Effects of irreversible different parameters on performance of air standard Otto cycle

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Abstract: An *irreversible* air standard *Otto cycle* model is proposed in this paper. The performance of an air-standard *Otto cycle* with heat transfer loss and *variable specific heats* of working fluid is analyzed by using finite-time thermodynamics. They are generalized formulas for internal combustion engines because they include the performance characteristic of special cases of Otto engines. The objective of this study is to analyze the effects of heat loss characterized 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. A more realistic and precise relationship between the fuel's chemical energy and the *heat leakage* that is based on a pair of inequalities is derived through the resulting temperature. The power output and the working range of the cycle increase with the increase of specific heats of the working fluid, while the efficiency decreases with the increase of specific heats of the working fluid. The *friction* loss has a negative effect on the performance. The results obtained in the present study are of importance to provide good guidance for performance evaluation and improvement of practical Otto engines. [Journal of American Science 2011;7(3):248-254]. (ISSN: 1545-1003).

Keywords: : *Otto cycle, Heat leakage, Friction, Irreversible, Variable specific heat.*

1. Introduction

A series of achievements have been made since finite-time thermodynamics was used to analyze and optimize real heat engines (Bejan, 1996). Preliminary models leading to a qualitative understanding of how engine losses could be reduced are introduced by Mozurkewich and Berry (1982). They are based on the optimal control theory (Chen, Wu, Sun, 1999). The losses are considered to consist of the friction forces in the crank shaft bearings and piston rings, the pressure drop or differentiation effect as the gas flows through the inlet valves, the heat leakage amount from the working fluid to the cylinder walls and also the time loss term containing the burning velocity (Chen, Sun, 2004). In practice, air standard analysis is useful for illustrating the thermodynamic aspects of an engine operation cycle. Meanwhile, it can provide approximate estimates of trends as major engine operating variables change (Hoffman, Watowich, Berry, 1985). Good approximations of power output, thermal efficiency and mep (mean effective pressure) can be expected. For an ideal engine cycle, heat losses do not occur, however, for a real engine cycle, heat losses indeed exist and should not be neglected. It is recognized that heat loss strongly affects the overall performance of the internal combustion engine (Chen, Lin, Luo, Sun, Wu, 2002). If it is neglected, the analysis will just depend on the ideal air standard cycle. Some attention has been paid to analyzing the effects of heat transfer losses on the performance of internal combustion engines (Brown, Fernandez, Diazpico, 1994).

The heat addition process for an air standard cycle has been widely described as subtraction of an arbitrary heat loss parameter times the average temperature of the heat addition period from the fuel's chemical energy (Orlov, Berry, 1993). That is, the heat transfer to the cylinder walls is assumed to be a linear function of the difference between the average gas and cylinder wall temperatures during the energy release process. However, the heat leakage parameter and the fuel's energy depend on each other. Their valid ranges given in the literature affect the feasibility of air standard cycles (Wang, Chen, Sun, Wu, 2002). If they are selected arbitrarily, they will present unrealistic results and make the air standard cycles unfeasible (Chen, Zheng, Sun, Wu, 2003). There by, the performance analysis of any internal combustion engine can be covered by a more realistic and valid range of the heat loss parameter and the fuel's energy (Chen, Sun, Wu, 2004). Moreover, his study was done without considering the effects of variable specific heats of the working fluid and friction (Ge, Chen, Sun, 2005). In particular, no performance analysis is available in the literature with emphasis on the Otto cycle with considerations of variable specific heats of the working fluid, friction and heat leakage characterized by a percentage of the fuel's energy (Al-Sarkhi, Jaber, Abuqudais, Probert, 2006). This study is aimed at analyzing these effects (i.e. variable specific heats of working fluid, friction and heat loss characterized by a percentage of the fuel's energy) on the net work output and the indicated thermal efficiency of an air standard Otto cycle (Al-Sarkhi, Jaber, Probert, 2006). In the present study, we relax the assumptions that there are no heat losses during combustion, that there are no friction losses of the piston for the cycle, and that specific heats of the working fluid are constant (Ge, Chen, Sun, Wu, 2006). In other words, heat transfer between the working fluid and the environment through the cylinder wall is considered and characterized by a percentage of the fuel's energy; friction loss of the piston in all the processes of the cycle on the performance is taken into account. Furthermore, we consider the variable specific heats of the working fluid that is significant in practical cycle analysis. The results obtained in the study may offer good guidance for design and operation of the Otto cycle engine (Ozsoysal, 2006).

