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Influence of Heat Loss on the Second-Law Efficiency of an Otto Cycle

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Abstract

The second-law efficiency is one of the most important criteria for measuring the performance of a thermodynamic system 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. In this article, the second-law efficiency of an air-standard Otto cycle with consideration of heat losses is analyzed, using finite-time thermodynamics (FTT). Moreover, influence of various design parameters such as the initial temperature of the working fluid and the constants related to combustion and heat transfer through the cylinder wall, on variation curve of the second-law efficiency versus first-law efficiency are indicated. The obtained results of this article also can be a useful guide for the analysis and comparison of real engines.

Keywords: Finite-time thermodynamics; Otto cycle, Heat loss; First-law efficiency; Second-law efficiency

1. Introduction

The Otto cycle is one of the types of air-standard cycles; also is an ideal cycle for internal combustion engines. The second-law efficiency is the ratio of the actual thermal efficiency (first-law efficiency) to the maximum possible (reversible) thermal

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efficiency under the same conditions [1]. For the work-producing devices, the second-law efficiency can also be expressed as the ratio of the useful work output and the maximum possible (reversible) work output [1, 2]. In recent years, many attentions have been paid in order to analyzing the performances of the Otto cycle and exergy analysis for it. Effects of heat loss as percentage of fuel's energy, friction and variable specific heats of working fluid on performance of air standard Otto cycle are investigated by Lin *et al.* [3]. Finite-time thermodynamic modeling and analysis of an irreversible Otto cycle is done by Ge *et al.* [4]. Heat transfer effects on the net work output and efficiency characteristics for an air-standard Otto cycle is studied by Chen *et al.* [5]. Thermodynamic simulation of performance of an Otto cycle with heat transfer and variable specific heats of working fluid performed by Ge *et al.* [6]. performance analysis of an Irreversible Otto Cycle using Finite Time Thermodynamics is done by Mehta *et al.* [7]. Finally, first and second law analysis of an ejector expansion Joule-Thomson cryogenic refrigeration cycle and first and second-laws analysis of an air-standard Dual cycle with heat loss consideration are investigated by Rashidi *et al.* [8, 9]. In this paper, the second-law analysis of an air-standard Otto cycle with heat transfer is investigated.

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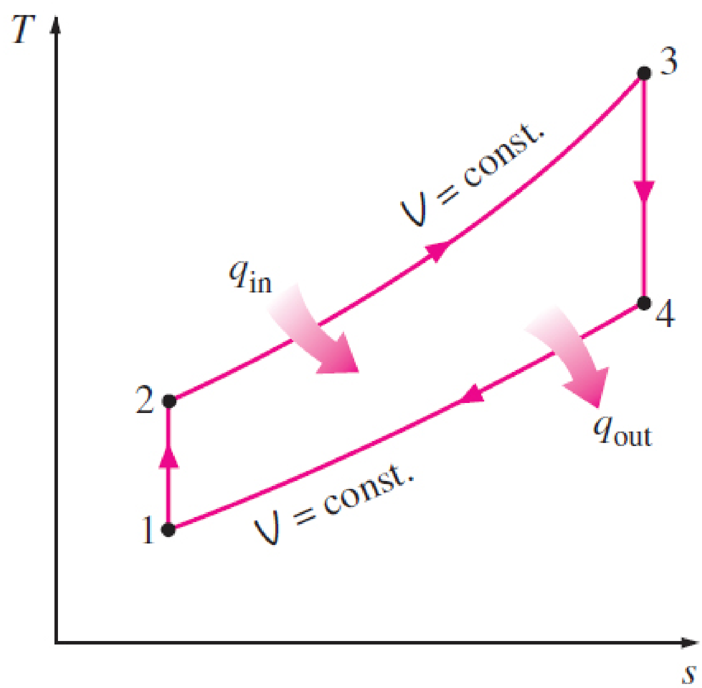

Figure 1: The *T-s* diagram of air-standard Otto cycle [1].

