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TECHNICAL NOTE

Comment on "The effects of heat transfer on the exergy efficiency of an air-standard Otto cycle" by Hakan Özcan, Heat and Mass Transfer (2011) 47:571–577

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Abstract In this letter, it is shown that the applied relations in the paper by Hakan Özcan [H. Özcan, The effects of heat transfer on the exergy efficiency of an air-standard Otto cycle, Heat and Mass Transfer (2011) 47:571–577] are erroneous and thus the reported results are invalid. These incorrect relations [Eqs. (8), (9), (10), (14) and (16) of HÖ2011] are replaced by correct ones. Moreover, the obtained results (graphs and tables) are modified based on the correct relations. Finally, to achieve more realistic results, the internal irreversibility described by using the compression and expansion efficiencies is added to the analysis.

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1 Introduction

The Otto cycle is an ideal cycle for spark-ignition reciprocating engines. It is named after Nikolaus A. Otto, who built a successful four-stroke engine in 1876 in Germany using the cycle proposed by Frenchman Beau de Rochas in 1862 [1]. A schematic of a *T-s* diagram for an air-standard Otto cycle is shown in Fig. 1. Process $(1 \rightarrow 2)$ is an isentropic compression of the air as the piston moves from bottom dead center (BDC) to top dead center (TDC). Heat is then added at constant volume, $(2 \rightarrow 3)$, while the piston is momentarily at rest at TDC (This process corresponds to the ignition of the fuel–air mixture by the spark and the subsequent burning in the actual engine). Process $(3 \rightarrow 4)$ is an isentropic expansion and process $(4 \rightarrow 1)$ is the rejection of heat from the air while the piston is at BDC [2]. In recent years, much attention has been paid to the performance of internal combustion engines for different cycles. Comparison of performances of air-standard Atkinson and Otto cycles with heat transfer considerations is investigated by Hou [3]. Heat transfer effect on the net work output and efficiency characteristics for an air-standard Otto cycle is studied by Chen et al. [4]. Reciprocating heat-engine cycles are performed by Ge et al. [5]. Som et al. [6] investigated thermodynamic irreversibilities and energy balance in combustion processes. Exergy analysis for a Braysson cycle is studied by Zheng et al. [7]. Lior et al. [8] performed the Energy, exergy, and second-law performance criteria. Second-law analyses applied to internal combustion engines operation are done by Rakopoulos et al. [9]. Rashidi et al. [10] investigated first and second-laws analysis of an air-standard Dual cycle with heat loss consideration. Chen et al. [11] studied heat transfer effects on net work and/or power as functions of efficiency for air-standard Diesel cycles. Effect of heat

