EFFECTS OF ENGINE SPEED ON PERFORMANCE OF AN IRREVERSIBLE DUAL CYCLE

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Abstract - This study is aimed at investigating the effect of engine speed on the Dual cycle performance with considerations of heat transfer 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 show that the maximum power output and the power output at maximum thermal efficiency increase with increasing engine speed and then decrease. The results also 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. This paper provides an additional criterion for use in the evaluation of the performance and the suitability of a dual engine.

Index terms - Finite time thermodynamics; Performance analysis; Dual cycle

I. INTRODUCTION

Since finite-time thermodynamics [1-3] 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 [4–7]. Ghatak and Chakrabort [8] analyzed the effects of the variable specific heat of working fluid and heat transfer loss on the performance of Dual cycle. Sahin et al. [9] optimized the performance of a new combined power cycle based on power density analysis of the dual cycle. Ust [10] made a comparative performance analysis and optimization of irreversible Atkinson cycle under maximum power density and maximum power conditions. Chen et al. [11] derived the characteristics of power output and thermal efficiency for Dual cycle with heat transfers and friction like term losses. Blank and Wu [12] and Lin et al. [13] considered the effect of heat transfer through a cylinder wall on the work output of the dual cycle. Chen et al. [14] investigated the effects of cylinder wall heat transfer and global losses lumped in a friction like term on the performance of a dual cycle during a finite time. Zhao and Chen [15] derived and optimized the power output and efficiency of the cycle. The results obtained are general, so that the optimal performance of irreversible Otto and Diesel cycles are included in two special cases of the Dual cycle and may be directly derived from that of the Dual heat engine. Wang et al. [16] modeled Dual cycle with friction like loss during a finite time. Lin et al. [17] derived the relations between net power and the efficiency of the Dual cycle with due consideration of the heat transfer loss. Ust et al. [18] performed an ecological performance analysis for an irreversible Dual cycle based on an ecological coefficient of performance criterion. Wang et al. [19] proposed a simplified irreversible model for an air-standard Dual cycle by using the finite time thermodynamic approach. Ge et al. [20, 21] 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. All of the above mentioned research, the specific heats at constant pressure and volume of working fluid are assumed to be constants or functions of temperature alone and have the linear and or the non-linear forms. But when calculating the chemical heat released in combustion at each instant of time for internal combustion engine, the specific heat ratio is generally modeled as a linear function of mean charge temperature [22, 23]. The model has been used and the phenomena that it takes into account are well known [24]. However, since the specific heat ratio has a great influence on the heat release peak and on the shape of the heat release curve [25], some researchers have elaborated different mathematical equations to describe the dependence of specific heat ratio from temperature [22, 24-29]. It should be mentioned here that the most important thermodynamic property used in the heat release calculations for engines is the specific heat ratio [29]. Also, the investigation of the effect of engine speed on performance of Dual cycle does not appear to have been published. Therefore, the objective of this study is to examine the effect engine speed on the power output and the thermal efficiency of air standard Dual cycle.

II. THERMODYNAMIC ANALYSIS

An air standard Dual cycle model is shown in Fig. 1. The compression $(1 \rightarrow 2)$ 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.

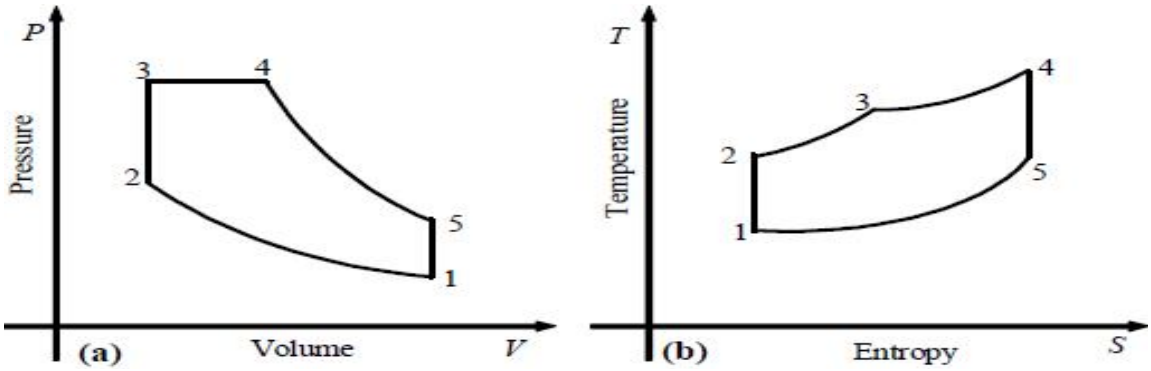

Figure 1. (a) $P-V$ **diagram; (b)** $T-S$ **diagram for the air standard Dual cycle**