2. Thermodynamic analysis

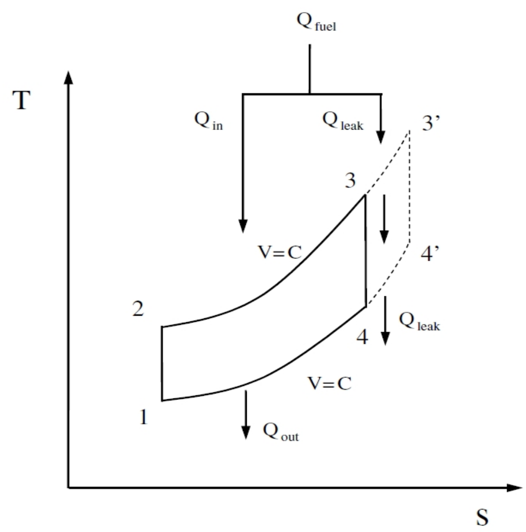

Figure 1. T–s diagram of an air standard Otto cycle model

Fig.1 shows the limitation of the maximum cycle temperature due to heat leakage in the temperature-entropy diagram of an air standard Otto cycle model.

Thermodynamic cycle 1–2–3′–4′–1 denotes the air standard Otto cycle without heat leakage, while cycle 1–2–3-4-1 designates the air standard Otto cycle with heat leakage. Process 1–2 is an isentropic compression from BDC (bottom dead center) to TDC (top dead center). The heat addition takes place in process 2–3, which is isochoric. The isentropic expansion process, 3–4, is the power or expansion stroke.

The cycle is completed by an isochoric heat rejection process, 4–1. The heat added to the working fluid per unit mass is due to combustion. The temperature at the completion of the constant volume combustion (T_3) depends on the heat input due to combustion and the heat leakage through the cylinder wall. In this study, the amount of heat leakage is considered to be a percentage of the delivered fuel's energy (Mozurkewich and Berry, 1982). The fuel's energy then is the sum of the actual fuel energy transferred to the working fluid and the heat leakage through the cylinder walls. If any heat leakage occurs, the maximum cycle temperature (T_3) remains less than that of the no heat leakage case ($T_{3'}$).

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. Assuming that the heat engine is operated at the rate of N cycles per second, the total energy of the fuel per second input into the engine can be given by

$$Q_{fuel}=Nm_f Q_{LHV} \tag{1}$$

And then the heat leakage per second is

$$Q_{leak}=a\,Q_{fuel}=a\,Nm_f Q_{LHV} \tag{2}$$

where m_f is the delivered fuel mass into the cylinder, Q_{LHV} is the lower heating value of the fuel and a is an unknown percentage parameter having a value between 0 and 1.

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_{leak},

$$Q_{in}=Q_{fuel}-Q_{leak}=(1-a)Nm_f Q_{LHV} \tag{3}$$

In practical internal combustion engine cycles, constant pressure and constant volume specific heats of the working fluid are variable, and these variations will greatly affect the performance of the cycle. it can be assumed that the specific heats of the working fluid are functions of temperature alone and have the following linear forms:

$$C_{pm}=a_p+k_1T \tag{4}$$

and

$$C_{vm} = b_v + k_1 T \tag{5}$$

where C_{pm} and C_{vm} are respectively, the specific heats with respect to constant pressure and volume. a_p, b_v and k_1 are constants. Accordingly, the gas constant (R) of the working fluid can be expressed as

$$R = C_{pm} - C_{vm} = a_p - b_v \tag{6}$$

The temperature is restricted as the maximum temperature in the cycle is T_3, and the available energy Q_{in} during the heat addition per second can be written as

$$Q_{in} = Nm_a \int_{T_2}^{T_3} C_{vm} dT = Nm_a \int_{T_2}^{T_3} (b_v + k_1 T) dT = Nm_a [b_v (T_3 - T_2) + k_1 bv(T_3^2 - T_2^2). \tag{7}$$

