First and second-laws analysis of an air-standard Dual cycle with heat loss consideration

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Abstract

In this article, the first and second-laws analysis of the thermodynamic Dual cycle with considering of heat loss are investigated by using finite-time thermodynamics (FTT). The influences of various factors (e.g. initial temperature of working fluid, constants related to combustion and heat transfer through the cylinder wall) on the performance of Dual cycle are analyzed. As well as, the curve of the first-law efficiency versus compression ratio, the net work output versus compression ratio, the second-law efficiency versus compression ratio and the second-law efficiency versus the first-law efficiency are indicated. The finding results in this article can be useful for analysis of Dual cycles.

Keywords: Finite-time thermodynamics; Air-standard Dual cycle; Heat loss; Compression ratio; Net work output; First-law efficiency; Second-law efficiency

1. Introduction

The air-standard Dual cycle (or Dual combustion cycle) is a thermodynamic cycle that is a combination of the Otto cycle and the Diesel cycle, first introduced by Russian-German engineer Gustav Trinkler. The first-law efficiency (thermal efficiency) is an important thermodynamic parameter for analysis of a cycle. The Second-law analysis is a good benchmark for the availability of systems. The second-law efficiency is the ratio of the actual first-law efficiency to the maximum possible (reversible) thermal 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 to the maximum possible (reversible) work output [1, 2]. In recent years, many attentions have been paid in order to analyzing the performances of the Dual cycle, other air-standard cycles and second-law analysis. Ozsoysal [3] investigated effects of combustion efficiency on a Dual cycle. Chen *et al.* [4] studied effects of heat transfer, friction and variable specific heats of working fluid on performance of an irreversible Dual cycle. Ge *et al.* [5] investigated finite-time thermodynamic modeling and analysis for an irreversible Dual cycle. Hou [6] studied heat transfer effect on the performance of an air standard Dual cycle. Chen *et al.* [7] studied optimal performance of an irreversible Dual-cycle. Ust *et al.* [8] investigated optimizations of a Dual cycle cogeneration system based on a new exergetic performance criterion. Wang *et al.* [9] investigated the effects of friction on the performance of an air standard Dual cycle. Ahmadi *et al.* [10] studied thermodynamic analysis and thermoeconomic optimization of a Dual pressure combined cycle power plant with a supplementary firing unit. Parlak [11] investigated comparative performance analysis of irreversible Dual and Diesel cycles under maximum power conditions. Gahruei *et al.* [12] studied Mathematical modeling and comparison of air standard Dual and Dual-Atkinson cycles with friction, heat transfer and variable specific heats of the working fluid. Lior *et al.* [13] investigated energy, exergy, and second Law performance criteria and Rashidi *et al.* [14] studied first and second law analysis of an ejector expansion Joule–Thomson cryogenic refrigeration cycle. In this article, the first and the second-laws analysis of an air-standard Dual cycle with heat loss is investigated.

2. Thermodynamic analysis

The T-s and the P-v diagrams of an air-standard Dual cycle are shown in **Figures 1** and **2**. The Dual cycle consists of five processes, isentropic compression (1→2), heat addition occur at two processes, constant-volume (2→3) and constant-pressure (3→4), isentropic expansion (4→5) and constant-volume heat rejection (5→1).

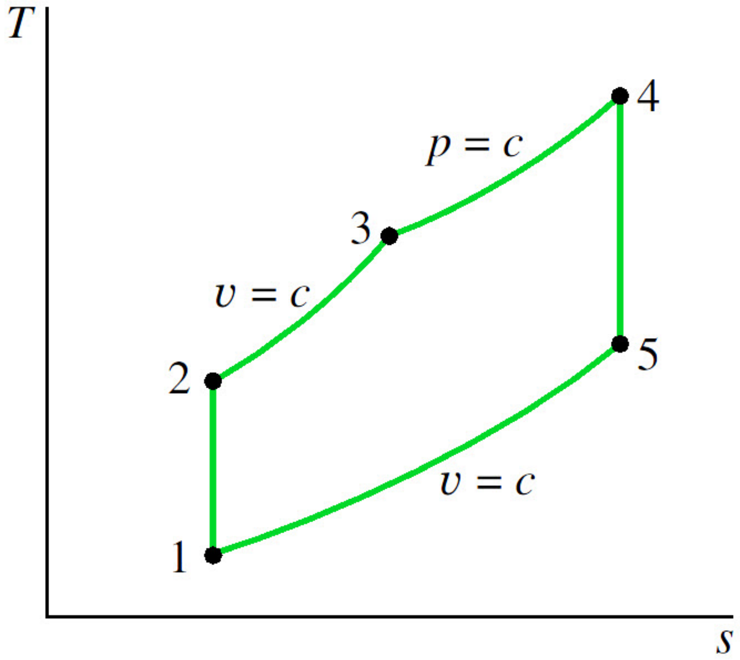

Figure 1. The T-s Diagram of an air-standard Dual cycle.

For an ideal Dual cycle, the heat added per unit mass of the working fluid during constant-volume and constant-pressure processes is defined as:

$$q_{in} = C_v\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right), \tag{1}$$

where, C_v and C_p are the constant-volume and constant-pressure specific-heat, respectively. For a real Dual cycle, the heat loss between the working fluid and the cylinder wall is not negligible. It is assumed that the heat loss through the cylinder wall is proportional to the average temperature of both the working fluid and the cylinder wall and that the wall temperature is assumed constant. The heat added per unit mass of the working fluid by combustion is defined as [4]:

