Effects of Equivalence Ratio and Mean Piston Speed on Performance of an Irreversible Dual Cycle

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In the present study, the performance of an air standard dual cycle is analyzed using finite-time thermodynamics. 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 show that the maximum power output and the power output at the maximum efficiency point increase and then decrease as the equivalence ratio and/or the mean piston speed increases. The results also show that the optimal compression ratio corresponding to maximum power output point and the working range of the cycle remain constant as the mean engine speed is increased, but they increase and then decrease as the equivalence ratio is increased. It is noteworthy that the results obtained in the present study are of significance for providing guidance with respect to the performance evaluation of practical internal combustion engines.

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1. Introduction

In practice, combustion occurs neither at constant volume nor at constant pressure. This leads to the dual, limited pressure, or mixed cycle which has heat addition in two stages, firstly at constant volume, and secondly at constant pressure.

Significant achievements have ensued since finite-time thermodynamics was developed in order to analyze and optimize the performances of real heat engines [1–6].

Blank and Wu [7] carried out the effect of combustion on the work or power-optimized dual cycle. They derived the maximum work or power and the corresponding efficiency bounds. The power versus efficiency relationship is a very important characteristic for heat engines.

Lin et al. [8] derived the relations between the net power and the efficiency for the dual cycle with due consideration of the heat transfer losses.

Sahin et al. [9] carried out a performance analysis and optimization based on the ecological criterion for an air standard dual cycle coupled to constant temperature thermal reservoirs. The optimal performances and design parameters such as compression ratio, pressure ratio, cut-off ratio and thermal conductance allocation ratio which maximize the ecological objective function are investigated.

Wang et al. [10] modeled dual cycle with friction like term loss during a finite time and studied the effect of friction like term loss on cycle performance.

Sahin et al. [11] studied the optimal power density characteristics for Atkinson, Miller and dual cycles without any such losses.

Parlak et al. [12, 13] optimized the performance of irreversible dual cycle, gave the experimental results, and compared the performance of dual and Diesel cycles under the maximum power output.

Hou [14] 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.

Ust et al. [15] 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.

Chen et al. [16] and Ghatak and Chakraborty [17] analyzed the effect of variable specific heats and heat transfer loss on the performance of the dual cycle when variable specific heats of working fluid are linear functions of its temperature.

Zhao and Chen [18] defined the internal irreversibility by using the compression and expansion efficiencies and analyzed the performances of dual cycle.

Ge and Chen [19, 20] 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 piston speed, the internal irreversibility described by using the compression and expansion efficiencies, and the heat transfer loss are considered.

Ebrahimi [21] studied the effects of cut-off ratio on performance of an irreversible dual cycle with considering

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the variable specific heat ratio of the working fluid.

As can be seen in the relevant literature, the effects of equivalence ratio and mean piston speed on performance of a dual cycle do not appear to have been investigated previously by others. Therefore, the objective of this study is to examine the effects of equivalence ratio and mean piston speed on performance of an irreversible dual cycle.

2. Thermodynamic analysis

The temperature–entropy (T–S) diagram of an irreversible dual heat engine is shown in Fig. 1, where T_1, T_{2s}, T_2, T_3, T_4, T_{4s} and T_5 are the temperatures of the working substance in state points 1, 2s, 2, 3, 4, 4s and 5. 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 additions are an isochoric process $2 \to 3$ and an isobaric process $3 \to 4$. The process $4 \to 5s$ is a reversible adiabatic expansion, while $4 \to 5$ 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 $5 \to 1$.

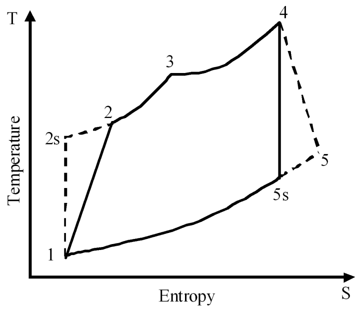

Fig. 1. T–S diagram of a dual cycle.