2. Thermodynamic Analysis

The *T-s* diagram for an air-standard Otto cycle is shown in figure 1. The Otto cycle consists of four processes; isentropic compression (1→2), constant-volume heat addition (2→3), isentropic expansion (3→4) and constant-volume heat rejection (4→1). For an ideal air-standard Otto cycle, the heat added per unit mass of the working fluid during the constant-volume process (2→3) is defined as:

$$q_{\mathrm{in}} = C_{\mathrm{V}} \left(T_3 - T_2 \right), \tag{1}$$

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where, C_V is the constant-volume specific heat. The temperatures within the combustion chamber of an internal combustion engine reach values approximately 2700 (K) and up. Materials in the engine cannot tolerate this kind of temperature and would quickly fail if proper heat transfer did not occur. Thus, because of keeping an engine and engine lubricant 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 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 complicated, so it is approximately assumed to be proportional to the average temperature of both the working fluid and cylinder wall and that, during the operation, the wall temperature remains approximately invariant. The heat added per unit mass of the working fluid of the cycle by combustion is given by the following linear relation [6]:

$$q_{\text{in}} = A - B\left(T_2 + T_3\right), \tag{2}$$

where, A and B are two constants related to the combustion and the heat transfer, respectively. Parameter A is the total heat energy of the fuel and parameter B is the waste energy by heat loss. Combining equations (1) and (2) yields:

$$T_3 = \frac{\left[A + \left(C_V - B\right)T_2\right]}{\left(C_V + B\right)}. \tag{3}$$

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

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$$T_2 = T_1 r_c^{k-1}, \tag{4}$$

and

$$T_4 = T_3 r_c^{1-k}, \tag{5}$$

where, r_c is the compression ratio, (V_1/V_2), and k is the specific heat ratio (C_P/C_V). For the air-standard Otto cycle, the heat rejected per unit mass of the working fluid during the constant-volume process (4→1) is defined as:

$$q_{out} = C_V\left(T_4 - T_1\right). \tag{6}$$

The net work output per unit mass of the working fluid for the Otto cycle is given by the following equation:

$$w_{net} = q_{in} - q_{out} = C_V\left(T_3 - T_2\right) - C_V\left(T_4 - T_1\right), \tag{7}$$

where, T_1, T_2, T_3 and T_4 are absolute temperatures at states 1, 2, 3 and 4. By substituting equations (3), (4) and (5) into equations (1) and (7), the first-law efficiency (thermal efficiency) of the Otto cycle will be obtained as:

$$\eta_I = \frac{w_{net}}{q_{in}} = 1 - r_c^{k-1}. \tag{8}$$

According to References [1, 2], the second-law efficiency of a system is defined as:

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$$\eta_{\mathrm{II}} = \frac{w_{\mathrm{net}}}{w_{\max}}, \tag{9}$$

where, $w_{\max}$ is the maximum possible work that the system can produce which is defined as:

$$w_{\max} = \eta_{\max} \cdot q_{\mathrm{in}}, \tag{10}$$

where, $\eta_{\max}$ is the maximum possible thermal efficiency (Carnot cycle efficiency) which is defined as following:

$$\eta_{\max} = \left(1 - \frac{T_1}{T_3}\right). \tag{11}$$

Substituting equations (1) and (11) into equation (10) yields:

$$w_{\max} = C_{\mathrm{V}}\left(T_3 - T_2\right)\left(1 - \frac{T_1}{T_3}\right). \tag{12}$$

Finally, by some simple algebraic calculations, w_{net}, $w_{\max}$ and η_{II} will be obtained as follows:

$$w_{\mathrm{net}} = \eta_{\mathrm{I}} \cdot q_{\mathrm{in}} = \frac{C_{\mathrm{V}}\eta_{\mathrm{I}}\left[A - 2BT_1/\left(1-\eta_{\mathrm{I}}\right)\right]}{\left(C_{\mathrm{V}} + B\right)}, \tag{13}$$