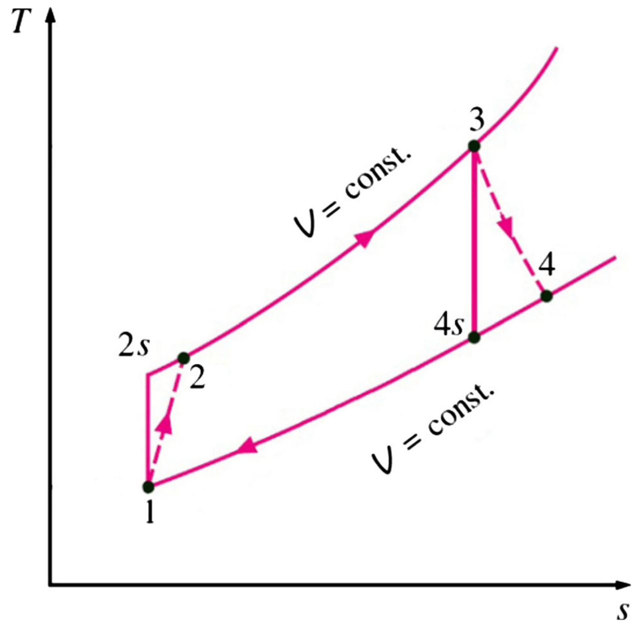

Fig. 1 The *T-s* diagram of air-standard Otto cycle

transfer on the performance of an air-standard Diesel cycle is performed by Akash [12]. Heat transfer effects on the performance of an air-standard Dual cycle are done by Hou [13]. Performance analysis and parametric optimum criteria of an irreversible Atkinson heat-engine is done by Zhao et al. [14]. Finite-time thermodynamic modeling and analysis for an irreversible Dual cycle is studied by Ge et al. [15]. Effects of friction and temperature-dependent specific-heat of the working fluid on the performance of a Diesel engine are investigated by Al-Sarkhi et al. [16]. Chet et al. [17, 18] discussed effects of heat transfer, friction and variable specific heats of working fluid on performance of an irreversible Dual cycle and thermodynamic simulation of performance of an Otto cycle with heat transfer and variable specific heats of working fluid.

In this article, the effect of heat loss on the exergy of an air-standard Otto cycle studied.

2 Thermodynamic analysis

For an air-standard Otto cycle, the heat added per unit mass of the working fluid of the cycle during the constant-volume process (2s → 3) is represented by the following equation:

$$q_{\text{in}} = C_{\text{v}}(T_3 - T_{2\text{s}}) \tag{1}$$

The temperatures within the combustion chamber of an internal combustion engine can reach values on the order of 2,700 [K] and above. Materials in the engine cannot tolerate this kind of temperature and would quickly fail if proper heat transfer did not occur. Therefore, to prevent the engine and engine lubricants from thermal failure, the interior maximum temperature of the combustion chamber must be limited to much lower values by heat fluxes through the cylinder wall during the combustion period. Since during the other processes of the operating cycle, the heat flux is essentially quite small and negligible due to both the lower temperature ratios in comparison with combustion process and the very short time involved for the processes, it is assumed that the heat loss through the cylinder wall occurs only during combustion. The calculation of actual heat transfer through the cylinder wall occurring during combustion is quite complex; so it is assumed to be proportional to the average temperature of both the working fluid and the cylinder wall, and that, during the operation, the wall temperature remains approximately constant. The heat added per unit mass of the working fluid of the cycle by combustion is given by the following linear relation [11–19]:

$$q_{\text{in}} = A - B(T_{2\text{s}} + T_3), \tag{2}$$

where, A and B are two constants related to combustion and heat transfer, respectively. Combining Eqs. (1) and (2) gives:

$$T_3 = \frac{[A + (C_{\text{v}} - B)T_{2\text{s}}]}{(C_{\text{v}} + B)}. \tag{3}$$

For processes (1 → 2s) and (3 → 4s), we have:

$$T_{2\text{s}} = T_1 r_{\text{c}}^{k-1}, \tag{4}$$

and

$$T_{4\text{s}} = T_3 r_{\text{c}}^{1-k}. \tag{5}$$

Second-law efficiency analysis is a good benchmark for the availability of systems that is described as the ratio of the actual thermal efficiency (first-law efficiency) to the maximum possible (reversible) thermal efficiency under the same conditions. This procedure is also known as entropy generation minimization and was pioneered by Bejan [20]. For work-producing devices, the second-law efficiency can also be expressed as the ratio of the useful work output to the maximum possible (reversible) work output, as follow:

$$\eta_{\text{II}} = \frac{w_{\text{net}}}{w_{\text{max}}}, \tag{6}$$

where, the net work output per unit mass of the working fluid of the cycle, w_{net}, and the maximum possible work output of the cycle that the system can produce, w_{max}, are defined as follows:

$$w_{\text{net}} = \eta_{\text{I}} \cdot q_{\text{in}} = \frac{C_{\text{v}}\eta_{\text{I}}[A - 2BT_1/(1 - \eta_{\text{I}})]}{(C_{\text{v}} + B)}, \tag{7}$$

and

$$w_{\max} = \eta_{\max} \cdot q_{\text{in}}, \tag{8}$$

where, the maximum possible (reversible) thermal efficiency (Carnot cycle efficiency) is given by:

