HEAT TRANSFER EFFECTS ON THE PERFORMANCE OF AN AIR STANDARD OTTO CYCLE

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Abstract
There are heat losses during the cycle of a real engine that are neglected in ideal air standard analysis. In this paper the performance of an air-standard Otto cycle with heat transfer loss and variable specific heats of working fluid is analyzed by using finite-time thermodynamics. Heat transfer from the unburned mixture to the cylinder walls has a negligible effect on the performance for the compression process. Additionally, the heat transfer rates to the cylinder walls during combustion are the highest and extremely important. Therefore, we assume that the compression and power processes proceed instantaneously so that they are reversible adiabatics, and the heat losses during the heat rejection process can be neglected. The heat loss through the cylinder wall is assumed to occur only during combustion and is further assumed to be proportional to the average temperature of both the working fluid and the cylinder wall. The results show that the effects of heat transfer loss and variable specific heats of working fluid on the cycle performance are obvious, and they should be considered in practice cycle analysis. Higher heat transfer to the combustion chamber walls lowers the peak temperature and pressure and reduces the work per cycle and the efficiency. The effects of other parameters, in conjunction with the heat transfer, including combustion constants and intake air temperature, are also reported. The results are of importance to provide good guidance for the performance evaluation and improvement of practical real engines.
Keywords: Otto cycle, finite-time thermodynamics, heat transfer, air-standard

1.Introduction

A series of achievements have been made since finite-time thermodynamics was used to analyze and optimize real heat-engines [1]. Chen et al. [2] derived the relations between the net power and the efficiency of the Otto-cycle with heat-transfer loss. Chen et al. [3] analyzed and derived the characteristics of power and efficiency for an Otto-cycle with heat-transfer and friction-like term losses. Ge et al. [4, 5] considered the effect of variable specific heats on the cycle process and studied the performance characteristics of endoreversible and irreversible Otto cycles when variable specific heats of working fluid are linear functions of the temperature. The friction loss defined in Ref.[6] , and was studied on the performance of an irreversible Otto-cycle when heat-transfer, friction and internal irreversibility losses are considered. Hou [7] analyzed the effects of heat transfer on the net work output and the indicated thermal efficiency of an air standard Dual cycle. Klein [8] studied the effect of heat transfer on the performance of the Otto-cycle. Ebrahimi [9]studied the relations between the work output and the compression ratio, between the thermal efficiency and the compression ratio for an endoreversible Otto cycle are derived with variable specific heat ratio of working fluid.

2. Cycle model

An air standard Otto-cycle model is shown in Fig. 1. Process 1 → 2S is a reversible adiabatic compression, while process 1 → 2 is an irreversible adiabatic process that takes into account the internal irreversibility in the real compression process. The heat addition is an isochoric process 2 → 3. Process 3 → 4S is a reversible adiabatic expansion, while 3 → 4 is an irreversible adiabatic process that takes into account the internal irreversibility in the real expansion process. The heat rejection is an isochoric process 4 → 1.

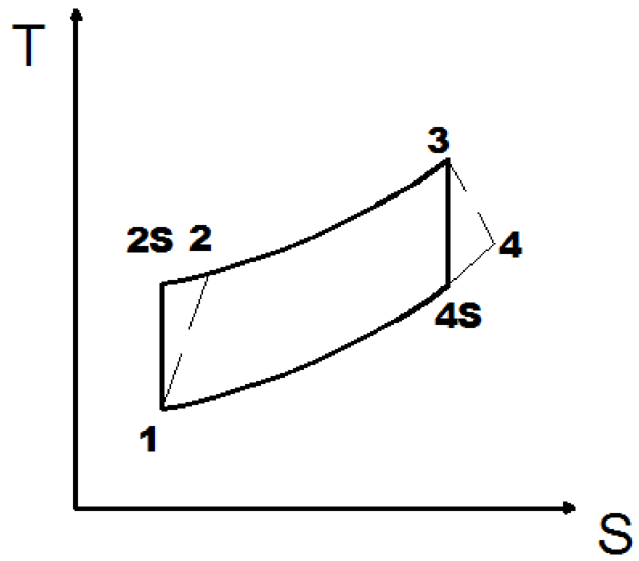

Fig.1. 1 *T-S* diagram for the cycle model

According to Ref. [9] the specific heat ratio of the working fluid is a function of temperature alone and has the linear forms given as follows:

$$\gamma = \gamma_0 - k_1 T \tag{1}$$

Where γ the specific heat ratio, c_p/c_v , and T is the absolute temperature, and γ_0 and k_1 are constants.

