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Comparison of Performances for Air-Standard Atkinson and Dual Combustion Cycles with Heat Transfer Considerations

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There are heat losses during the cycle of real engine that are neglected in ideal air-standard analysis. In this paper, the effect of heat transfer on the net output work is shown and thermal efficiency of the air-standard Atkinson and the Dual combustion cycles are analyzed. Comparison of performances of the air-standard Atkinson and the Dual combustion cycles with heat transfer considerations are also discussed. We assumed that the compression and power processes are adiabatic and reversible and any convective, conductive and radiative heat transfer to cylinder wall during the heat rejection process may be ignored. The heat loss through the cylinder wall is assumed to occur only during combustion and is further assumed to be proportional to average temperature of both the working fluid and cylinder wall. The results show that the net work output versus efficiency and the maximum net work output and corresponding efficiency bounds are influenced by the magnitude of heat transfer. The results are of importance to provide guidance for the performance evaluation of practical engines.

Keywords: Thermodynamics, Efficiency, Atkinson cycle, Dual combustion cycle, Heat transfer

1 Introduction

The Atkinson cycle engine is a type of internal combustion engines, which was designed and built by James Atkinson in (1882). The cycle is also called the Sargent cycle by several physic-oriented thermodynamic books. By the use of clever mechanical linkages, Atkinson's engine carried the expansion branch farther than any other existing engines. The Dual combustion cycle is another type of internal combustion engine. In recent years, many attentions have been paid to analyzing the performance of internal combustion cycles. Comparison of performances for the air-standard Atkinson and the Otto cycles with heat

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transfer considerations and heat transfer effects on the performance of an air-standard Dual combustion cycle did by Hou [1] and [2]. Performance analyzing and parametric optimum criteria of an irreversible Atkinson heat-engine did by Zhao and Chen [3]. Optimization of the Dual combustion cycle considering the effect of combustion on power did by Chen [4]. Performance of an endoreversibe Atkinson cycle with variable specific heat ratio of working fluid did by Ebrahimi [5]. The results obtained in this work can help us to understand how the net-work output and efficiency are influenced by heat transfer during combustion, or the constant volume heat addition process.

2 Cycle model

The T-s diagrams of the Atkinson and the Dual combustion cycles are shown in Figures (1.a) and (1.b). In the Atkinson cycle, the heat added in the isochoric process $(2 \rightarrow 3)$ and the heat rejected in the isobaric process $(4 \rightarrow 1)$. In the Dual combustion cycle the heat added in the isochoric and isobaric process $(2 \rightarrow 3)$ and $(3 \rightarrow 4)$, the heat rejected in the isochoric process $(5 \rightarrow 1)$.

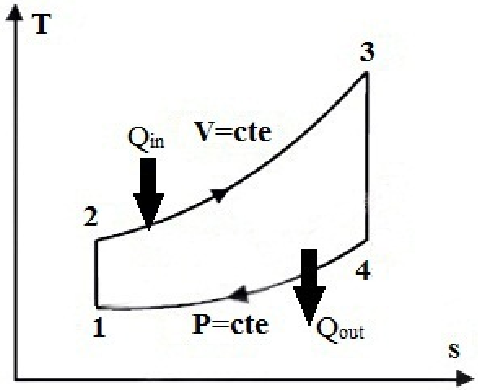

Figure 1.a T-s diagram for the Atkinson cycle

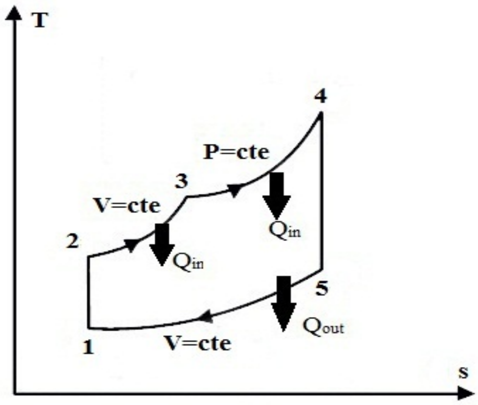

Figure 1.b T-s diagram for the Dual combustion cycle

2. 1 Thermodynamics analysis of the air-standard Atkinson cycle

Following the assumption described above, process $(1 \rightarrow 2)$ is an isentropic compression from bottom dead center (BDC) to top dead center (TDC). The heat addition takes place in process $(2 \rightarrow 3)$, which is isochoric. The isentropic expansion process $(3 \rightarrow 4)$, is the power or expansion stroke. The cycle is completed by an isobaric heat rejection process $(4 \rightarrow 1)$. The heat added to the working fluid per unit mass is due to combustion. Assuming constant specific heats, the net work output per unit mass of the working fluid is given by the first-law of thermodynamics:

$$w = C_v(T_3 - T_2) - C_p(T_4 - T_1), \tag{1}$$

Where C_p and C_v are the constant pressure and constant volume specific heat, respectively; and T_1, T_2, T_3 and T_4 are the absolute temperatures at states 1, 2, 3 and 4. For the isentropic process $(1 \rightarrow 2)$ and $(3 \rightarrow 4)$, we have

$$T_2 = T_1 r_c^{\,k-1}, \tag{2}$$

$$\frac{T_4}{T_3} = (\frac{V_3}{V_4})^{k-1} = (\frac{V_3}{V_1})^{k-1}(\frac{V_1}{V_4})^{k-1} = (\frac{V_2}{V_1})^{k-1}(\frac{V_1}{V_4})^{k-1} = r_c^{\,1-k}(\frac{V_1}{V_4})^{k-1}, \tag{3}$$

