Engine speed effects on the characteristic performance of Otto engines

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Abstract: The performance of an air-standard Otto cycle is analyzed using finite-time thermodynamics. In the model, the linear relation between the specific heat ratio 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. The relations between the power output and the compression ratio, between the power output and the thermal efficiency are derived by detailed numerical examples. The results shows that if compression ratio is less than certain value, the power output increases with increasing engine speed, while if compression ratio exceeds certain value, the power output first increases and then starts to decrease with increasing engine speed. With further increase in compression ratio, the increase of engine speed results in decreasing the power output. The results obtained in this paper may provide guidance for the design of practical internal-combustion engines. [Journal of American Science 2010;6(1):123-128]. (ISSN: 1545-1003).

Key words: Otto cycle; heat resistance; internal irreversibility; performance optimization

1. Introduction

Since finite-time thermodynamics (Andresen et al., 1984; Bejan, 1996; Aragon-Gonzalez et al., 2000) 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 (Aragon-Gonzalez et al., 2006; Chen et al., 2007; Aragon-Gonzalez et al., 2008; Ge et al., 2008a). Mozurkewich and Berry (1982) used mathematical techniques, developed for optimal-control theory, to reveal the optimal motions of the pistons in Diesel and Otto cycle engines, respectively. Left (1987) calculated the maximum work output and efficiency of an Otto heat engine. Wu and Blank (1993) also optimized the endoreversible Otto cycle with respect to both net power output and mean effective pressure. Angulo-Brown et al. (1994) modeled the behaviors of Otto with friction loss during finite times. Chen et al. (1998) derived the relations between the net work-output and the efficiency for Otto cycle with due consideration of heat-transfer losses. 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. Wang et al. (2002) optimized the power output of Diesel and Otto engines with friction loss during finite times. Rostovtsev et al. (2003) considered how to improve the efficiency of an ideal Otto heat engine. Chen et al. (2003) derived the characteristics of power output and thermal efficiency for Otto cycle with heat transfers and friction like term losses. Ge et al. (2005a, 2005b) studied the effects of variable specific heats of the working fluid on the performances of an Otto cycle with heat transfer loss and with heat transfer and friction losses, respectively. Chen et al. (2006) investigated the performance of an Otto heat engine by considering the irreversibility resulting from the compression and expansion processes, finite-time processes and heat loss through the cylinder wall. Ozsoysal (2006) gave the valid ranges of the heat transfer loss parameters of the Otto and diesel cycles with consideration of the heat loss as a percentage of the fuel's energy. Hou (2007) compared the performances of air standard Atkinson and Otto cycles with heat transfer loss considerations. Ge et al. (2008a; 2008b) analyzed the performance of an air standard Otto and Diesel 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. Lin and Hu (2008) analyzed the effects of heat loss by a percentage of the fuel's energy, friction and variable specific heats of working fluid on the performance of an air standard Otto cycle with a restriction of maximum cycle temperature. Gumus (2009) studied the performance analysis for an Otto cycle based on alternative performance criteria namely maximum power, maximum power density and maximum efficient power.

As can be seen in the relevant literature, the investigation of the effect of engine speed on performance of Otto 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 Otto cycle.

2. Thermodynamic analysis

The temperature-entropy diagram of an irreversible Otto heat engine is shown in Fig. 1, where T_1, T_{2s}, T_2, T_3, T_4 and T_{4s} are the temperatures of the working substance in state points 1, 2s, 2, 3, 4 and 4s. Process $1\rightarrow 2s$ is a reversible adiabatic compression, while process $1\rightarrow 2$ is an irreversible adiabatic process that takes into account the internal irreversibility in the real compression process. The heat addition is an isochoric process $2\rightarrow 3$. Process $3\rightarrow 4s$ is a reversible adiabatic expansion, while $3\rightarrow 4$ is an irreversible adiabatic process that takes into account the internal irreversibility in the real expansion process. The heat-removing process is the reversible constant volume $4\rightarrow 1$.