According to the relation between specific heat with constant pressure and specific heat with constant volume

$$c_p - c_v = R \tag{2}$$

The heat added to the working fluid during process 2 → 3 is

$$Q_{in} = Q_{23} = NM \int_{T2}^{T3} c_v dT \tag{3}$$

$$Q_{23} = NM \int_{T2}^{T3} \frac{R}{\gamma_0 - k_1 T - 1} dT \tag{4}$$

$$Q_{in} = -NM \frac{R}{k1} ln\left(\frac{\gamma_0 - k_1 T3 - 1}{\gamma_0 - k_1 T2 - 1}\right) = NM \frac{R}{k_1} ln\left(\frac{\gamma_0 - k_1 T2 - 1}{\gamma_0 - k_1 T3 - 1}\right) \tag{5}$$

The heat rejected by the working fluid during the process $4 \rightarrow 1$ is

$$Q_{out} = Q_{41} = NM \int_{T1}^{T4} c_v dT \quad (6)$$

$$Q_{41} = NM \int_{T1}^{T4} \frac{R}{\gamma_0 - k_1 T - 1} dT \quad (7)$$

$$Q_{out=} - NM \frac{R}{k_1} ln\left(\frac{\gamma_0 - k_1 T4 - 1}{\gamma_0 - k_1 T1 - 1}\right) = NM \frac{R}{k_1} ln\left(\frac{\gamma_0 - k_1 T1 - 1}{\gamma_0 - k_1 T4 - 1}\right) \quad (8)$$

Where M is the molar number of the working fluid. For the two adiabatic processes $1 \rightarrow 2$ and $3 \rightarrow 4$, the compression and expansion efficiencies can be defined.

$$\eta c = \frac{T2s - T1}{T2 - T1} \quad (9)$$

$$\eta e = \frac{T4 - T3}{T4s - T3} \quad (10)$$

These two efficiencies can be used to describe the internal irreversibilities of the processes. Because Cp and Cv are dependent on temperature, the adiabatic exponent given before as the specific heat ratio $\gamma = \frac{c_p}{c_v}$ will vary with temperature as well. Therefore, the equation often used for reversible adiabatic processes with constant k cannot be used for reversible adiabatic processes with variable k. The equation for reversible adiabatic process with variable k can be written as follows [2,3]

$$TV^{\gamma - 1} = (T + dT)(V + dV)^{\gamma_0 - 1} \quad (11)$$

Re-arranging Eqs. (1) and (11), we get the following equation

$$T_j(\gamma_0 - k_1 T_i - 1)\left(\frac{V_j}{V_i}\right)^{\gamma_0 - 1} = T_i(\gamma_0 - k_1 T_j - 1) \quad (12)$$

The compression ratio is defined as

$$r_c = \frac{V1}{V2} \quad (13)$$

The equations for processes ($1 \rightarrow 2s$) and ($3 \rightarrow 4s$) are shown

$$T_1(\gamma_0 - k_1 T_{2s} - 1)(r_c)^{\gamma_0 - 1} = T_{2s}(\gamma_0 - k_1 T_1 - 1) \quad (14)$$

$$T_3(\gamma_0 - k_1 T_{4s} - 1) = T_{4s}(\gamma_0 - k_1 T_3 - 1)(r_c)^{\gamma_0 - 1} \quad (15)$$

Using Eqs. (5) and (8), one can derive the expressions of the work output and efficiency as:

$$W_{out} = Q_{in} - Q_{out} \quad (16)$$

$$W_{out} = NM \frac{R}{k_1} ln\left(\frac{\gamma_0 - k_1 T2 - 1}{\gamma_0 - k_1 T3 - 1}\right) - NM \frac{R}{k_1} ln\left(\frac{\gamma_0 - k_1 T1 - 1}{\gamma_0 - k_1 T4 - 1}\right) \quad (17)$$