Where r_c is the compression ratio (V_1/V_2) and k is the specific heat ratio (C_p/C_v). Additionally, since process $(4 \rightarrow 1)$ is isobaric, we have

$$\frac{V_1}{V_4} = \frac{T_1}{T_4} \tag{4}$$

Substitution of equations (2) and (4) into equation (3) yields

$$T_4 = T_1(\frac{T_3}{T_2})^{\frac{1}{k}} \tag{5}$$

The heat added per unit mass of the working fluid during the constant volume process $(2 \rightarrow 3)$ per cycle is represented by the first-law of thermodynamics:

$$q_{in} = C_v(T_3 - T_2) \tag{6}$$

Also the heat added to the working fluid during the constant volume combustion process can be given in the following linear expression [2]:

$$q_{in} = \alpha - \beta(T_3 + T_2) \tag{7}$$

Where α, β are two constants related to combustion process and heat transfer, respectively [1]

Combining equations (6) and (7) yields

$$T_3 = \frac{\alpha + (C_v - \beta)T_2}{C_v + \beta} \tag{8}$$

Substitution of equation (2) into equation (8) gives

$$T_3 = \frac{[\alpha + (C_v - \beta)T_1 r_c^{\,k-1}]}{C_v + \beta} \tag{9}$$

Substituting of equations (2) and (9) into equation (5) gives T_4 as a function of T_1

$$T_4 = \left\{\frac{[\alpha T_1^{-1} r_c^{\,1-k} + (C_v - \beta)]}{(C_v + \beta)}\right\}^{\frac{1}{k}} T_1 \tag{10}$$

By combining the results obtained from equations (2), (9) and (10) into equation (1), the net work output per unit mass of the working fluid can be expressed in terms of T_1 as

$$w = C_v \left\{ \frac{\alpha + (C_v - \beta) T_1 r_c^{k-1}}{C_v + \beta} - r_c^{k-1} T_1 \right\} - C_p T_1 \left\{ \left[\frac{\alpha T_1^{-1} r_c^{1-k} + (C_v - \beta)}{(C_v + \beta)} \right]^{\frac{1}{k}} - 1 \right\} \tag{11}$$

Similarly, substitution equations (2), (9) and (10) into equation (7) yields

$$q_{in} = \alpha - \beta \left\{ r_c^{k-1} T_1 + \left[\frac{\alpha + (C_v - \beta) r_c^{k-1} T_1}{C_v + \beta} \right] \right\} \tag{12}$$

Equation (11) divided by equation (12) gives the indicated thermal efficiency,

$$\eta = \frac{w}{q_{in}} = \frac{C_v \left\{ \frac{\alpha + (C_v - \beta) T_1 r_c^{k-1}}{C_v + \beta} - r_c^{k-1} T_1 \right\} - C_p T_1 \left\{ \left[\frac{\alpha T_1^{-1} r_c^{1-k} + (C_v - \beta)}{(C_v + \beta)} \right]^{\frac{1}{k}} - 1 \right\}}{\alpha - \beta \left\{ r_c^{k-1} T_1 + \left[\frac{\alpha + (C_v - \beta) r_c^{k-1} T_1}{C_v + \beta} \right] \right\}} \tag{13}$$

Then, differentiating with respect to r_c and seeking a maximum work output, w_{max}, by setting

$$\frac{dw}{dr_c} = 0 \tag{14}$$

We finally get

$$a_1 r_c^{2k} + a_2 r_c^{1+k} - a_3 = 0, \tag{15}$$

$$a_1 = \frac{C_v - \beta}{C_v + \beta}, \tag{16}$$

$$a_2 = \frac{\alpha}{(C_v + \beta) T_1}, \tag{17}$$

$$a_3 = \left(\frac{1 - a_1}{a_2} \right)^{\frac{1}{1-k}} \tag{18}$$

Note that w_{max} can be obtained by substituting $r_c = r_{cm}$ into equation (11). Furthermore, the corresponding thermal efficiency at maximum work output η_m, can be obtained by substituting r_{cm} into equation (13). [1]

2. 2 Thermodynamics analysis of the air-standard Dual combustion cycle

Following the assumption described at the cycle model, Figure (1.b) shows the temperature-entropy (T-s) diagrams for the thermodynamic processes of an air-standard Dual combustion cycle. Process $(1 \rightarrow 2)$ is an isentropic compression from BDC to TDC. The heat addition takes place in two steps: process $(2 \rightarrow 3)$ is isochoric and process $(3 \rightarrow 4)$ isobaric. The isentropic expansion process $(4 \rightarrow 5)$ is the power or expansion stroke. The cycle completed by an isochoric heat rejection process $(5 \rightarrow 1)$. Assuming constant specific heats, the net work output per unit mass of the working fluid is given by the first-law of thermodynamics:

$$w = C_v (T_3 - T_2) + C_p (T_4 - T_3) - C_v (T_5 - T_1), \tag{19}$$

Where C_p and C_v are the constant pressure and constant volume specific heat, respectively; and T_1, T_2, T_3, T_4 and T_5 are the absolute temperatures at states 1, 2, 3, 4 and 5. For the isentropic processes $(1 \rightarrow 2)$ and $(4 \rightarrow 5)$, we have

$$T_2 = T_1 r_c^{\,k-1}, \tag{20}$$

$$T_5 = T_4 \left(\frac{r}{r_c} \right)^{k-1}, \tag{21}$$

Where r_c and r are the compression ratio (V_1 / V_2) and the cut-off ratio (V_4 / V_3), and k is the specific heat ratio (C_p / C_v). The overall heat input per unit mass of working fluid per cycle can be represented by the first-law of thermodynamics:

$$q_{in} = C_v (T_3 - T_2) + C_p (T_4 - T_3) \tag{22}$$

The heat added to the working fluid during the total combustion process can be given in the following linear expression [2]

$$q_{in} = \alpha - \beta (T_2 + T_3) + \alpha - \beta (T_3 + T_4) \tag{23}$$

Combining equations (22) and (23) yields

$$T_3 = \frac{\alpha + (C_v - \beta) T_2}{C_v + \beta} = \frac{\left[\alpha + (C_v - \beta) T_1 r_c^{k-1} \right]}{C_v + \beta}, \tag{24}$$

and

$$T_4 = \frac{\alpha + (C_p - \beta) T_3}{C_p + \beta} = \frac{\alpha + (C_p - \beta) \left[\alpha + (C_v - \beta) T_1 r_c^{k-1} \right] / (C_v + \beta)}{C_p + \beta} \tag{25}$$

Substituting of equation (24) into equation (21) gives T_5 as a function of T_1

$$T_5 = \left\{ \frac{\alpha + (C_p - \beta)\left[\alpha + (C_v - \beta) T_1 r_c^{k-1}\right] / (C_v + \beta)}{C_v + \beta} \right\} \left(\frac{r}{r_c} \right)^{k-1} \tag{26}$$

By combining the results obtained from equations (20), (24), (25) and (26) into equation (1), the net work output per unit mass of the working fluid can be expressed in terms of T_1 as

$$\begin{aligned} w = &\frac{C_v(\alpha - 2\beta r_c^{k-1} T_1)}{C_v + \beta} + \frac{\alpha C_p}{C_p + \beta} - \frac{2\beta C_p}{(C_p + \beta)} \left[\frac{\alpha + (C_v - \beta) r_c^{k-1} T_1}{(C_v + \beta)} \right] \\ &- C_v \left\{ \frac{(\alpha r^{k-1} r_c^{1-k})}{(C_p + \beta)} + \frac{r^{k-1}(C_p - \beta)}{(C_p + \beta)^2} \left[\alpha r_c^{1-k} + (C_v - \beta) T_1 \right] - T_1 \right\} \end{aligned} \tag{27}$$

Similarly, substituting equations (20), (24) and (25) into equation (23) yields

$$\begin{aligned} q_{in} = &\alpha - \beta \left\{ r_c^{k-1} T_1 + \left[\frac{\alpha + (C_v - \beta) r_c^{k-1} T_1}{C_v + \beta} \right] \right\} + \alpha - \beta \left[\frac{\alpha + (C_v - \beta) r_c^{k-1} T_1}{C_v + \beta} \right] \\ &- \beta \left\{ \frac{\alpha + (C_p - \beta)\left[\alpha + (C_v - \beta) r_c^{k-1} T_1\right] / (C_v + \beta)}{C_p + \beta} \right\} \end{aligned} \tag{28}$$

Equation (27) divided by equation (28) gives the indicated thermal efficiency:

$$\eta = \frac{w}{q_{in}} \tag{29}$$

Then, differentiating w with respect to r_c and seeking a maximum work output, w_{max}, by setting

$$\frac{\partial w}{\partial r_c} = 0 \tag{30}$$

We finally get

$$r_{cm} = \left[\frac{b_3 (b_4 + b_5)}{b_1 + b_2} \right]^{\frac{1}{(2k-2)}}, \tag{31}$$

Where

$$b_1 = \frac{2 C_v \beta (k-1) T_1}{C_v + \beta}, \tag{32}$$

$$b_2 = \frac{2 C_p \beta (C_v - \beta)(k-1) T_1}{(C_p + \beta)^2}, \tag{33}$$

$$b_3 = \frac{C_v r^{k-1}}{C_p + \beta}, \tag{34}$$

$$b_4 = \alpha(k-1), \tag{35}$$

$$b_5 = \frac{\alpha(C_p - \beta)(k-1)}{C_p + \beta} \tag{36}$$

Hence, w_{max} occurs at r_{cm} (the corresponding compression ratio at the maximum work output condition). In other words, w_{max} can be obtained by substituting $r_c = r_{cm}$ into equation (27). Furthermore, the corresponding thermal efficiency at maximum work output η_m, can be obtained by substituting r_{cm} into equation (29). [2]

3 Results and discussion

The net-work output versus efficiency characteristic and efficiency bound η_m at maximum work depend on α, β and T_1. The ranges for α, β, r and T_1 are 3000-3500 kJ/kg, 0.3-1.5 kJ/kg-K, 1.8 and 300-400 K, respectively. Additionally, C_p = 1.003 kJ/kg-K, C_v = 0.716 kJ/kg-K and k=1.4.