$$\eta_{\max} = \left(1 - \frac{T_1}{T_3}\right) \cdot \tag{9}$$

Accordingly, Written expression for $w_{\max}$ in Eq. (8) of HÖ2011 that has been mentioned as follow:

$$w_{\max} = C_v(T_3 - T_1), \tag{10}$$

is completely incorrect and should be replaced by:

$$w_{\max} = C_v(T_3 - T_{2s})\left(1 - \frac{T_1}{T_3}\right) \cdot \tag{11}$$

Consequently, Eqs. (9) and (10) of HÖ2011 that were explained as follow:

$$w_{\max} = C_v\left[\frac{A(1-\eta_I) - 2BT_1 + T_1\eta_I(C_v + B)}{(1-\eta_I)(C_v + B)}\right], \tag{12}$$

and

$$\eta_{II} = \frac{\eta_I[A(1-\eta_I) - 2BT_1]}{A(1-\eta_I)2BT_1 + T_1\eta_I(C_v + B)}, \tag{13}$$

should be replaced by:

$$w_{\max} = C_v\left(\frac{A(1-\eta_I) - 2BT_1}{(1-\eta_I)(C_v + B)}\right) \times \left(\frac{A(1-\eta_I) + \eta_I(C_v + B)T_1 - 2BT_1}{A(1-\eta_I) + (C_v - B)T_1}\right), \tag{14}$$

and

$$\eta_{II} = \frac{\eta_I[A(1-\eta_I) + (C_v - B)T_1]}{A(1-\eta_I) + \eta_I(C_v + B)T_1 - 2BT_1} \cdot \tag{15}$$

On the other hand, the exergy change of a closed system as it undergoes a process from state i to state j becomes:

$$e_{ij} = e_j - e_i = (u_j - u_i) + P_0(v_j - v_i) - T_0(s_j - s_i) \cdot \tag{16}$$

Finally, the exergy analysis for the processes of air-standard Otto cycle with heat-transfer losses consideration can be obtained as following results.

For the isentropic compression process (1 → 2s), the exergy change is obtained as:

$$e_{12} = C_v T_1 \times \left\{\left(\frac{\eta_I}{1-\eta_I}\right) - (k-1)\left(1 - (1-\eta_I)^{\left(\frac{1}{k-1}\right)}\right)\right\} \cdot \tag{17}$$

In HÖ2011, according to Eq. (14), it's assumed that, the internal energy during the heat addition process, to be constant, whereas during the heat addition process of an air-standard Otto cycle, the volume remains invariant and thus, there is no reason for this assumption. So, Eq. (19) is replaced instead Eq. (18) [Eq. (14) of HÖ2011] as follow:

$$e_{23} = -C_v T_0 \ln\left[\frac{A(1-\eta_I) + (C_v - B)T_1}{(C_v + B)T_1}\right], \tag{18}$$

and

$$e_{23} = C_v\left\{\begin{array}{l}\left(\dfrac{A(1-\eta_I) - 2BT_1}{(1-\eta_I)(C_v + B)}\right) \\ -T_0 \ln\left(\dfrac{A(1-\eta_I) + (C_v - B)T_1}{(C_v + B)T_1}\right)\end{array}\right\} \cdot \tag{19}$$

For the isentropic expansion process (3 → 4s), the exergy change is obtained as:

$$e_{34} = -C_v\left\{\begin{array}{l}\left(\dfrac{A\eta_I(1-\eta_I) + \eta_I T_1(C_v - B)}{(C_v + B)(1-\eta_I)}\right) \\ -(k-1)T_1\left(1 - (1-\eta_I)^{\left(\frac{1}{k-1}\right)}\right)\end{array}\right\} \cdot \tag{20}$$