Combining Eqs. (3) and (7) yields

$$Nm_a [b_v (T_3 - T_2) + k_1 bv(T_3^2 - T_2^2) = (1-a) Nm_f Q_{LHV} \tag{8}$$

Dividing Eq. (8) by the amount of air mass m_a, we have

$$a = 1 - \frac{l(m_a/m_f)_s}{Q_{LHV}} [b_v (T_3 - T_2) + k_1 bv(T_3^2 - T_2^2)] \tag{9}$$

Or

$$T_2 = \frac{-b_v + \sqrt{b_v^2 + 2k_1[0.5 k_1 T_3^2 + b_v T_3 - (1-a)\frac{Q_{LHV}}{l(m_a/m_f)_s}}}{k_1} \tag{10}$$

where l is the excess air coefficient defined as $l = (m_a/m_f)/(m_a/m_f)_s$, (m_a/m_f) is the air–fuel ratio and the subscripts a, f, and s, respectively, denote air, fuel and the stoichiometric condition.
The first condition for realizing a feasible cycle is

$T_2 \le T_3 (= T_{\max})$, so that

$$a \le 1 \tag{11}$$

The upper limit for the percentage of heat leakage is then found as $a_{\max} = 1$. The second condition, $T_2 \ge T_1 (= T_{\min})$, is utilized to determine the lower limit as follows

$$a \ge 1 - \frac{l(m_a/m_f)_s}{2Q_{LHV}} [k_1 (T_3^2 - T_1^2) + 2b_v (T_3 - T_1)] \tag{12}$$

Hence, the minimum value of a is expressed as

$$a_{\min} = 1 - \frac{l(m_a/m_f)_s}{2Q_{LHV}} [k_1 (T_3^2 - T_1^2) + 2b_v (T_3 - T_1)] \tag{13}$$

The heat rejected per second by the working fluid (Q_{out}) during process $4 \to 1$ is

$$Q_{out} = Nm_a \int_{T_1}^{T_4} C_{vm} dT = Nm_a \int_{T_1}^{T_4} (b_v + k_1 T) dT = Nm_a [b_v (T_4 - T_1) + 0.5 k_1 (T_4^2 - T_1^2)]. \tag{14}$$

The adiabatic exponent $k = C_{pm}/C_{vm}$ will vary with temperature since both C_{pm} and C_{vm} are dependent on temperature.

Accordingly, the equation often used in reversible adiabatic processes with constant k cannot be used in reversible adiabatic processes with variable k. however, a suitable engineering approximation for reversible adiabatic processes with variable k can be made, i.e. this process can be divided into infinitesimally small processes and for each of these processes, the adiabatic exponent k can be regarded as constant. For instance, for any reversible adiabatic process between states I and II, we can regard the process as consist of numerous infinitesimally small processes with constant k. For any of these processes, when small changes in temperature dT and volume dV of the working fluid take place, the equation for a reversible adiabatic process with variable k can be written as follows:

$$TV^{k-1} = (T + dT)(V + dV)^{k-1} \tag{15}$$

Rearranging Eqs. (4)-(6) and (15), we get the following equation

$$dT/T + [R/(b_v + k_1 T)](dV/V) = 0 \tag{16}$$

Integrating Eq. (16) from state I to state II, we obtain

$$k_1 (T_j - T_i) + b_v \ln(T_j/T_i) = -R \ln(V_j/V_i) \tag{17}$$

The compression ratio (g_c) is defined as $g_c = V_1/V_2$. Therefore, the equations for processes $1 \to 2$ and $3 \to 4$ are shown, respectively, by the following equations:

$$k_1 (T_2 - T_1) + b_v \ln(T_2/T_1) = R \ln g_c \tag{18}$$

and

$$k_1 (T_3 - T_4) + b_v \ln(T_3/T_4) = R \ln g_c \tag{19}$$

From Eqs. (7) and (14), the power output without friction losses is given by:

$$P_R = Q_{in} - Q_{out} = Nm_a[b_v(T_3 + T_1 - T_2 - T_4) + 0.5k_1(T_3^2 + T_1^2 - T_2^2 - T_4^2)] \tag{20}$$