and

$$w_{\max} = C_{\mathrm{V}}\left[\frac{A(1-\eta_{\mathrm{I}})-2BT_1}{(1-\eta_{\mathrm{I}})(C_{\mathrm{V}}+B)}\right]\times\left[\frac{A(1-\eta_{\mathrm{I}})+\eta_{\mathrm{I}}(C_{\mathrm{V}}+B)T_1-2BT_1}{A(1-\eta_{\mathrm{I}})+(C_{\mathrm{V}}-B)T_1}\right], \tag{14}$$

and

$$\eta_{\mathrm{II}} = \frac{\eta_{\mathrm{I}}\left[A(1-\eta_{\mathrm{I}})+(C_{\mathrm{V}}-B)T_1\right]}{A(1-\eta_{\mathrm{I}})+\eta_{\mathrm{I}}(C_{\mathrm{V}}+B)T_1-2BT_1}. \tag{15}$$

3. Results and Discussions

In this paper, the following parameters are used: $A = 2500 \rightarrow 4500$ (kJ/kg.K), $B = 0.5 \rightarrow 1.2$ (kJ/kg.K), $k = 1.35$, $T_1 = 280 \rightarrow 320$ (K). Figure 2 illustrates the effects of parameters A, B and T_1 on the second-law efficiency for a fixed value of the first-law efficiency. Based on this figure, the second-law efficiency increases, with decreasing A, and increasing B and T_1. Figures 3, 4 and 5 indicate effects of parameters A, B and T_1 on variation curve of the second-law efficiency versus the first-law efficiency. It can be seen that the first and second law efficiencies increase with increasing the compression ratio, r_c. Moreover, it is found that, for a given compression ratio, the second-law efficiency increases, with the decrease of A, and the increase of B and T_1. This means that, for these variations, the maximum possible thermal efficiency will decrease that, finally, cause slope of above curve to increase. It can be seen that, the results of the figure 2, are confirmed by figures 3, 4 and 5. Note that some values of the second-law efficiency might be insufficient for a feasible Otto cycle.

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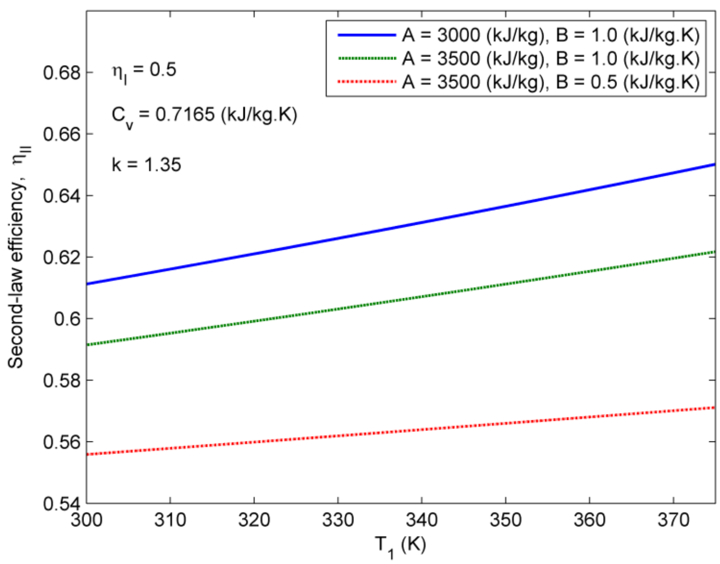

Figure 2: Effects of A and B on the $\eta_{\mathrm{II}} - T_1$ characteristics.