Finally, during the constant-volume heat rejection process (4s → 1), the exergy change according to Eq. (21) [Eq. (16) of HÖ2011], is modified to Eq. (22):

$$e_{41} = C_v\left\{\left[\frac{A(1-\eta_I) - 2BT_1}{(C_v + B)}\right]\right\}, \tag{21}$$

and

$$e_{41} = -C_v\left\{\begin{array}{l}\left(\dfrac{A(1-\eta_I) - 2BT_1}{(C_v + B)}\right) \\ -T_0 \ln\left(\dfrac{A(1-\eta_I) + (C_v - B)T_1}{(C_v + B)T_1}\right)\end{array}\right\} \cdot \tag{22}$$

As it can be seen in Fig. 1, in the ideal air-standard Otto cycle, for the reversible adiabatic compression (1 → 2s) and expansion (3 → 4s) processes, the entropy generation and thus the entropy change of the working fluid is zero; while as in a real Otto cycle, the internal irreversibilities cause the entropy of the working fluid to increase, during the irreversible adiabatic compression (1 → 2) and expansion (3 → 4) processes. Therefore, the following compression and expansion efficiencies can be used to describe the internal irreversibilities of the compression and expansion processes, respectively.

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}, \tag{23}$$

and

$$\eta_e = \frac{T_3 - T_4}{T_3 - T_{4s}} \cdot \tag{24}$$

3 Numerical results

According to HÖ2011, the following constants and parameters have been used in the calculations: $A = 2{,}500 \rightarrow 4{,}500$ [kJ/kg], $T_1 = 300 \rightarrow 375$ [K], $B = 0.5 \rightarrow 1.2$ [kJ/kg.K],

$C_v = 0.7165$ [kJ/kg.K] and $\eta_c = \eta_e = 0.96 \rightarrow 1$. The effects of constants related to combustion, A, and heat transfer, B, and initial temperature, T_1, on the second-law efficiency, η_{II}, are indicated in Figs. 2, 3, 4, 5 and 6. It can be seen that, η_{II} increases with decreasing A and increasing B and T_1, for the fixed values of first-law efficiency, η_I, and then compression ratio, r_c. In addition, Figs. 4, 5 and 6 show η_I and η_{II} increase with the increase of r_c. Note that some values of the second-law efficiency might be insufficient for a feasible Otto cycle.

Figures 7, 8 and 9 and Tables 1 and 2 illustrate the influence of parameters A, B and η_I on the exergy change (useful work potential) during each process of cycle. The results show that the effect of parameters A and B on the exergy change during the compression, e_{12}, is not significant, since these parameters do not affect the temperature change during compression. When the combustion process starts, the parameters A and B have a significant impact on

