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Comparison of performances of air standard Atkinson and Dual cycles with heat transfer considerations

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Abstract

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 and indicated thermal efficiency of air standard Atkinson and Dual cycles are analyzed. Comparison of performances of air standard Atkinson and Dual cycles with heat transfer considerations are also discussed. We assume 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 and improvement of practical Atkinson and Diesel engines.

Keywords: Thermodynamics; Atkinson cycle; Dual cycle; Heat transfer; Diesel engine

1. Introduction

The Atkinson cycle engine is a type of internal combustion engine, which was designed and built by James Atkinson in 1882 [1]. 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 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 of air standard Atkinson and Otto cycles with heat transfer considerations and Heat transfer effects on the performance of an air standard Dual cycle by Hou [1] and [2]. Performance analyzing and parametric optimum criteria of an irreversible Atkinson heat-engine by Zhao and Chen [3]. Optimization of Dual cycle considering the effect of combustion on power by Chen [4]. Performance of an endoreversibe Atkinson cycle with variable specific heat ratio of working fluid by Ebrahimi [5]. The results obtained in this

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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.

Nomenclature

a_1	constant defined in Eq. (15)
a_2	constant defined in Eq. (15)
a_3	constant defined in Eq. (15)
b_1	constant defined in Eq. (31)
b_2	constant defined in Eq. (31)
b_3	constant defined in Eq. (31)
b_4	constant defined in Eq. (31)
b_5	constant defined in Eq. (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 to working fluid (kJ / kg)
r	cut-off ratio
r_c	compression ratio
r_{cm}	compression ratio at maximum work
s	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 (kJ / kg)
w_{max}	maximum work output per unit mass of working fluid per cycle (kJ / kg)

Greek

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

Subscripts

Max	maximum
1, 2, 3, 4	state points

2. Cycle model

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

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 following equation:

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

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 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 Eqs. (2) and (4) into Eq. (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 following equation:

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

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

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

Combining Eqs. (6) and (7) yields

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

Substitution of Eq. (2) into Eq. (8) gives

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

Substituting of Eqs. (2) and (9) into Eq. (5) gives T_4 as a function of T_1

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

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

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

Similarly, substitution Eqs. (2), (9) and (10) into Eq. (7) yields

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

Eq. (11) divided by Eq. (12) gives the indicated thermal efficiency,

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

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

$$\frac{dw}{dr_c} = 0. \quad (14)$$

we finally get

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

$$a_1 = \frac{C_v - \beta}{C_v + \beta}, \quad (16)$$

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

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

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

2.2. Thermodynamics analysis of the air standard Dual cycle

Following the assumption described at cycle model, Figure 1 shows the temperature-entropy (T-s) diagrams for the thermodynamic processes of an air standard Dual 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 following equation:

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

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 absolute temperatures at states 1, 2, 3, 4 and 5. For the isentropic process ($1 \rightarrow 2$) 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 following equation:

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

The heat added to the working fluid during the constant volume combustion process can be given in the following linear expression

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

Combining Eqs. (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 Eq. (24) into Eq. (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 Eqs. (20), (24), (25) and (26) into Eq. (1), the net work output per unit mass of the working fluid can be expressed in terms of T_1 as

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

Similarly, substituting Eqs. (20), (24) and (25) into Eq. (23) yields

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

Eq. (27) divided by Eq. (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

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

where

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

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

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

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

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

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 Eq. (27). Furthermore, the corresponding thermal efficiency at maximum work output η_m can be obtained by substituting r_{cm} into Eq. (29). [2]

3. Figures

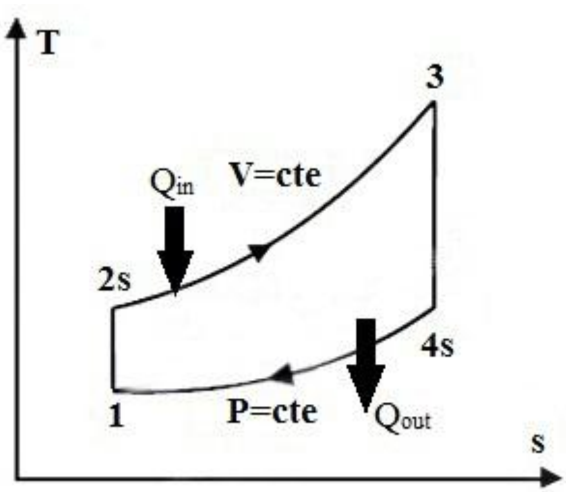


Fig. 1.a. T-s diagram for Atkinson cycle.