The effect of β on the w-η characteristic curves for the Atkinson and Dual combustion cycles at α = 3000 kJ/kg, and T_1 = 350 K is indicated in Figure (2) increasing β corresponds to enlarging the heat loss and, thus, decreasing the amount of heat added to the engine. Accordingly, the maximum work and efficiency decrease with increasing β.

The effect of α on the w-η characteristic curves for the Atkinson and Dual combustion cycles at β = 0.5 kJ/kg-K and T_1 = 350 K is depicted in Figure (3) increasing α increase the amount of heat added to the engine due to combustion.

Figure (4) shows the effect of intake temperature, T_1, on the w-η characteristic curves for α =3500 kJ/kg and β = 0.5 kJ/kg-K. The results show that the maximum work and efficiency decrease as T_1 increases, and for a given T_1, the maximum net-work of Atkinson cycle is higher than for the Dual combustion cycle.

The compression ratios (r_{cm}) that result in maximum work as a function of α and β are plotted in Figure (5) for a fixed β, r_{cm} increases as α increase. Note that the compression ratios that maximize the work of the Dual combustion cycle are always higher than those for the Atkinson cycle at the same operating conditions.

The effects of α and β on the maximum work output, w_{max}, and the corresponding efficiency at w_{max} η_m, are demonstrated in Figure (6) and (7). Figure (6), Figure (7) Shows that an increase in β results in a decrease of w_{max} (η_m).

The effects of β and T_1 on the maximum work output and the corresponding efficiency at the maximum work output are shown in Figures (8) and (9), respectively. It is seen that the heat loss parameter has a strong effect on the performance of the cycle. Both w_{max} and η_m decrease as β and T_1 increases.

The effects of α and T_1 on w_{max} and η_m are shown in Figures (10) and (11). It is found that both w_{max} and η_m increase as the constant α increase.

