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Exergoeconomic optimization of an irreversible regenerated air refrigerator with constant-temperature heat reservoirs

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Abstract
Based on the finite time exergoeconomic method, the performance analysis and optimization of an irreversible regenerated air refrigerator cycle are carried out by taking the profit rate as the optimization objective. The profit rate is defined as the difference between the revenue rate of output exergy and the cost rate of input exergy. The analytical expression for profit rate is derived, taking into account several irreversibilities, such as heat resistance, losses due to the pressure drop and the effects of non-isentropic expansion as well as compression. The influences of several parameters such as the temperature ratio of reservoirs, the efficiencies of both compressor and expander, the pressure recovery coefficient and so on are discussed by numerical examples. According to the simulation results, the double-maximum profit rate can be achieved when the pressure ratio and the distributions of heat conductance reach their optimal values respectively. By varying the price ratio, the relationship between the profit rate objective and other objectives can be established and the implementation of profit rate as objective can achieve higher COP compared to the cases using ecological function and cooling load as objectives.

Keywords: Finite-time thermodynamics; Exergoeconomic performance; Irreversible regenerated air.

1. Introduction

The world's energy reserves are decreasing as the intensive consumption and exhaustion of resources, leading to the rising costs of energy. Hence, from the economic perspective, optimizations of the performance of thermodynamic cycles are urgently required. In order to obtain the result closer to the real device, finite-time thermodynamics [1-8], as a powerful tool, is often used to optimize thermodynamic performances. For the Finite-time thermodynamic analyses of refrigeration cycles, the cooling load [9-14], the coefficient of performance (COP) [15-17] and exergy efficiency [18, 19] are often selected as optimization objectives. As conventional refrigerants contain chlorofluorocarbons (CFCs) which are implicated in ozone depletion, the environment friendly air refrigerator is becoming a popular topic of research with its application spreading to aviation industry, food storage and other cooling processes in modern industries [20-22]. Based on the theory of finite-time thermodynamics, the performance of air refrigeration cycle (inverse Brayton cycle) is analyzed and optimized with the consideration of external irreversibility (heat resistance between the reservoir and internal cycle) and internal irreversibilities (losses due to pressure-drop in the piping and nonisentropic expansion as well as compression). For simple Brayton refrigeration cycles, Chen *et al*. [11] derived the analytical relations between cooling load and pressure ratio and between COP and pressure ratio with the consideration of non-isentropic expansion as well as compression and heat resistance losses. Also the results show the cooling load has a parabolic dependence on COP. Luo *et al.* [23] optimized the allocation of heat exchanger inventory for maximizing the cooling load and the COP of the irreversible air refrigeration cycles. Zhou *et al.* [24-27] analyzed and optimized simple endoreversible and irreversible Brayton refrigeration cycles coupled to both constant- and variable-temperature heat reservoirs by taking the cooling load density, i.e., the ratio of cooling load to the maximum specific volume, as the optimization objective. Tu *et al.* [28] optimized simple endoreversible Brayton refrigeration cycles coupled to constant- temperature heat reservoirs by taking cooling load, ecological function and exergy efficiency as optimization objectives. The performance analyses and optimizations considering these objectives were also compared. Ust [29] optimized an simple irreversible air refrigeration cycle based on ecological coefficient of performance (ECOP) criterion which is defined as the ratio of cooling load to the loss rate of availability. Compared with simple Brayton refrigeration cycles, regenerated cycles are more common in industrial applications. Chen *et al*. [30] optimized the performance of an externally and internally irreversible regenerated Brayton refrigerator by taking the cooling load as an objective. Zhou *et al.* [31, 32] carried out the performance analyses and optimizations for regenerated air refrigeration cycles coupled to constant- and variable-temperature heat-reservoirs by taking cooling load density as an optimization objective. Tu *et al*. [33, 34] optimized cooling load, COP and exergy efficiency for real regenerated air refrigerator. Tyagi *et al.* [35] optimized the performance of an irreversible regenerative Brayton refrigerator cycle by taking the cooling load per unit cost as an optimization objective. Ust [36] compared the performance analyses and optimizations by taking ecological coefficient of performance, exergetic efficiency and COP as optimization objectives for an irreversible regenerative air refrigerator cycle.

