Thermodynamic modeling of performance of an irreversible Diesel cycle with engine speed

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Abstract: Finite-time thermodynamic analysis of an air-standard internal-combustion Diesel cycle is performed in this paper. The relations between the power output and the compression ratio, between the power output and the thermal efficiency are derived. The maximum net power output and the corresponding efficiency limit of the cycle with friction losses are also found. Detailed numerical examples are given. This paper provides an additional criterion for use in the evaluation of the performance and the suitability of a Diesel engine. [Nature and Science. 2009;7(9):78-82]. (ISSN: 1545-0740).

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

1. Introduction

Today, practical engineers make use of the air-standard Diesel cycle to provide a short description of the Diesel engine. Conventional analysis of the air-standard Diesel cycle is performed using reversible thermodynamics. The development of finite-time thermodynamics (Curzon and Ahlborn, 1975; Aragon-Gonzalez et al., 2000; Chen et al., 2008), a new discipline of modern thermodynamics, provides a new and powerful tool for the performance analysis of practical engineering cycles. Thus, 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., 2006; Ge et al., 2008a). Chen et al. (1982) and Aizenbud and Band (1993) determined the optimal motion of a piston fitted to a cylinder. The cylinder contained a gas pumped with a given heating rate and coupled to a heat bath during finite times. 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. Chen et al. (1996) derived the relations between the power output and the thermal efficiency for the Diesel cycle with the consideration of the heat-transfer losses. Bhattacharyya (2000) proposed a simplified irreversible model for an air-standard Diesel cycle by using the finite time thermodynamic approach. In his study, global thermal and friction losses are lumped into an equivalent friction term, which is linear in the piston velocity. Rocha-Martinez et al. (2002) investigated the effect of variable specific-heats on the Otto and Diesel cycle performance. Chen et al. (2002) modeled the behaviors of Diesel cycle, with friction losses, over a finite period. Chen et al. (2003, 2004) determined the characteristics of power and efficiency for Otto and Dual cycles with heat transfer and friction losses. Fischer and Hoffman (2004) concluded that a quantitative simulation of an Otto-engine's behavior can be accurately achieved by a simple Novikov model with heat leaks. Hou (2004) derived the performance characteristic of Dual cycle with only heat transfer loss and studied the effects of heat transfer loss on the performance of the cycle. 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. 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. Parlak et al. (2006) defined the internal irreversibility by using entropy production, and analyzed the effect of the internal irreversibility on the performance of irreversible Dual cycle. Al-Sarkhi et al. (2006) studied the effects of variable specific heats of the working fluid on the performances of the Diesel and Miller cycles. Ge et al. (2007) studied the effects of variable specific heats of the working fluid on the performances of the Diesel cycle. Zhao and Chen (2007) performed analysis and parametric optimum criteria of an irreversible Atkinson heat engine using finite time thermodynamics. Zhao and Chen (2007) also defined the internal irreversibility by using compression and analyzed the performance of

Dual cycle when the maximum temperature of the cycle is fixed and the efficiency has a new definition. 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 diesel cycle.

As can be seen in the relevant literature, the investigation of the effect of engine speed 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. Irreversible Diesel cycle

The Diesel cycle shown in figure 1 approximates the compression stroke up to ignition with the adiabatic reversible (isentropic) process $1 \rightarrow 2$; it assumes that the combustion process is represented by the reversible constant pressure process $2 \rightarrow 3$; it approximates the power stroke with the isentropic expansion process $3 \rightarrow 4$; and it assumes that the heat-removing process is the reversible constant volume process $4 \rightarrow 1$.

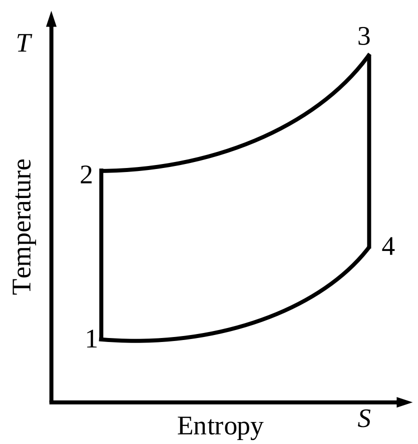

Figure 1. $T-S$ diagram of a Diesel cycle

The combustion heat input is Q_{23} and the heat rejected by the working fluid is Q_{41}. Thus, assuming that the heat engine is operated at the rate of N cycles per second, the reversible power output per second is:

$$P_{rev} = N\left(Q_{23} - Q_{41}\right) = NM\left[c_p\left(T_3 - T_2\right) - c_v\left(T_4 - T_1\right)\right] \quad (1)$$

where c_v is the constant volume specific heat, c_p is the constant pressure specific heat, M is the molar number of the working fluid and T is the absolute temperature.