$$\eta_{th} = \frac{W_{out}}{Q_{in}} \quad (18)$$

$$\eta_{th} = \frac{NM\frac{R}{k_1}ln\left(\frac{\gamma_0-k_1T2-1}{\gamma_0-k_1T3-1}\right)-NM\frac{R}{k_1}ln\left(\frac{\gamma_0-k_1T1-1}{\gamma_0-k_1T4-1}\right)}{NM\frac{R}{k_1}ln\left(\frac{\gamma_0-k_1T2-1}{\gamma_0-k_1T3-1}\right)} \quad (19)$$

$$\eta_{th} = \frac{ln\left[\left(\frac{\gamma_0-k_1T2-1}{\gamma_0-k_1T3-1}\right)\left(\frac{\gamma_0-k_1T4-1}{\gamma_0-k_1T1-1}\right)\right]}{ln\left(\frac{\gamma_0-k_1T2-1}{\gamma_0-k_1T3-1}\right)} \quad (20)$$

For an ideal Otto-cycle model, there are no heat-transfer losses. However, for a real Otto-cycle, heat-transfer irreversibility between the working fluid and the cylinder wall is not negligible. The heat loss through the cylinder wall is assumed to be proportional to the average temperature of both the working fluid and the cylinder wall and the wall temperature is constant. The heat leak is given by the following linear relation [2]

$$Q_{leak} = MB(T_2 + T_3 - 2T_0) \quad (21)$$

where B is a constant related to heat-transfer, T_0 is the average temperature of the cylinder wall.

Friction force is

$$f_\mu = \mu v = \mu\frac{dx}{dt} \quad (22)$$

For a four stroke cycle engine, the total distance the piston travels per cycle is

$$4L = 4(x_1 - x_2) \quad (23)$$

The piston's mean velocity is

$$\bar{v} = 4LN \quad (24)$$

where x_2 is the piston's position corresponding to the minimum volume of the trapped gases, $\Delta t1\to2$ is the time spent in the power stroke.

Then, the lost power is

$$P_\mu = \frac{dW_\mu}{dt} = \mu\frac{dx}{dt}\frac{dx}{dt} = \mu v^2 \quad (25)$$

The power output is

$$P_{otto} = Q_{in} - Q_{out} - P_\mu \quad (26)$$

$$P_{otto} = NM\frac{R}{k_1}ln\left(\frac{\gamma_0-k_1T2-1}{\gamma_0-k_1T3-1}\right) - NM\frac{R}{k_1}ln\left(\frac{\gamma_0-k_1T1-1}{\gamma_0-k_1T4-1}\right) - \mu16(LN)^2 \quad (27)$$

The efficiency of the cycle is

$$\eta_{otto} = \frac{Q_{in}-Q_{out}-P_\mu}{Q_{in}+Q_{leak}} \quad (28)$$

$$\eta_{otto} = \frac{NM\frac{R}{k_1}ln\left(\frac{\gamma_0-k_1T2-1}{\gamma_0-k_1T3-1}\right)-NM\frac{R}{k_1}ln\left(\frac{\gamma_0-k_1T1-1}{\gamma_0-k_1T4-1}\right)-\mu16(LN)^2}{NM\frac{R}{k_1}ln\left(\frac{\gamma_0-k_1T2-1}{\gamma_0-k_1T3-1}\right)+MB(T_2+T_3-2T_0)} \quad (29)$$

When $r_c, T_1, T_3, \eta_c$and are $\eta_e$given, T_{2S} can be obtained from Eq. (14), then, substituting T_{2S} into Eq. (9) yields T_2, T_{4s} can be obtained from Eq. (15) and T_4 can be deduced by substituting T_{4s} into Eq. (10). Substituting $T_{1,}$ T_2, T_3 and T_4 into Eqs. (26) and (29) yields the power and efficiency. Then, the relations between the power output and the compression ratio, between the thermal efficiency and the compression ratio, as well as the optimal relation between power output and the efficiency of the cycle can be obtained.