The combustion efficiency of a gasoline type fuel, such as octane, can be expressed in terms of excess air factor from measured data as [22]:

$$\eta_{com} = -1.44738 + 4.18581/\phi - 1.86876/\phi^2, \quad 0.83 < \phi < 1.3\,, \tag{1}$$

where ϕ is the equivalence ratio. The range of effective ϕ values spans normal spark ignition combustion, i.e., from about 0.83 to about 1.33 [22].

The total value of the heat that can be released from the combustion of the quantity $\dot{m}_f$ of fuel is Q_{fuel} and is controlled by the combustion efficiency of the fuel in question [22, 23]:

$$Q_{fuel} = \eta_{com}\dot{m}_f Q_{LHV}\,, \tag{2}$$

where Q_{LHV} is the lower calorific value of the fuel and $\dot{m}_f$ is the mass flow rate of the fuel.

Substituting Eq. (1) in Eq. (2) gives

$$Q_{fuel} = \left(-1.44738 + 4.18581/\phi - 1.86876/\phi^2\right) \times \dot{m}_f Q_{LHV}\,. \tag{3}$$

The heat loss through the cylinder wall is given in the following linear expression [18, 20]:

$$Q_{ht} = \dot{m}_t B(T_2 + T_4 - 2T_0), \tag{4}$$

where $\dot{m}_t$ is the mass flow rate of the air–fuel mixture, B is constant and T is the absolute temperature.

Since the total energy of the delivered fuel Q_{fuel} is assumed to be the sum of the heat added to the working fluid Q_{in} and the heat leakage Q_{ht},

$$Q_{in} = Q_{fuel} - Q_{ht} = \left(-1.44738 + 4.18581/\phi - 1.86876/\phi^2\right) \times \dot{m}_f Q_{LHV} - \dot{m}_t B(T_2 + T_4 - 2T_0). \tag{5}$$

The compression ratio, r_c, and the cut-off ratio, β, are defined as:

$$r_c = \frac{V_1}{V_2} \tag{6}$$

and

$$\beta = \frac{V_4}{V_3} = \frac{T_4}{T_3}\,. \tag{7}$$

For the processes $1 \to 2s$ and $4 \to 5s$, one finds that

$$T_{2s} = T_1 r_c^{\gamma-1} \tag{8}$$

and

$$T_{5s} = T_4\left(\frac{\beta}{r_c}\right)^{\gamma-1}, \tag{9}$$

where γ is the specific heat ratio.

Some internal irreversibilities caused by turbulence or viscous stresses, can be quantified with a parameter known as isentropic efficiency. To describe these internal irreversibilities of the processes $1 \to 2$ and $4 \to 5$, the compression and expansion efficiencies can be defined as [19, 20, 24–28]:

$$\eta_c = (T_{2s} - T_1)/(T_2 - T_1) \tag{10}$$

and

$$\eta_e = (T_5 - T_4)/(T_{5s} - T_4). \tag{11}$$

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

Assuming constant specific heats, the heat added to the working fluid and the heat rejected by the working fluid are defined as follows from the first law of thermodynamics [7, 18]:

$$Q_{in} = \dot{m}_t[c_V(T_3 - T_2) + c_p(T_4 - T_3)] = \frac{\dot{m}_t R_{air}}{(\gamma - 1)}[T_3 - T_2 + \gamma(T_4 - T_3)] \tag{12}$$

and

$$Q_{\rm out} = \dot{m}_{\rm t} c_V (T_5 - T_1) = \frac{\dot{m}_{\rm t} R_{\rm air}}{(\gamma - 1)} (T_5 - T_1), \tag{13}$$

where c_V and $c_{\rm p}$ are the specific heat at constant volume and at constant pressure, respectively, and $R_{\rm air}$ is the gas constant for the working fluid.