The compression ratio, r_c, is defined as:

$$r_c = V_1/V_2 \quad (2)$$

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

$$T_2 = T_1 r_c^{\gamma-1} \quad (3)$$

and

$$T_4 = T_1^{\gamma-1} T_3^{\gamma} r_c^{(1-\gamma)\gamma} \quad (4)$$

Where γ is the ratio of specific heats, $\gamma = c_p/c_v$. Thus, Equation (1) becomes:

$$P_{rev} = NM \frac{R}{\gamma-1}\left[\gamma\left(T_3 - T_1 r_c^{\gamma-1}\right) - T_1^{\gamma-1} T_3^{\gamma} r_c^{(1-\gamma)\gamma} + T_1\right] \quad (5)$$

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

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).

$$f_{\mu} = -\mu v = -\mu \frac{dx}{dt} \quad (6)$$

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

$$P_{los} = \frac{dW_{los}}{dt} = -\mu\left(\frac{dx}{dt}\right)^2 = -\mu\left(S_p\right)^2 \quad (7)$$

Thus, the lost power is

$$P_{los} = -\mu\left(\bar{S}_p\right)^2 \quad (8)$$

where $\bar{S}_p$ is the mean velocity of the piston.

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

$$\bar{S}_p = 4LN \quad (9)$$

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

The resulting power output (P) the Diesel cycle engine can be written as:

$$P = P_{rev} - P_{los} = NM \frac{R}{\gamma-1}\left[\gamma\left(T_3 - T_1 r_c^{\gamma-1}\right) - T_1^{\gamma-1} T_3^{\gamma} r_c^{(1-\gamma)\gamma} + T_1\right] - 16\mu\left(LN\right)^2 \quad (10)$$

The efficiency of the Diesel cycle engine is expressed by

$$\eta_{th} = P/Q_{in} \quad (11)$$

Where

$$Q_{in} = NMc_p\left(T_3 - T_2\right) \quad (12)$$

Hence, the irreversible cycle efficiency is

$$\eta_{th} = \frac{MR\left[\gamma\left(T_3 - T_1 r_c^{\gamma-1}\right) - T_1^{\gamma-1} T_3^{\gamma} r_c^{(1-\gamma)\gamma} + T_1\right] - 16\mu N\left(\gamma-1\right)L^2}{MR\gamma\left(T_3 - T_2\right)} \quad (13)$$

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

When r_c , T_1 and T_3 are given, the power output and thermal efficiency of the Diesel cycle engine can be obtained from Eqs. (12) and (13), respectively. 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: $T_3 = 2200\,K$, $T_1 = 360\,K$, $L = 95\,mm$, $\mu = 12.9\,Nsm^{-1}$, $N = 3000 \rightarrow 7000\,rpm$, $r_c = 1 \rightarrow 70$, $\gamma = 1.4$ and $M = 1.57\times10^{-5}\,kmol$ (Chen et al. 2006; Ghatak and Chakraborty, 2007; Ge et al., 2009; Ebrahimi, 2009b). 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 and 3 show the effects of the variable engine speed on the cycle performance with heat resistance, internal irreversibility and friction losses (the dashed lines in the figures denote where the cycle cannot work normally). 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 Diesel cycle engine.