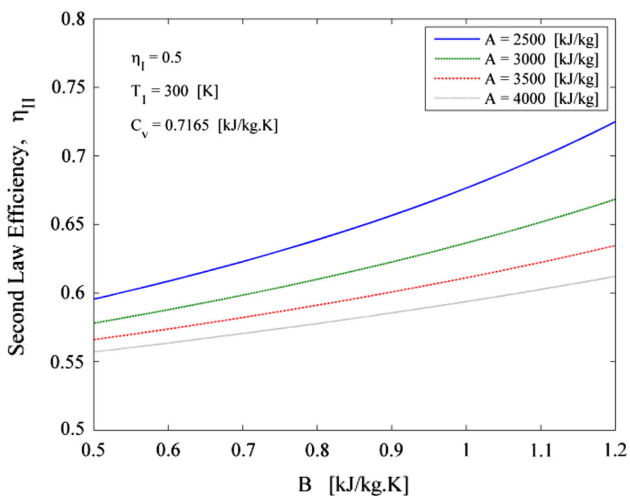

Fig. 2 Effect of A on the $\eta_{II} - B$ characteristics

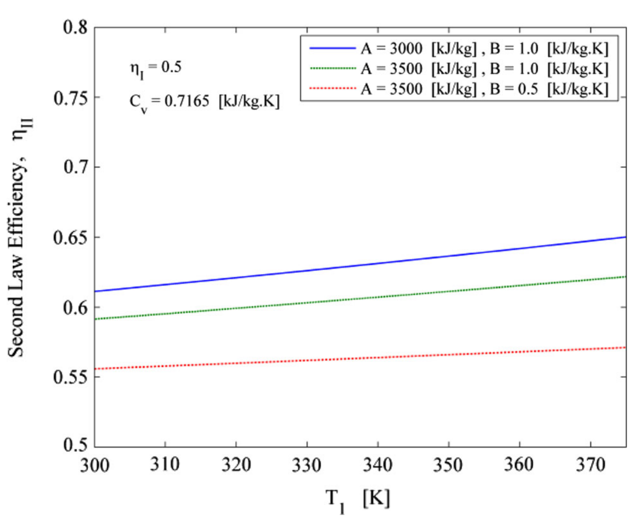

Fig. 3 Effects of A and B on the $\eta_{II} - T_1$ characteristics

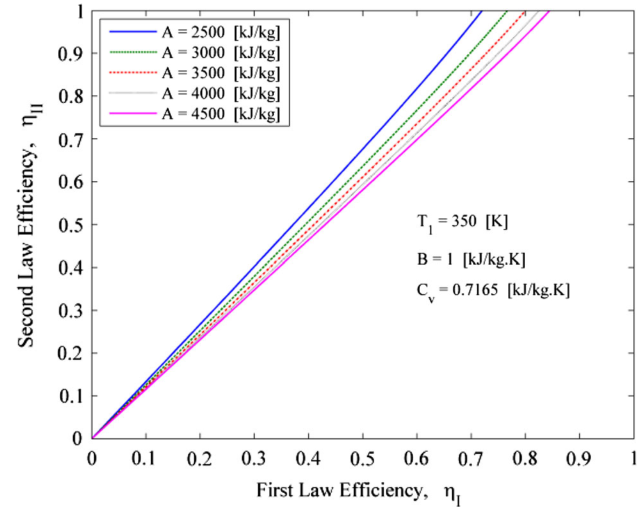

Fig. 4 Effect of A on the $\eta_{II} - \eta_I$ characteristics

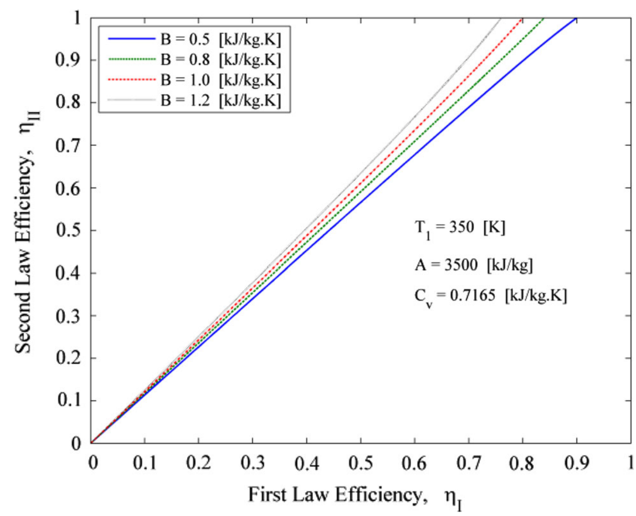

Fig. 5 Effect of B on the $\eta_{II} - \eta_I$ characteristics

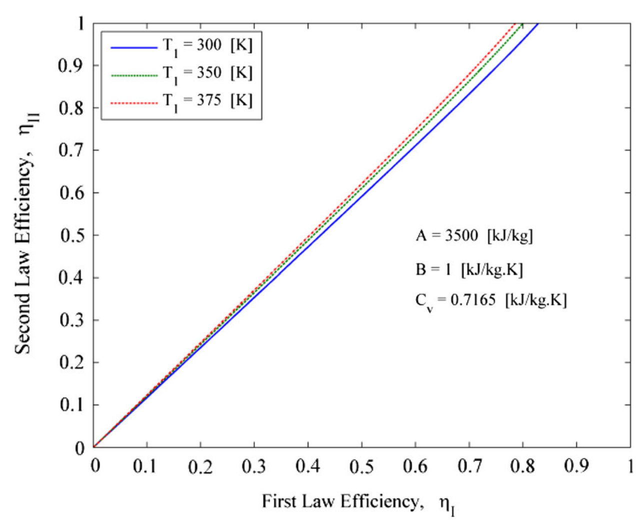

Fig. 6 Effect of T_1 on the $\eta_{II} - \eta_I$ characteristics

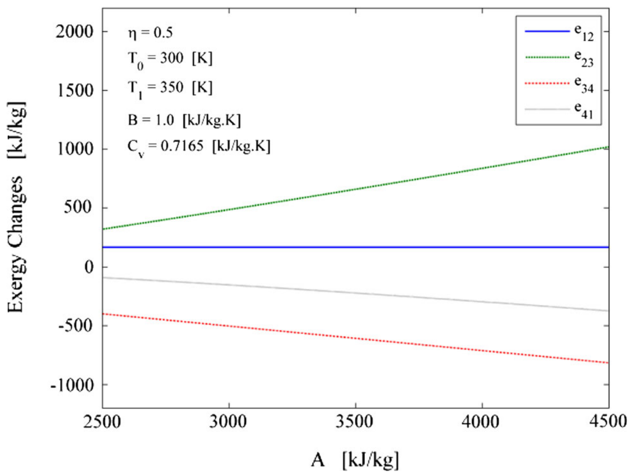