The relations between $\dot{m}_{\rm a}$ and $\dot{m}_{\rm f}$, between $\dot{m}_{\rm a}$ and $\dot{m}_{\rm t}$ are defined as [23]:

$$\dot{m}_{\rm a} = \dot{m}_{\rm f} (m_{\rm a}/m_{\rm f})_{\rm s}/\phi \tag{14}$$

and

$$\dot{m}_{\rm t} = \dot{m}_{\rm f} [1 + (m_{\rm a}/m_{\rm f})_{\rm s}/\phi], \tag{15}$$

where $m_{\rm a}/m_{\rm f}$ is the air–fuel ratio and the subscript s denotes stoichiometric conditions.

Combining Eqs. (8) and (10) gives the temperature T_2 as a function of T_1:

$$T_2 = \frac{T_1}{\eta_{\rm c}} \left(r_{\rm c}^{\gamma-1} + \eta_{\rm c} - 1 \right). \tag{16}$$

Substituting Eqs. (5), (7), (14), (15) and (16) into Eq. (12), the T_3 can be found as

$$T_3 = \frac{(\gamma - 1)\left[\frac{(-1.44738\phi + 4.18581 - 1.86876/\phi) Q_{\rm LHV}}{(m_{\rm a}/m_{\rm f})_{\rm s} + \phi} + 2BT_0\right] + \frac{T_1}{\eta_{\rm c}} (R_{\rm air} - B\gamma + B)\left(r_{\rm c}^{\gamma-1} + \eta_{\rm c} - 1\right)}{B\beta(\gamma - 1) + R_{\rm air} + R_{\rm air}\gamma(\beta - 1)}. \tag{17}$$

Combining Eqs. (5) and (7) yields

$$T_4 = \frac{\left\{\beta(\gamma - 1)\left[\frac{(-1.44738\phi + 4.18581 - 1.86876/\phi) Q_{\rm LHV}}{(m_{\rm a}/m_{\rm f})_{\rm s} + \phi} + 2BT_0\right] + \frac{\beta T_1}{\eta_{\rm c}} (R_{\rm air} - B\gamma + B)\left(r_{\rm c}^{\gamma-1} + \eta_{\rm c} - 1\right)\right\}}{B\beta(\gamma - 1) + R_{\rm air} + R_{\rm air}\gamma(\beta - 1)}. \tag{18}$$

The temperature T_5 can be found by substituting Eqs. (5) and (6) into Eq. (7)

$$T_5 = \frac{\left[\eta_{\rm e}\left(\frac{\beta}{r_{\rm c}}\right)^{\gamma} + 1 - \eta_{\rm e}\right]\left\{\beta(\gamma - 1)\left[\frac{(-1.44738\phi + 4.18581 - 1.86876/\phi) Q_{\rm LHV}}{(m_{\rm a}/m_{\rm f})_{\rm s} + \phi} + 2BT_0\right] + \frac{\beta T_1}{\eta_{\rm c}} (R_{\rm air} - B\gamma + B)(r_{\rm c}^{\gamma-1} + \eta_{\rm c} - 1)\right\}}{B\beta(\gamma - 1) + R_{\rm air} + R_{\rm air}\gamma(\beta - 1)}. \tag{19}$$

Every time the piston moves, friction acts to retard the motion. It should be noted here that the friction is another source of irreversibility. Considering the friction effects on the piston in all the processes of the cycle, we assume a dissipation term represented by a friction force that is linearly proportional to the velocity of the piston, which can be written as follows [2, 20]:

$$f_\mu = -\mu S_{\rm p} = -\mu \frac{{\rm d}x}{{\rm d}t}, \tag{20}$$

where μ is the coefficient of friction, which takes into account the global losses, x is the piston's displacement and $S_{\rm p}$ is the piston's velocity. Therefore, the lost power due to friction is

$$P_\mu = \frac{{\rm d}W_\mu}{{\rm d}t} = -\mu \left(\frac{{\rm d}x}{{\rm d}t}\right)^2 = -\mu (S_{\rm p})^2. \tag{21}$$