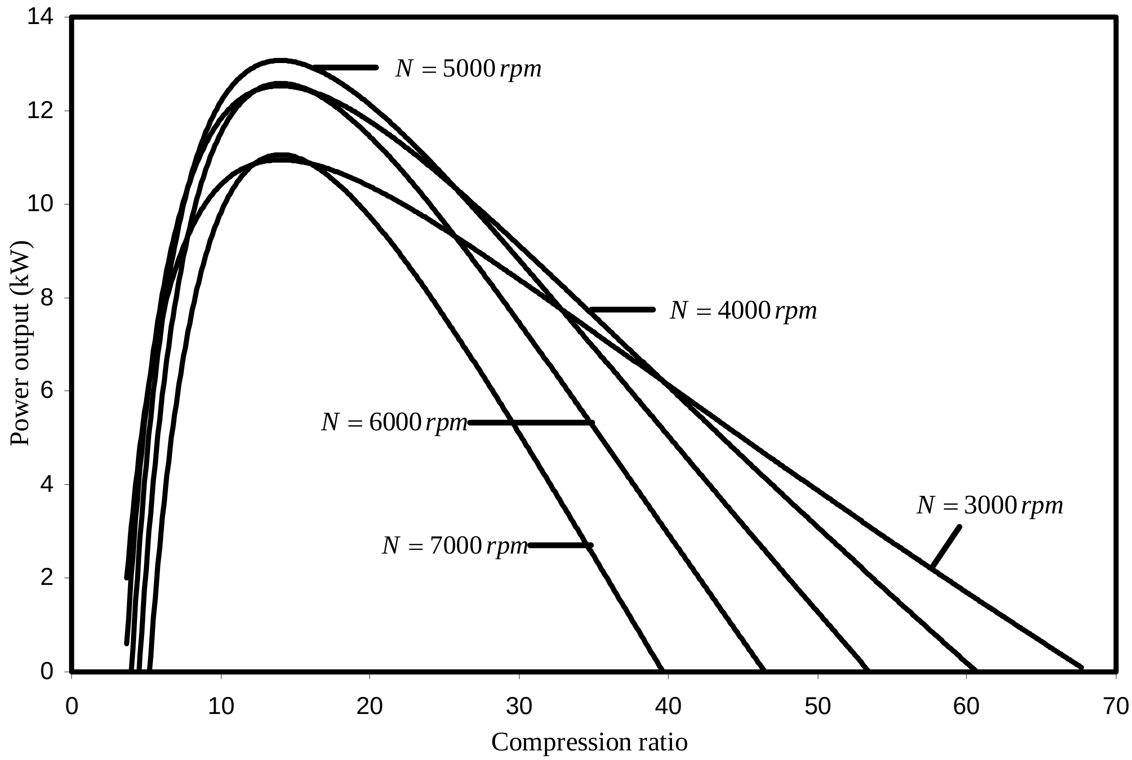

Figure 2. Effect of engine speed on the variation of the power with compression ratio

Figure 2 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 $N = 5000\,rpm$ 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

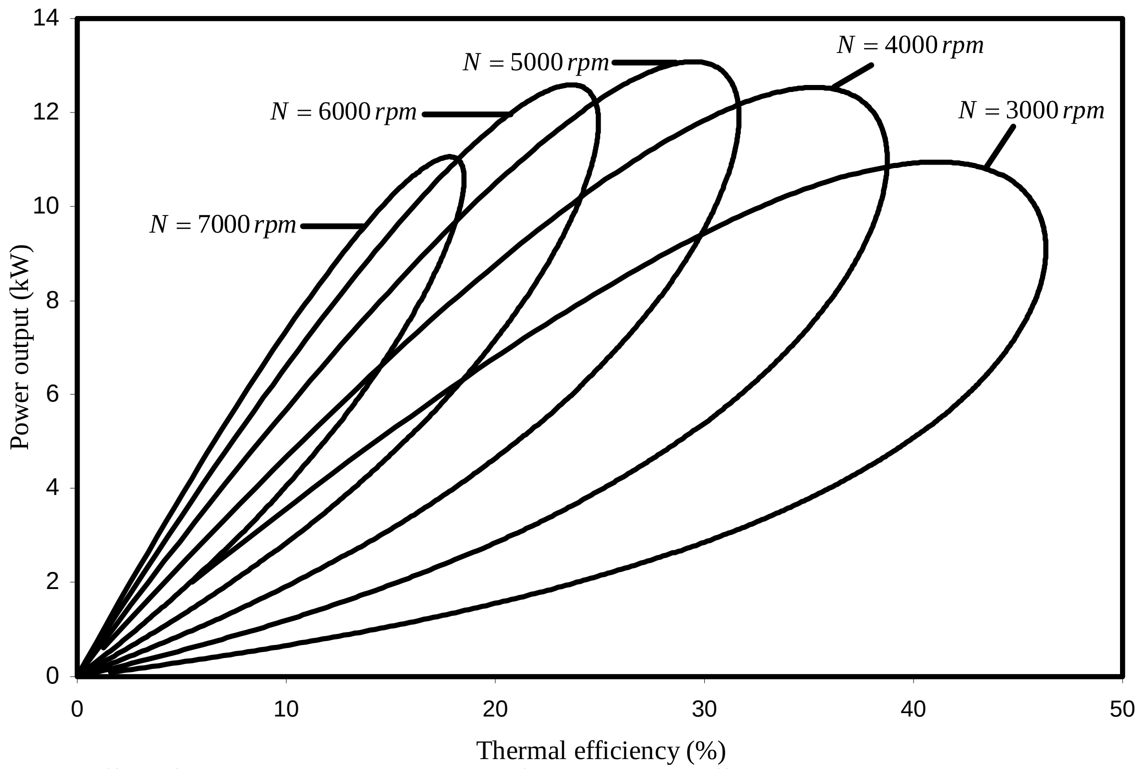

Figure 3. Effect of engine speed on the variation of the power with efficiency

increase of engine speed. 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 $N = 5000\,rpm$ when $r_c \le 12.3$ or $r_c > 16$ and is for $N = 3000\,rpm$ when $12.3 < r_c \le 16$ and also the largest power output is for $N = 3000\,rpm$ when $r_c \le 5.2$ or $r_c > 39.5$, is for $N = 4000\,rpm$ when $5.2 < r_c \le 7.8$ or $25.8 \le r_c \le 39.5$ and is for $N = 5000\,rpm$ when $7.8 < r_c < 25.8$.

The influence of the engine speed on the power output versus thermal efficiency is displayed in figure 3. 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 3000 to around 5000 rpm. With further increase in mean 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 3000 to 7000 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

The effect of finite rate process and friction on the irreversible Diesel cycle is determined in this paper. The relations between net power output, efficiency, compression ratio, and the engine speed are derived. The effects of engine speed on the power output and the efficiency were analyzed by detailed numerical examples. Our above analysis provides a new theoretical basis for the performance evaluation, improvement, and optimization of practical Diesel engines.

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