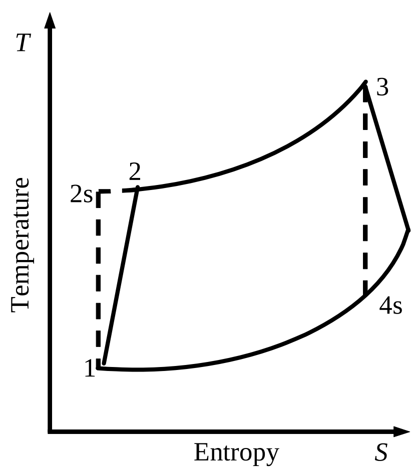

Figure 1. temperature-entropy diagram for the air standard Otto cycle

In a real cycle, the specific heat ratio is generally modeled as the first order equation of mean charge temperature (Gatowski, et al., 1984; Brunt, et al., 1998; Ebrahimi, 2006). Thus, it can be supposed that the specific heat ratio of the working fluid is function of temperature alone and has the first order equation forms:

$$\gamma = \gamma_o - k_1 T \tag{1}$$

where γ is the specific heat ratio and T is the absolute temperature. γ_o and k_1 are constants.

The heat added per second in the isobaric ($2\rightarrow 3$) heat addition process may be written as:

$$Q_{in} = M_n \int_{T_2}^{T_3} c_v dT = M_n \int_{T_2}^{T_3} \left(\frac{R}{\gamma_o - k_1 T - 1} \right) dT = \frac{M_n R}{k_1} \ln\left(\frac{\gamma_o - k_1 T_2 - 1}{\gamma_o - k_1 T_4 - 1} \right) \tag{2}$$

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

The heat rejected per second in the isochoric heat rejection process ($4\rightarrow 1$) may be written as:

$$Q_{out} = M_n \int_{T_1}^{T_4} c_v dT = M_n \int_{T_1}^{T_4} \left(\frac{R_{air}}{\gamma_o - k_1 T - 1} \right) dT = \frac{M_n R}{k_1} \ln\left(\frac{\gamma_o - k_1 T_1 - 1}{\gamma_o - k_1 T_4 - 1} \right) \tag{3}$$

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

$$\eta_c = (T_{2s} - T_1)/(T_2 - T_1) \tag{4}$$

and

$$\eta_e = (T_4 - T_3)/(T_{4s} - T_3) \tag{5}$$

These two efficiencies can be used to describe the internal irreversibility of the processes.

Since c_p and c_v are dependent on temperature, the adiabatic exponent γ will vary with temperature as well. Therefore, the equation often used in a reversible adiabatic process with constant γ cannot be used in a reversible adiabatic process with variable γ. However, according to Refs. (Ge et al. 2007; Chen et al., 2008), the equation for a reversible adiabatic process with variable γ can be written as follows:

$$TV^{\gamma-1} = (T + dT)(V + dV)^{\gamma-1} \tag{6}$$

From Eq. (6), one gets

$$T_i (\gamma_o - k_1 T_j - 1) = T_j (\gamma_o - k_1 T_i - 1)(V_j / V_i)^{\gamma_o - 1} \tag{7}$$

The compression ratio, r_c, is defined as:

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

Therefore, the equations for processes ($1\rightarrow 2s$) and ($3\rightarrow 4s$) are shown, respectively, by the following:

$$T_1 (\gamma_o - k_1 T_{2s} - 1)(r_c)^{\gamma_o - 1} = T_{2s} (\gamma_o - k_1 T_1 - 1) \tag{9}$$

and

$$T_3\left(\gamma_o - k_1 T_{4s} - 1\right) = T_{4s}\left(\gamma_o - k_1 T_3 - 1\right)\left(r_c\right)^{\gamma_o - 1} \tag{10}$$

The energy transferred to the working fluid during combustion is given by the following linear relation (Chen et al., 2008; Ebrahimi, 2009b)

$$Q_{in} = M_n\left[A - B\left(T_2 + T_4\right)\right] \tag{11}$$

where A and B are two constants related to combustion and heat transfer which are function of engine speed. From equation (10), it can be seen that Q_{in} contained two parts: the first part is $M_n A$, the released heat by combustion per second, and the second part is the heat leak loss per second, $M_n B(T_2 + T_4)$.