$$q_{in} = A - B\left(T_2 + T_4\right), \tag{2}$$

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

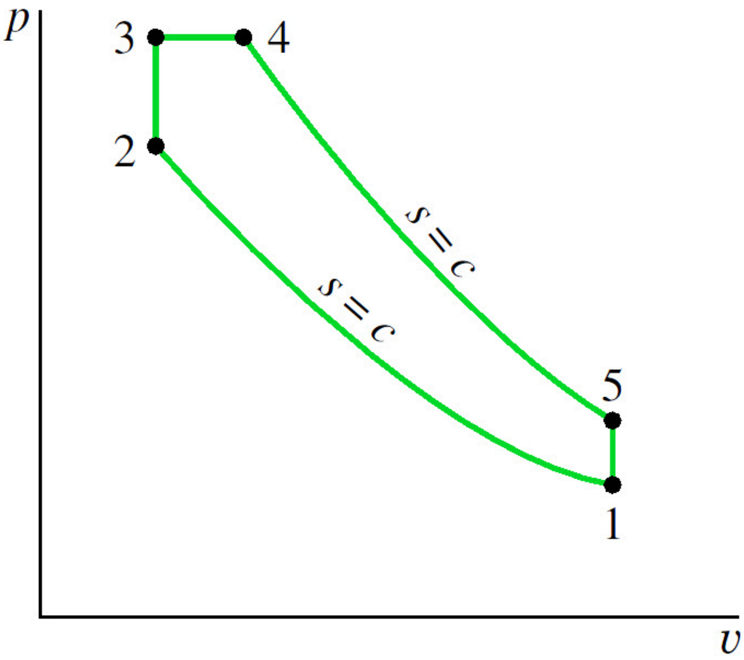

Figure 2. The P-v Diagram of an air-standard Dual cycle.

$$T_4 = \frac{\left[A + T_2\left(C_v - B\right) - T_3\left(C_v - C_p\right)\right]}{\left(C_p + B\right)}. \tag{3}$$

Defining the compression ratio, r_c, the pressure ratio, r_p, and the cut-off ratio, r, as follow:

$$r_c = \frac{V_1}{V_2}, \tag{4}$$

$$r_p = \frac{T_3}{T_2}, \tag{5}$$

and

$$r = \frac{V_4}{V_3} = \frac{T_4}{T_3} \cdot \tag{6}$$

For the isentropic process (1⟶2), we have:

$$T_2 = T_1 r_c^{k-1}. \tag{7}$$

According to Eq. (5), the temperature of the state 3 is defined as:

$$T_3 = T_2 r_p. \tag{8}$$

For the isentropic process (4→5), we have:

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{k-1} = \left(\frac{V_4}{V_2}\cdot\frac{V_2}{V_5}\right)^{k-1} = \left(\frac{r}{r_c}\right)^{k-1}. \tag{9}$$

Thus,

$$T_5 = T_4\left(\frac{r}{r_c}\right)^{k-1}. \tag{10}$$

For an air-standard Dual cycle, the heat rejected per unit mass by the working fluid during constant-pressure process is defined:

$$q_{out} = C_v\left(T_5 - T_1\right). \tag{11}$$

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

$$w_{net} = q_{in} - q_{out} = C_v\left(T_3 + T_1 - T_2 - T_5\right) + C_p\left(T_4 - T_3\right). \tag{12}$$

Finally, the first-law efficiency is defined:

$$\eta_I = \frac{w_{net}}{q_{in}} = \frac{C_v\left(T_3 + T_1 - T_2 - T_5\right) + C_p\left(T_4 - T_3\right)}{C_v\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right)}. \tag{13}$$

The second-law 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. For the work-producing devices, the second-law efficiency can also be expressed as the ratio of the useful net work output to the maximum possible (reversible) net work output [1, 2]. According to above description, the second-law efficiency of an air-standard Dual cycle is defined as:

$$\eta_{II} = \frac{w_{net}}{w_{max}}, \tag{14}$$

where, w_{max} is maximum possible net work of the Dual cycle. On the other hand, w_{max} is defined as follow:

$$w_{max} = q_{in}\,\eta_{max}, \tag{15}$$

where, η_{max} is maximum efficiency (the Carnot efficiency) and for the Dual cycle defined as:

$$\eta_{II} = \left(1 - \frac{T_1}{T_4}\right). \tag{16}$$

So, according to Eqs. (1) and (16), Eq. (15) can be written as:

$$w_{max} = \left[C_v\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right)\right]\left(1 - \frac{T_1}{T_4}\right). \tag{17}$$

Finally, by substituting Eqs. (12) and (17) into Eq. (14), the second-law efficiency of an air-standard Dual cycle is as follows:

$$\eta_{II} = \frac{C_v\left(T_3 + T_1 - T_2 - T_5\right) + C_p\left(T_4 - T_3\right)}{\left[C_v\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right)\right]\left(1 - \dfrac{T_1}{T_4}\right)}. \tag{18}$$

3. Numerical calculations and discussion

In this paper, the following parameters are used:

$T_1 = 280 \rightarrow 320\ (K),\ A = 3500 \rightarrow 4500\ (kJ/kg),\ B = 0.8 \rightarrow 1.2\ (kJ/kg.K),\ r_c = 1 \rightarrow 55,$
$C_p = 1.003\ (kJ/kg.K),\ C_v = 0.716\ (kJ/kg.K),\ r = 1.6,\ r_p = 1.2.$

Figures 3-5, show that the effects of parameters A, B and T_1 on curve of the first-law efficiency versus the compression ratio. A and B are the total input heat and the loss heat respectively. According to these graphs and Eq. (2), the first-law efficiency increases with increasing A and with decreasing B and T_1.

Figures 6-8, show that the effects of parameters A, B and T_1 on curve of the net work output versus the compression ratio. It can be seen that the net work output increases with increasing A and with decreasing B and T_1. The maximum value of the net work output occur at the compression ratio less 20.

Figures 9-11, show that the effects of parameters A, B and T_1 on curve of the net work output versus the first-law efficiency. It can be seen that these figures are loop-shaped. The net work output and first-law efficiency increase with increasing A and with decreasing B and T_1.

Effects of parameters A, B and T_1 on curve of the second-law efficiency versus the compression ratio illustrate in **figures 12-14**. The second-law efficiency versus the compression ratio under variation of parameters A, B and T_1 behave like the first-law efficiency at the same condition (**figures 3-5**).

