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Performance analysis of an air-standard Miller cycle with considerations of heat loss as a percentage of fuel's energy, friction and variable specific heats of working fluid

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Abstract

This study is aimed at examining the effects of heat loss characterized by a percentage of fuel's energy, friction and variable specific heats of working fluid on the performance of an air-standard Miller cycle under the restriction of maximum cycle temperature. A more realistic and precise relationship between the fuel's chemical energy and the heat leakage that is constituted on a pair of inequalities is derived through the resulting temperature. The variations in power output and thermal efficiency with compression ratio, and the relations between the power output and the thermal efficiency of the cycle are presented. The results show that the power output as well as the efficiency where maximum power output occurs will increase with the increase of maximum cycle temperature. The temperature-dependent specific heats of working fluid have a significant influence on the performance. The power output and the working range of the cycle increase while the efficiency decreases with the increase of specific heats of working fluid. The friction loss has a negative effect on the performance. Therefore, the power output and efficiency of the cycle decrease with increasing friction loss. Comparison of the performance of air-standard Miller and Otto cycles are also discussed. Miller cycle has larger power output and efficiency than Otto cycle does, i.e., Miller cycle is more efficient than Otto cycle. It is noteworthy that the effects of heat loss characterized by a percentage of fuel's energy, friction and variable specific heats of working fluid on the performance of a Miller-cycle engine are significant and should be considered in practice cycle analysis. The results obtained in the present study are of importance to provide a good guidance for the performance evaluation and improvement of practical Miller engines.

Keywords: Miller cycle; Heat leakage; Friction; Irreversible; Variable specific heat

1. Introduction

Concerns about energy saving, pre-ignition engine knock, emission of pollutants and carbon dioxide production have result in modifications in the internal combustion Otto engine. There are several methods proposed by Simmons [1] to modify the spark-ignited internal combustion Otto cycle. One of the major alternatives of the Otto cycle is to shorten the compression process relative to the expansion process by early close of intake valve, which has great potential for increased efficiency and net work power in the spark ignited internal combustion engine [2–4]. The modified Otto cycle is called Miller cycle [5].

To make the analysis of the engine cycle much more manageable, air standard cycles are used to describe the major processes occurring in internal combustion engines. Air is assumed to behave as an ideal gas, and all processes are considered to be reversible [2,3]. In practice, air-standard analysis is useful for illustrating the thermodynamic aspects of an engine operation cycle. For an ideal engine cycle, the heat losses do not occur, however, for a real engine cycle, the heat losses indeed exist and should not be negligible. It is recognized that heat loss strongly affects the overall performance of the internal combustion engine. If it is neglected, the analysis will just depend on the ideal air standard cycle.

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Nomenclature

a_p	constant, defined in Eq. (4)
b	friction-like term loss, $b = \mu(Ncx_2)^2$
b_v	constant, defined in Eq. (5)
C_{pm}	molar specific heat at constant pressure
C_{vm}	molar specific heat at constant volume
c	constant, defined in Eq. (21); $c = 2$ for a two stroke engine cycle, while $c = 4$ for a four stroke engine cycle
f_μ	friction force, defined in Eq. (20)
k	specific heat ratio, $k = C_{pm}/C_{vm}$
k_1	constant, defined in Eqs. (4) and (5)
L	the total distance the piston travels per cycle
m_a	mass of air per cycle
m_f	mass of fuel per cycle
N	cycles per second
P	net actual power output of the cycle, defined in Eq. (22)
P_R	power output without friction losses, defined in Eq. (19)
P_μ	lost power due to friction, defined in Eq. (21)
Q_{fuel}	total energy of the fuel per second input into the engine
Q_{LHV}	lower heating value of the fuel
Q_{leak}	heat leakage per second
Q_{in}	heat input
Q_{out}	heat reject
R	gas constant of working fluid
T	temperature
T_1, T_2, T_3, T_4, T_5	temperatures at state points 1, 2, 3, 4, 5
V	volume
v	piston's velocity
$\bar{v}$	piston's mean-velocity
x	piston's displacement
x_1	piston position corresponding to the volume V_1 of the trapped gases
x_2	piston position corresponding to the volume V_2 of the trapped gases

Greek symbols

α	heat leakage percentage
γ	another compression ratio, $\gamma = V_5/V_1$
γ_c	compression ratio, $\gamma_c = V_1/V_2$
η	efficiency of the cycle
λ	excess air coefficient
μ	coefficient of friction

Some effort has been paid to analyze the effects of heat transfer losses on the performance of internal combustion engines [6–10]. Klein [6] examined the effect of heat transfer through a cylinder wall on work output of the Otto and Diesel cycles. Chen et al. [7,8] derived the relations between net power output and the efficiency of the Diesel and Otto cycles with considerations of heat loss through the cylinder wall. Hou [9] studied the effect of heat transfer through a cylinder wall on performance of the Dual cycle.

Besides heat loss, friction has a significant effect on the performance but it is omitted in ideal engine cycles. Taking into account the friction loss of the piston, Angulo-Brown et al. [10], Chen et al. [11] and Wang et al. [12] modeled Otto, Diesel and Dual cycles with friction-like loss, respectively. Furthermore, Chen et al. [13,14] and Ge et al. [15] derived the characteristics of power and efficiency for Otto, Dual and Miller cycles with considerations of heat transfer and friction-like term losses. The above studies [6–15] were done without considering the variable specific heats of working fluid. However, in the real engine cycle, the specific heat of working fluid is not a constant and should be considered in practice cycle analysis [16–19].

In those studies [6–19], the heat addition process for an air standard cycle has been widely described as subtraction from the fuel's chemical energy of an arbitrary heat loss parameter times the average temperature of the heat adding period. In other words, 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. If they are selected arbitrarily, they will present unrealistic results and make the air standard cycles unfeasible [20]. For this reason, a more realistic and precise relationship between the fuel's chemical energy and the heat leakage needs to be derived through the resulting temperature [20]. Therefore, 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.