Using the above equations, the power output of the dual cycle can be expressed in terms of T_1 as below

$$\begin{aligned} P_{\rm dual} = Q_{\rm in} - Q_{\rm out} - p_\mu = \frac{\dot{m}_{\rm f}[\phi + (m_{\rm a}/m_{\rm f})_{\rm s}] R_{\rm air}}{\phi(\gamma - 1)} \\ \times \Bigg\{ T_1 - \frac{T_1}{\eta_{\rm c}} \left(r_{\rm c}^{\gamma-1} + \eta_{\rm c} - 1 \right) \\ + \left[\gamma\beta - \gamma - \eta_{\rm e}\beta \left(\frac{\beta}{r_{\rm c}}\right)^{\gamma-1} + \beta\eta_{\rm e} - \beta\right] \\ \times \Bigg[\frac{(\gamma - 1)(-1.44738\phi + 4.18581 - 1.86876/\phi) Q_{\rm LHV}}{(m_{\rm a}/m_{\rm f})_{\rm s} + \phi} \\ + 2BT_0(\gamma - 1) + \frac{T_1}{\eta_{\rm c}} (R_{\rm air} - B\gamma + B) \\ \times \left(r_{\rm c}^{\gamma-1} + \eta_{\rm c} - 1 \right) \Bigg] \Bigg\} \\ \times \{B\beta(\gamma - 1) + R_{\rm air} + R_{\rm air}\gamma(\beta - 1)\}^{-1} - \mu \left(\bar{S}_{\rm p}\right)^2, \end{aligned} \tag{22}$$

where $\bar{S}_{\rm p}$ is the mean velocity of the piston.

The efficiency of the dual cycle engine is expressed by

$$\eta_{\rm dual} = \frac{p_{\rm out}}{Q_{\rm in}} = \frac{N}{D},$$

where

$$\begin{aligned} N = \Bigg\{ T_1 - \frac{T_1}{\eta_{\rm c}} \left(r_{\rm c}^{\gamma-1} + \eta_{\rm c} - 1 \right) \\ + \left[\gamma\beta - \gamma - \eta_{\rm e}\beta \left(\frac{\beta}{r_{\rm c}}\right)^{\gamma-1} + \beta\eta_{\rm e} - \beta\right] \end{aligned}$$

$$\times\left[2BT_0(\gamma-1)+\frac{(\gamma-1)(-1.44738\phi+4.18581-1.86876/\phi)Q_{\mathrm{LHV}}}{(m_{\mathrm{a}}/m_{\mathrm{f}})_{\mathrm{s}}+\phi}+\frac{T_1}{\eta_{\mathrm{c}}}(R_{\mathrm{air}}-B\gamma+B)\left(r_{\mathrm{c}}^{\gamma-1}+\eta_{\mathrm{c}}-1\right)\right]\right\}$$

$$\times\left\{B\beta(\gamma-1)+R_{\mathrm{air}}+R_{\mathrm{air}}\gamma(\beta-1)\right\}^{-1}-\frac{\phi\mu\left(\bar{S}_{\mathrm{p}}\right)^2(\gamma-1)}{\dot{m}_{\mathrm{f}}[\phi+(m_{\mathrm{a}}/m_{\mathrm{f}})_{\mathrm{s}}]R_{\mathrm{air}}},$$

and

$$D=\left\{[1+\gamma\beta-\gamma]\left[\frac{(\gamma-1)(-1.44738\phi+4.18581-1.86876/\phi)Q_{\mathrm{LHV}}}{(m_{\mathrm{a}}/m_{\mathrm{f}})_{\mathrm{s}}+\phi}+2BT_0(\gamma-1)\right.\right.$$

$$\left.\left.+\frac{T_1}{\eta_{\mathrm{c}}}(R_{\mathrm{air}}-B\gamma+B)\left(r_{\mathrm{c}}^{\gamma-1}+\eta_{\mathrm{c}}-1\right)\right]\right\}\left\{B\beta(\gamma-1)+R_{\mathrm{air}}+R_{\mathrm{air}}\gamma(\beta-1)\right\}^{-1}-\frac{T_1}{\eta_{\mathrm{c}}}\left(r_{\mathrm{c}}^{\gamma-1}+\eta_{\mathrm{c}}-1\right). \tag{23}$$

