline with the increase of β. With the increase of β, the optimal compression ratio corresponding to maximum thermal efficiency increases while the maximum thermal efficiency and the optimal thermal efficiency corresponding to maximum power output decrease. The optimal compression ratio corresponding to maximum power output first increases with increasing β and then starts to decline.

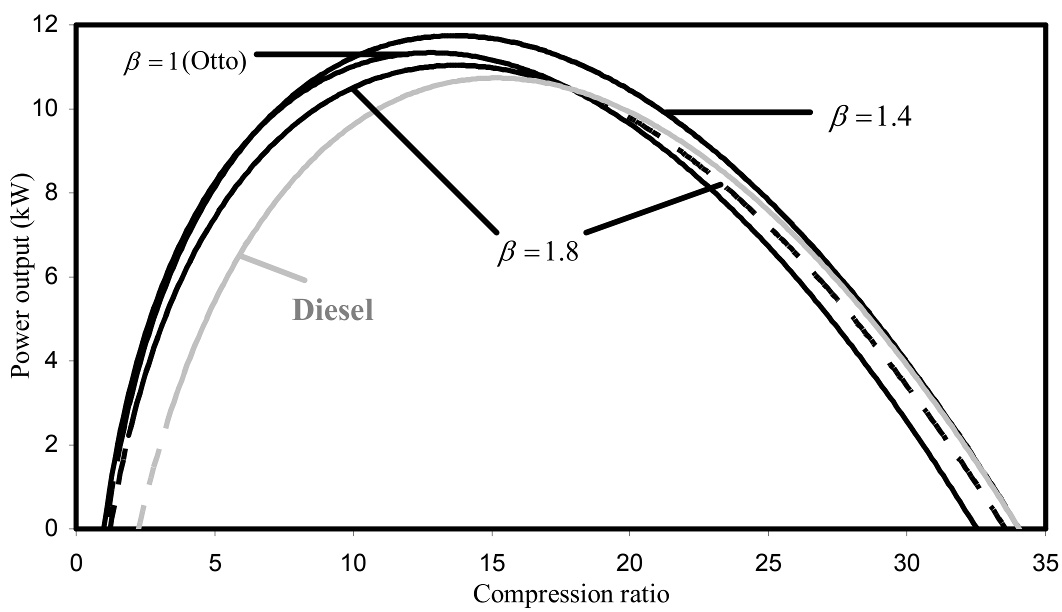

Fig. 2. Effect of β on the $P_{out} - r_c$ characteristic

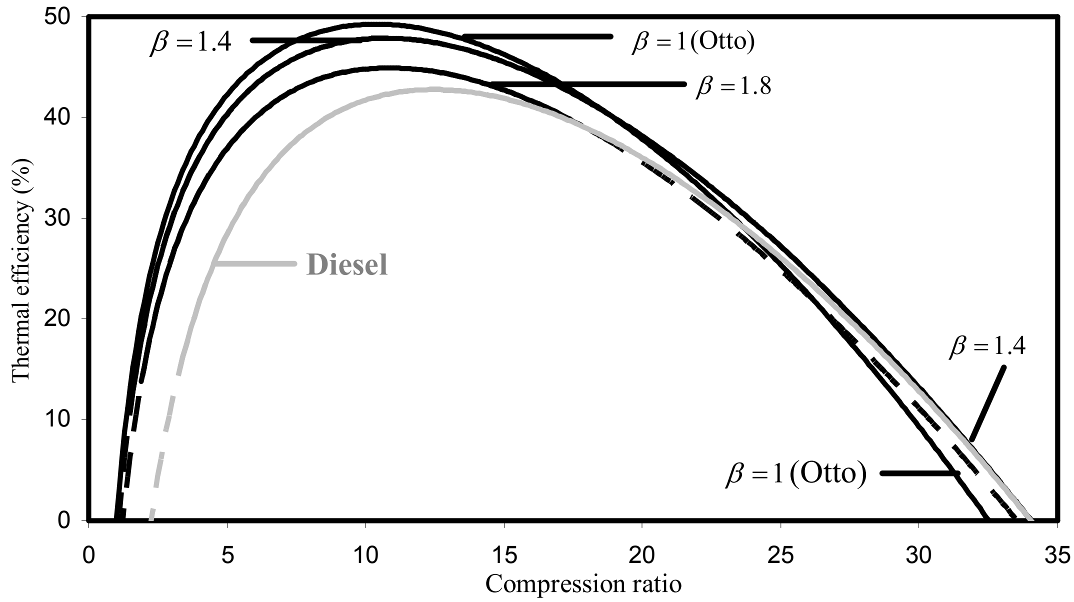

Fig. 3. Effect of β on the $\eta_{th} - r_c$ characteristic

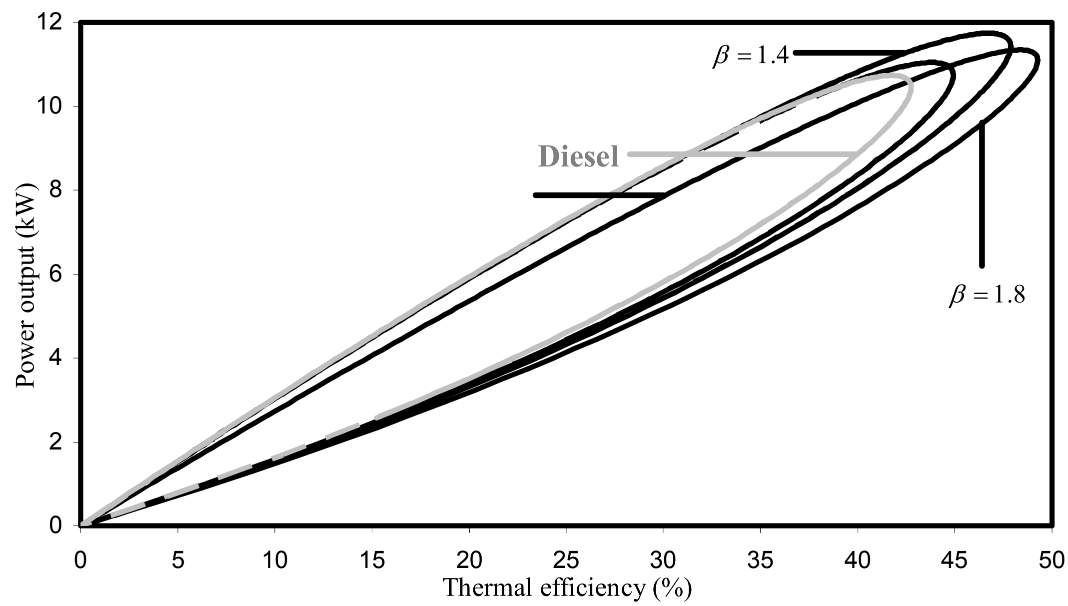

Fig. 4. Effect of β on the $\eta_{at} - r_c$ characteristic

4. Conclusion

In this paper, an air standard dual cycle model with considerations of heat transfer loss, friction loss computed according to the mean velocity of the piston and variable specific heat ratio is presented. The effects of cut-off ratio on the power output and the thermal efficiency were analyzed by detailed numerical examples. The results can also be applied to the performance analysis of the Diesel and Otto cycles. The results show that the effects of cut-off ratio on the cycle performance are obvious, and they should be considered in practical cycle analysis. The results can provide significant guidance for the performance evaluation and improvement of real Dual engines. It would be more meaningful if one considers experimental results. This will be a next work in the near future

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