However, Ozsoysal's study [20] was only focused on the temperature limitations and no performance analysis was presented. Moreover, his study was done without considering the effects of variable specific heats of working fluid and friction. In particular, no performance analysis is available in the literature with emphasis on the Miller cycle with considerations of variable specific heats of working fluid, friction and heat leakage characterized by a percentage of fuel's energy.

In the present study, we aim at examining these effects (i.e., variable specific heats of working fluid, friction and heat loss characterized by a percentage of fuel's energy) on the net work output and the indicated thermal efficiency of an air standard Miller cycle. We relax the assumptions that there are not heat losses during combustion, that there are not friction losses of the piston for the cycle, and that specific heats of working fluid are constant. In other words, heat transfer between the working fluid and the environment through the cylinder wall is considered and characterized by a percentage of fuel's energy; friction loss of the piston on the power output is taken into account.

Moreover, we consider the variable specific heats of working fluid that is significant in practice cycle analysis. The results obtained in the study may offer good guidance for the design and operation of the Miller-cycle engine.

2. Thermodynamic analysis

Fig. 1 shows the limitation of the maximum cycle temperature due to heat leakage in temperature-entropy diagram of an air standard Miller cycle model. Thermodynamic cycle 1–2–3′–4′–5′–1 denotes the air standard Miller cycle without heat leakage, while cycle 1–2–3–4–5–1 designates the air standard Miller cycle with heat leakage. Process 1–2 is an isentropic compression. 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 4–5 and an isobatic 5–1 heat rejection processes. The heat added to the working fluid per unit mass is due to combustion. The temperature at the completion of constant-volume combustion (T_3) depends on the heat input due to combustion and 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 [20]. 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 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, then the total energy of the fuel per second input into the engine can be given by

$$Q_{\text{fuel}} = N m_f Q_{\text{LHV}} \tag{1}$$

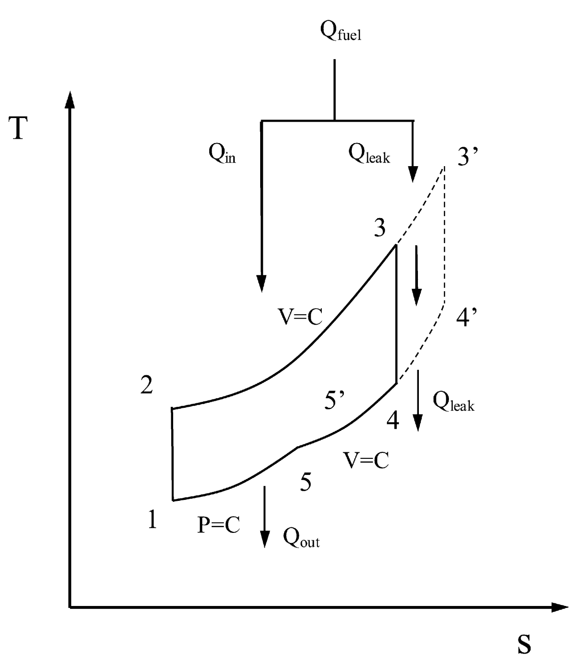

Fig. 1. $T-s$ diagram of an air standard Miller cycle model.

and thus the heat leakage per second is

$$Q_{\text{leak}} = \alpha Q_{\text{fuel}} = \alpha N m_f Q_{\text{LHV}} \tag{2}$$

where m_f is the delivered fuel mass into the cylinder, Q_{LHV} is the lower heating value of the fuel and α 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},

$$Q_{\text{in}} = Q_{\text{fuel}} - Q_{\text{leak}} = (1-\alpha) N m_f Q_{\text{LHV}} \tag{3}$$

In practical internal combustion engine cycle, constant-pressure and constant-volume specific heats of the working fluid are variable and these variations will greatly affect the performance of the cycle. According to Ref. [16], it can be assumed that the specific heats of the working fluid are functions of temperature alone and have the following linear forms:

$$C_{pm} = a_p + k_1 T \tag{4}$$

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 as T_3, the available energy Q_{in} during the heat addition per second can be written as

$$Q_{\text{in}} = N m_a \int_{T_2}^{T_3} C_{vm}\, \mathrm{d}T = N m_a \left[b_v (T_3 - T_2) + 0.5 k_1 \left(T_3^2 - T_2^2\right)\right] \tag{7}$$

Combining Eqs. (3) and (7), and then dividing by the amount of air mass m_a, we have

$$\alpha = 1 - \frac{\lambda (m_a/m_f)_s}{Q_{\text{LHV}}} \left[b_v (T_3 - T_2) + 0.5 k_1 \left(T_3^2 - T_2^2\right)\right] \tag{8}$$

or

$$T_2 = \frac{-b_v + \sqrt{b_v^2 + 2k_1 [0.5 k_1 T_3^2 + b_v T_3 - (1-\alpha) \frac{Q_{\text{LHV}}}{\lambda (m_a/m_f)_s}]}}{k_1} \tag{9}$$

where λ is the excess air coefficient defined as $\lambda = (m_a/m_f)/(m_a/m_f)_s$, (m_a/m_f) is air-fuel ratio and the subscripts a, f, and s, respectively, denotes air, fuel and the stoichiometric condition.

The first condition for realizing a feasible cycle is $T_2 \leqslant T_3 (= T_{\max})$, so that

$$\alpha \leqslant 1 \tag{10}$$

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

$$\alpha \geqslant 1 - \frac{\lambda(m_a/m_f)_s}{2Q_{LHV}}\left[k_1\left(T_3^2 - T_1^2\right) + 2b_v(T_3 - T_1)\right] \tag{11}$$

Hence, the minimum value of α is expressed as

$$\alpha_{min} = 1 - \frac{\lambda(m_a/m_f)_s}{2Q_{LHV}}\left[k_1\left(T_3^2 - T_1^2\right) + 2b_v(T_3 - T_1)\right] \tag{12}$$

The heat rejected per second by the working fluid (Q_{out}) during processes $4 \to 5 \to 1$ is

$$\begin{aligned} Q_{out} &= Nm_a\left[\int_{T_5}^{T_4} C_{vm}\,dT + \int_{T_1}^{T_5} C_{pm}\,dT\right] \\ &= Nm_a\left[b_v(T_4 - T_5) + a_p(T_5 - T_1) + 0.5k_1\left(T_4^2 - T_1^2\right)\right] \end{aligned} \tag{13}$$