As already mentioned in the previous section, it can be supposed that the specific heat ratio of the working fluid is function of temperature alone and has the linear forms:

$$\gamma = \gamma_\circ - k_1 T \quad (1)$$

where γ is the specific heat ratio and T is the absolute temperature. $\gamma_\circ$ and k_1 are constants.

Assuming that the heat engine is operated at the rate of N cycles per second, the heat added to the working fluid, during processes ($2 \to 3$) and ($3 \to 4$) is

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

$$NM\int_{T_2}^{T_3}\left(\frac{R}{\gamma_\circ - k_1 T - 1}\right)dT + NM\int_{T_3}^{T_4}\left(\frac{(\gamma_\circ - k_1 T)R}{\gamma_\circ - k_1 T - 1}\right)dT = \quad (2)$$

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

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 ($5 \to 1$) is

$$Q_{out} = NM\int_{T_1}^{T_5} c_v dT =$$

$$M\int_{T_1}^{T_5}\left(\frac{R}{\gamma_\circ - k_1 T - 1}\right)dT = \frac{NMR}{k_1}\ln\left(\frac{\gamma_\circ - k_1 T_1 - 1}{\gamma_\circ - k_1 T_5 - 1}\right) \quad (3)$$

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

The equation often used in reversible adiabatic process, with constant γ, cannot be used in reversible adiabatic process with a variable γ. According to Ref. [30], however, 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:

$$TV^{\gamma-1} = (T + \mathrm{d}T)(V + \mathrm{d}V)^{\gamma-1} \quad (4)$$

From Eq. (6), one can get the following equation

$$T_i(\gamma_\circ - k_1 T_j - 1) = T_j(\gamma_\circ - k_1 T_i - 1)(V_j / V_i)^{\gamma_\circ - 1} \quad (5)$$

The compression ratio, r_c, and pressure ratio, α, are defined as

$$r_c = V_1 / V_2 \quad (6)$$

and

$$\alpha = T_3 / T_2 \quad (7)$$

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

$$T_1(\gamma_\circ - k_1 T_2 - 1)(r_c)^{\gamma_\circ - 1} = T_2(\gamma_\circ - k_1 T_1 - 1) \quad (8)$$

$$T_4(\gamma_\circ - k_1 T_5 - 1) = T_5(\gamma_\circ - k_1 T_4 - 1)\left(\frac{T_3}{T_4} r_c\right)^{\gamma_\circ - 1} \quad (9)$$

For an ideal Dual cycle model, there are no losses. However, for a real internal combustion engine cycle, the heat transfer irreversibility between the working fluid, the cylinder wall and friction like term loss are not negligible. 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. The energy transferred to the working fluid during combustion is given by the following linear relation [11, 14].

$$Q_{in} = NM\left[A - B(T_2 + T_4)\right] \quad (10)$$

where A and B are two constants related to combustion and heat transfer which are functions of engine speed. From equation (10), it can be seen that Q_{in} contained two parts: the first part is NMA, the

released heat by combustion per second, and the second part is the heat leakage loss per second, $Q_{leak} = NMB(T_2 + T_4)$.

Taking into account the friction loss of the piston, as deduced by Chen et al. [11, 14] for the Dual cycle, and a dissipation term represented by a friction force which in a linear function of the velocity gives

$$f_\mu = -\mu v = -\mu \frac{\mathrm{d}x}{\mathrm{d}t} \tag{11}$$

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

$$P_\mu = \frac{dW_\mu}{dt} = -\mu\left(\frac{\mathrm{d}x}{\mathrm{d}t}\right)^2 = -\mu v^2 \tag{12}$$

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

$$\bar{v} = 4LN \tag{13}$$

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

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

$$P_{out} = \frac{W}{\tau} - P_\mu = \frac{NMR}{k_1}\ln\left(\frac{(\gamma_\circ - k_1T_2 - 1)(\gamma_\circ - k_1T_5 - 1)}{(\gamma_\circ - k_1T_4 - 1)(\gamma_\circ - k_1T_1 - 1)}\right) + NMR(T_4 - T_3) - 16\mu(LN)^2 \tag{14}$$