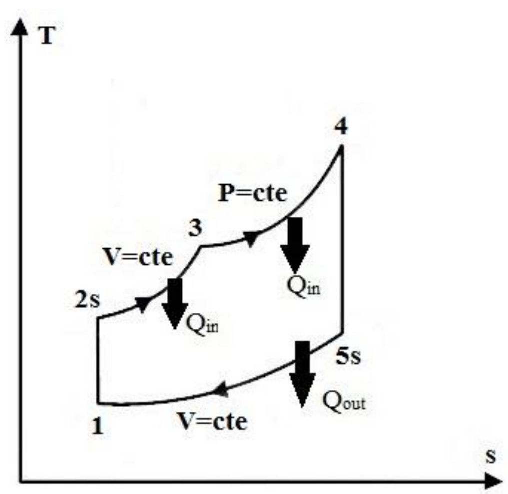


Fig. 1.b. T-s diagram for Dual cycle.

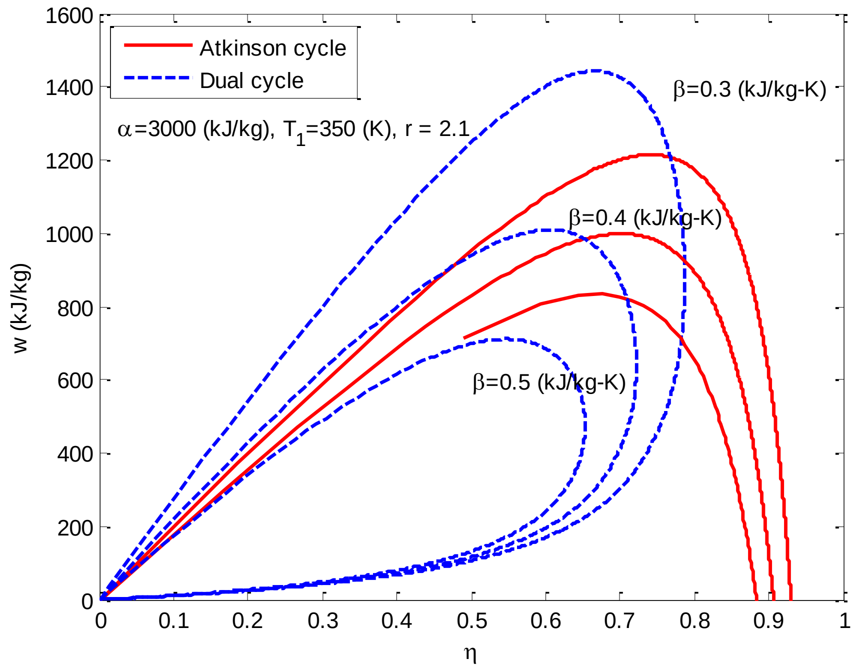


Fig. 2. Effect of β on the w versus η characteristics.