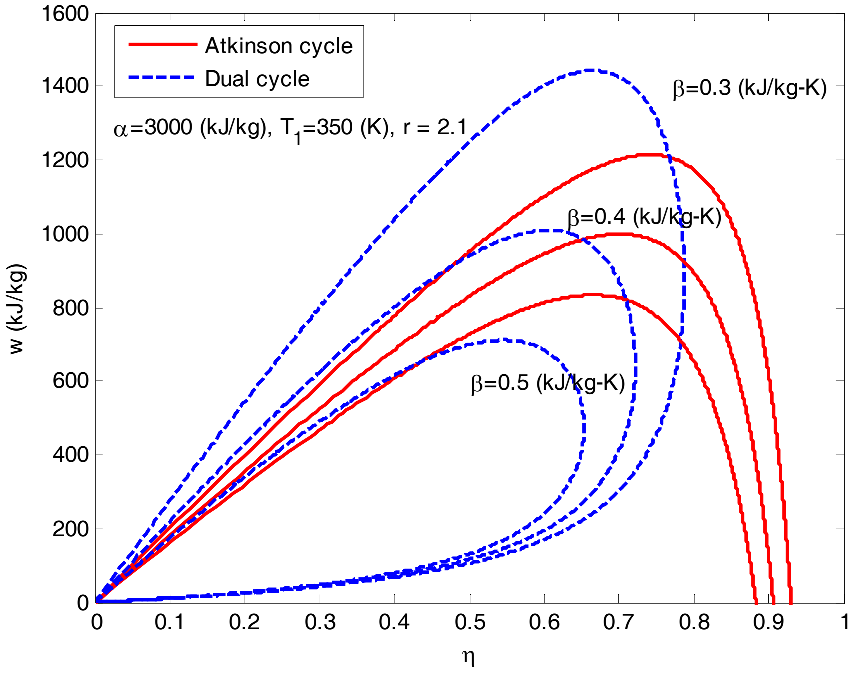

Figure 2 Effect of β on the w versus η characteristics

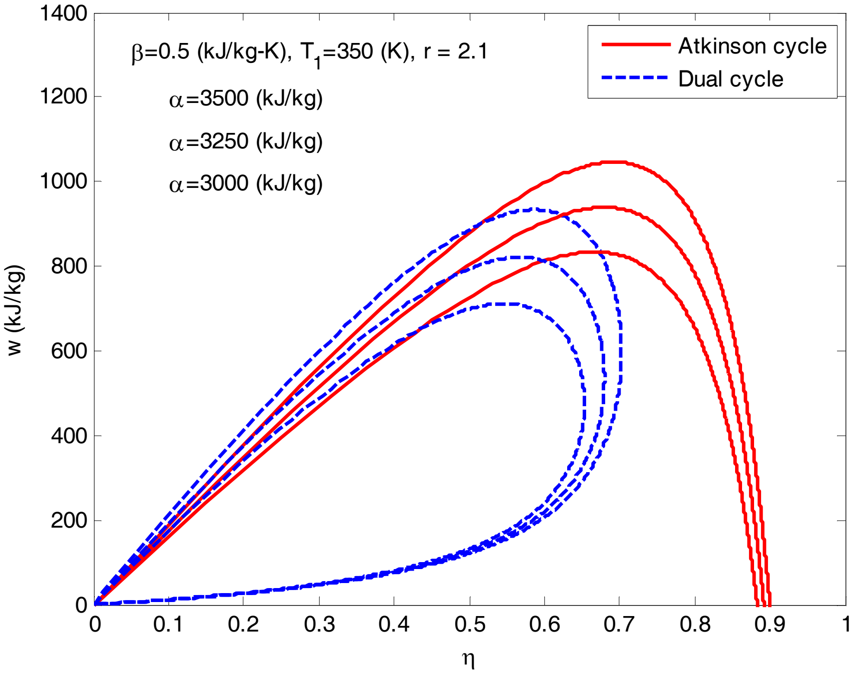

Figure 3 Effect of α on the w versus η characteristics.

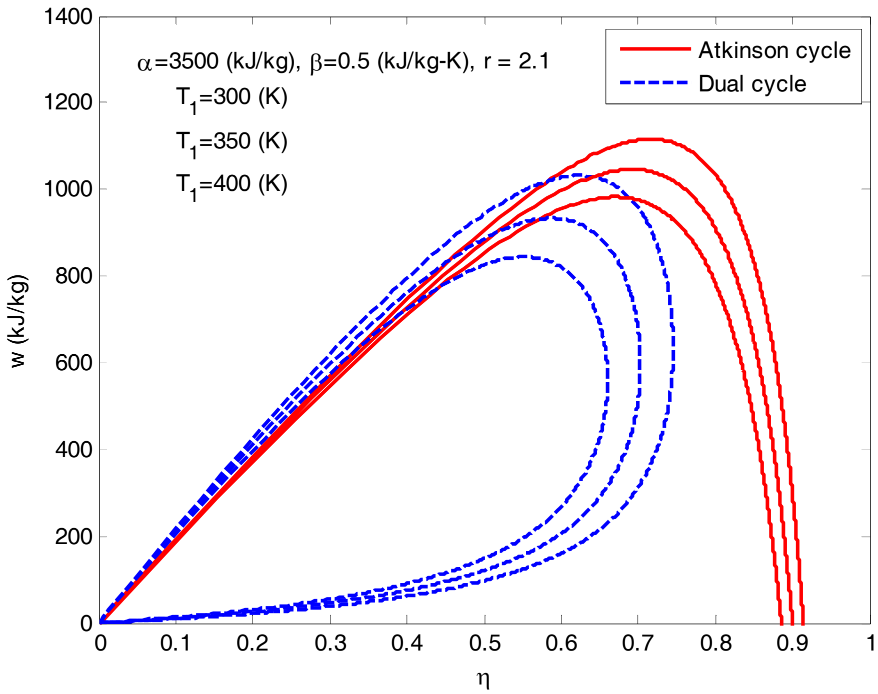

Figure 4 Effect of T_1 on the w versus η characteristics

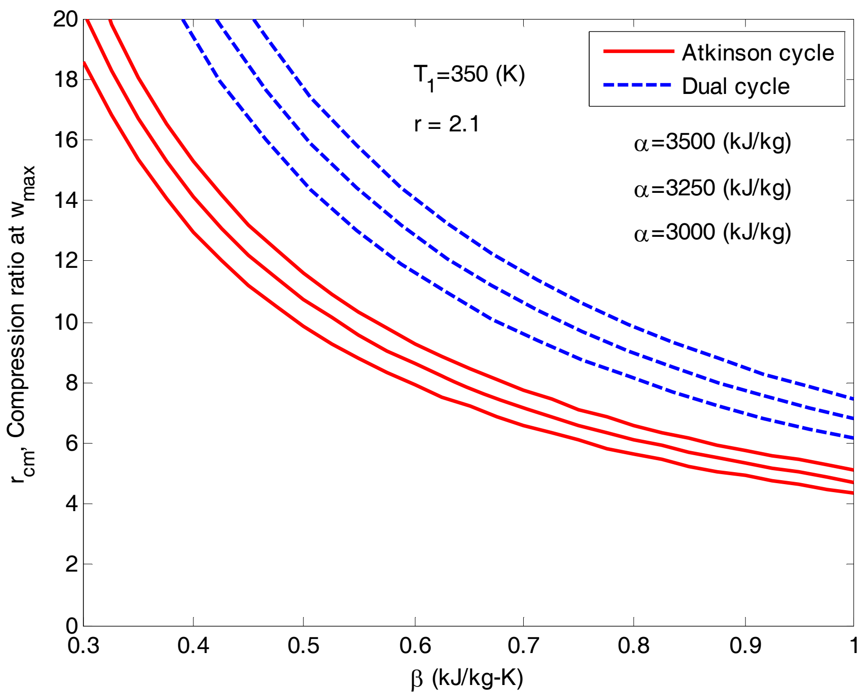

Figure 5 Comparison ratios at maximum net work for various values of α and β at T_1 = 350 K.