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)

$$f_\mu = -\mu v = -\mu \frac{dx}{dt} \tag{12}$$

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 power due to friction is

$$P_\mu = \frac{dW_\mu}{dt} = -\mu\left(\frac{dx}{dt}\right)^2 = -\mu v^2 \tag{13}$$

Running at N cycles per second, the mean velocity of the piston is

$$\bar{v} = 4LN \tag{14}$$

where L is the total distance the piston travels per cycle.

Thus, the power output of the Otto cycle engine can be written as

$$P_{out} = Q_{in} - Q_{out} - P_\mu = \frac{M_n R}{k_1}\left[\ln\left(\frac{\gamma_o - k_1 T_2 - 1}{\gamma_o - k_1 T_3 - 1}\right) - \ln\left(\frac{\gamma_o - k_1 T_1 - 1}{\gamma_o - k_1 T_4 - 1}\right)\right] - 16\mu\left(LN\right)^2 \tag{15}$$

The efficiency of the Otto cycle engine is expressed by

$$\eta_{th} = \frac{Q_{in} - Q_{out} - P_\mu}{Q_{in}} = \frac{M_n R\left[\ln\left(\frac{\gamma_o - k_1 T_2 - 1}{\gamma_o - k_1 T_3 - 1}\right) - \ln\left(\frac{\gamma_o - k_1 T_1 - 1}{\gamma_o - k_1 T_4 - 1}\right)\right] - 16 k_1 \mu\left(LN\right)^2}{M_n R \ln\left(\frac{\gamma_o - k_1 T_2 - 1}{\gamma_o - k_1 T_3 - 1}\right)} \tag{16}$$

When r_c, η_c, η_e and T_1 are given, T_{2s} can be obtained from Eq. (9), then, substituting T_{2s} into Eq. (4) yields T_2. T_3 can be deduced by substituting Eq. (2) into Eq. (11). T_{4s} can be found from Eq. (10), and T_4 can be deduced by substituting T_{4s} into Eq. (5). Substituting T_1, T_2, T_3 and T_4 into Eqs. (15) and (16), respectively, the power output and thermal efficiency of the Otto 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 parameters have been used in this exercise: $M_n = 1.57E-5 \times N\ kmols^{-1}$, $\eta_c = 0.97$, $\eta_e = 0.97$, $k_1 = 7.18 \times 10^{-5}\ K^{-1}$, $\gamma_o = 1.41$, $A = 60000\ J.mol^{-1}$, $L = 70\ mm$, $B = 28\ J.mol^{-1}K^{-1}$, $b = -9.7617 \times 10^{-5}\ K^{-1}$, $N = 2000 - 6000\ rpm$, $r_c = 1 - 70$, $\mu = 12.9\ Nsm^{-1}$ and $T_1 = 300\ K$ (Heywood, 1988; Chen et al. 2007; Ghatak and Chakraborty, 2007; Ge et al., 2008b; Ebrahimi, 2009a). 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 engine speed can be plotted. Numerical examples are shown as follows.

Figures 2 and 3 show the effects of the variable engine speed on the cycle performance with heat resistance, internal irreversibility and friction losses. From these figures, it can be found that the engine speed plays important roles on the power output. It is clearly seen that the effect of engine speed on the power output is related to compression ratio. They reflect the performance characteristics of a real irreversible Otto cycle engine. It should be noted that the heat added and the heat rejected by the working fluid increase with increasing engine speed (see Eqs. (2) and (3)).

Figure 3 indicates the effects of the engine speed on the power output of the cycle for different values of the compression ratio. It can be seen that the power output versus compression ratio characteristic is approximately parabolic like curves. In other word, the power output increases with increasing compression ratio, reach their maximum values and then decreases with further increase in compression ratio. The maximum power output increases with increasing engine speed up to about 5000 rpm where it reaches its peak value then starts to decline as the engine speed increases. This is consistent with the experimental results in the internal combustion engine (Mercier, 2006).