Every time the piston moves, friction acts to retard the motion. Considering the friction effects on the piston in all the processes of the cycle, we assume a dissipation term represented by a friction force (f_m) that is linearly proportional to the velocity of the piston, which can be written as follows:

$$f_m = -mu = -m\frac{dx}{dt} \tag{21}$$

where m is the coefficient of friction, which takes into account the global losses on the power output, x is the piston's displacement and v is the piston's velocity. Therefore, the power lost due to friction is

$$P_m = f_m v = -m(\frac{dx}{dt})^2 = -mv^2 \tag{22}$$

for a four stroke cycle engine, the total distance the piston travels per cycle is

$$4L = 4(x_1 - x_2) = 4x_2(x_1/x_2 - 1) = 4x_2(g_c - 1) \tag{23}$$

where x_1 and x_2 are the piston's position corresponding to the maximum and minimum volume, respectively, and L is the stroke of the piston. Running at N cycles per second, the mean velocity of the piston is

$$\bar{v} = 4LN \tag{24}$$

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

$$P = P_R - |P_m| = Nm_a[b_v(T_3 + T_1 - T_2 - T_4) + 0.5k_1(T_3^2 + T_1^2 - T_2^2 - T_4^2)] - 16m[Nx_2(g_c - 1)^2] \tag{25}$$

The efficiency of the Otto cycle engine is expressed by

$$h = \frac{P}{Q_{in}} = \{m_a[b_v(T_3 + T_1 - T_2 - T_4) + 0.5k_1(T_3^2 + T_1^2 - T_2^2 - T_4^2)] - 16mN[x_2(g_c - 1)]^2\} \times \{m_a[b_v(T_3 - T_2) + 0.5k_1(T_3^2 - T_2^2)]\}^{-1} \tag{26}$$

When T_1, T_3 and g_c are given, T_2 can be obtained from Eq. (18) and T_4 can be found from Eq. (19). Finally, by substituting T_1, T_2, T_3 and T_4 into Eqs. (25) and (26), respectively, the power output and the efficiency of the Otto cycle engine can be obtained. Therefore, the relations between the power output, the efficiency and the compression ratio can be derived.

3. Results and discussion

The following constants and ranges of parameters are used in the calculations: b_v = 0.6858-0.8239 kJ/kg K, $m_a = 1.26 \times 10^{-3}$ kg, T_1 = 300-400 K, $k_1 = 0.000133 - 0.00034 kJ/kg.K^2$, x_2 = 0.01 m, N = 30, Q_{LHV} = 44000 kJ/kg and m = 0.0129-0.0169 kN s/m. This study focuses on the limitation of the maximum cycle temperature T_3 instead of $T_{3'}$ due to the varying heat leakage conditions. Numerical examples are shown as follows.

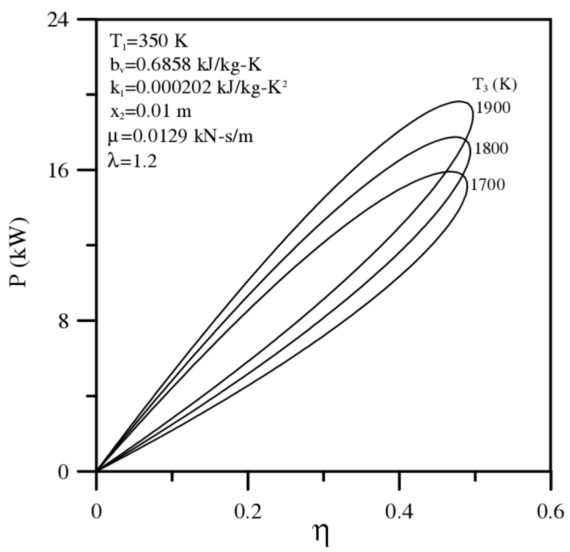

Figure 2. The influence of maximum cycle temperature (T_3) on the power output (P) versus efficiency (h) characteristic curves

We obtain the loop shaped power output versus efficiency curves, which reflect the performance characteristics of a real irreversible Otto cycle engine (Fig 2). It is depicted that the maximum power output, the maximum efficiency, the power at maximum efficiency and the efficiency at maximum power will increase with the increase of T_3.