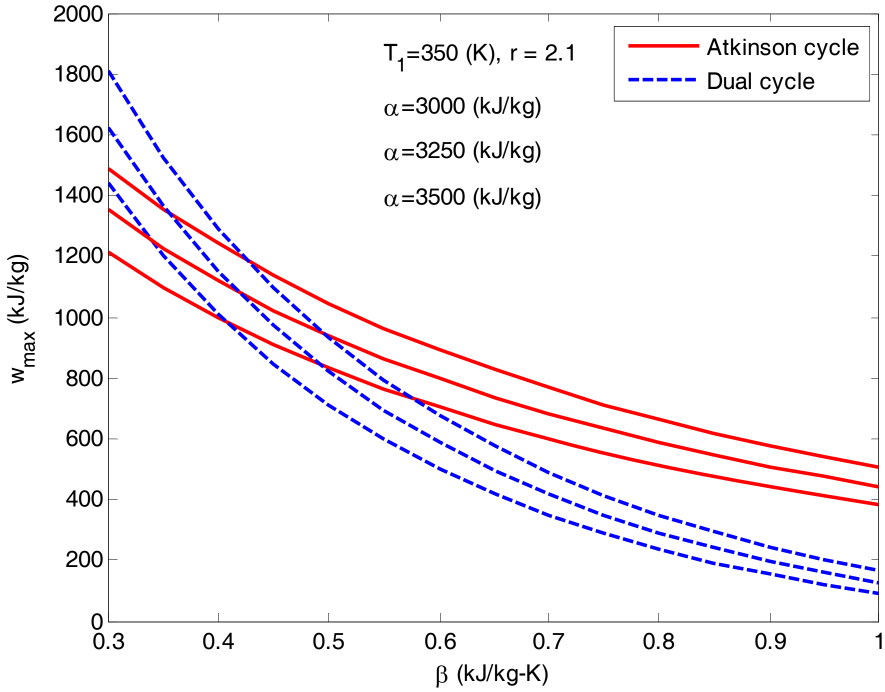

Figure 6 Effects of α and β on w_{max}.

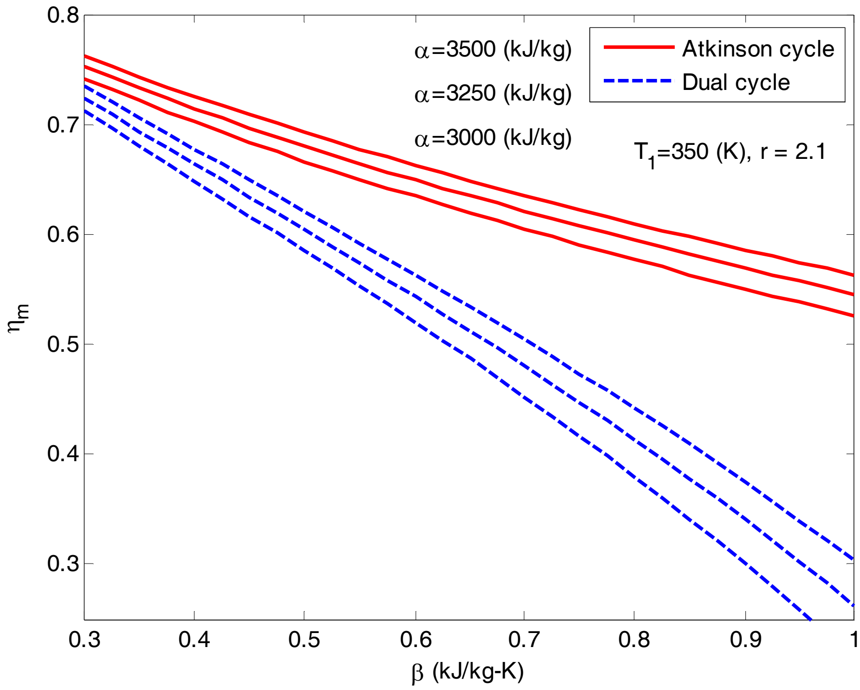

Figure 7 Effects of α and β on η_m.

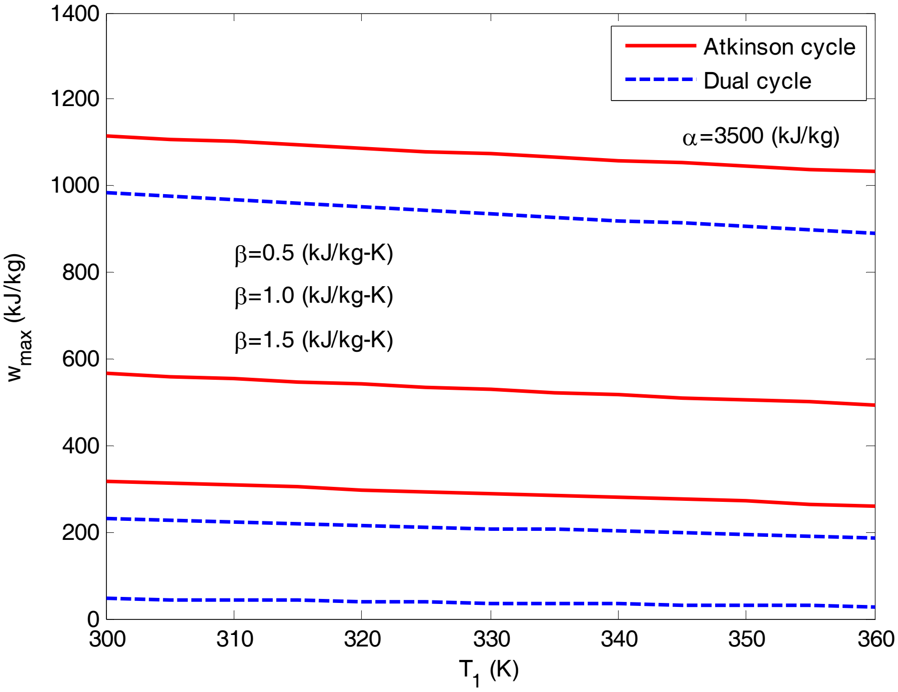

Figure 8 Effects of β and T_1 on w_{max}.

