Effects of gasoline-air equivalence ratio on performance of an Otto engine

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Abstract: The effects of equivalence ratio on the performance of an Otto cycle during the finite time are investigated. In the cycle model, the friction loss computed from the empirical correlation, the specific heat ratio of the working fluid supposed constant, 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, and between the thermal efficiency and the compression ratio are derived. Moreover, the effects of equivalence ratio on the cycle performance are analyzed. The results show that the power output, the thermal efficiency, the optimal compression ratio corresponding to maximum power output point, the optimal compression ratio corresponding to maximum thermal efficiency point and the working range of the cycle increase and then decrease as the equivalence ratio increases. [Journal of American Science 2010;6(2):131-135]. (ISSN: 1545-1003).

Key words: equivalence ratio; Otto cycle; internal irreversibility; performance

1. Introduction

A study of gas cycles as the models of internal combustion engines is useful for illustrating some of the important parameters influencing engine performance. In the last two decades, by using finite time thermodynamics theory, many optimization studies based on various performance criteria have been carried out for endoreversible and irreversible heat engine models [Sahin et al., 2002, Parlak et al., 2004; ebrahimi, 2009a]. Leff (1987) showed that some model engines (e.g., Otto, Diesel, Joule-Brayton, and Atkinson), operating reversibly without any loss at maximum work output per cycle, have efficiencies equal to, or well approximated by, the Novikov–Chambadal–Curzon Ahlborn (NCCA) efficiency. Orlov and Berry (1993) deduced the power and efficiency upper limits for internal-combustion engines. They derived the maximum work or power and the corresponding efficiency bounds. Bera and Bandyopadhyay (1998) studied the effect of combustion on the thermoeconomic performances of Otto and Joule–Brayton engines. 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. Parlak et al., (2005) 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. Chen et al., (2003) determined the characteristics of power and efficiency for Otto cycle with heat transfer and friction losses. 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 and Sahin (2006) defined the internal irreversibility by using entropy production and analyzed the effect of the internal irreversibility on the performance of the irreversible reciprocating heat engine cycle. Ge et al., (2008) analyzed the performance of an air standard Otto cycle. 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.

As can be seen in the relevant literature, the investigation of the effect of equivalence ratio on performance of Otto cycle does not appear to have been published. Therefore, the objective of this study is to examine the effect of the equivalence ratio on performance of air standard Otto cycle.

2. Otto cycle model

An air-standard Otto cycle model is shown in figure 1. 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 rejection is an isochoric process $4 \rightarrow 1$. 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 (Heywood, 1988):

$$Q_{in} = \dot{m}_t c_v \left(T_3 - T_2\right) \quad (1)$$

and

$$Q_{out} = \dot{m}_t c_v \left(T_4 - T_1\right) \tag{2}$$

Therefore, the power output is

$$P_{otto} = Q_{in} - Q_{out} - p_{fri} = \dot{m}_t c_v \left(T_3 - T_2\right) - \dot{m}_t c_v \left(T_4 - T_1\right) - p_{fri} = \frac{R_{air}\dot{m}_t}{\gamma - 1}\left(T_1 - T_2 + T_3 - T_4\right) - p_{fri} \tag{3}$$

where $\dot{m}_t$ is the mass flow rate of the air–fuel mixture, R_{air} is the gas constant, c_v is the specific heat at constant volume for the working fluid, T is the absolute temperature, p_{fri} is the friction power and γ is the specific heat ratio, $\gamma = c_p/c_v$.

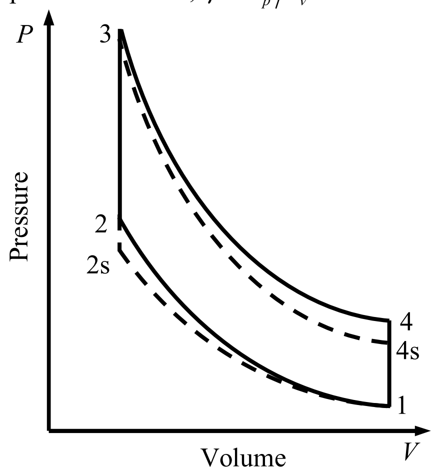

Figure 1. $P-V$ diagram for the air standard Otto cycle

The compression ratio, r_c , is defined as:

$$r_c = V_1/V_2 \tag{4}$$

where V is the volume of the gas in the cylinder.