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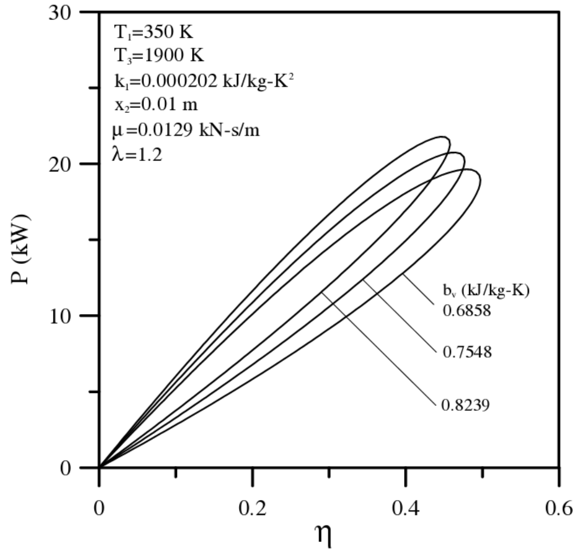

Figure 3. The influence of b_v on the power output (P) versus efficiency (h) characteristic curves

Figs. 3 show the influence of the parameter b_v related to the variable specific heats of the working fluid on the performance of the Otto cycle. For a fixed k_1 , a larger b_v corresponds to a greater value of the specific heat with constant volume (C_{vm}) or the specific heat with constant pressure (C_{pm})

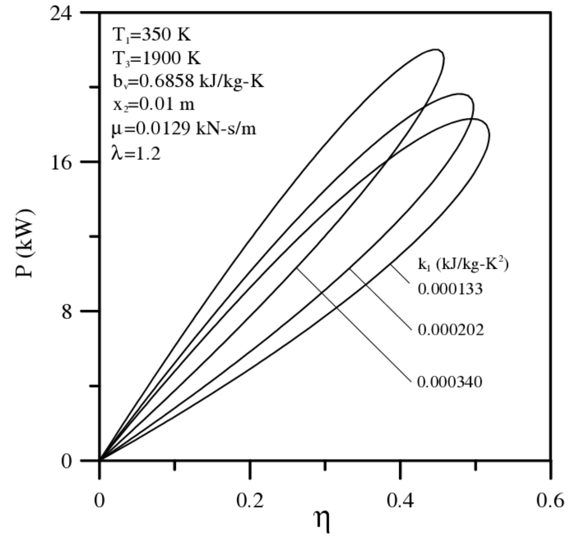

Figure 4. The influence of k_1 on the power output (P) versus efficiency (h) characteristic curves

Figs. 4 represent the influence of the parameter k_1 related to the variable specific heats of the working fluid on the performance of the Otto cycle. For a given b_v , a larger k_1 corresponds to a greater value of the specific heats with constant volume (C_{vm}) or the specific heat with constant pressure (C_{pm}).

With the increase of k_1 , the maximum power output and the power at maximum efficiency increase, while the maximum efficiency and the efficiency at maximum power output decrease, as shown in Fig. 4.