Nowadays, systems are analyzed and designed based on the consideration of both thermodynamic parameters and cost accounting requirements after the research of Salamon and Nitzan [37, 38] which is to maximize the profit rate of an endoreversible heat engine. In order to distinguish this method from the analysis optimizing pure thermodynamic objectives, Chen *et al.* [39-45] analyzed the profit rate of thermal systems by attributing costs to input and output exergy and termed this method as finite-time exergoeconomic analysis, and its performance bound at maximum profit rate as finite-time exergoeconomic performance bound. Other researches seeking for best economic performance of thermal systems were carried out on endoreversible heat engines, refrigerators and heat pumps by Ibrahim *et al*. [46], De Vos [47, 48] and Bejan [49], with the only irreversibility restricted to the heat transfer between the working fluid and the heat reservoirs. De Vos [47, 48] applied the Newtonian (linear) heat transfer law to derive the relation between the optimal efficiency and economic returns when carrying out thermoeconomics analysis for heat engine. Chen *et al*. [50] investigated the endoreversible thermoeconomic performance of heat engine with the heat transfer between the working fluid and the heat reservoirs obeying linear phenomenological law. Sahin *et al.* [51-53] proposed an optimization criterion considering thermodynamic parameters per unit total cost.

Based on the exergoeconomic analysis of Carnot cycle [39] and the study for a regenerated air refrigeration cycle with cooling load as objective [30], the profit rate optimization for an irreversible regenerated refrigerator is investigated in this paper.

2. Irreversible regenerated air refrigeration cycle

The model of an irreversible regenerated air (Brayton) refrigerator with constant-temperature (hot reservoir temperature T_H and cold reservoir temperature T_L) heat reservoirs to be considered in this paper is shown in Figures 1 and 2. The heat reservoirs are assumed to have infinite thermal capacitance rates and the working fluid is considered as ideal gas with constant thermal capacitance rate C_{wf}. Process 5-2 is a heat addition process with air flowing through the regenerator. In process 2-3, air is compressed non-isentropically considering the irreversibility effect of the compressor. In process 3-6, heat is rejected to the heat sink in hot-side heat exchanger when air flowing through the latter cooler. Process 6-4 is a heat rejection process with air flowing through the regenerator. In process 4-1 air is expanded non-isentropically considering the irreversibility effect of the expander. Process1-5 is a heat addition process with air flowing through the regenerator. Processes 2-3_s and 4-1_s are the isentropic compression and expansion processes in ideal Brayton cycle corresponding to the processes 2-3 and 4-1, respectively. In order to analytically express the compression and expansion irreversibilities, the efficiencies of the compressor and expander are introduced and defined as:

$$\eta_c = (T_{3s} - T_2)/(T_3 - T_2), \eta_t = (T_4 - T_1)/(T_4 - T_{1s}) \tag{1}$$

Counter-flow heat-exchanger model is applied to all the heat-exchangers including the hot- and cold-side heat-exchangers as well as the regenerator. Their heat conductance rates (the product of heat-transfer coefficient k and the heat-exchange surface area A) can be expressed as $U_H = k_H A_H$, $U_L = k_L A_L$ and $U_R = k_R A_R$, respectively. The pressure drop in the piping for low pressure part and for high pressure part can be expressed by the pressure recovery coefficients $D_1 = P_2 / P_1$ and $D_2 = P_4 / P_3$ respectively.