Figures 15-17, show that the effects of parameters A, B and T_1 on curve of the net work output versus the second-law efficiency. It can be seen that these figures are loop-shaped too and the maximum value of the second-law efficiency occur at the maximum value of the net work output. The net work output and second-law efficiency increase with increasing A and with decreasing B and T_1.

Finally, the curves of the first-law efficiency versus the second-law efficiency with variation of parameters A, B and T_1 depict in **figures 18-20**. These curves are sharp loop-shaped and the maximum value of the second-law efficiency occur at the maximum value of the first-law efficiency.

4. Figures and results

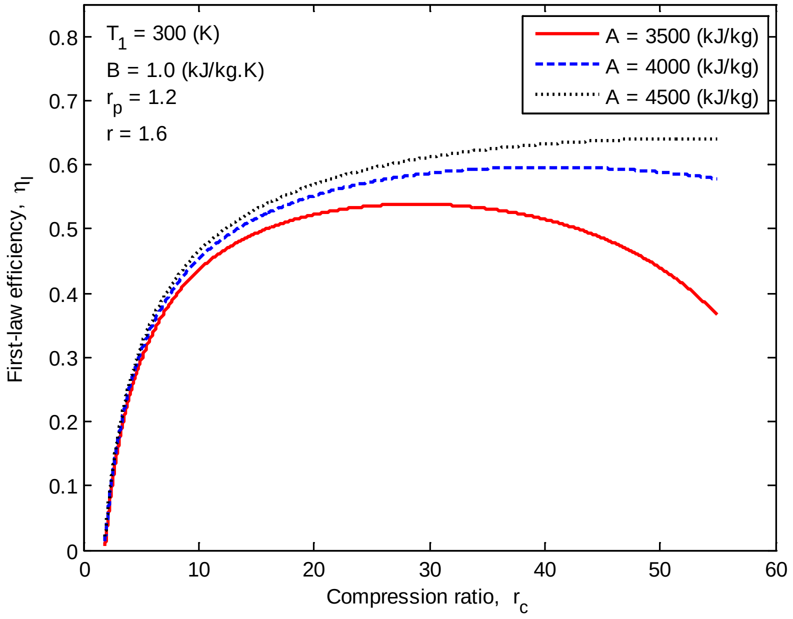

Figure 3. Effect of A on curves of the fist-law efficiency versus the compression ratio.

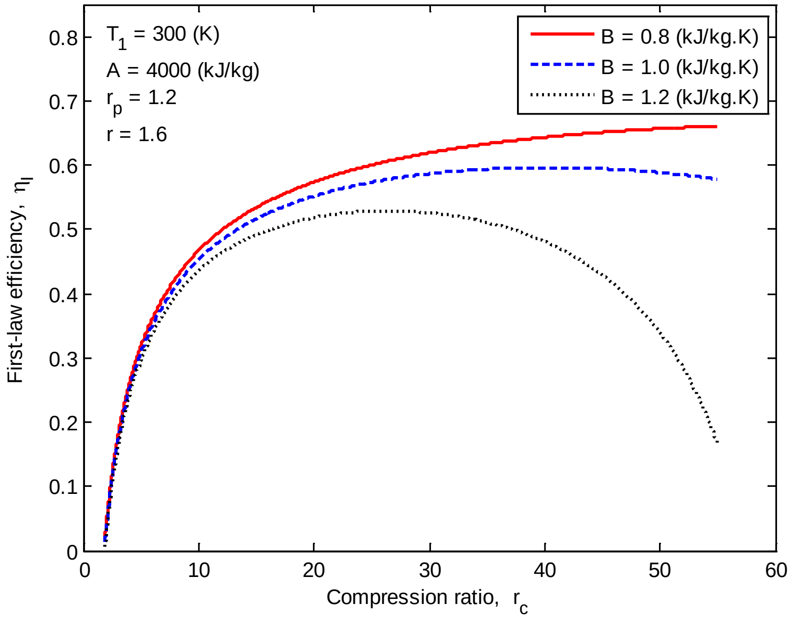

Figure 4. Effect of B on curves of the fist-law efficiency versus the compression ratio.

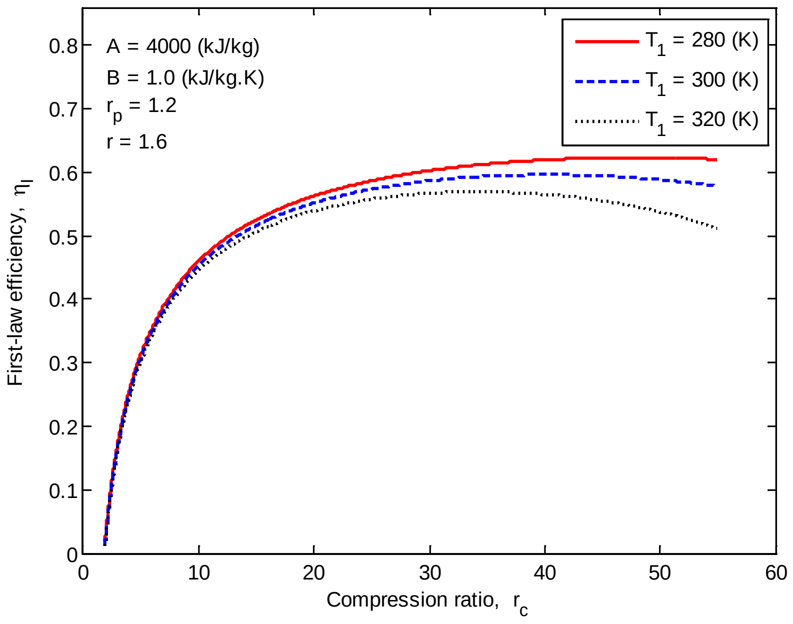

Figure 5. Effect of T_1 on curves of the fist-law efficiency versus the compression ratio.