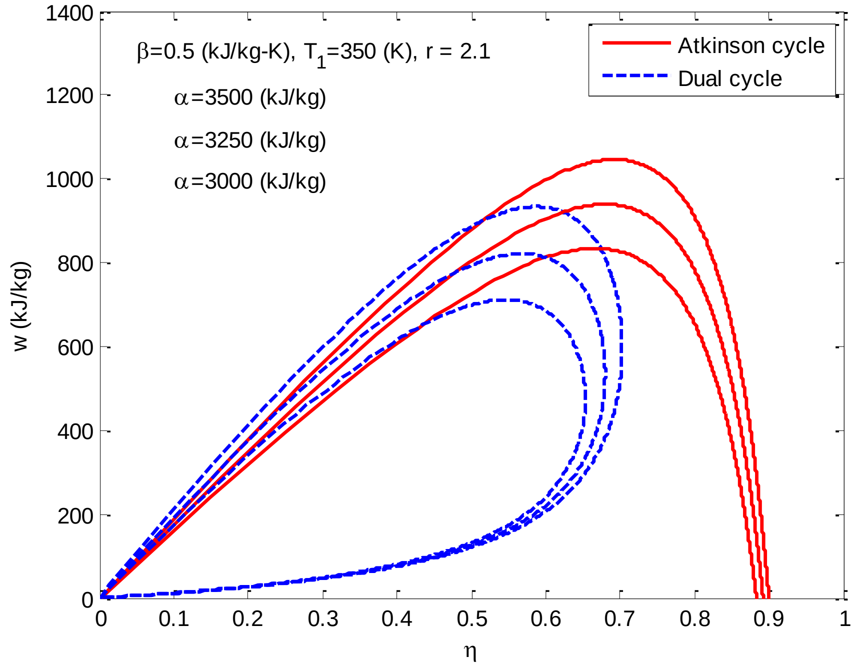
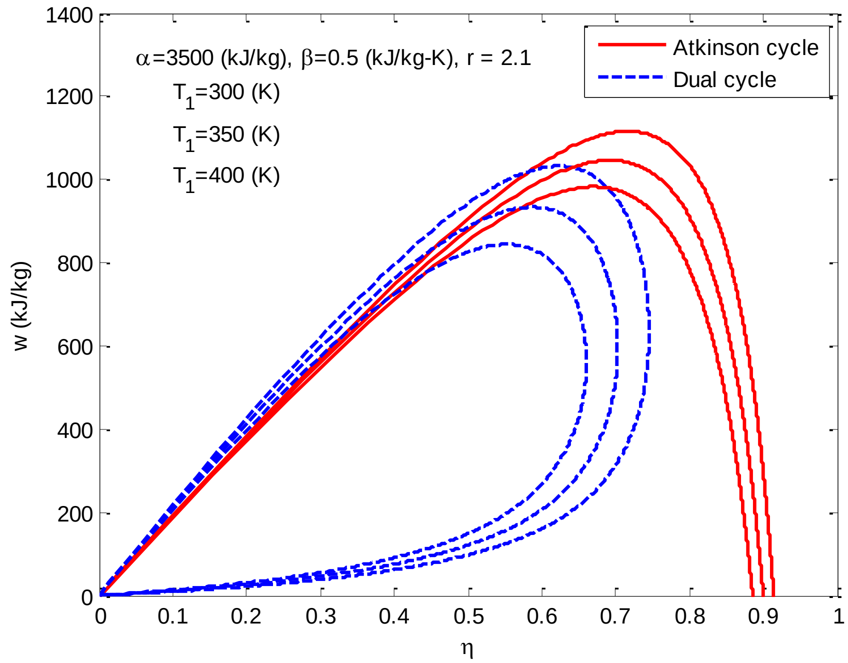


Fig. 3. Effect of α on the w versus η characteristics.



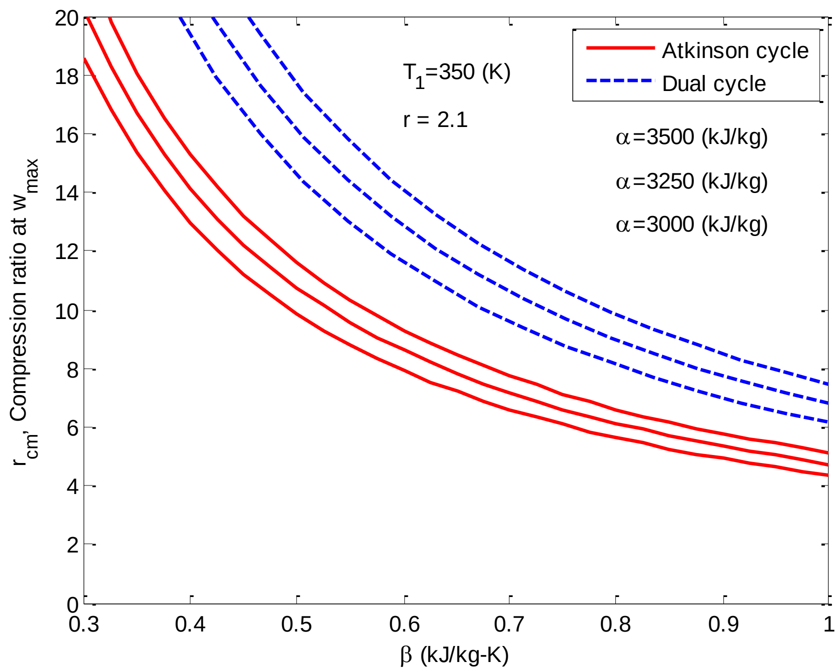


Fig. 4. Effect of T_1 on the w versus η characteristics.

Fig. 5. Compression ratios at maximum net work for various values of α and β at $T_1 = 350\ K$.