Fig. 7 Effect of A on the exergy changes

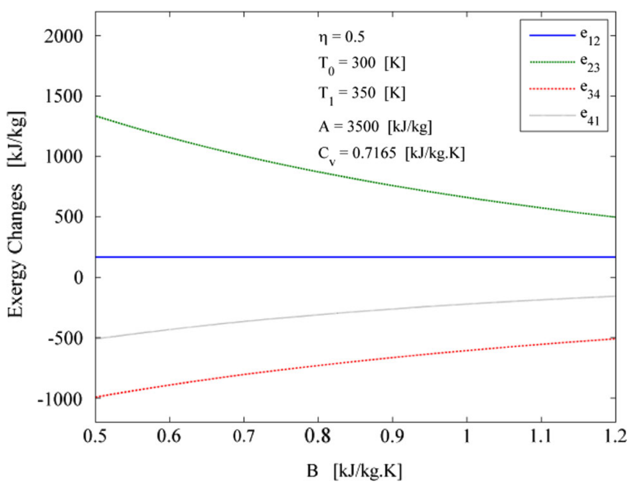

Fig. 8 Effect of B on the exergy changes

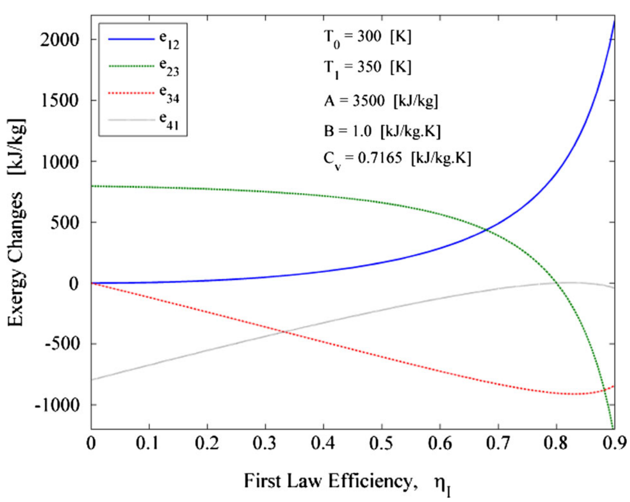

Fig. 9 The exergy changes with efficiency

the exergy change. The exergy changes during combustion, e_{23}, expansion, e_{34}, and heat rejection, e_{41}, increase with increasing value of parameter A, otherwise decrease with increasing value of parameter B, as shown in Figs. 7 and 8. This means as the maximum temperature of the mixture increases, its work potential increases.