3. Numerical examples and discussion

The following constants and ranges of parameters are used in the calculations: $T_1 = 350$ K, $T_3 = 2200$ K,$x_1 = 8 \times 10^{-2}$m, $x_2 = 1 \times 10^{-2}$m, $M = 1{,}57 \times 10^{-5}$kmol, $\gamma_0 = 1{,}30 \rightarrow 1{,}4, \eta_c = 0{,}97, \eta_e = 0{,}97, k_1 = 0{,}00003 \rightarrow 0{,}00009$ K^{-1},$r_c = 1 \rightarrow 40, N = 30$
According to Ref.5 $B = 25\, J/molK$

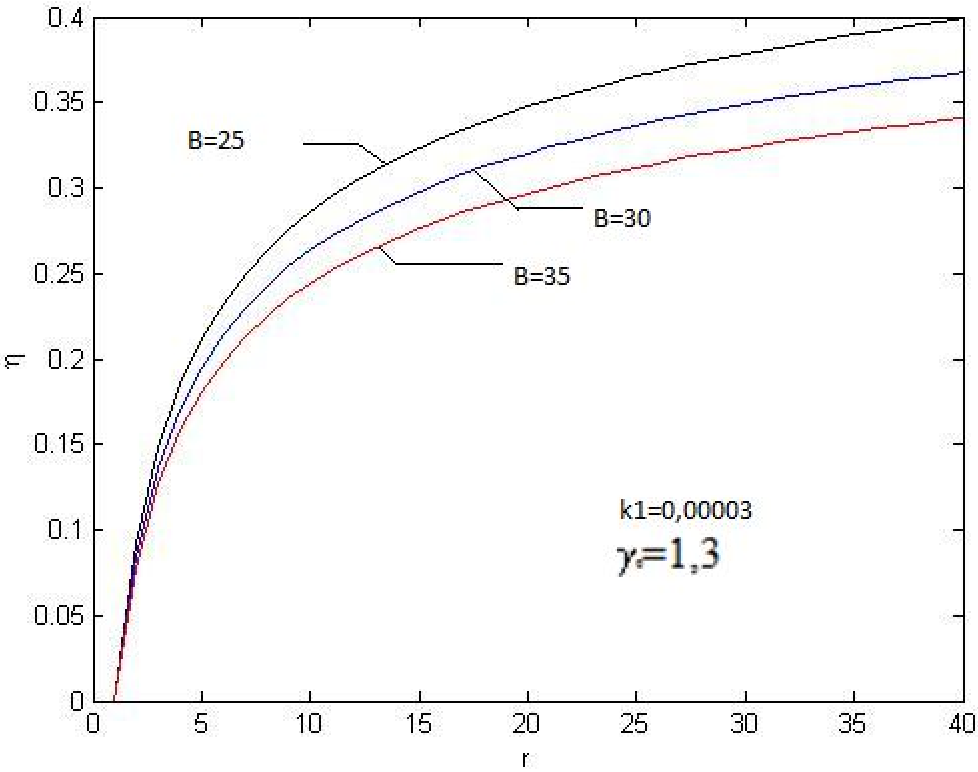

Fig.3. 1The effect of B on cycle efficiency

Figure 3.1 show the effects of B on the performance of the cycle at compression ratio. The thermal efficiency increase with increasing compression ratio and decrease value of a B. With the increasing heat transfer loss , the maximum thermal efficiency decreases while the compression ratio at maximum thermal efficiency point decreases.