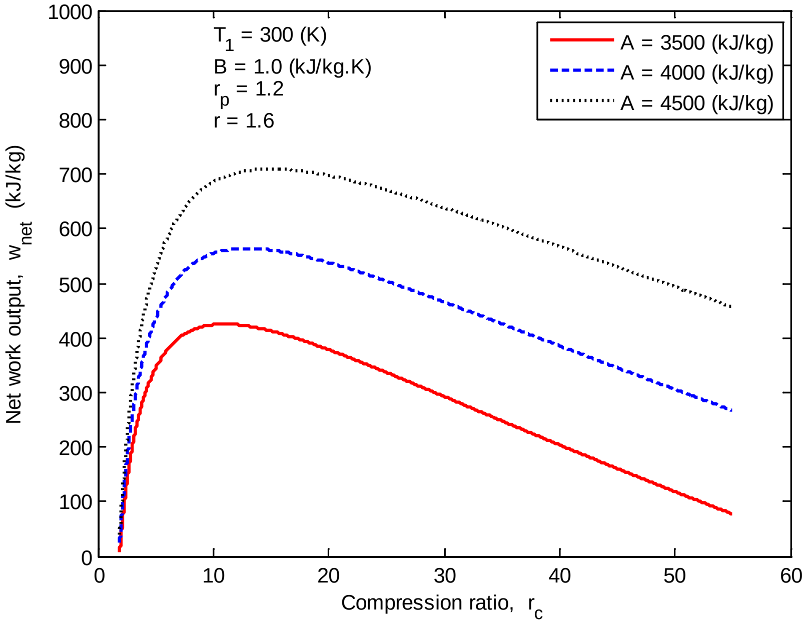

Figure 6. Effect of A on curves of the net work output versus the compression ratio.

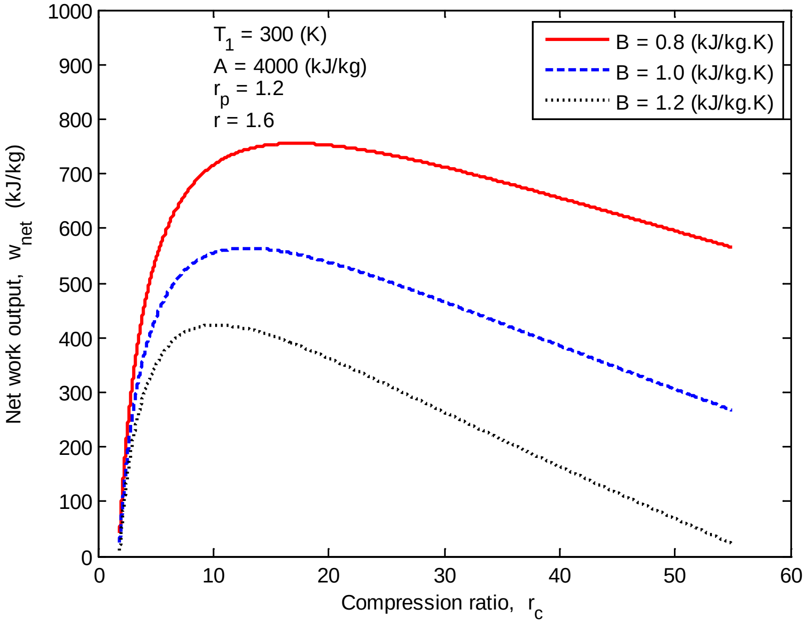

Figure 7. Effect of B on curves of the net work output versus the compression ratio.

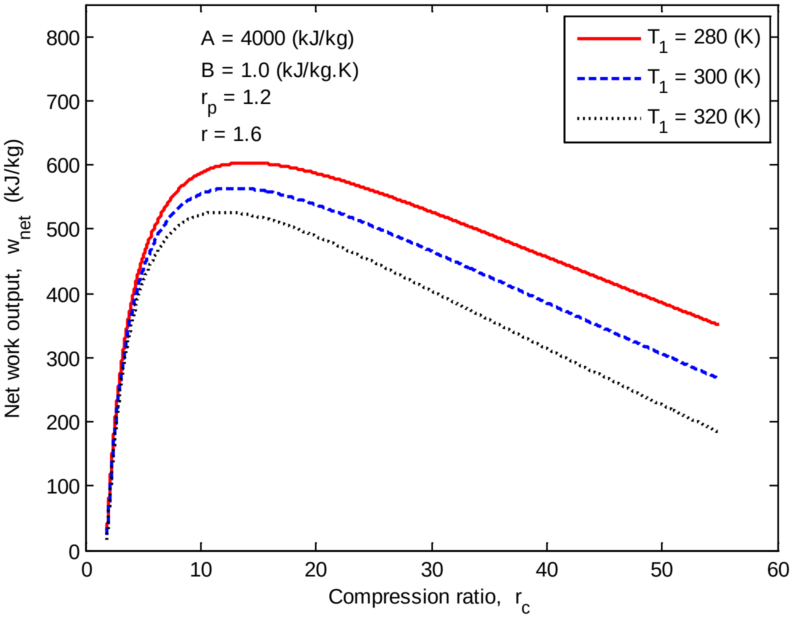

Figure 8. Effect of T_1 on curves of the net work output versus the compression ratio.