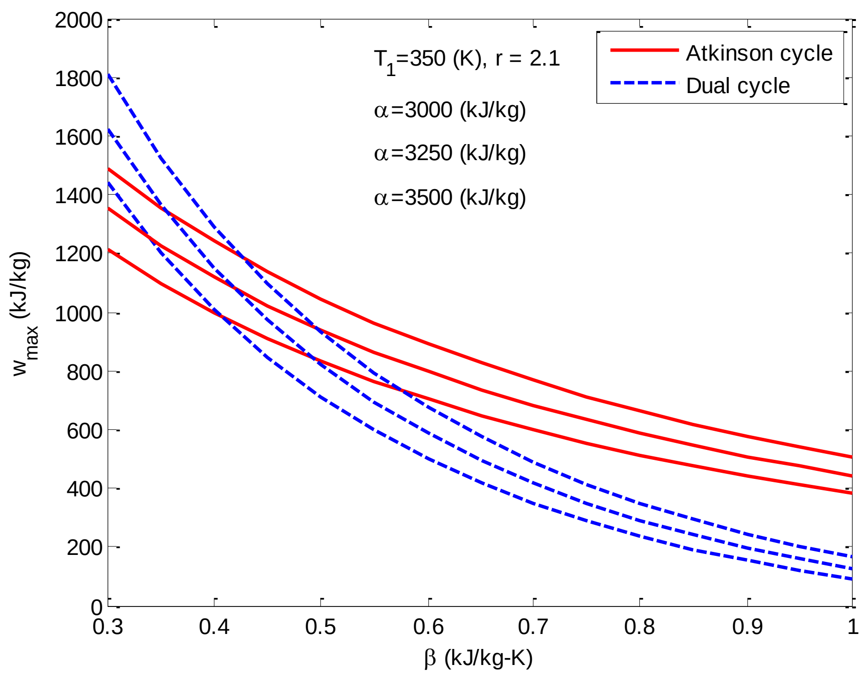


Fig. 6. Effect of α and β on w_{max} .

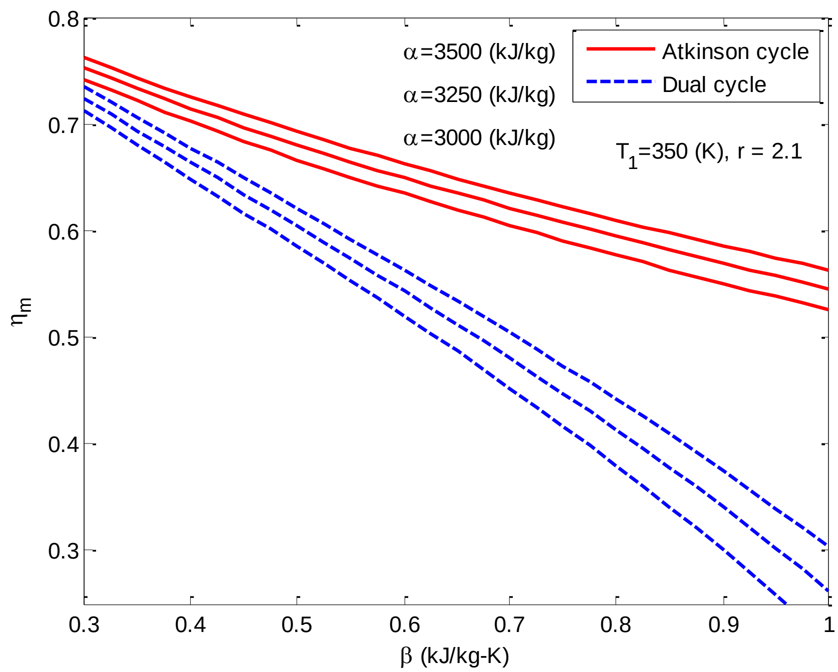


Fig. 7. Effect of α and β on η_m .