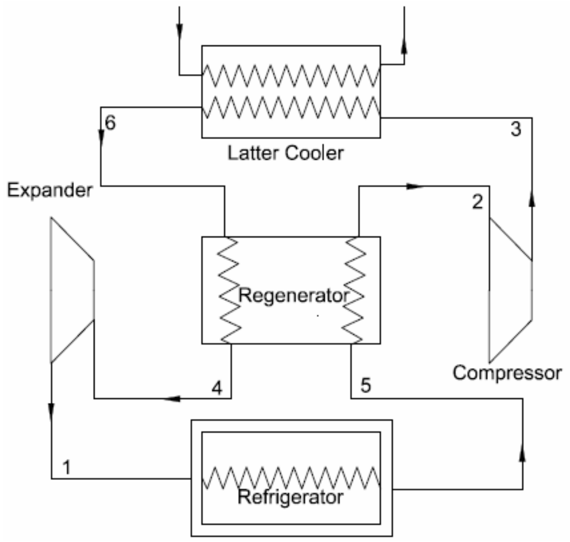

Figure 1. The schematic of a regenerated air refrigerator

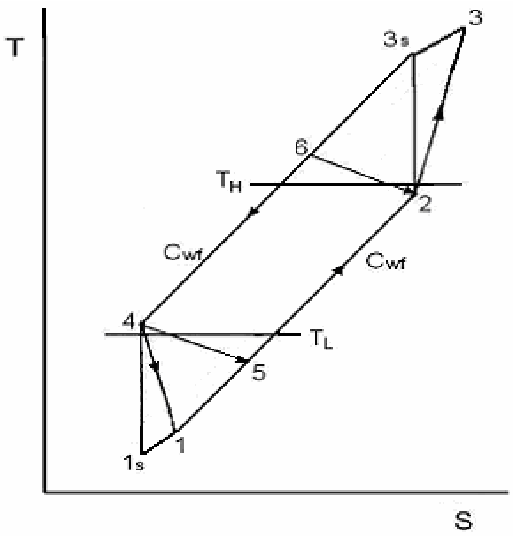

Figure 2. The temperature versus entropy diagram of a regenerated irreversible Brayton refrigeration cycle

3. Analytical expression for the profit rate

According to the properties of heat reservoir and working fluid, the heat transfer law (linear heat transfer law is applied between the heat reservoir and the working fluid) and the theory of heat exchangers, the rate of heat transfer (Q_H) released to the heat sink, the rate of heat transfer (Q_L) supplied by the heat source (the cooling rate R), and the rate of heat transfer happened in the regenerator (Q_R) can be expressed as, respectively,

$$\begin{aligned} Q_H &= U_H[(T_3 - T_H) - (T_6 - T_H)]/\ln[(T_3 - T_H)/(T_6 - T_H)] \\ &= C_{wf}(T_3 - T_6) = C_{wf} E_H (T_3 - T_H) \end{aligned} \tag{2}$$

$$\begin{aligned} R = Q_L &= U_L[(T_L - T_5) - (T_L - T_1)]/\ln[(T_L - T_5)/(T_L - T_1)] \\ &= C_{wf}(T_5 - T_1) = C_{wf} E_L (T_L - T_1) \end{aligned} \tag{3}$$

$$Q_R = C_{wf}(T_6 - T_4) = C_{wf}(T_2 - T_5) = C_{wf} E_R (T_6 - T_5) \tag{4}$$

where E_H, E_L and E_R are the effectivenesses of the hot- as well as cold-side heat exchangers and the regenerator, respectively, and are defined as:

$$E_H = 1 - \exp(-N_H)\,, E_L = 1 - \exp(-N_L)\,, E_R = N_R/(1+N_R) \tag{5}$$

where N_H , N_L and N_R are the heat transfer units for the hot- as well as cold-side heat exchangers and the regenerator, respectively, and are defined as:

$$N_H = U_H / C_{wf}\,, N_L = U_L / C_{wf}\,, N_R = U_R / C_{wf} \tag{6}$$

The temperature ratio of the compressor operating isentropically is,

$$x = T_{3s}/T_2 = (P_3/P_2)^m = \pi^m\,, x \ge 1 \tag{7}$$

where $m = (k-1)/k$ with k being the adiabatic index, π is the pressure ratio of the compressor and P is the pressure. By defining the total pressure recovery coefficient as $D = D_1 D_2$, the temperature ratio of the expandor operating isentropically is given as,

$$T_4/T_{1s} = (P_4/P_1)^m = D^m x \tag{8}$$

The rate of exergy input to the system is equal to the net power input, and the first law of thermodynamics gives,

$$E_{in} = Q_H - Q_L \tag{9}$$

As the refrigerator is utilized to absorb heat from the cold space, the rate of output exergy is given as

$$E_{out} = Q_L(T_0/T_L - 1) - Q_H(T_0/T_H - 1) \tag{10}$$

The profit rate, defined as the difference between the revenue rate of output exergy and the cost rate of input exergy, is selected as the objective function of this refrigeration system, and is expressed as

$$M = \psi_1 E_{out} - \psi_2 E_{in} \tag{11}$$

where ψ_1 and ψ_2 are the prices of exergy output rate and power input, respectively.
Combining Eqs. (1)-(11) gives,

$$M = \frac{\begin{array}{l} C_{wf}E_L\left[\psi_1(T_0/T_L-1)+\psi_2\right]\{[\eta_c-(x+\eta_c-1)(1-E_H)E_R]T_L-(D^{-m}x^{-1}\eta_t-\eta_t \\ +1)\{[E_R\eta_c+(x+\eta_c-1)(1-E_H)(1-2E_R)]T_L+(1-E_R)E_H T_H\eta_c\}\} \\ -C_{wf}E_H\left[\psi_1(T_0/T_H-1)+\psi_2\right]\{(x+\eta_c-1)(1-E_R)E_L T_L+[(x+\eta_c-1)E_R-\eta_c]T_H \\ +(D^{-m}x^{-1}\eta_t-\eta_t+1)(1-E_L)[(x+\eta_c-1)(1-2E_R)+E_R\eta_c]T_H\} \end{array}}{\begin{array}{l} \eta_c-(x+\eta_c-1)(1-E_H)E_R-(D^{-m}x^{-1}\eta_t-\eta_t+1)(1 \\ -E_L)[E_R\eta_c+(x+\eta_c-1)(1-E_H)(1-2E_R)] \end{array}} \tag{12}$$

By defining the heat reservoir temperature ratio as $\tau_1 = T_H/T_L$ and the ratio of hot-side heat reservoir temperature to the ambient temperature as $\tau_2 = T_H/T_0$,yields the expression of the dimensionless profit rate as,

$$\overline{M} = M / (C_{wf} T_L \psi_1)$$

$$= \frac{\begin{array}{l} E_L \left[\tau_1/\tau_2 - (1-\psi_2/\psi_1)\right]\{\eta_c - (x+\eta_c-1)(1-E_H)E_R - (D^{-m}x^{-1}\eta_t - \eta_t + 1)\{[E_R\eta_c \\ +(x+\eta_c-1)(1-E_H)(1-2E_R)] + (1-E_R)E_H\tau_1\eta_c\}\} \\ -E_H\left[1/\tau_2 - (1-\psi_2/\psi_1)\right]\{(x+\eta_c-1)(1-E_R)E_L + [(x+\eta_c-1)E_R - \eta_c]\tau_1 \\ +(D^{-m}x^{-1}\eta_t - \eta_t+1)(1-E_L)[(x+\eta_c-1)(1-2E_R) + E_R\eta_c]\tau_1\} \end{array}}{\begin{array}{l} \eta_c - (x+\eta_c-1)(1-E_H)E_R - (D^{-m}x^{-1}\eta_t - \eta_t + 1)(1 \\ -E_L)[E_R\eta_c + (x+\eta_c-1)(1-E_H)(1-2E_R)] \end{array}} \quad (13)$$