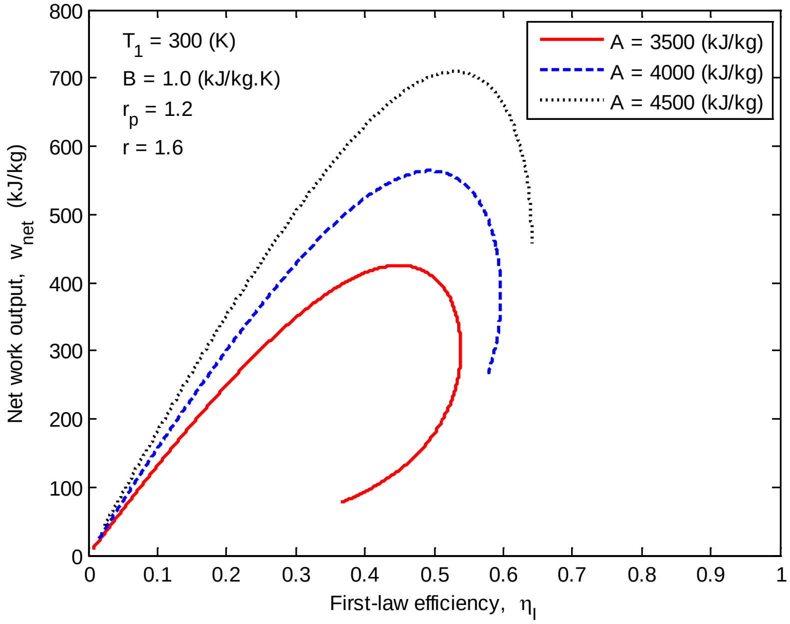

Figure 9. Effect of A on curves of the net work output versus the first-law efficiency.

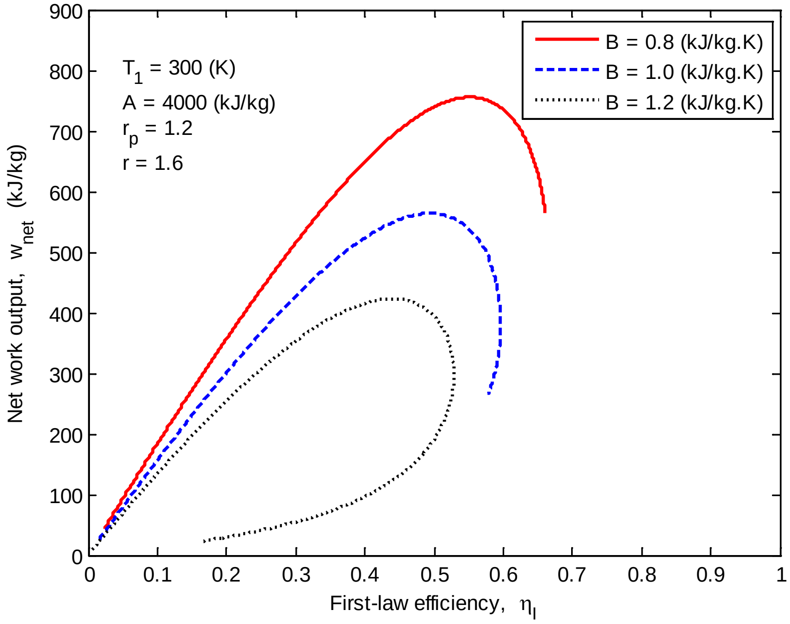

Figure 10. Effect of B on curves of the net work output versus the first-law efficiency.

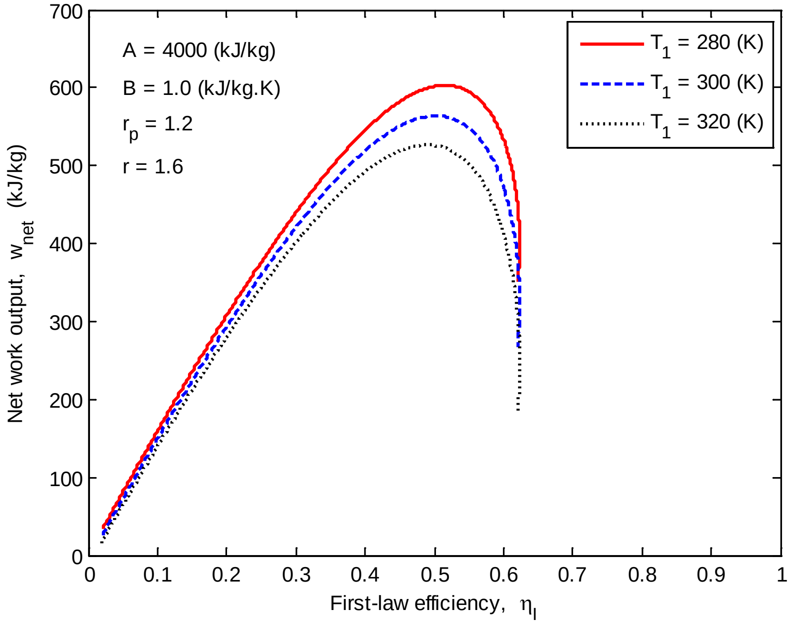

Figure 11. Effect of T_1 on curves of the net work output versus the first-law efficiency.

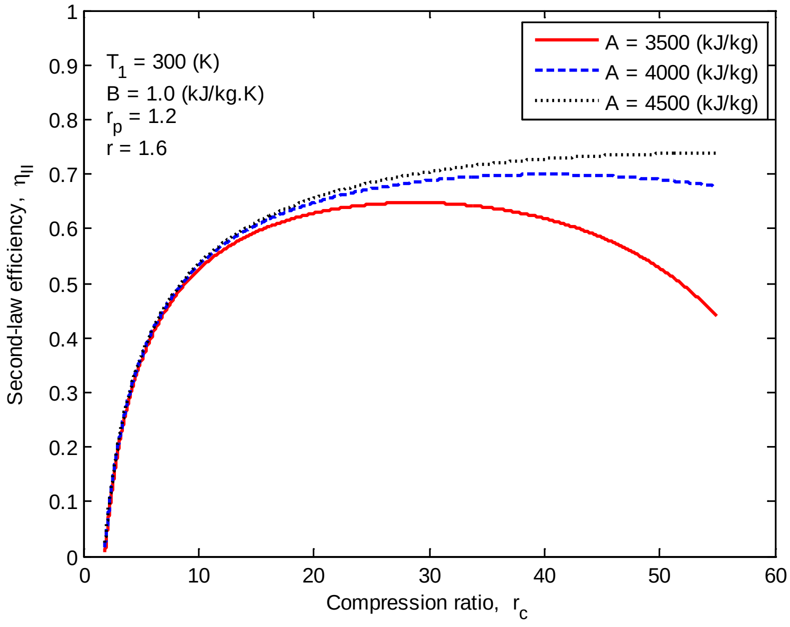

Figure 12. Effect of A on curves of the second-law efficiency versus the compression ratio.