3. Results and discussion

The effects of equivalence ratio and mean piston speed on the performance of the dual cycle with $\eta_{\mathrm{e}}=0.97$, $\eta_{\mathrm{c}}=0.97$, $Q_{\mathrm{LHV}}=44000$ kJ kg^{-1}, $(m_{\mathrm{a}}/m_{\mathrm{f}})_{\mathrm{s}}=14.6$, $T_0=350$ K, $T_1=280$ K, $\bar{S}_{\mathrm{p}}=8$–17 m s^{-1}, $\phi=0.85\to1.3$, $\gamma=1.4$, $\beta=1.4$, $r_{\mathrm{c}}=1\to60$, $\mu=12.9$ N s m^{-1}, $\dot{m}_{\mathrm{f}}=0.00004\bar{S}_{\mathrm{p}}$ kg s^{-1} and $B=0.92$ kJ kg^{-1} K^{-1} [19, 23, 25–29] are shown in Figs. 2–5. Using the above constants and range of parameters, the power output versus compression ratio characteristic and the power output versus efficiency characteristic under variable mean piston speed and equivalence ratio can be plotted. Numerical examples are shown as follows.

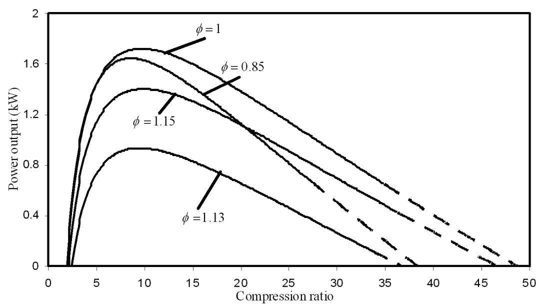

Fig. 2. Effect of equivalence ratio on the variation of the power output with compression ratio ($\bar{S}_{\mathrm{p}}=11$ m s^{-1}).

Figures 2–5 show the effects of the equivalence ratio and the mean piston 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 equivalence ratio and the mean piston speed play important roles on the performance of the dual engine. It is clearly seen that the effects of the equivalence ratio and the mean piston speed on the performance of the cycle are related to compression ratio. They reflect the performance characteristics of a real irreversible dual cycle engine. The power output versus compression ratio characteristic is approximately parabolic-like curve.

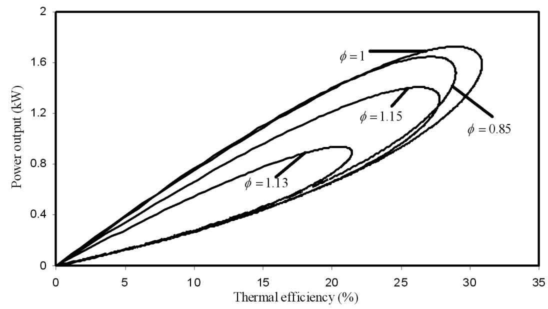

Fig. 3. Effect of equivalence ratio on the variation of the power output with thermal efficiency ($\bar{S}_{\mathrm{p}}=11$ m s^{-1}).

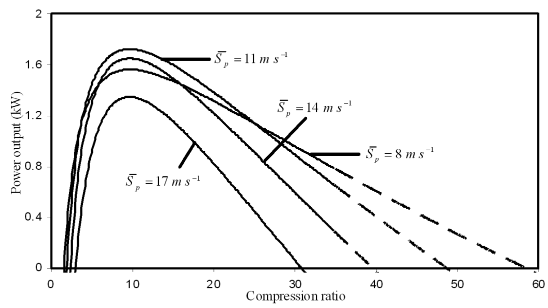

Fig. 4. Effect of equivalence ratio on the variation of the power output with compression ratio ($\phi=1$).