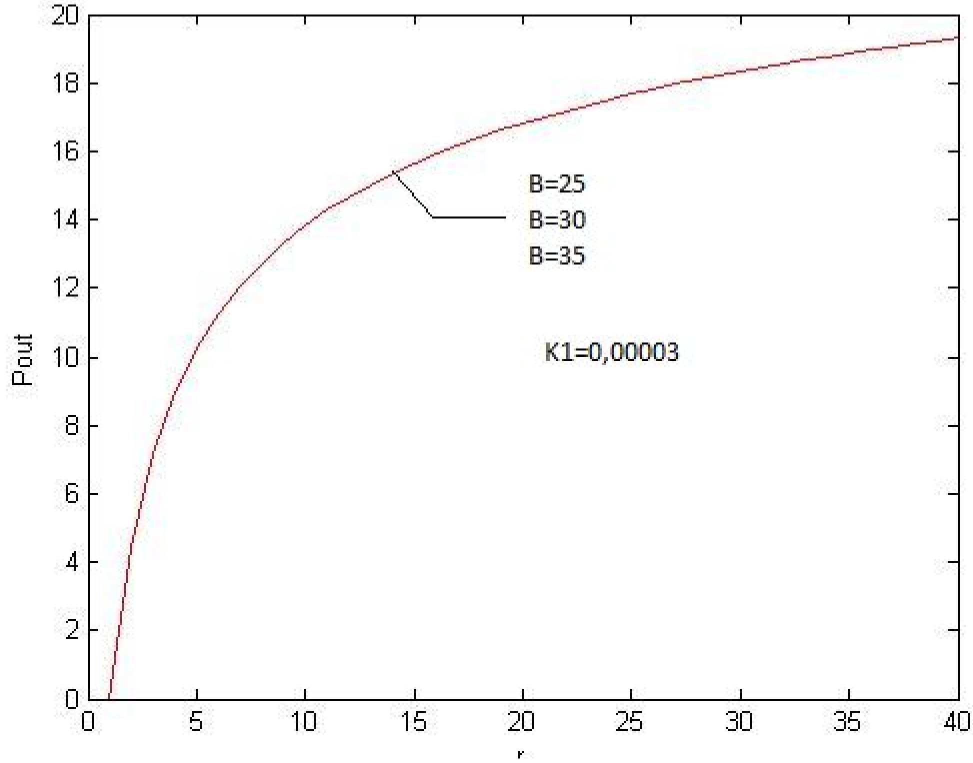

Fig.3. 2The effect of B on cycle power output

Figure 3.2 The power output increases with increasing compression ratio. The power output based on Eqs (26) and (27),in this reason power output is constant with increasing heat loss.

Figure 3.3 show the effects of γ_0 on the performance of the cycle at compression ratio. The thermal efficiency increases with increasing compression ratio and increasing value of a γ_0.

Figure 3.4 show the effects of γ_0 on the performance of the cycle at compression ratio. The power output decrease with increasing compression ratio and increasing value of a γ_0.

Figure 3.5 show the effects of k_1 on the performance of the cycle at compression ratio. The power output increase with increasing compression ratio and increasing value of a k_1.

Figure 3.6 show the effects of k_1 on the performance of the cycle at compression ratio. The thermal efficiency increase with increasing compression ratio and decreses value of a k_1. This is due to the increase of it he heat rejected by the working fluid and the heat added by the working fluid.The magnitude of the thermal efficiency becomes much smaller when the parameter $k1$ increases