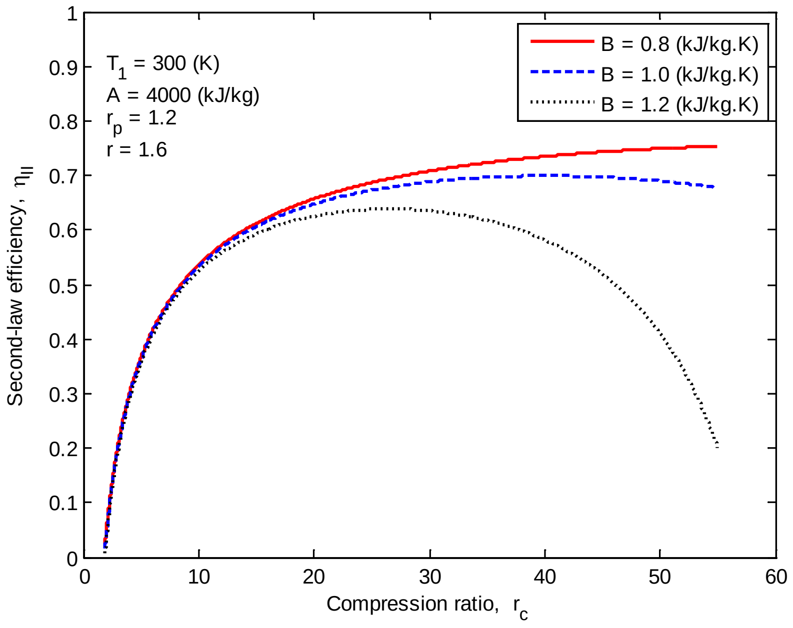

Figure 13. Effect of B on curves of the second-law efficiency versus the compression ratio.

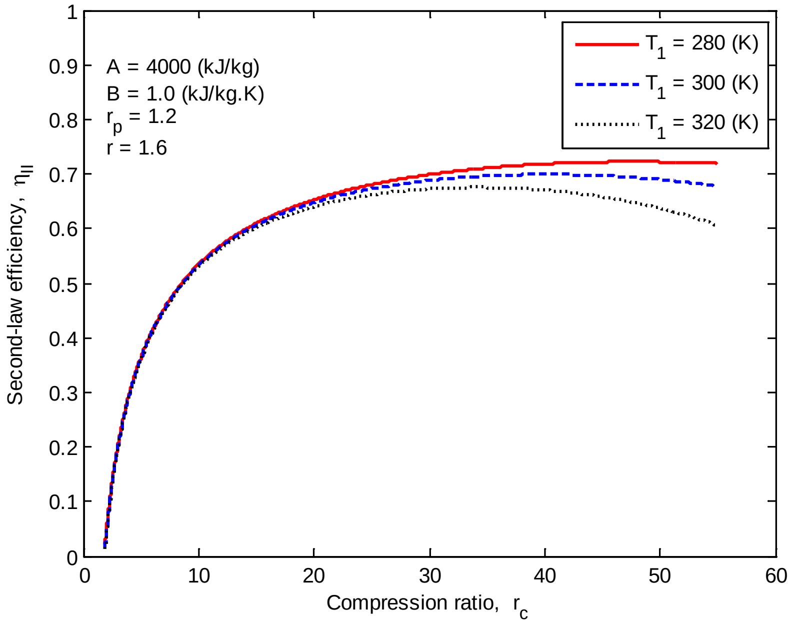

Figure 14. Effect of T_1 on curves of the second-law efficiency versus the compression ratio.

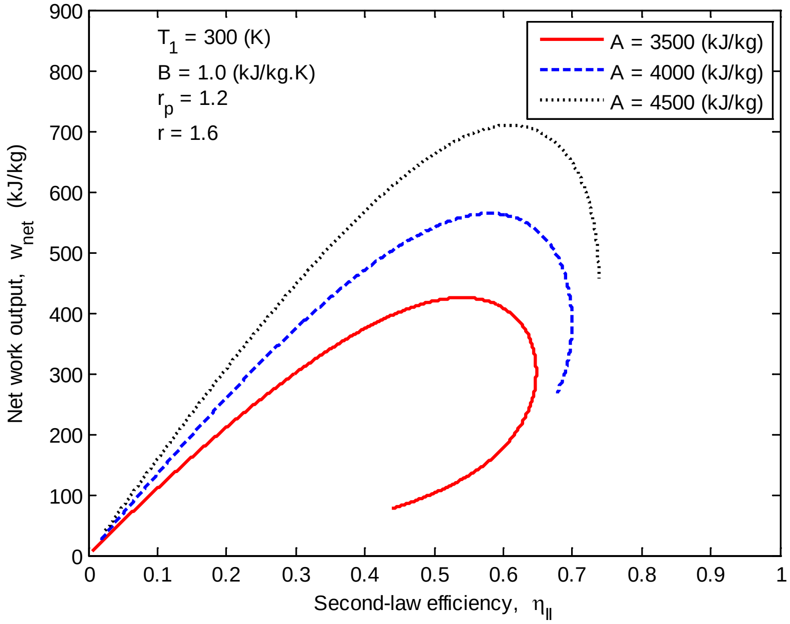

Figure 15. Effect of A on curves of the net work output versus the second-law efficiency.