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 process with constant k cannot be used in reversible adiabatic process with variable k. According to Ref. [16], however, a suitable engineering approximation for reversible adiabatic process with variable k can be made, i.e., this process can be divided into infinitesimally small processes, for each of these processes, adiabatic exponent k can be regarded as constant. For instance, for any reversible adiabatic process between states i and j, we can regard the process as that it consists of numerous infinitesimally small processes with constant k. For any of these processes, when small changes in temperature dT, and in volume dV of the working fluid take place, the equation for reversible adiabatic process with variable k can be written as follows:

$$TV^{k-1} = (T + dT)(V + dV)^{k-1} \tag{14}$$

From Eq. (14), we get the following equation

$$k_1(T_j - T_i) + b_v \ln(T_j/T_i) = -R\ln(V_j/V_i) \tag{15}$$

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

$$k_1(T_2 - T_1) + b_v \ln(T_2/T_1) = R\ln\gamma_c \tag{16}$$

and

$$k_1(T_3 - T_4) + b_v \ln(T_3/T_4) = R\ln(\gamma \cdot \gamma_c) \tag{17}$$

where the another compression ratio (γ) is defined as $\gamma = V_5/V_1 = T_5/T_1$. From the definition of γ, then T_5 can be expressed as

$$T_5 = \gamma \cdot T_1 \tag{18}$$

The power output without friction losses is given by:

$$\begin{aligned} P_R &= Q_{in} - Q_{out} \\ &= Nm_a\left[b_v(T_3 + T_5 - T_2 - T_4) - a_p(T_5 - T_1)\right. \\ &\quad \left. + 0.5k_1\left(T_3^2 + T_1^2 - T_2^2 - T_4^2\right)\right] \end{aligned} \tag{19}$$

Considering the friction of the piston and assuming a dissipation term represented by a friction force (f_μ) which is linearly proportional to the velocity of the piston [10–12], then we have

$$f_\mu = -\mu v = -\mu\frac{dx}{dt} \tag{20}$$

where μ 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

$$\begin{aligned} P_\mu &= f_\mu v = -\mu\left(\frac{dx}{dt}\right)^2 = -\mu v^2 = -\mu(LN)^2 \\ &= -\mu\left[Ncx_2(\gamma_c - 1)\right]^2 \end{aligned} \tag{21}$$

where L is the total distance the piston travels per cycle, x_2 is the piston's position corresponding to the volume V_2, and c is a constant ($c = 2$ for a two stroke engine cycle, while $c = 4$ for a four stroke engine cycle).

Accordingly, the net actual power output of the Miller-cycle engine can be written as:

$$\begin{aligned} P &= P_R - |P_\mu| \\ &= Nm_a\left[b_v(T_3 + T_5 - T_2 - T_4) - a_p(T_5 - T_1)\right. \\ &\quad \left. + 0.5k_1\left(T_3^2 + T_1^2 - T_2^2 - T_4^2\right)\right] - b(\gamma_c - 1)^2 \end{aligned} \tag{22}$$

where $b = \mu(Ncx_2)^2$. Also, the efficiency of the Miller-cycle engine can be expressed by:

$$\begin{aligned} \eta &= \frac{P}{Q_{in}} \\ &= \left\{Nm_a\left[b_v(T_3 + T_5 - T_2 - T_4) - a_p(T_5 - T_1)\right.\right. \\ &\quad \left.\left. + 0.5k_1\left(T_3^2 + T_1^2 - T_2^2 - T_4^2\right)\right] - b(\gamma_c - 1)^2\right\} \\ &\quad \times \left\{Nm_a\left[b_v(T_3 - T_2) + 0.5k_1\left(T_3^2 - T_2^2\right)\right]\right\}^{-1} \end{aligned} \tag{23}$$

When T_1, T_3, γ_c and γ are given, then T_2 can be obtained from Eq. (16), T_4 can be found from Eq. (17) and T_5 can be got from Eq. (18). Finally, by substituting T_1, T_2, T_3, T_4 and T_5 into Eqs. (22) and (23), respectively, the power output and the efficiency of the Miller-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

In the calculations, in order to approach to the real engine conditions, we use the same values of the coefficients a_p, b_v, and k_1 (related to the variable specific heats of the working fluid), and the piston position (x_2) at minimum volume of the trapped gases as in Refs. [16,19,21]. Also, in the analysis, the fuel used is gasoline with a lower heating value (Q_{LHV}) of 44 000 kJ kg^{-1} [20], and the same value of N (cycles per second) reported by Chen et al. [19] is used. In addition, we use the suitable value of mass of gas in the power stroke [4] and the appropriate ranges of intake temperature and friction loss in the cycle [16,19,21]. Therefore, according to those references [4,16,19–21], the following constants and ranges of parameters are chosen in the calculations to illustrate the proceeding analysis which can achieve a reasonable maximum gas temperature of 1500 to 3000 K and obtain a feasible range of temperatures in the cycle analysis, as observed in practice: $b_v = 0.6858-0.8239\ \text{kJ}\,\text{kg}^{-1}\,\text{K}^{-1}$, $m_a = 1.26 \times 10^{-3}$ kg, $k_1 = 0.000133-0.00034\ \text{kJ}\,\text{kg}^{-1}\,\text{K}^{-2}$, $Q_{\text{LHV}} = 44\,000\ \text{kJ}\,\text{kg}^{-1}$, $b = 0.018-0.03$ kW, $N = 30$, $T_1 = 300-400$ K, and $x_2 = 0.01$ m. In this study, we focus on the limitation of maximum cycle temperature T_3 instead of $T_{3'}$ due to the varying heat leakage conditions. Numerical examples are shown as follows.