In other words, the power output increases with the increasing compression ratio, attains its maximum value and then decreases with further increases in compression ratio. As can be found from these figures, the power output versus thermal efficiency characteristics exhibit loop shaped curves as is common to almost all real heat engines [30].

Referring to Fig. 2, it can be concluded that if compression ratio is less than certain value, the power output decreases with the increasing equivalence ratio. This can be attributed to the fact that the difference between heat added and heat rejected decreases with the increas-

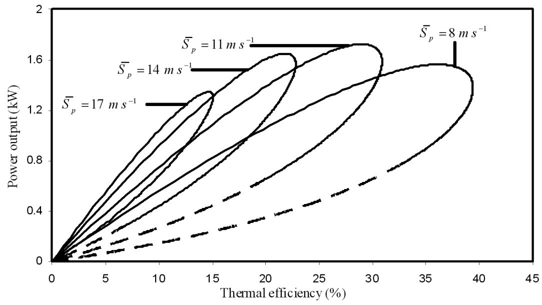

Fig. 5. Effect of equivalence ratio on the variation of the power output with thermal efficiency ($\phi = 1$).

ing equivalence ratio. But if compression ratio exceeds certain value, the power output first increases and then starts to decrease with the increasing equivalence ratio. This can be attributed to the fact that the difference between heat added and heat rejected first increases and then starts to decrease with the increasing equivalence ratio. It can also be concluded that, for example, the power output at compression ratio of 9.6 first increases and then decreases as the equivalence ratio is increased. This result is consistent with the experimental results in the internal combustion engine [31]. Figures 2 and 3 show that the optimal compression ratio corresponding to maximum power output point, the working range of the cycle, the thermal efficiency at the maximum power output point, the maximum power output and the power output at the maximum efficiency point increase and then decrease as the equivalence ratio increases. Numerical calculation shows that for any same compression ratio, the smallest power output is for $\phi = 1.3$ and also the largest power output is for $\phi = 0.85$ when $r_c \le 5$, is for $\phi = 1$ when $r_c > 5$.

Referring to Fig. 4, it can be concluded that if compression ratio is less than certain value, the power output decreases with the increasing mean engine speed, while if compression ratio exceeds certain value, the power output first increases and then starts to decrease with the increasing mean engine speed. With further increase in compression ratio, the increase of mean piston speed results in decreasing the power output. It can also be concluded that, for example, the power output at compression ratio of 9.6 first increases and then decreases as the mean piston speed is increased. This is consistent with the experimental results in the internal combustion engine [31]. The figure also shows that the optimal compression ratio corresponding to maximum power output point and the working range of the cycle remain constant as the mean engine speed is increased. Figures 4 and 5 show that the maximum power output and the power output at maximum thermal efficiency increase with the increasing mean piston speed up to about $\bar{S}_p = 11$ m s^{-1} where they reach their peak value then start to decline as the mean piston speed is increased.

Figure 5 shows that the thermal efficiency at maximum power output decreases when the mean piston speed increases from $\bar{S}_p = 8$ m s^{-1} to $\bar{S}_p = 17$ m s^{-1}. Numerical calculation shows that for any same compression ratio, the smallest power output is for $\bar{S}_p = 17$ m s^{-1} and also the largest power output is for $\bar{S}_p = 8$ m s^{-1} when $r_c \le 3.7$ or $r_c > 25.3$, is for $\bar{S}_p = 11$ m s^{-1} when $3.7 < r_c \le 25.3$.

According to above analysis, it can be found that the effects of the equivalence ratio and the mean piston 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

In this paper, an irreversible air standard dual cycle model which is closer to practice is established. The relations between power output, thermal efficiency, compression ratio, equivalence ratio and mean piston speed are derived. The maximum power output and the corresponding efficiency and the maximum efficiency and the corresponding power output are also calculated. The detailed effect analyses are shown by numerical examples. The analysis helps us to understand the strong effects of mean piston speed and equivalence ratio on the performance of the dual cycle. This paper provides an additional criterion for use in the evaluation of the performance and the suitability of a dual engine.

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