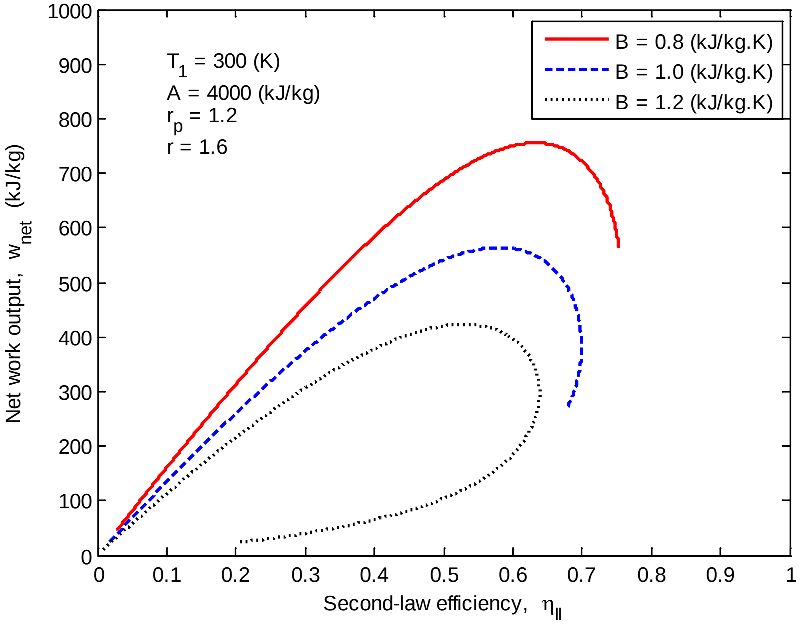

Figure 16. Effect of B on curves of the net work output versus the second-law efficiency.

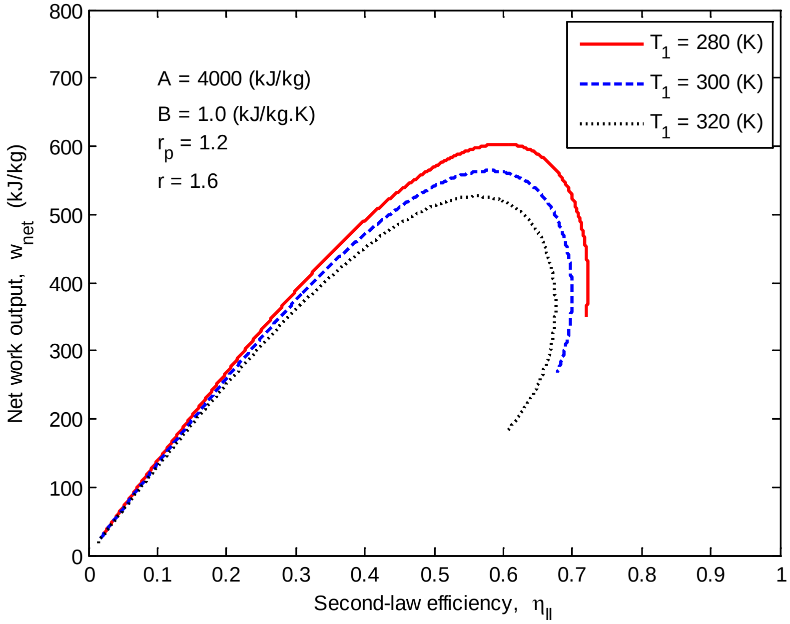

Figure 17. Effect of T_1 on curves of the net work output versus the second-law efficiency.

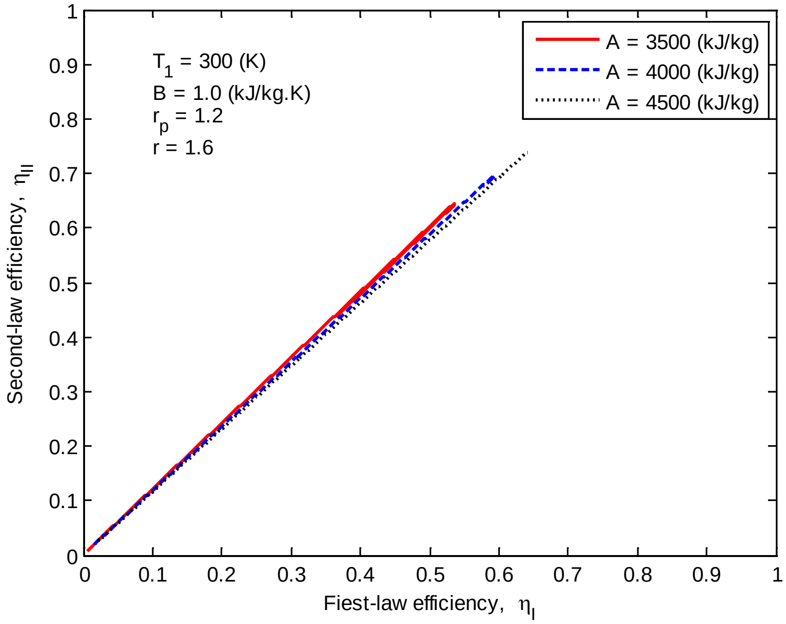

Figure 18. Effect of A on curves of the second-law efficiency versus the first-law efficiency.