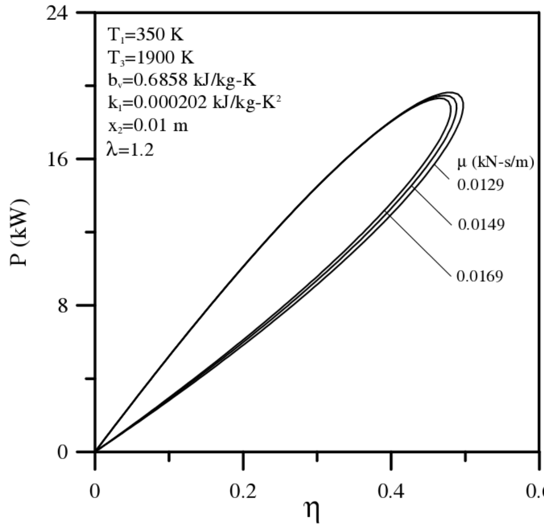

Figure 5. The influence of m on the power output (P) versus efficiency (h) characteristic curves

Figs. 5 show the influence of the friction like term loss (m) on the performance of the Otto cycle. It is clear that the parameter m has a negative effect on the performance. Fig5 shows that the maximum power output, the maximum efficiency, the power at maximum efficiency and the efficiency at maximum power will decrease with the increase of m .

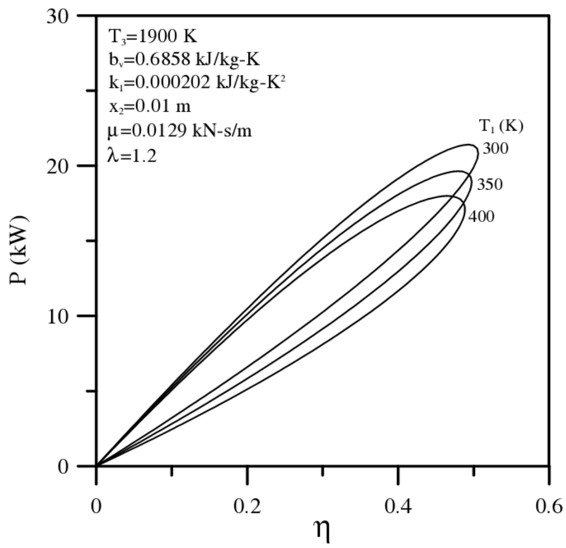

Figure 6. The influence of intake temperature (T_1) on the power output (P) versus efficiency (h) characteristic curves

Fig 6. depict the influence of intake temperature (T_1) on the performance of the Otto cycle. Fig6. It is also found that as T_1 increases, the maximum power output, the maximum efficiency, the efficiency at maximum power output and the power output at maximum efficiency decrease.

4. Conclusions

The effects of heat loss as a percentage of the fuel's energy, friction and variable specific heats of the working fluid on the performance of an Otto engine under the restriction of maximum cycle temperature are presented in this study. The results are summarized as follows.

(1) The maximum power output, the maximum efficiency, the power at maximum efficiency, the efficiency at maximum power and the value of the compression ratio when the power output or the efficiency is maximum increase with the increase of maximum cycle temperature T_3.

(2) The parameters b_v and k_1 related to the variable specific heats of the working fluid have a significant influence on the performance of the Otto cycle. For a fixed k_1 (or b_v), a larger b_v (or k_1) corresponds to a greater value of the specific heats with constant volume (C_{vm}). For a given compression ratio g_c in a feasible range, the power output of the cycle increase with the increase of the parameter b_v or k_1, nevertheless, the efficiency decreases with the increase of b_v or k_1. Furthermore, with the increase of b_v, the maximum power output and the power at maximum efficiency increase, while the maximum efficiency and the efficiency at maximum power output decrease.

(3) The influence of the friction like term loss m has a negative effect on the performance. Therefore, the maximum power output, the maximum efficiency, the power at maximum efficiency and the efficiency at maximum power will decrease with the increase of m.

(4) The maximum efficiency, the compression ratio at maximum power output and the compression ratio at maximum efficiency of the Otto cycle decrease with the increase of intake temperature T_1. The efficiency at maximum power output and the power output at maximum efficiency decrease with increasing T_1.

(5) It is noteworthy that the effects of heat loss as a percentage of the fuel's energy and friction loss on the performance of an Otto cycle engine with considerations of variable specific heats of working fluid are significant and should be considered in practical cycle analysis. The results obtained in the present study are of importance to provide good guidance for performance evaluation and improvement of practical Otto engines.