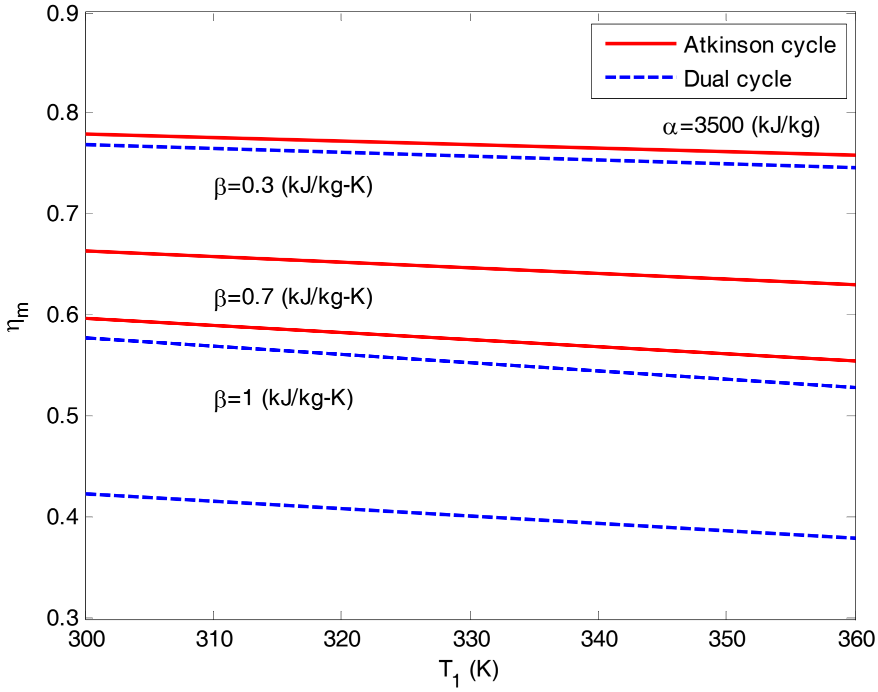

Figure 9 Effects of β and T_1 on η_m.

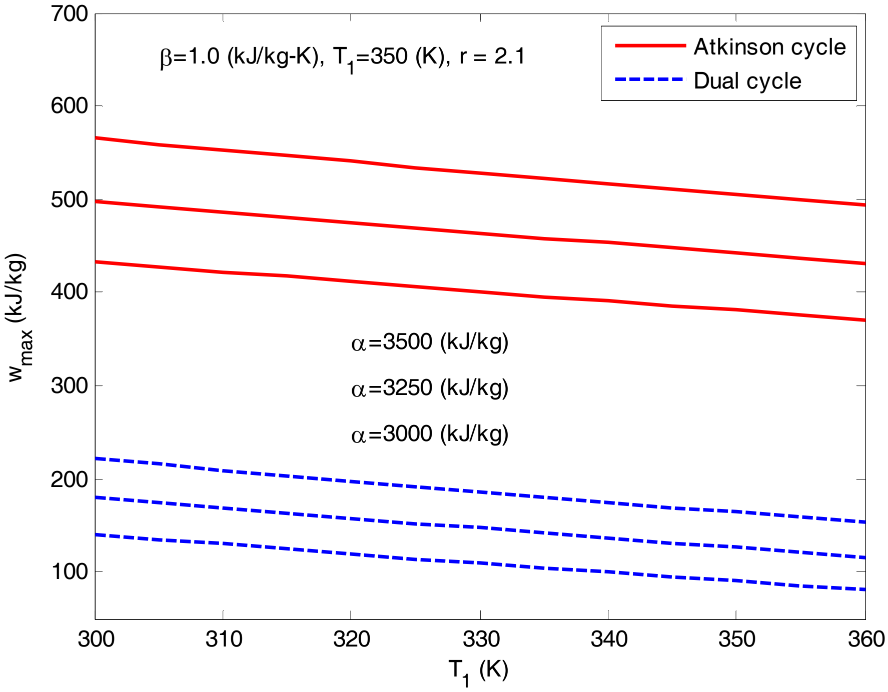

Figure 10 Effects of α and T_1 on w_{max}.