The variations of the heat leakage percentage (α) with respect to the maximum cycle temperature (T_3) and the volumetric compression ratio (γ_c) are shown in Fig. 2. It can be seen that the maximum cycle temperature plays a dominant role on the quantity of heat leakage. For a fixed compression ratio, the maximum cycle temperature increases with decreasing heat leakage percentage parameter. To achieve a fixed cycle maximum temperature, the heat leakage percentage parameter increases with the increase of compression ratio. Note that some values of the heat leakage percentage might be insufficient for a feasible Miller cycle. Consequently, acceptable values for α could only be achieved from the definition of $\alpha_{\min}$ given by Eq. (12). These significant characteristics are similar to those of Ozsoysal's study [20].

The variations of the heat leakage percentage (α) with respect to the air–fuel ratio (m_a/m_f)or the excess air coefficient (λ) and the volumetric compression ratio (γ_c) are demonstrated in Fig. 3. It can be found that the excess air coefficient also plays an important role on the quantity of heat leakage. To obtain a fixed maximum cycle temperature ($T_3 = 1900$ K), the heat leakage percentage parameter increases with the increase of compression ratio. For fixed maximum cycle temperature and compression ratio, heat leakage percentage parameter decreases with increasing excess air coefficient. Similar to Fig. 2, suitable values for α could only be achieved from the definition of $\alpha_{\min}$ given by Eq. (12).

Fig. 4 illustrates the influence of maximum cycle temperature (T_3) on the cycle performance. The power output given by Eq. (22) is a convex function with a single maximum for the optimum compression ratio, as shown in Fig. 4(a). Increase in compression ratio first results in an increase in power output, and after reaching a peak, then the net power output decreases dramatically with further increase in compression ratio. It can be seen from Fig. 4(b) that the behavior of the efficiency versus compression ratio plot is similar to that for the power output. Fig. 4 also depicts that increasing T_3 corresponds to enlarging the amount of heat added to the engine due to combustion, and, therefore, T_3 has a positive effect on the $P-\gamma_c$ and $\eta-\gamma_c$ characteristic curves. In other words, for a given γ_c the power output and efficiency increase with the increase of T_3, and the maximum power output and its corresponding efficiency increase with the increase of T_3. Additionally, it is found that the values of γ_c at the maximum power output or at the maximum efficiency increase with the increase of T_3.

It has been reported that for a real heat engine, the maximum power and maximum efficiency operating points are usually relatively close [21]. This is reflected through loop-shaped power versus efficiency plots. As shown in Fig. 5, we also obtain the loop-shaped power output versus efficiency curves which reflect the performance characteristics of a real irreversible Miller-cycle engine. It can be 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.

Figs. 6 and 7 demonstrate the influence of the parameter b_v related to the variable specific heats of the working fluid on the performance of the Miller cycle. For a fixed k_1, a larger b_v corresponds to a greater value of the specific heat with constant

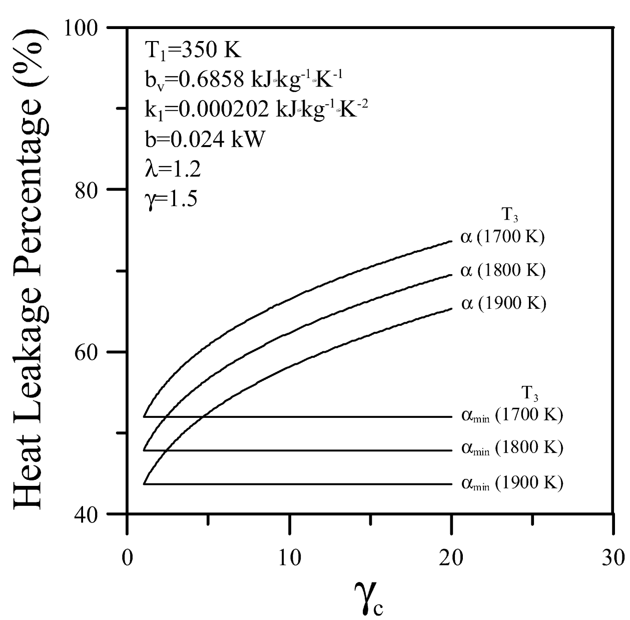

Fig. 2. The variation of the heat leakage percentage (α) with respect to the maximum cycle temperature (T_3) and the volumetric compression ratio (γ_c).

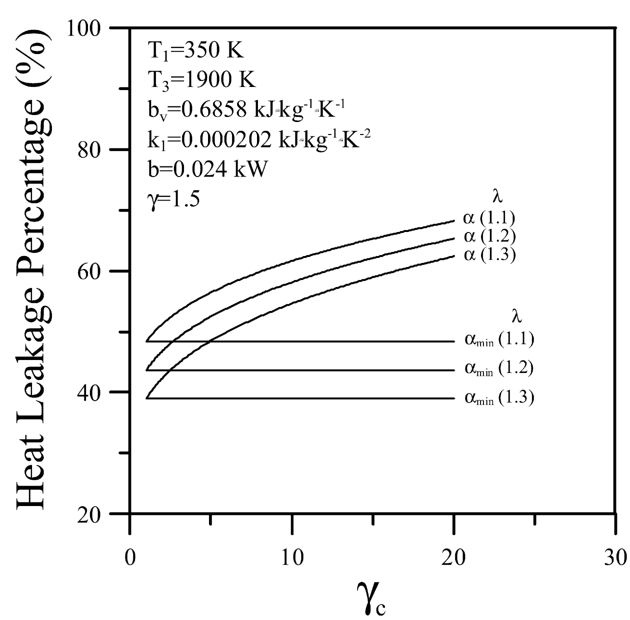

Fig. 3. The variation of the heat leakage percentage (α) with respect to the air–fuel ratio (m_a/m_f) or the excess air coefficient (λ) and the volumetric compression ratio (γ_c).