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

$$T_{2s} = T_1 r_c^{\gamma-1} \tag{5}$$

and

$$T_{4s} = T_3 r_c^{1-\gamma} \tag{6}$$

For the two reversible adiabatic processes $1 \to 2s$ and $3 \to 4s$, the compression and expansion efficiencies can be defined as (Chen, 2004; Ebrahimi, 2009c):

$$\eta_c = \left(T_{2s} - T_1\right)/\left(T_2 - T_1\right) \tag{7}$$

and

$$\eta_e = \left(T_4 - T_3\right)/\left(T_{4s} - T_3\right) \tag{8}$$

Substituting equation (5) into equation (7) yields:

$$T_2 = \frac{T_1\left(r_c^{\gamma-1} + \eta_c - 1\right)}{\eta_c} \tag{9}$$

When the total energy of the fuel is utilized, the maximum cycle temperature reaches undesirably high levels with regard to structural integrity. Hence, engine designers intend to restrict the maximum cycle temperature. The total energy of the fuel per second input into the engine can be given by: (Heywood, 1988)

$$Q_{fuel} = \eta_{com}\dot{m}_f Q_{LHV} \tag{10}$$

The heat loss through the cylinder wall is given in the following linear expression (Chen et al., 2008)

$$Q_{ht} = \dot{m}_t B\left(T_2 + T_3\right) \tag{11}$$

where B is constant.

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} ,

$$Q_{in} = Q_{fuel} - Q_{ht} = \eta_{com}\dot{m}_f Q_{LHV} - \dot{m}_t B\left(T_2 + T_3\right) \tag{12}$$

The relations between $\dot{m}_a$ and $\dot{m}_f$, between $\dot{m}_a$ and $\dot{m}_t$ are defined as (Heywood, 1988):

$$\dot{m}_f = \frac{\dot{m}_a \phi}{\left(m_a/m_f\right)_s} \tag{13}$$

and

$$\dot{m}_t = \dot{m}_a\left(1 + \frac{\phi}{\left(m_a/m_f\right)_s}\right) \tag{14}$$

where ϕ is the equivalence ratio, m_a/m_f is the air–fuel ratio and the subscript s denotes stoichiometric conditions.

Combining equations (1) and (12) gives:

$$T_3 = \frac{\left(\eta_c \eta_{com} Q_{LHV}\right)/\left(\phi\left(m_a/m_f\right)_s\right) - T_1\left(r_c^{\gamma-1} + \eta_c - 1\right)\left(B + c_v\right)}{\eta_c\left(B + c_v\right)} \tag{15}$$

Substituting equations (15) and (6) into equation (8) yields:

$$T_4 = \left[\frac{\left(\eta_c \eta_{com} Q_{LHV}\right)/\left(\phi\left(m_a/m_f\right)_s\right) - T_1\left(r_c^{\gamma-1} + \eta_c - 1\right)\left(B + c_v\right)}{\eta_c\left(B + c_v\right)}\right]\left(r_c^{\gamma-1} + 1 - \eta_e\right) \tag{16}$$

The combustion efficiency, η_{com} , of a gasoline type fuel, such as octane, can be expressed in terms of equivalence ratio factor from measured data as (Abd Alla, 2002):

$$\eta_{com} = \eta_{com\,\max}\left(-1.6082 + 4.6509/\phi - 2.0764/\phi^2\right) \tag{17}$$

where the maximum possible value of the combustion efficiency, $\eta_{com\,\max}$, is typically 0.9 in a spark ignition engine using a gasoline fuel. The range of effective ϕ values spans normal spark ignition combustion, i.e., from about 0.83 to about 1.33.

The data of total motored friction mean effective pressure for several four stroke cycle, four cylinder spark ignition engines between 845 and 2000 cm^3 displacement, at wide open throttle, as a function of engine speed (Abd Alla, 2002) are well correlated by an equation of the form:

$$fmep = 97 + 0.9N + 0.18N^2 \quad (18)$$

where N is in revolutions per second. The unit of $fmep$ is kpa .