The thermal efficiency of the Dual cycle engine is expressed by

$$\eta_{th} = \frac{P_{out}}{Q_{in}} = \frac{\frac{NMR}{k_1}\ln\left(\frac{(\gamma_\circ - k_1T_2 - 1)(\gamma_\circ - k_1T_5 - 1)}{(\gamma_\circ - k_1T_4 - 1)(\gamma_\circ - k_1T_1 - 1)}\right) + NMR(T_4 - T_3) - 16\mu(LN)^2}{\frac{NMR}{k_1}\ln\left(\frac{\gamma_\circ - k_1T_2 - 1}{\gamma_\circ - k_1T_4 - 1}\right) + NMR(T_4 - T_3)} \tag{15}$$

When the values of r_c, α and T_1 are given, T_2 can be obtained from Eq. (8) and T_3 can be found from Eq. (7), then, substituting Eq. (2) into Eq. (10) yields T_4, and the last, T_5 can be worked out using Eq. (9). Substituting T_1, T_2, T_3, T_4 and T_5 into Eqs. (14) and (15), 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.

IV. RESULTS AND DISCUSSIONS

The following constants and parameter values have been used in this exercise: $T_1 = 350\,K$, $N = 1000 - 5000\,rpm$, $k_1 = 0.00006\,K^{-1}$, $\gamma_\circ = 1.41$, $L = 0.07\,m$, $\mu = 0.0129\,kNsm^{-1}$, $M = 1.57\times10^{-5}\,kmol$, $A = 60000\,J.mol^{-1}$ and $B = 25\,J.mol^{-1}K^{-1}$ [8-9, 22, 31].

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.

The variations in the temperatures T_2, T_3, T_4 and T_5 Fig. 2. It is found that T_2, T_3 and T_4 increase with the increase of compression ratio, and T_5 decreases with the increase of compression ratio. In Fig. 2, there are two special states: one is the state with $T_5 \geq T_4$, the another is the state with $T_3 \geq T_4$. In the two special states, the cycle cannot work. Figs. 7-8 show the influence of the parameter N on the Dual cycle performance. It can be seen that the engine speed plays significant role on the Dual cycle performance.The results also 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.

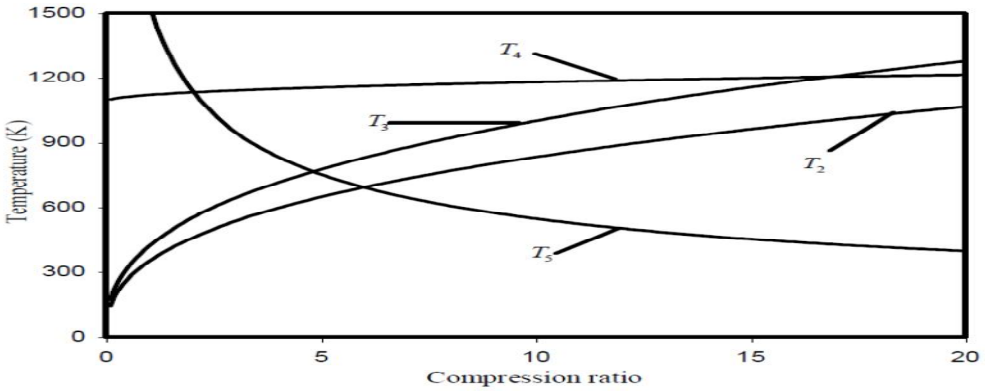

Figure 2. The temperature versus compression ratio for $N = 1800\,rpm$

The maximum power output increases with increasing engine speed up to about 4000 rpm where it reaches its peak value then starts to decline as the engine speed increases. The optimal compression ratio corresponding to maximum power output point remains constant with increase of engine speed. Also, the working range of the cycle remains constant with increasing engine speed up to about 3000 rpm then starts to decline as the speed increases. It can also be concluded that, for example, at the compression ratio of 11, the power output increases with increase of engine speed and then decreases. This is consistent with the experimental results in the internal combustion engine [32]. It can be seen that the power output at maximum thermal efficiency improves with increasing engine speed from 1000 to around 4000 rpm. With further increase in engine speed, the power output at maximum thermal efficiency decreases. But, the thermal efficiency at maximum power decreases with increase of engine speed from 1000 to 5000 rpm.