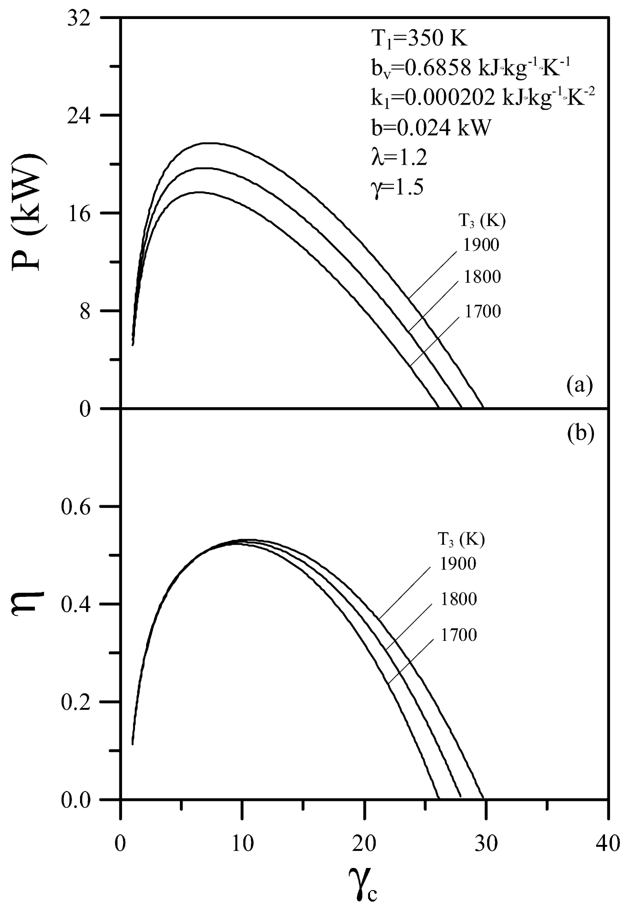

Fig. 4. (a) The influence of maximum cycle temperature (T_3) on the variation of power output (P) with compression ratio (γ_c); (b) The influence of maximum cycle temperature (T_3) on the variation of efficiency (η) with compression ratio (γ_c).

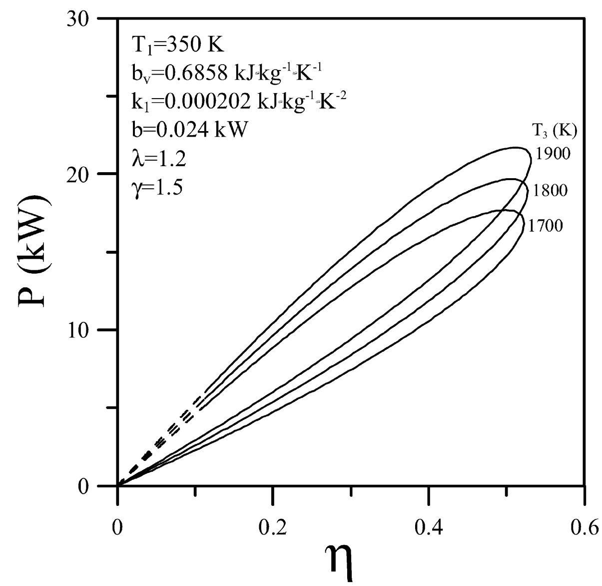

Fig. 5. The influence of maximum cycle temperature (T_3) on the power output (P) versus efficiency (η) characteristic curves.

volume (C_{vm}) or the specific heat with constant pressure (C_{pm}). Fig. 6(a) shows that for a given γ_c, the maximum power output and the working range of the cycle increase with the increase of b_v, nevertheless, Fig. 6(b) indicates that the maximum efficiency decreases with increasing b_v. It is noteworthy that the parameter b_v has a significant influence on the compression ratio where the maximum power or efficiency takes place. The values of γ_c at the maximum power output or at the maximum efficiency increase with the increase of b_v, as shown in Fig. 6. It can be observed from Fig. 7 that the curves of power output versus efficiency are also loop-shaped. It reveals that, 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.

Figs. 8 and 9 display the influence of the parameter k_1 related to the variable specific heats of the working fluid on the performance of the Miller 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}). It can be found from Fig. 8 that k_1 has the same influence as b_v (shown in Fig. 6) on the performance of the cycle. In other words, for a given γ_c, the power output and the working range of the cycle increase with the increase of k_1, as depicted

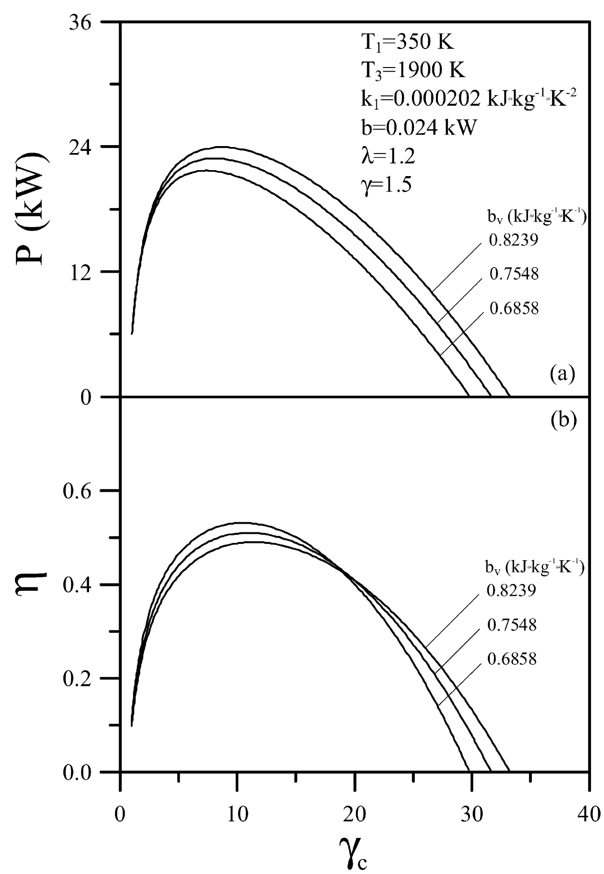

Fig. 6. (a) The influence of b_v on the variation of power output (P) with compression ratio (γ_c); (b) The influence of b_v on the variation of efficiency (η) with compression ratio (γ_c).

in Fig. 8(a), while, the efficiency decreases with the increase of k_1, as shown in Fig. 8(b). Additionally, it can be observed that the parameter k_1 has a significant influence on the loop-shaped curves for the power output versus efficiency plots. With in-