Therefore, the lost power due to friction is

$$p_{fri} = \frac{fmepV_dN}{2} = V_dN\left(45.5 + 0.45N + 0.09N^2\right) \quad (19)$$

Therefore, the net actual power output of the Otto cycle engine can be written as:

$$P_{otto} = \frac{R_{air}\dot{m}_a\left[\left(m_a/m_f\right)_s + \phi\right]}{\eta_c(\gamma-1)\left(m_a/m_f\right)_s}\left[T_1\left(1-r_c^{\gamma-1}\right) + \left[\begin{array}{l}\left(\eta_c\eta_{com}Q_{LHV}\right)/\left(\phi\left(m_a/m_f\right)_s\right) - \\ \dfrac{T_1\left(r_c^{\gamma-1}+\eta_c-1\right)\left(B+c_v\right)}{B+c_v}\end{array}\right]\left(\eta_e - r_c^{\gamma-1}\right)\right] - \quad (20)$$

$$V_dN\left(45.5+0.45N+0.09N^2\right)$$

The thermal efficiency of the Otto cycle engine is expressed by

$$\eta_{th} = P_{otto}/Q_{in} = \frac{1}{\eta_c\left(T_3-T_2\right)}\left[T_1\left(1-r_c^{\gamma-1}\right) + \left[\begin{array}{l}\left(\eta_c\eta_{com}Q_{LHV}\right)/\left(\phi\left(m_a/m_f\right)_s\right) - \\ \dfrac{T_1\left(r_c^{\gamma-1}+\eta_c-1\right)\left(B+c_v\right)}{B+c_v}\end{array}\right]\left(\eta_e - r_c^{\gamma-1}\right)\right] - \quad (21)$$

$$V_dN(45.5+0.45N+0.09N^2)/\left(T_3-T_2\right)$$

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

3. Numerical examples and discussions

As it can be concluded from Eqs. (20) and (21), the power output and the thermal efficiency of the Otto cycle are dependent on the equivalence ratio. In order to illustrate the effect of this parameter, the relations between the power output and the compression ratio, between the thermal efficiency and the compression ratio, and the optimal relation between power output and the efficiency of the cycles presented in figures 2–4. According to references (Ebrahimi, 2009b; Chen et al., 2008; Ge et al., 2008) , the following parameters are used: $\eta_e = 0.97$, $\eta_c = 0.97$, $Q_{hv} = 44000\,kJ/kg$, $T_1 = 300\,K$, $N = 3000\,rpm$, $V_d = 366.5\,cm^3$, $\gamma = 1.4$, $\phi = 0.9 \rightarrow 1.3$, $r_c = 1 \rightarrow 100$, $\left(m_a/m_f\right)_s = 14.5$, $\dot{m}_a = 0.0146\,kg/s$ and $B = 0.57\,kJ/kg\,K$.

Figures 2-3 show the effect of the equivalence ratio on the cycle performance with heat resistance, internal irreversibility and friction losses. From these figures, it can be found that the equivalence ratio plays important roles on the performance of the Otto engine. It is clearly seen that the effects of equivalence ratio on the performance of the cycle is related to compression ratio. They reflect the performance characteristics of a real irreversible Otto cycle engine. The power output versus compression ratio characteristic and the thermal efficiency versus compression ratio characteristic are approximately parabolic like curves. In other word, the power output and the thermal efficiency increase with increasing compression ratio, attain their maximum values and then decrease with further increases in compression ratio. It should be noted that the heat added and the heat rejected by the working fluid first increase and then start to decrease as the equivalence ratio increases (see Eqs. (2) and (3)). Figures 2 and 3 show that the power output and the thermal efficiency increase with increasing equivalence ratio up to about $\phi = 1.1$ where they reach their peak value. This can be attributed to the fact that the ratio of the heat added by the working fluid to the heat rejected by the working fluid increase with the increasing equivalence ratio. With further increase in equivalence ratio, the power output and the thermal efficiency start to decline as the equivalence ratio increases. It can be attributed to the decrease in the ratio of the heat added by the working fluid to the heat rejected by the working fluid. This result is consistent with the experimental results in the internal combustion engine (Mercier, 2006). The results also revealed that the optimal compression ratio corresponding to maximum power output point, the optimal compression ratio corresponding to maximum thermal efficiency point and the working range of the cycle increase and then decrease as the equivalence ratio increases. Numerical calculation shows that for any same compression ratio, the smallest power output and the smallest thermal efficiency are for $\phi = 0.9$ and the

largest power output and the largest thermal efficiency are for $\phi = 1.1$ when the equivalence ratio increases from $\phi = 0.9$ to $\phi = 1.3$.