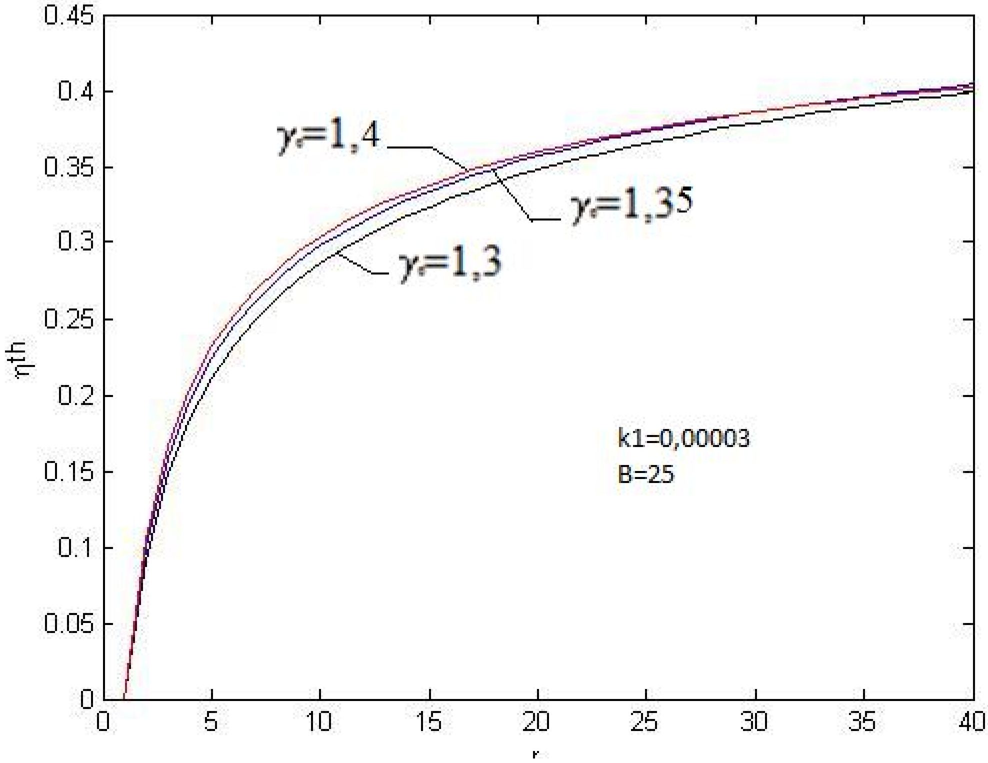

Fig.3. 3 The effect of γ_0 on cycle efficiency

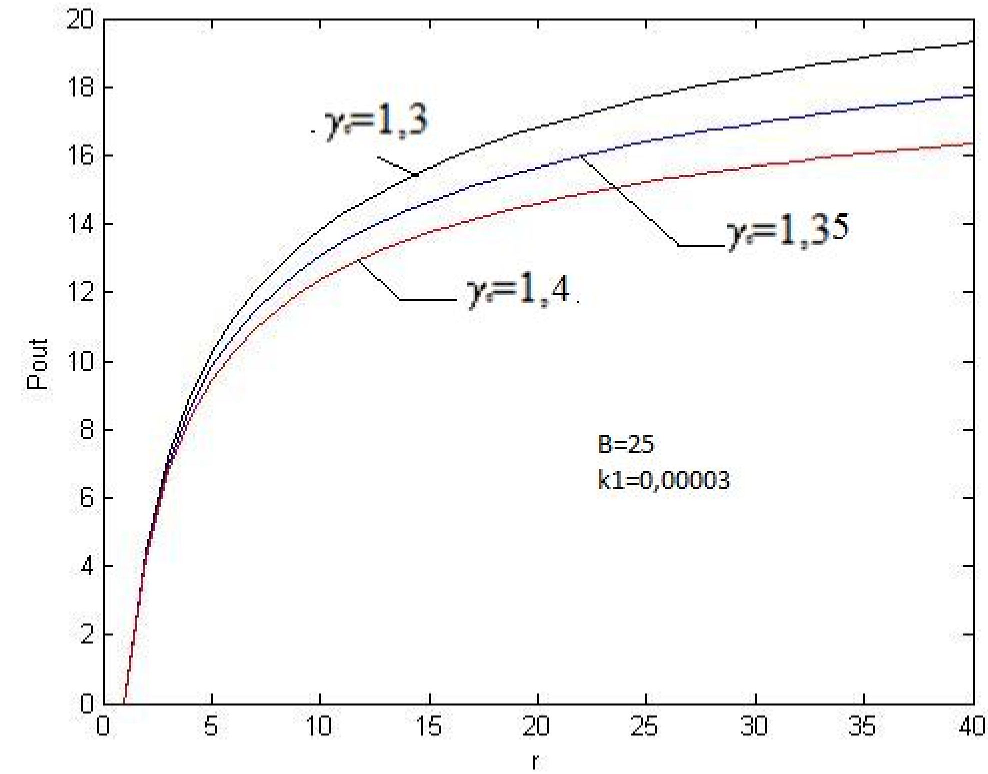

Fig.3. 4 The effect of γ_0 on cycle power output

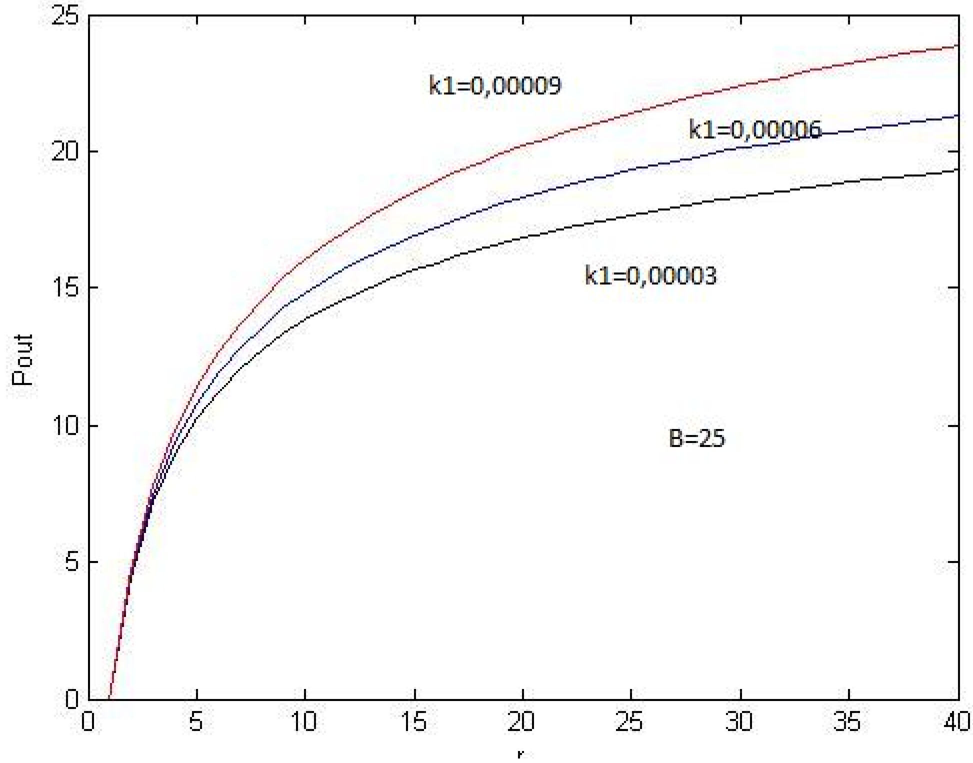

Fig.3.5 The effect of k_1 on cycle power output

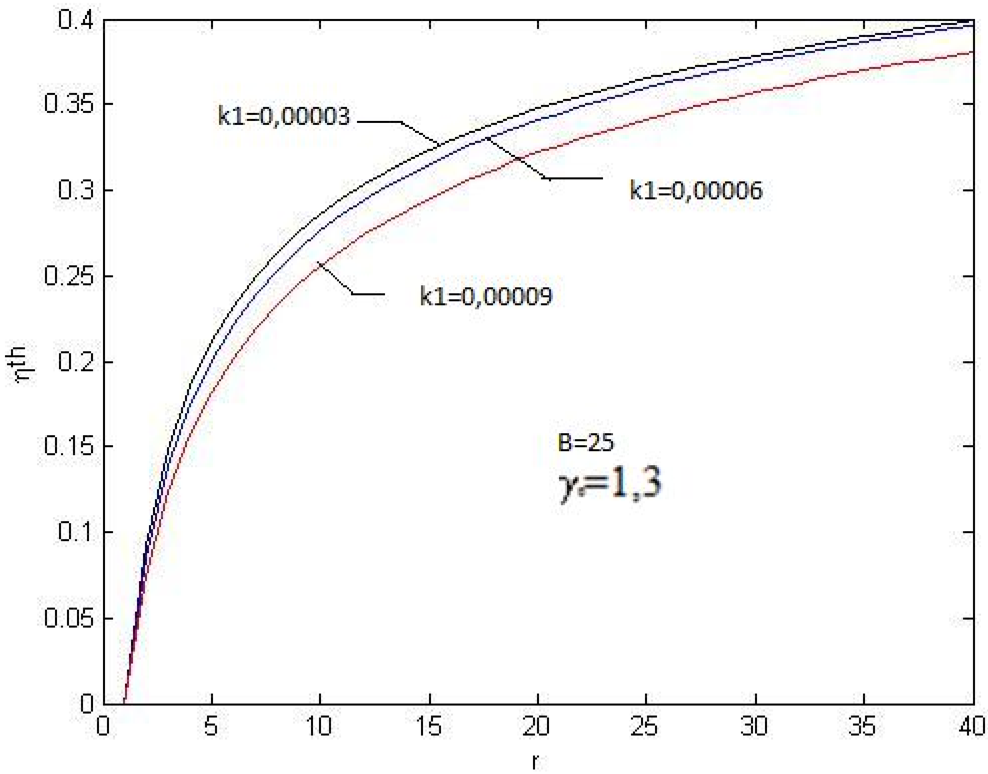

Fig.3.6 The effect of k_1 on cycle efficiency

According to the analysis, it can be concluded that the effects of the temprature dependent specific heat of the working fluid on the cycle performance are significant,and should be considered carefully in pratical cycle analysis design.

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