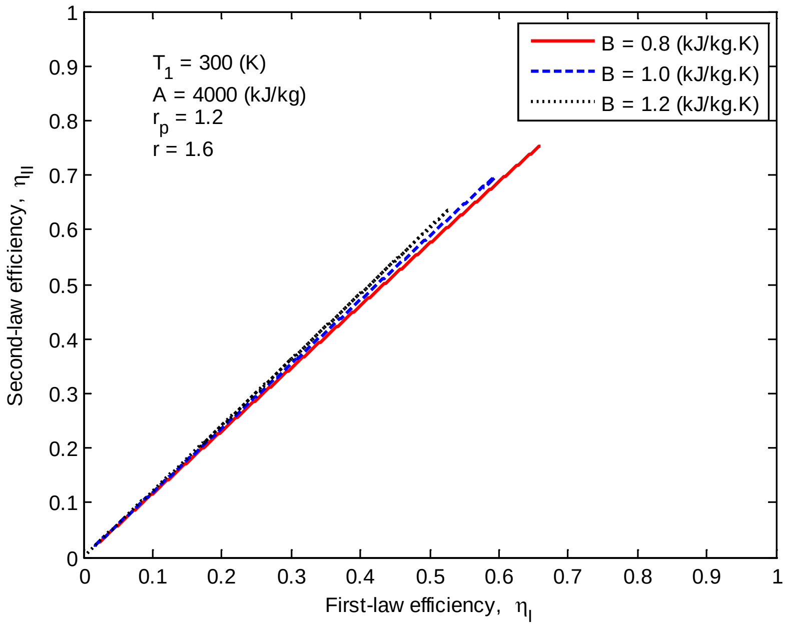

Figure 19. Effect of B on curves of the second-law efficiency versus the first-law efficiency.

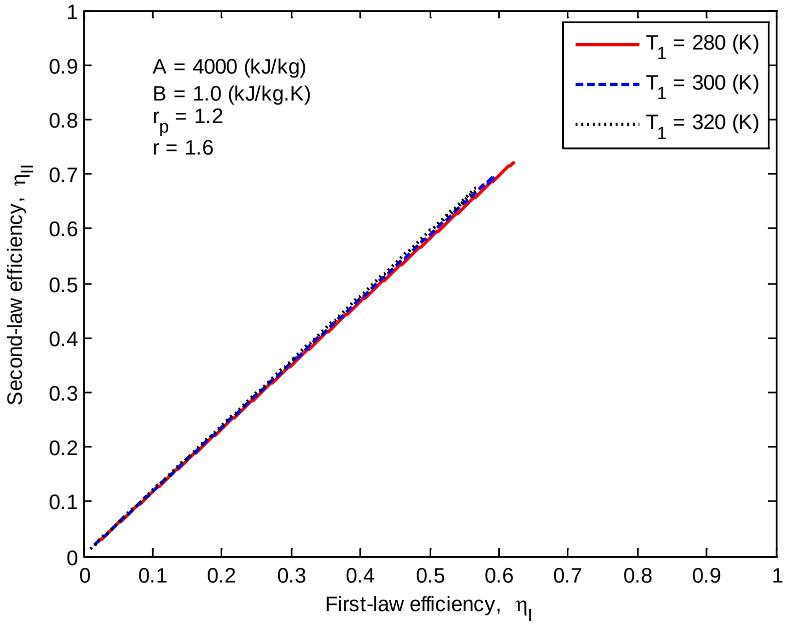

Figure 20. Effect of T_1 on curves of the second-law efficiency versus the first-law efficiency.

5. Conclusion

In this article, the first and second-laws analysis of an air-standard Dual cycle with consideration of heat loss have been analyzed. The effects of initial temperature, combustion and heat transfer factors on the first-law efficiency, second-law efficiency and the net work output have been shown. The finding results of this paper are obvious and should be considered in practical air-standard cycle analysis.

6. ACKNOWLEDGEMENTS

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12. References

[1] Y. A. Cengel, M. A. Boles, Thermodynamics: an engineering approach, 7th Ed, McGraw-Hill Book Company, (2010).

[2] A. Bejan, Advanced engineering thermodynamics, Hoboken, New Jersey: John Wiley & Sons INC, (2006).

[3] O. A. Ozsoysal. Effects of combustion efficiency on a Dual cycle. Energy Conversion and Management, 50 (2009), pp. 2400–2406.

[4] L. Chen, Y. Ge, F. Sun, C, Wu. Effects of heat transfer, friction and variable specific heats of working fluid on performance of an irreversible Dual cycle. Energy Conversion and Management, 47 (2006), pp. 3224–3234.

[5] Y. Ge, L. Chen, F. Sun. Finite-time thermodynamic modeling and analysis for an irreversible Dual cycle. Mathematical and Computer Modelling, 50 (2009), pp. 101_108.

[6] S. S. Hou. Heat transfer effects on the performance of an air standard Dual cycle. Energy Conversion and Management, 45 (2004), pp. 3003–3015.

[7] L. Chen, F. Sun, C Wu. Optimal performance of an irreversible Dual-cycle. Applied Energy, 79 (2004), pp. 3–14.

[8] Y. Ust, B. Sahin, A. Kodal. Optimization of a Dual cycle cogeneration system based on a new exergetic performance criterion. Applied Energy, 84 (2007), pp. 1079–1091.

[9] W. Wang, L. Chen, F. Sun, C. Wu. The effect of friction on the performance of an air standard dual cycle. Exergy, an International Journal, 2 (2002), pp. 340–344.

[10] P. Ahmadi, I. Dincer. Thermodynamic analysis and thermoeconomic optimization of a Dual pressure combined cycle power plant with a supplementary firing unit. Energy Conversion and Management, 52 (2011), pp. 2296–2308.

[11] A. Parlak. Comparative performance analysis of irreversible Dual and Diesel cycles under maximum power conditions. Energy Conversion and Management, 46 (2005), pp. 351–359.

[12] M. H. Gahruei. H. S. Jeshvaghani, S. Vahidi. L. Chen. Mathematical modeling and comparison of air standard Dual and Dual-Atkinson cycles with friction, heat transfer and variable specific-heats of the working fluid. Applied Mathematical Modelling, xxx (2013), pp. xxx–xxx.

[13] N. Lior, N. Zhang. Energy, exergy, and Second Law performance criteria. Energy, 32 (2007), pp. 281–296.

[14] M. M. Rashidi, O. Anwar Beg, A. Habibzadeh. First and second law analysis of an ejector expansion Joule–Thomson cryogenic refrigeration cycle. International Journal of Energy Research, 36 (2012), pp. 231–240.

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