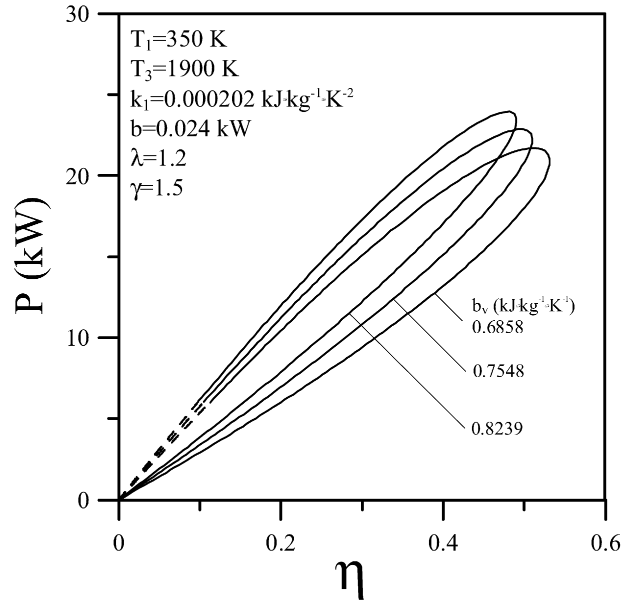

Fig. 7. The influence of b_v on the power output (P) versus efficiency (η) characteristic curves.

creasing 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. 9.

Figs. 10 and 11 show the influence of the friction-like term loss (b) on the performance of the Miller cycle. It is clear that the parameter b has a negative effect on the performance. The maximum power output, the maximum efficiency and the value of the compression ratio at maximum power output or at maximum efficiency will decrease with increasing b, as shown in Fig. 10. Furthermore, Fig. 11 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 b.

Note that the trends for power versus efficiency obtained in this study (as shown in Figs. 6–11) are similar to those reported in Refs. [16–19]. That is, the effects of heat loss, friction and variable specific heats of working fluid on the performance of an air standard cycle are obvious and significant. However, in those studies [16–19], the heat transfer to the cylinder walls was assumed to be a linear function of the difference between the average gas and cylinder wall temperatures during the energy release process. Because the heat leakage parameter and the fuel's energy depend on each other, their valid ranges given in the literature strongly affect the feasibility of air standard cycles. In other words, their unsuitable values will yield unrealistic results and then make the air standard cycles unfeasible, e.g. unrealistic maximum cycle temperature, unacceptable low maximum cycle temperature, and the temperature after the compression stroke higher than the maximum cycle temperature [20]. It is noteworthy that, in this study, the performance of an air standard Miller cycle is analyzed by considering a more realistic and precise relationship between the fuel's chemical energy and the heat leakage that is constituted on a pair of inequalities and is derived through the resulting temperature under the restriction of maximum cycle temperature. As a result, the minimum and maximum heat leakage levels are so formulated that their valid ranges restrict and satisfy the operational characteristics of Miller cycles.

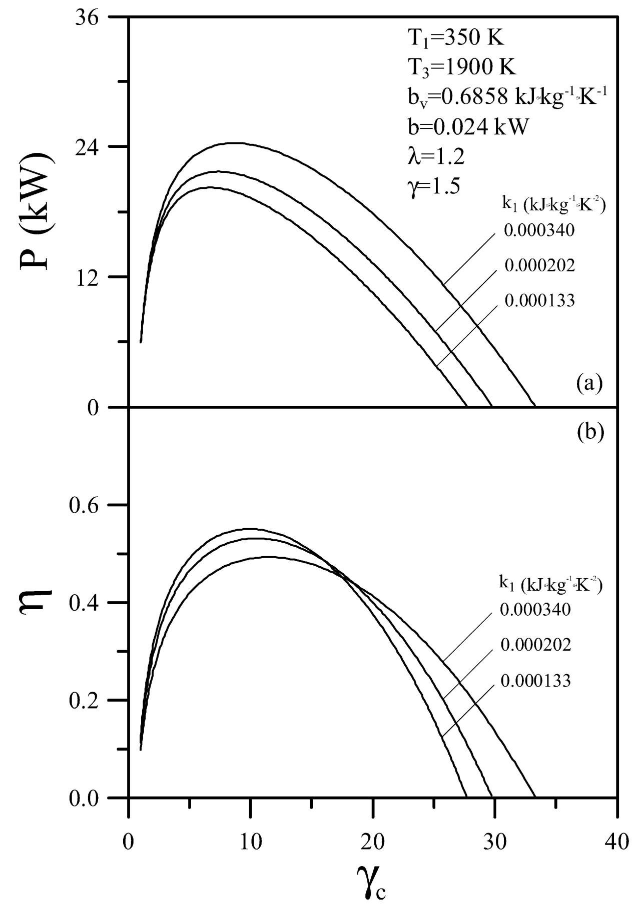

Fig. 8. (a) The influence of k_1 on the variation of power output (P) with compression ratio (γ_c); (b) The influence of k_1 on the variation of efficiency (η) with compression ratio (γ_c).

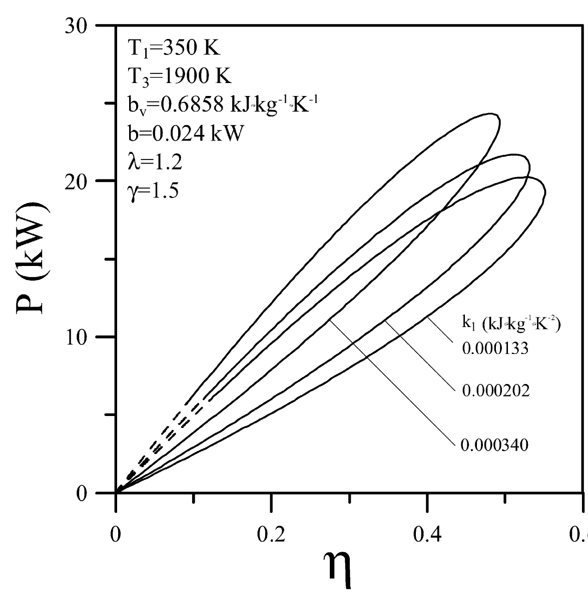

Fig. 9. The influence of k_1 on the power output (P) versus efficiency (η) characteristic curves.