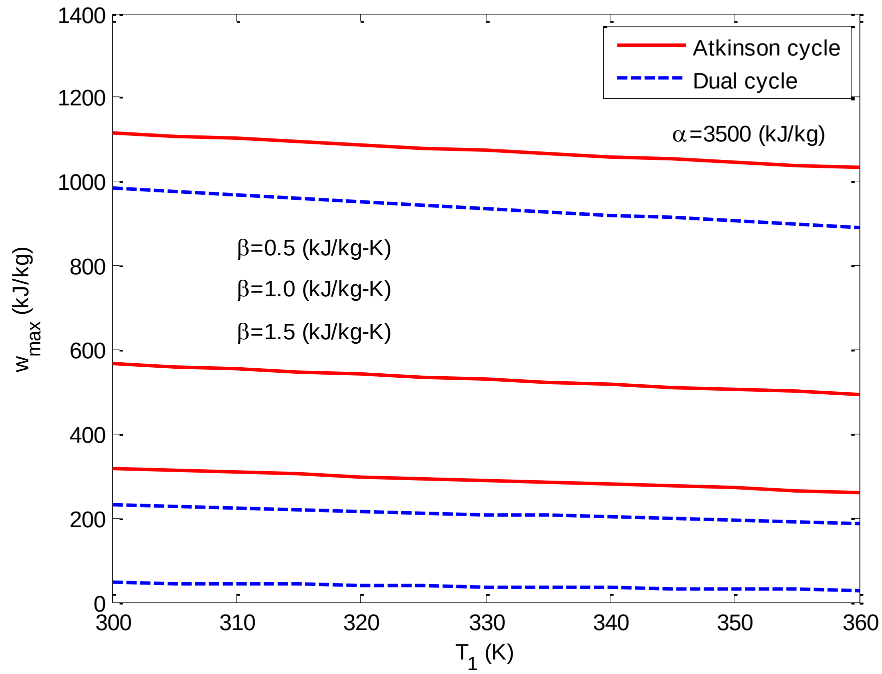


Fig. 8. Effect of β and T_1 on w_{max} .

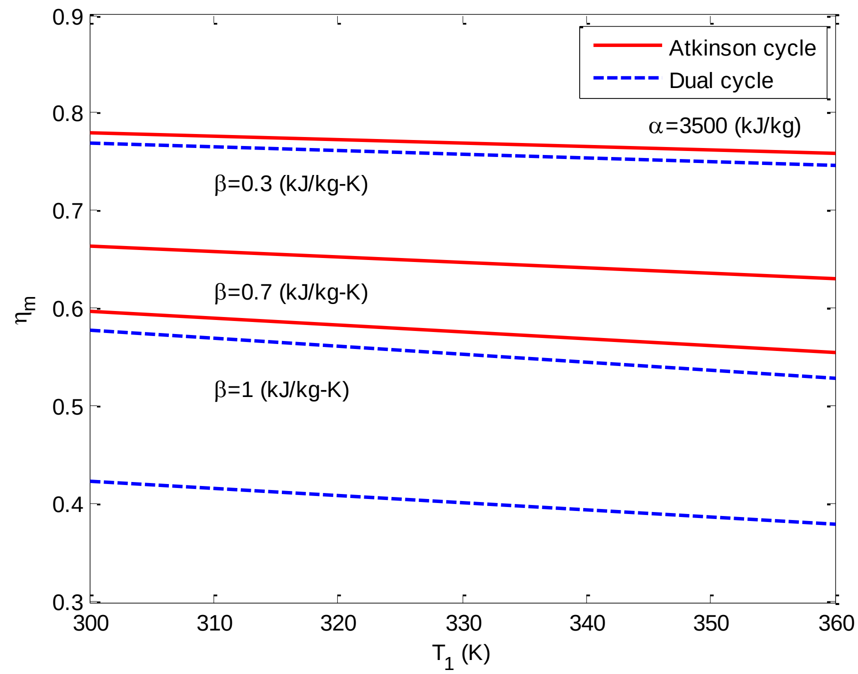


Fig. 9. Effect of β and T_1 on η_m .