CONCLUSION

An air-standard Dual-cycle model, assuming a temperature-dependent specific heat ratio of the working fluid, and with heat resistance and frictional irreversible losses, has been investigated numerically. The performance characteristics of the cycle were obtained by numerical examples. The general conclusions drawn from the results of this work are as follows:

(1) The results also 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.

(2) The maximum power output and the power output at maximum thermal efficiency increase with increase of engine speed and then decrease.

(3) The optimal compression ratio corresponding to maximum power output point remains constant with increase of engine speed.

The results can provide some significant guidelines for the performance evaluation and improvement of real Dual engines.

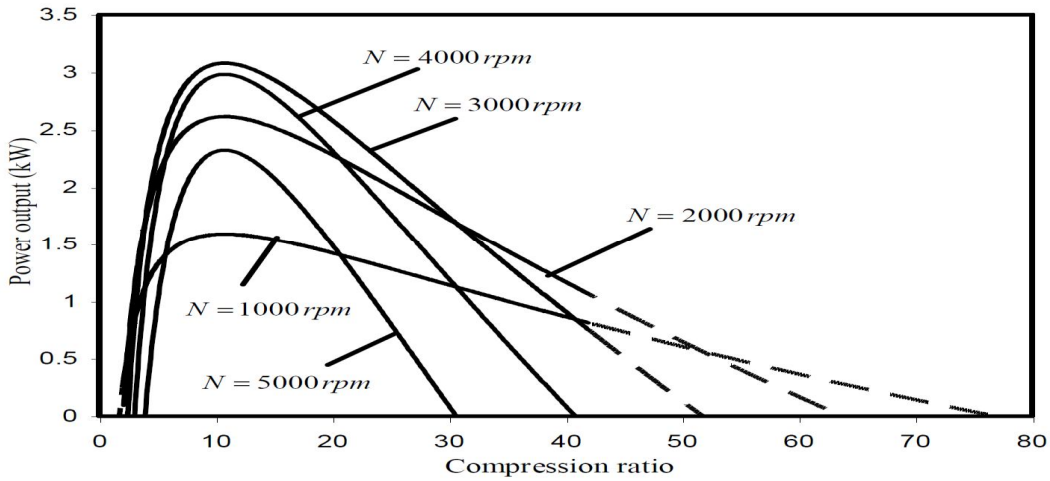

Figure 3. Effect of N on the $P_{out} - r_c$ characteristic

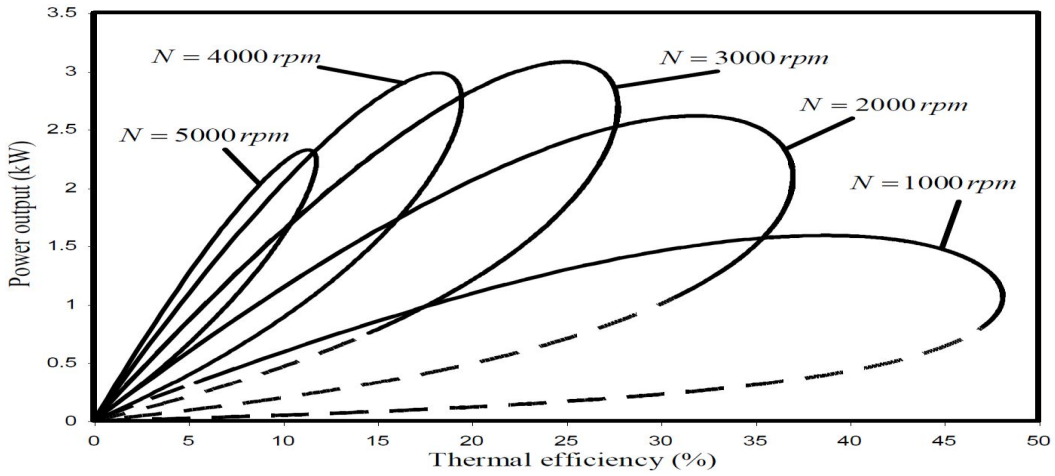

Figure4. Effect of N on the $P_{out} - \eta_{at}$ characteristic

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