Table 1 Effect of A on the exergy change $A = 3{,}000$ [kJ/kg] ($A = 3{,}750$ [kJ/kg])

Process or state	$e_j - e_i$[kJ/kg]	e_i[kJ/kg]
1		2,094 (2,094)
1 → 2	319.4 (319.4)	
2		2,413.4 (2,413.4)
2 → 3	376.8 (637.9)	
3		2,790.2 (3,051.3)
3 → 4	−634.6 (−832.0)	
4		2,155.6 (2,219.3)
4 → 1	−61.5 (−125.3)	
Fuel conversion efficiency $\eta_{f,i}$		0.1548 (0.2518)
Exergy conversion efficiency $\eta_{e,i}$		0.1505 (0.2448)

$C_v = 0.7165$ [kJ/kg], $r_c = 12$, $T_0 = 300$ [K], $T_1 = 333$ [K], $B = 1.0$ [kJ/kg.K]

Table 2 Effect of B on the exergy change $B = 0.5$ [kJ/kg.K] ($B = 1.0$ [kJ/kg.K])

Process or state	$e_j - e_i$[kJ/kg]	e_i[kJ/kg]
1		2,094 (2,094)
1 → 2	319.4 (319.4)	
2		2,413.4 (2,413.4)
2 → 3	1,269.6 (549.5)	
3		3,683.0 (2,962.9)
3 → 4	−1,284.4 (−766.2)	
4		2,398.6 (2,196.7)
4 → 1	−304.5 (−102.7)	
Fuel conversion efficiency $\eta_{f,i}$		0.4740 (0.2195)
Exergy conversion efficiency $\eta_{e,i}$		0.4608 (0.2134)

$C_v = 0.7165$ [kJ/kg], $r_c = 12$, $T_0 = 300$ [K], $T_1 = 333$ [K], $A = 3{,}500$ [kJ/kg]

Tables 1 and 2 also show that the effectiveness of parameters A and B on the exergy of the system at the beginning of the processes and the exergy change during each process in the form of numerical example. The positive values of exergy in each state represent the work potential in that state and the positive and negative values of exergy change during the processes indicate the type of processes of work-producing and work-consuming. The negative values mean that the exergy of system during each process is decreased and this means that work is done by the system. The positive values mean that the exergy of