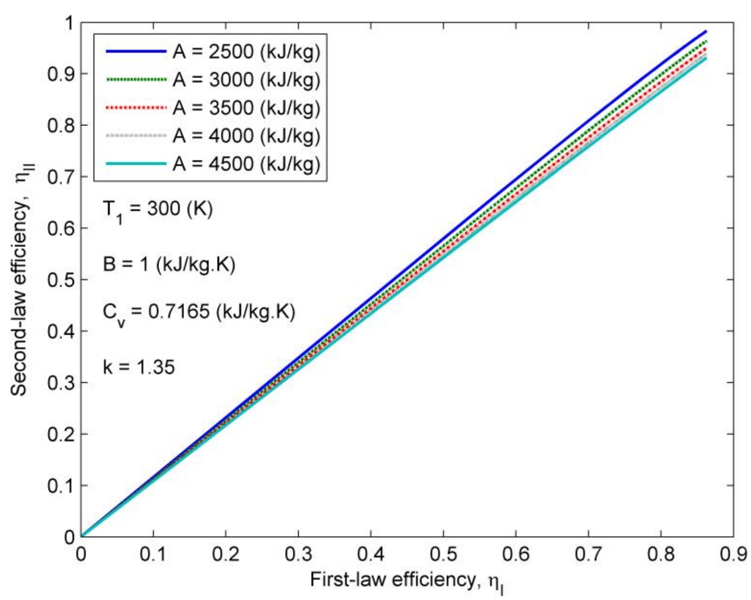

Figure 3: Effect of A on the η_{II} - η_{I} characteristics.

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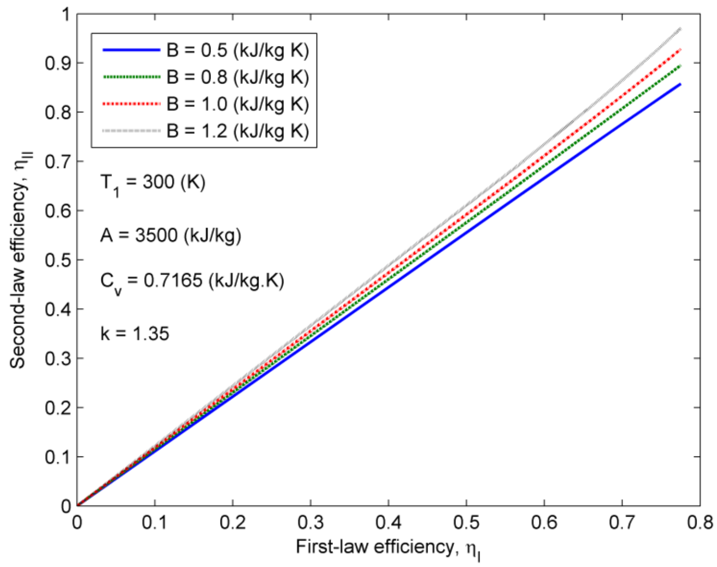

Figure 4: Effect of B on the η_{II} - η_{I} characteristics.

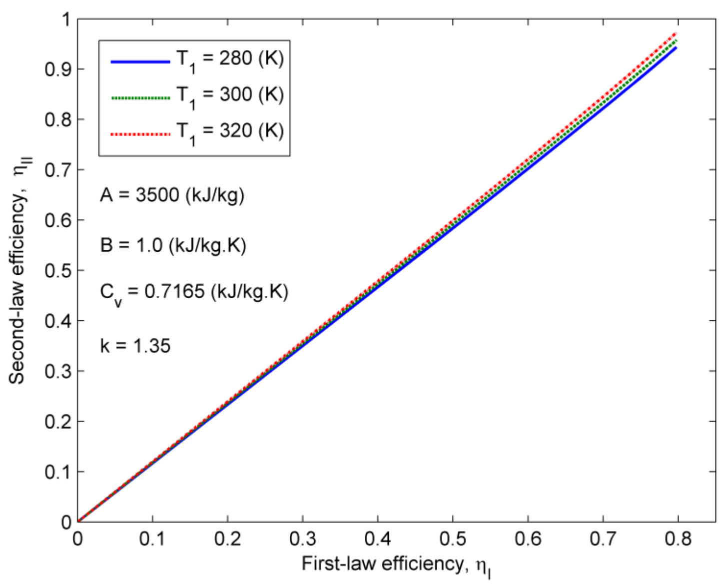

Figure 5: Effect of T_1 on the η_{II} - η_{I} characteristics.

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Conclusion

In this paper, the influence of heat loss through the cylinder wall on performance of an air-standard Otto cycle has been investigated, using first and second-law efficiencies. The results show that the effect of the heat loss on performance of the cycle is obvious, and they should be considered in practical cycle analysis. The finding in this paper may provide significant guidance for the design of practice internal combustion engines and improvements of real reciprocating engines.

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