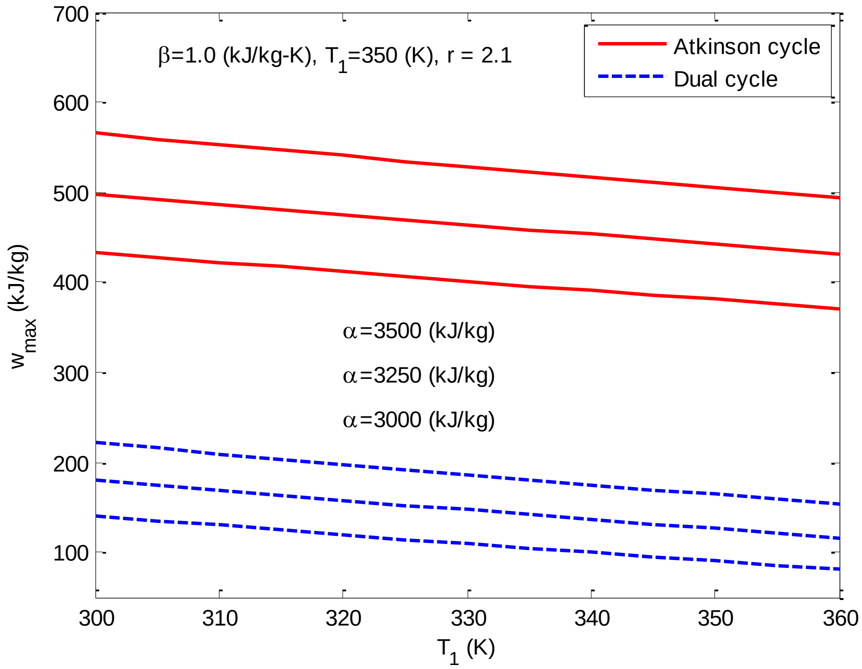


Fig. 10. Effect of α and T_1 on w_{max} .

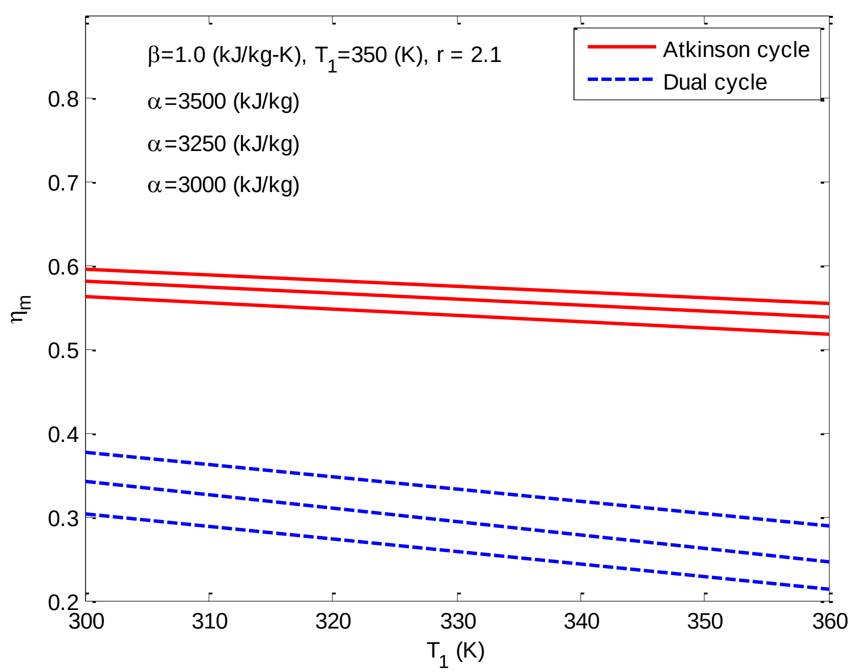


Fig. 11. Effect of α and T_1 on η_m .

4. 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 $w-\eta$ characteristic curves for the Atkinson and Dual cycles at $\alpha=3000\,kJ/kg$, and $T_1=350\,K$ is indicated in Fig. 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 $w-\eta$ characteristic curves for the Atkinson and Dual cycles at $\beta=0.5\,kJ/kg-K$ and $T_1=350\,K$ is depicted in Fig. 3. Increasing α increases the amount of heat added to the engine due to combustion.

Fig. 4. Shows the effect of intake temperature, T_1, on the $w-\eta$ characteristic curves for $\alpha=3500\,kJ/kg$ and $\beta=0.5\,kJ/kg-K$. 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 cycle.

The compression ratios (r_{cm}) that result in maximum work as a function of α and β are plotted in Fig. 5. For a fixed β, r_{cm} increases as α increases. Note that the compression ratios that maximize the work of the Dual 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 Fig. 6. and 7, Fig. 6. (Fig. 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 Fig. 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 Fig. 10 and 11. It is found that both $w_{\max}$ and η_m increase as the constant α increases.

5. Conclusion

The effects of heat transfer through the cylinder wall on the performance of Atkinson and Dual 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 maximum work output are also 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 air standard Atkinson and Dual 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 corresponding efficiency at maximum work output decrease as the heat transfer constant β increases.

2. The maximum work output and corresponding efficiency at maximum work output increase as the combustion constant α increases.

3. The maximum work output and corresponding efficiency at maximum work output decrease 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.

5. The compression ratios that maximize the work of the Dual cycle are always found to be higher than those for the Atkinson cycle at the same operating conditions.

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