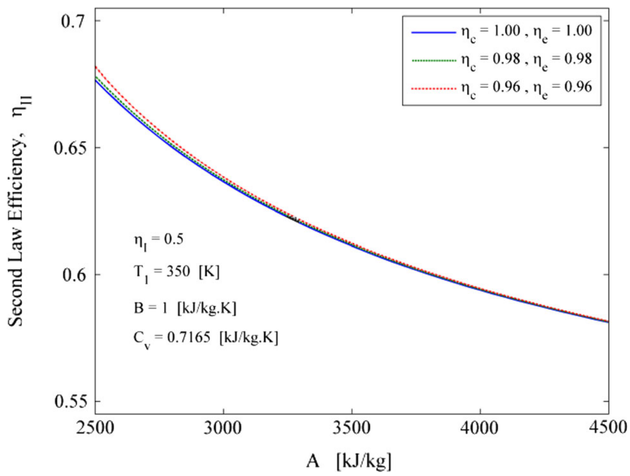

Fig. 10 Effects of η_c and η_e on the $\eta_{II} - A$ characteristics

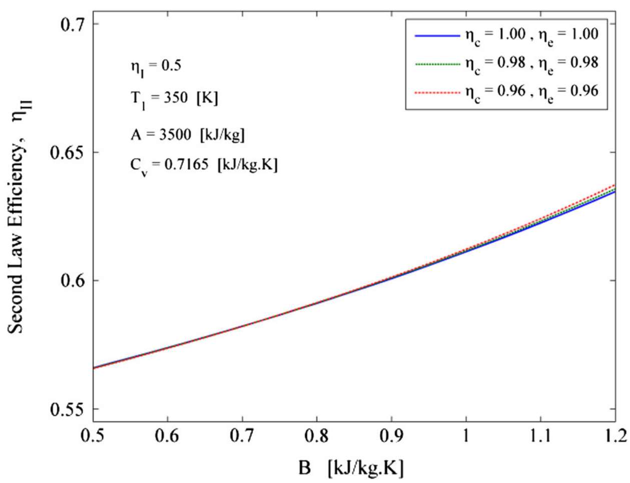

Fig. 11 Effects of η_c and η_e on the $\eta_{II} - B$ characteristics

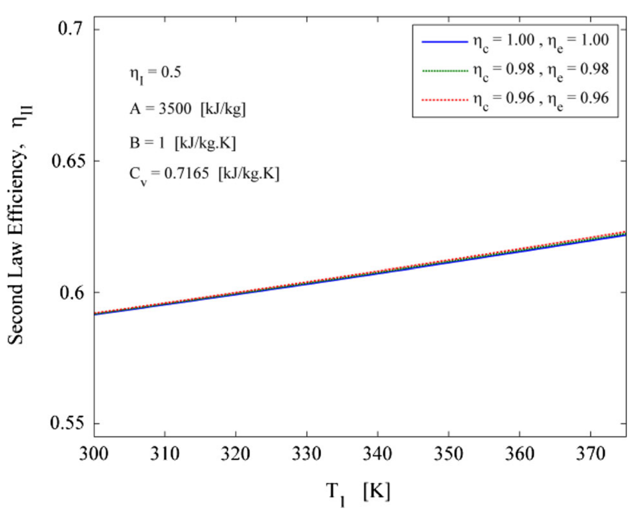

Fig. 12 Effects of η_c and η_e on the $\eta_{II} - T_1$ characteristics

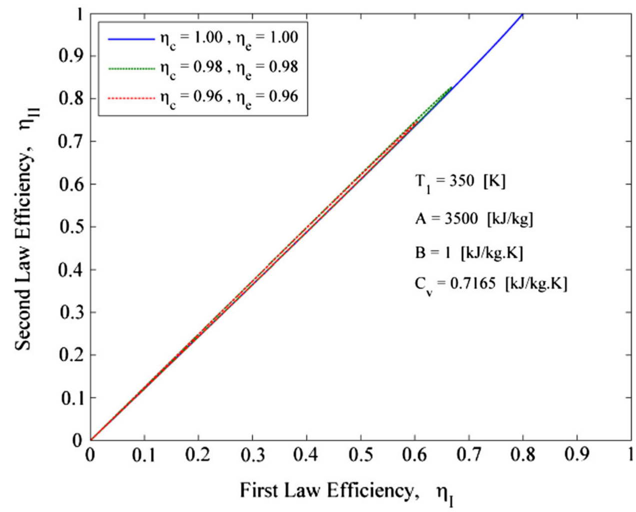

Fig. 13 Effects of η_c and η_e on the $\eta_{II} - \eta_I$ characteristics

system during each process is increased and work is done on the system.

Finally, Figs. 10, 11, 12 and 13 show the effect of the internal irreversibility (the compression efficiency, η_c, and expansion efficiencies, η_e) on the second-law efficiency. Based on these Figs., η_I and η_{II} increase with increasing η_c and η_e.

4 Conclusions

In this letter, we indicated that some used relations in HÖ2011 are incorrect and thus the reported results are invalid. In the following, we replaced wrong relations, graphs, tables and analysis by the correct ones and provided a more comprehensive analysis. The effects of the heat transfer and internal irreversibility on the second-law efficiency of Otto cycle are represented, separately. The results show, effects of the irreversibilities on performance of Otto cycle is obvious, and should be considered in analysis of practical cycles.

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