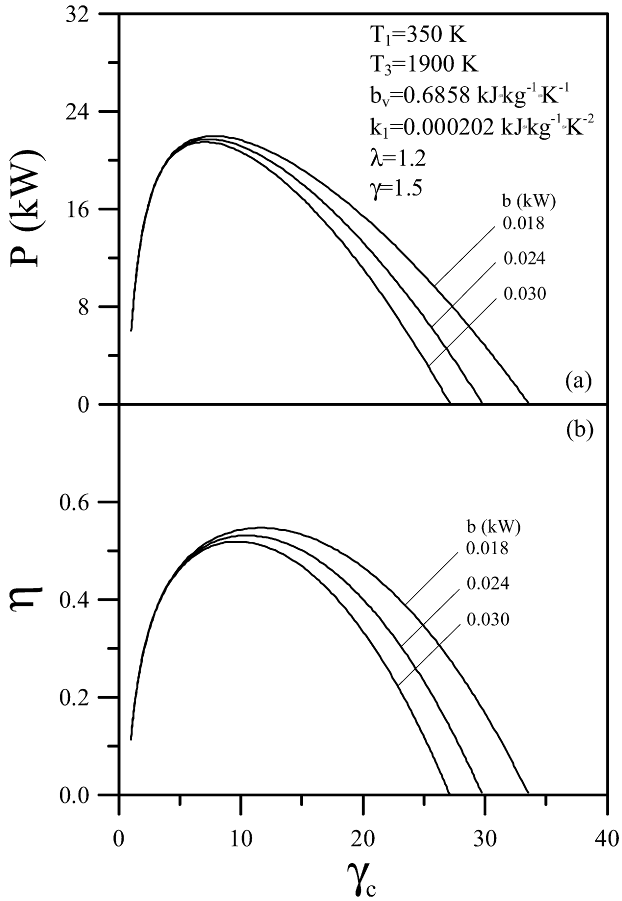

Fig. 10. (a) The influence of b on the variation of power output (P) with compression ratio (γ_c); (b) The influence of b on the variation of efficiency (η) with compression ratio (γ_c).

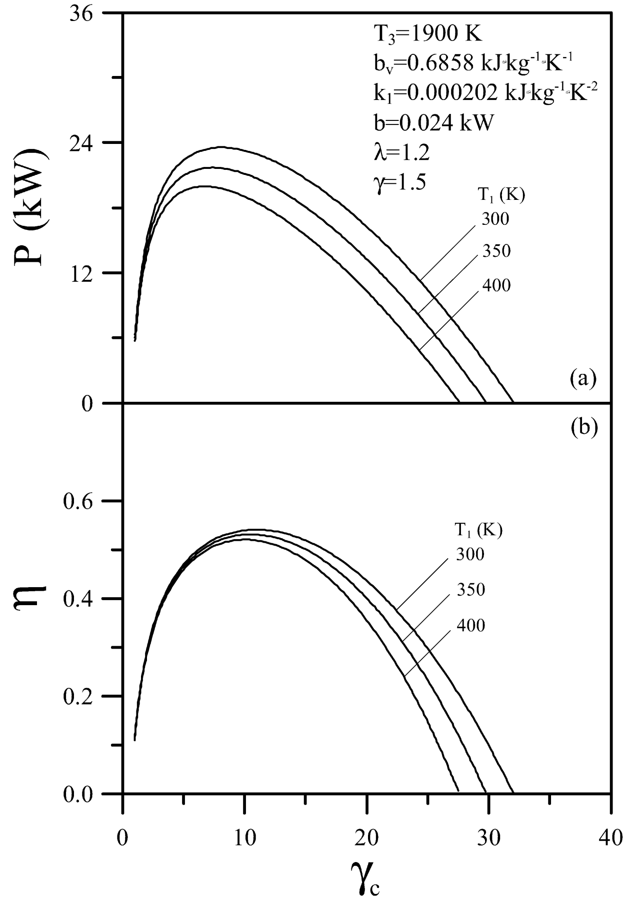

Fig. 12. (a) The influence of intake temperature (T_1) on the variation of power output (P) with compression ratio (γ_c); (b) The influence of intake temperature (T_1) on the variation of efficiency (η) with compression ratio (γ_c).

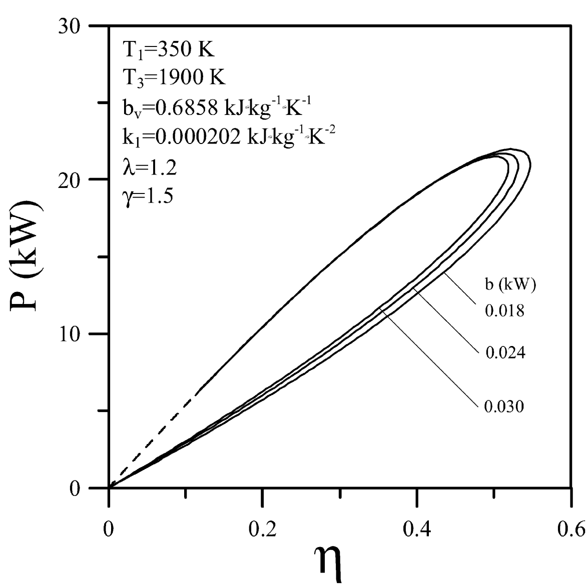

Fig. 11. The influence of b on the power output (P) versus efficiency (η) characteristic curves.

Figs. 12 and 13 depict the influence of intake temperature (T_1) on the performance of the Miller cycle. Fig. 12 displays that for a restricted the maximum cycle temperature $T_3 = 1900$ K, the maximum power output, the maximum efficiency, the compression ratio at maximum power output and the compression ratio at maximum efficiency of the Miller cycle decrease with the increase of T_1. Fig. 13 also demonstrates loop-shaped curves of power versus efficiency plots. Furthermore, it can be 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.