In view of the analytical results from this work, we realize that the understanding and development of engines and engine cycles should be further explored by considering a more realistic model with advanced theoretical and numerical techniques. For instance, in air standard analysis, the constant volume heat input process replaces the combustion of the real engine cycle, which takes place at close to constant volume conditions, and exhaust blow down in a real engine is almost, but not quite, constant volume. As expected, the maximum temperature in the cycle will depend on the crank angle at which the exhaust valve opens. Hence, a new type of cycle analysis is needed. In other words, conceiving a new model as a function of crank angle to help understand, correlate, and analyze the relation between the maximum temperature and the crank angle at which the exhaust valve opens in the cycle. Additionally, considering the combined effects of heat loss and friction on the performance of engine cycles, detailed comparisons between this work and numerical analysis (or experiments) are worthy of further study.

References

[1] Bejan A. Entropy generation minimization: The new thermodynamics of finite-size device and finite time processes, J. Appl. Phys. 1996; 79(3): 1191–1218.

[2] Mozurkewich M, Berry RS. Optimal paths for thermodynamic system: the ideal Otto cycle. J Appl Phys. 1982; 53(1):34–42.

[3] Chen L, Wu C, Sun F. Finite time thermodynamic optimization or entropy generation minimization of energy systems, J. Non-Equilib. Thermodyn. 1999; 24(4):327–359.

[4] Chen L, Sun F. Advances in Finite Time Thermodynamics: Analysis and Optimization, Nova Science, NewYork. 2004.

[5] Hoffman Kh, Watowich SJ, Berry Rs. Optimal paths for thermodynamic systems: the ideal Diesel cycle. J Appl Phys. 1985; 58(6): 25–34.

[6] Chen L, Lin J, Luo J, Sun F, Wu C. Friction effects on the characteristic performance of Diesel engines. Int J Energ Res. 2002; 26(10):65–71. H.N.(1992).

[7] Angulo-Brown F, Fernandez-Betanzos J, Diaz-Pico CA. Compression ratio of an optimized Otto cycle model. Eur J. Phys. 1994; 15(1):38–42.

[8] Orlov V. N, Berry R. S. Power and efficiency limits for internal combustion engines via methods of finite-time thermodynamics, J. Appl. Phys. 1993; 74(1043).17–22.

[9] Wang W, Chen L, Sun F, Wu C. The effects of friction on the performance of an air standard dual cycle. Exergy An Int J 2002; 2(4):340–344.

[10] Chen L, Zheng T, Sun F, Wu C. The power and efficiency characteristics for an irreversible Otto cycle. Int J Ambient Energ. 2003; 24(4):195–200.

[11] Chen L, Sun F, Wu C. The optimal performance of an irreversible dual cycle. Appl Energ. 2004; 79(1):3–14.

[12] Ge Y, Chen L, Sun F. Effects of heat transfer and friction on the performance of an irreversible air-standard Miller cycle. Int Commun Heat Mass. 2005; 32(1):45–56.

[13] Ge Y, Chen L, Sun F, Wu C. Thermodynamic simulation of performance of an Otto cycle with heat transfer and variable specific heats of working fluid. Int J Therm Sci. 2005; 44(5):506–511.

[14] Al-Sarkhi A, Jaber JO, Abuqudais M, Probert SD. Effects of friction and temperature-dependent specific-heat of the working fluid on the performance of a Diesel-engine. Appl Energ. 2006; 83(1):53–65.

[15] Al-Sarkhi A, Jaber JO, Probert SD. Efficiency of a Miller engine. Appl Energ. 2006; 83(3):3–5.

[16] Ge Y, Chen L, Sun F, Wu C. Effects of heat transfer, friction and variable specific heats of working fluid on performance of an irreversible dual cycle. Energ Convers Manage. 2006; 47(3):24–34.

[17] Ozsoysal OA. Heat loss as a percentage of fuel’s energy in air standard Otto and diesel cycles. Energ Convers Manage. 2006; 47(1):51–62.

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