According to above analysis, it can be found that the effects of the equivalence ratio 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, the effects of equivalence ratio on the performance of an Otto cycle during the finite time are investigated. The results show that the power output, the thermal efficiency, the optimal compression ratio corresponding to maximum power output point, the optimal compression ratio corresponding to maximum thermal efficiency point, the working range of the cycle, the power output at maximum thermal efficiency and the

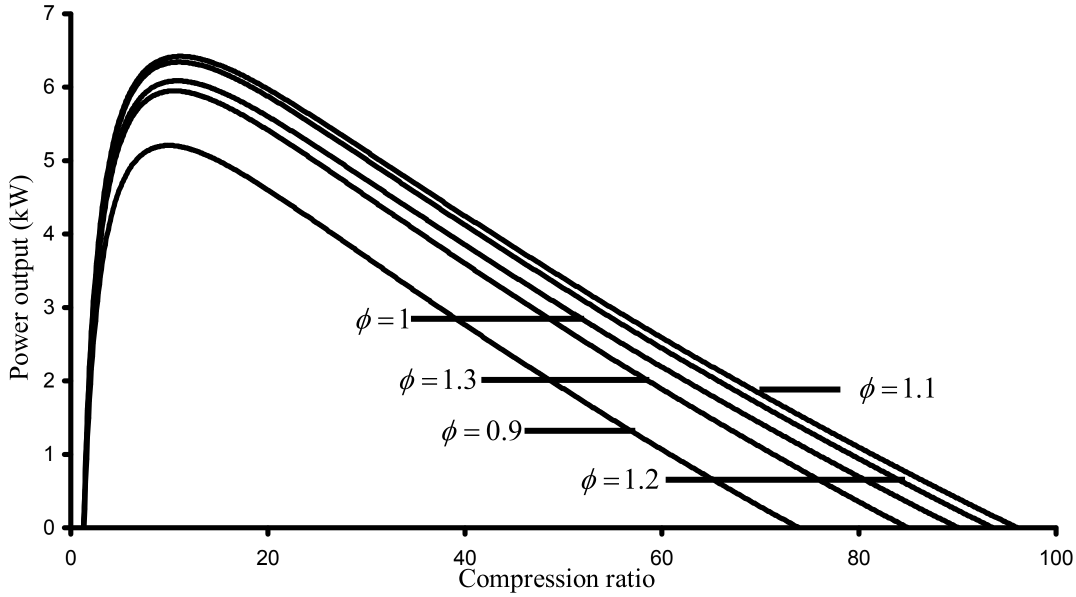

Figure 2. Effect of combustion efficiency on the variation of the power output with compression ratio

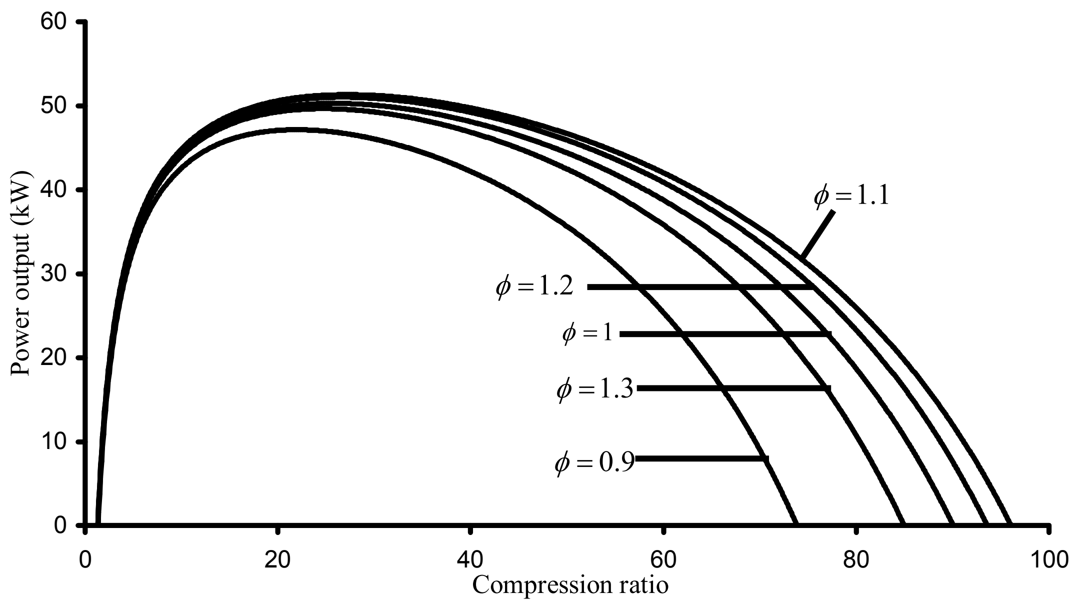

Figure 3. Effect of combustion efficiency on the variation of the thermal efficiency with compression ratio

thermal efficiency at maximum power output increase and then decrease as the equivalence ratio increases. The results of this investigation are of importance when considering the designs of actual Otto engines.

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