Figs. 14 and 15 show the effects of another compression ratio (γ) on the performance of Miller cycle. Note that for a special case, when $\gamma = 1$, Miller cycle becomes Otto cycle. One can see that the maximum power output, the maximum efficiency, the efficiency at maximum power output and the power output at maximum efficiency increase with the increase of γ. While the value of the compression ratio (γ_c) when the power output or the efficiency is maximum decreases as γ increases. Additionally, comparison of the performance of air-standard Miller and Otto cycles are also discussed. It can be found from Figs. 14 and 15 that Miller cycle has larger maximum power output and maximum efficiency than Otto cycle does. In other words, the Miller cycle is more efficient than the Otto cycle.

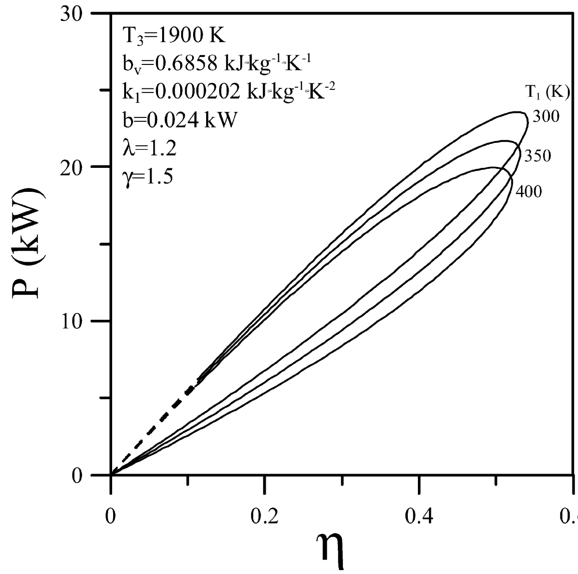

Fig. 13. The influence of intake temperature (T_1) on the power output (P) versus efficiency (η) characteristic curves.

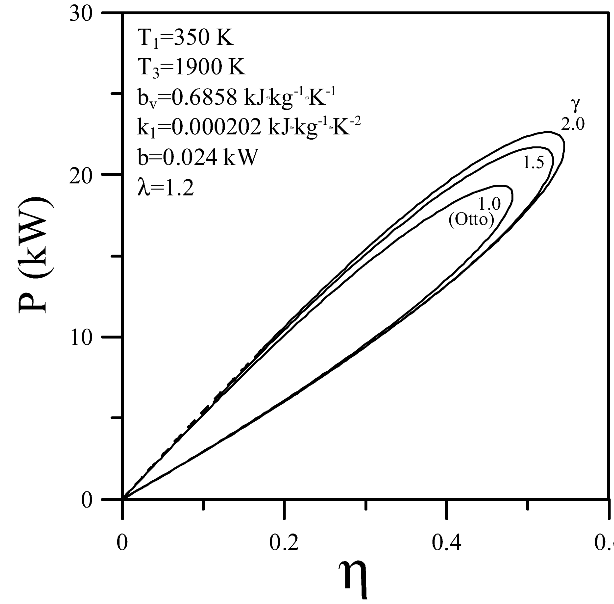

Fig. 15. Comparison of the performance of air-standard Miller and Otto cycles: the power output (P) versus efficiency (η) characteristic curves.

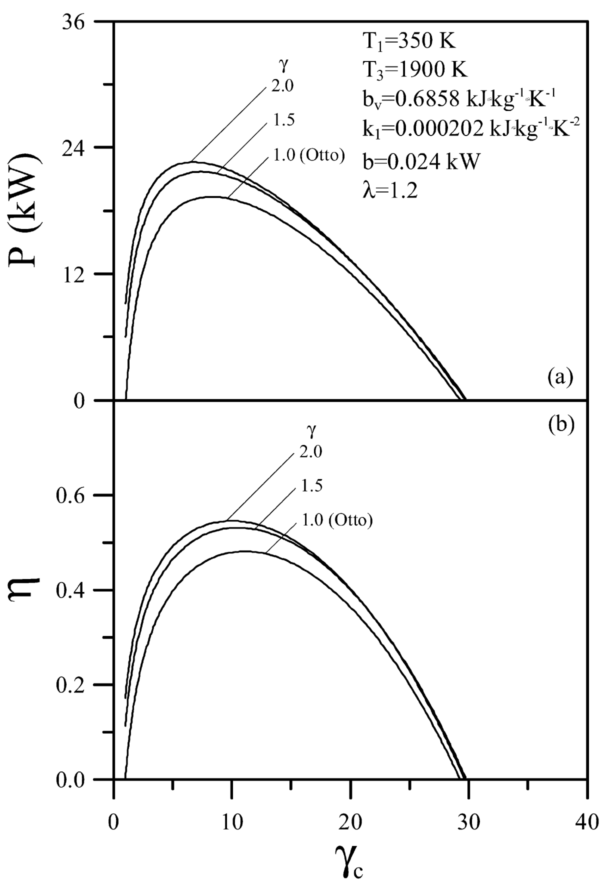

Fig. 14. Comparison of the performance of air-standard Miller and Otto cycles: (a) the power output (P) versus compression ratio (γ_c); (b) the efficiency (η) versus compression ratio (γ_c).

4. Conclusions

The effects of heat loss as a percentage of fuel's energy, friction and variable specific heats of working fluid on the performance of a Miller-cycle engine under the restriction of maximum cycle temperature are presented in this study. The results can be 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 Miller 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 γ_c, the power output and the working range 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 or 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.
(3) The influence of the friction-like term loss b 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 b.
(4) The maximum power output, the maximum efficiency, the compression ratio at maximum power output and the compression ratio at maximum efficiency of the Miller 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) The maximum power output, the maximum efficiency, the efficiency at maximum power output and the power output at maximum efficiency increase with the increase of another compression ratio γ. While the value of the com-

pression ratio (γ_c) when the power output or the efficiency is maximum decreases as γ increases. Furthermore, the Miller cycle can be found to be more efficient than the Otto cycle.

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

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