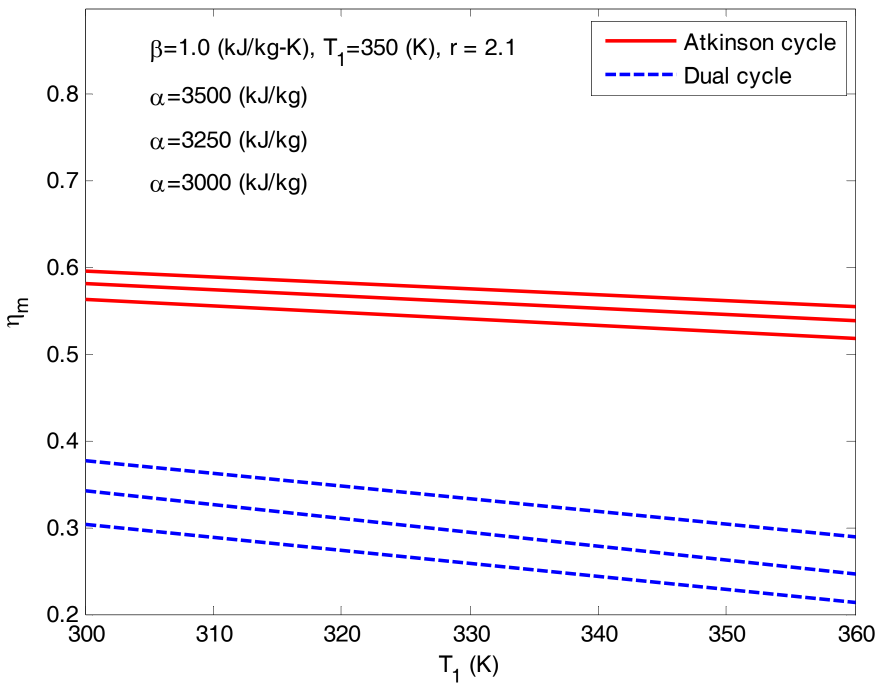

Figure 11 Effects of α and T_1 on η_m.

4 Conclusions

The effects of heat transfer through the cylinder wall on the performance of the Atkinson and the Dual combustion cycle are investigated in this study. The relation between net-work output and thermal efficiency is derived. Furthermore, the maximum work output and the corresponding thermal efficiency at the maximum work output are also are derived. In the analyses, the influence of four significant parameters, namely the heat transfer and combustion constant, compression ratio and intake air temperature on the net-work output versus efficiency characteristic, and the maximum work and the corresponding efficiency at maximum work are examined. Comparisons of the performances of the air-standard Atkinson and the Dual combustion cycles with heat transfer considerations are also discussed. The general conclusions drawn from the results of this work are as follows:

1. The maximum work output and the corresponding efficiency at maximum work output decreases as the heat transfer constant β increases.
2. The maximum work output and the corresponding efficiency at maximum work output increases as the combustion constant α increases.
3. The maximum work output and the corresponding efficiency at maximum work output decreases as the intake temperature (T_1) increases.
4. For a given value of heat release during combustion (α) an increase in heat loss (β) leads to a decrease of the compression ratio (r_{cm}) that maximizes the work of the Atkinson cycle.

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Nomenclatures

a_1: constant defined in equation (15)
a_2: constant defined in equation (15)
a_3: constant defined in equation (15)
b_1: constant defined in equation (31)
b_2: constant defined in equation (31)
b_3: constant defined in equation (31)
b_4: constant defined in equation (31)
b_5: constant defined in equation (31)
C_p: constant pressure specific heat (kJ/kg – K)
C_v: constant volume specific heat (kJ/kg – K)
k: $k = C_p/C_v$
q_{in}: heat added per unit mass to working fluid (kJ/kg)
r: cut-off ratio
r_c: compression ratio
r_{cm}: compression ratio at maximum work
s: specific entropy (kJ/kg – K)
r_p: pressure ratio
T_i: temperature at state i (K)
V_i: volume at state i (m^3)
w: net work output per unit mass (kJ/kg)
w_{max}: maximum work output per unit mass of working fluid per cycle (kJ/kg)

Greek symbols

α: constant related to combustion (kJ/kg)
β: constant related to heat transfer (kJ/kg – K)
η: efficiency
η_m: corresponding thermal efficiency at maximum work output

چکیده

در هنگام فرآیندهای موتورهای واقعی، مقداری گرمای تلف شده وجود دارد که در تحلیل ایده‌آل این موتورها در نظر گرفته نمی‌شود. در این مقاله، اثر انتقال حرارت روی کار خالص خروجی و بازده حرارتی چرخه استاندارد هوایی آتکینسون و چرخه احتراق دوگانه بررسی شده است. همچنین عملکرد چرخه‌های آتکینسون و دوگانه نیز با هم مقایسه شده است. فرض شده است که مراحل تراکم و قدرت بی‌دررو و برگشت‌پذیر و بدون هیچ نوع انتقال حرارتی (اعم از جابجایی، همرفت و تشعشعی) به دیواره سیلندر باشد. فقط اتلاف حرارتی صورت گرفته از طریق دیواره سیلندر در مرحله احتراق در نظر گرفته شده است. نتایج بدست آمده نشان می‌دهد که کار خالص خروجی در برابر بازده و کار خروجی بیشینه در برابر بازده وابسته به آن تا حد زیادی تحت تأثیر مقدار انتقال حرارت می‌باشد. نتایج این بررسی می‌تواند به عنوان یک راهنمای مناسب برای بررسی عملکرد موتورهای واقعی مورد استفاده قرار گیرد.