The optimal compression ratio corresponding to maximum power output point remains constant with increase of engine speed. The working range of the

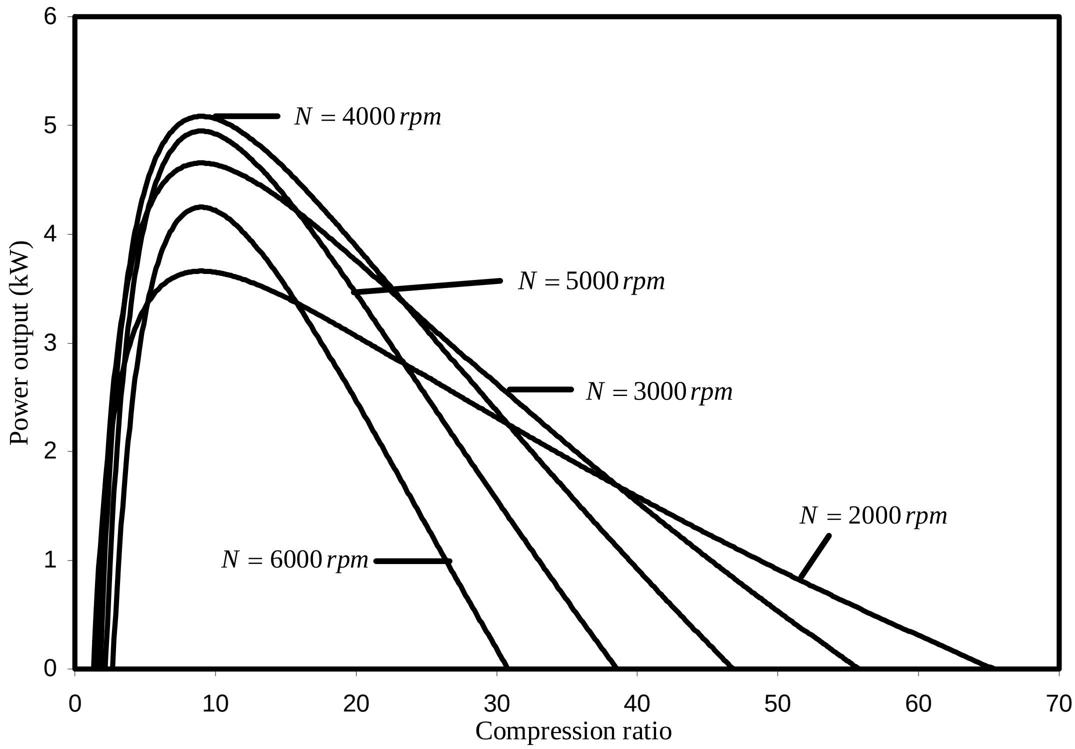

Figure 1. Effect of N on the $P_{out} - r_c$ characteristic

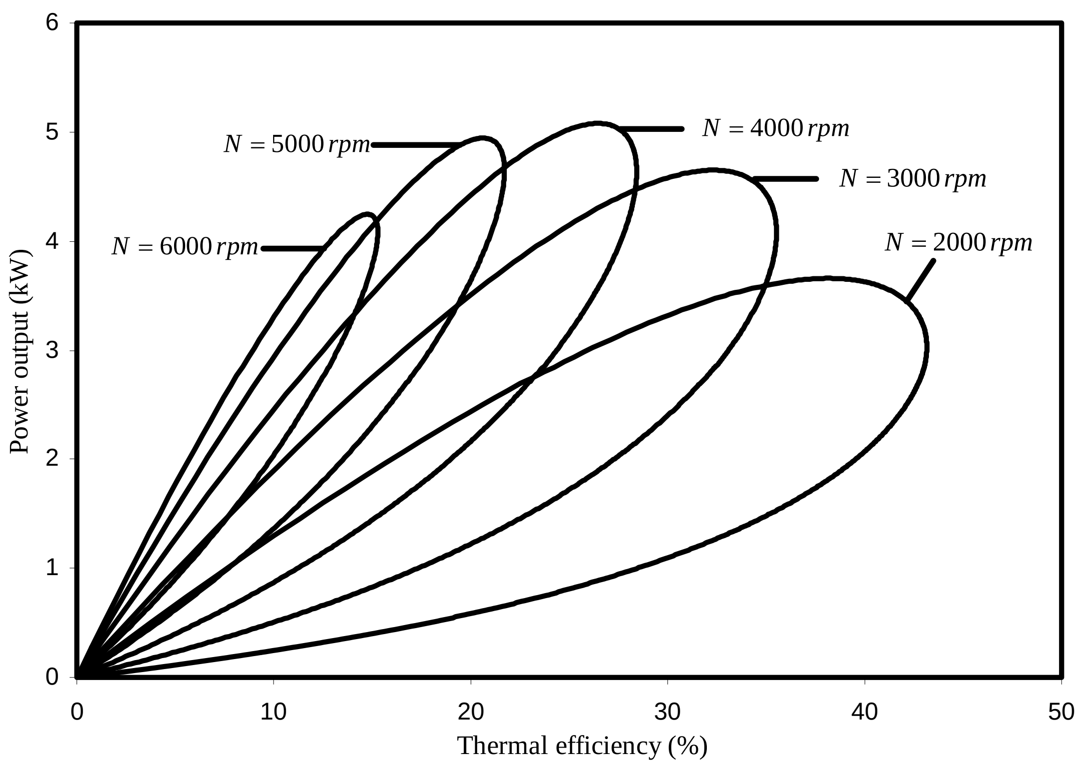

Figure 2. Effect of N on the $P_{out} - \eta_{th}$ characteristic

cycle decreases as the engine speed increases. The results shows that if compression ratio is less than certain value, the power output increases with increasing engine speed, while if compression ratio exceeds certain value, the power output first increases and then starts to decrease with increasing engine speed. With further increase in compression ratio, the increase of engine speed results in decreasing the power output. Numerical calculation shows that for any same compression ratio, the smallest power output is for $N = 6000\,rpm$ when $r_c \le 5.2$ or $r_c > 15.7$ and is for $N = 2000\,rpm$ when $5.2 < r_c \le 15.7$ and also the largest power output is for $N = 2000\,rpm$ when $r_c \le 2.2$ or $r_c > 38.6$, is for $N = 3000\,rpm$ when $2.2 < r_c \le 3.6$ or $23.4 \le r_c \le 38.6$ and is for $N = 6000\,rpm$ when $3.6 \le r_c < 23.4$.

The influence of the engine speed on the power output versus thermal efficiency is displayed in figure 4. As can be seen from this figure, the power output versus thermal efficiency is loop shaped one. It can be seen that the power output at maximum thermal efficiency improves with increasing engine speed from 2000 to around $N = 4000\,rpm$. With further increase in engine speed, the power output at maximum thermal efficiency decreases. It can also be seen that the thermal efficiency at maximum power decreases with increase of engine speed from 2000 to $N = 6000\,rpm$.

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

4. Conclusion

An air standard Otto cycle model, assuming a temperature dependent specific heat ratio of the working fluid, and heat resistance and frictional irreversible losses, has been investigated numerically. The performance characteristics of the cycle with varying engine speeds and compression ratios were obtained by numerical examples. The results show that if compression ratio is less than certain value, the power output increases with increasing engine speed, while if compression ratio exceeds certain value, the power output first increases and then starts to decrease with increasing engine speed. With further increase in compression ratio, the increase of engine speed results in decreasing the power output. The results also show that the maximum power output increase with increasing engine speed. With further increase in engine speed, the increase of engine speed results in decreasing the maximum power output. The analysis helps us to understand the strong effect of engine speed on the performance of the Otto cycle. Therefore, the results are of great significance to provide good guidance for the performance evaluation and improvement of real Otto engines.

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