4. Results and discussion

Eq. (13) indicates that the dimensionless profit rate for an irreversible regenerated air refrigerator is influenced by the pressure ratio (π), the heat transfer irreversibilities (E_H, E_L and E_R), the price ratio (ψ_1/ψ_2), the internal irreversibilities (η_c, η_R and D), the heat reservoir temperature ratio (τ_1) and the ratio of hot-side heat reservoir temperature to the ambient temperature(τ_2). As the air refrigeration cycle model is a steady flow model, the total heat transfer surface area of the heat exchangers is a constant, and the distribution of heat exchange surface area can be optimized. Also pressure ratio as a fundamental design parameter can be optimized. Hence, numerical simulation is carried out using Matlab to optimize the pressure ratio and the distribution of heat conductances of the heat exchangers. The effects of other parameters on the relationship between profit rate and the pressure ratio will be investigated. The relationship between the profit rate and other objectives will be discussed by varying the price ratio.

4.1 Optimal pressure ratio

The effects of the heat transfer irreversibilities (E_H, E_L and E_R), the internal irreversibilities (η_c, η_t and D), and two temperature ratios (τ_1 and τ_2) on the characteristic of profit rate versus the pressure ratio are shown in Figures 3-8 with the price ratio $\psi_1/\psi_2 = 20$. The shape of these curves is parabolic-like with one maximum profit rate ($\overline{M}_{\max,\pi}$) at the optimum pressure ratio (π_{opt}). The profit rate increases with the increasings of the effectivenesses of the hot-and cold-side heat exchangers (Figure 3), the efficiencies of the compressor and expander (Figure 5), the pressure recovery coefficient (Figure 6), the heat reservoir temperature ratio (Figure 7) and the ratio of hot-side heat reservoir temperature to the ambient temperature (Figure 8). In Figure 4, with the increasing of the effectiveness of the regenerator, the profit rate increases when the pressure ratio is relatively small, and decreases when the pressure ratio is large. Also, the optimal pressure ratio (π_{opt}) increases with the increasings of the effectiveness of the hot- and cold- side heat exchangers, the efficiencies of the compressor and the expander and two temperature ratios, while decreases with the increasings of the effectiveness of the regenerator and the pressure recovery coefficient.

4.2 Optimal distribution of heat conductance

For the fixed heat-exchanger inventory ($U_T = U_H + U_L + U_R$), the distribution of heat conductance will influence the performance of the air refrigerator with the profit rate being the objective. By defining the hot-side distribution of heat conductance (u_H) and cold-side distribution of heat conductance (u_L) as

$$u_H = U_H / U_T, u_L = U_L / U_T \quad (14)$$

The heat conductances of the hot- and cold-side heat exchanger as well as the regenerator can be expressed as,

$$U_H = u_H U_T, U_L = u_L U_T, U_R = (1 - u_H - u_L) U_T \quad (15)$$

When $k = 1.4$, $C_{wf} = 0.8 kW/K$, $U_T = 5 kW/K$, $\psi_1 / \psi_2 = 20$, $\eta_c = \eta_t = 0.95$, $\tau_1 = 1.25$, $\tau_2 = 1$, $D = 0.96$, $\pi = 5$ are set, a 3-D plot of the dimensionless profit rate versus the hot-side distribution of heat conductance (u_H) and cold-side distribution of heat conductance (u_L) is shown in Figure 9. Eq.(14) indicates that both u_H and u_L should be smaller than or equal to 1 and so as their summation. Hence, an angle-bisecting plane vertical to the u_H and u_L plane is created in Figure 9 as a limit for u_H and u_L, with points in this plane indicating the limiting case of air refrigerator without regenerator. It can be observed from Figure 9 that the surface is similar to a paraboloid, with one maximum point ($\overline{M}_{\max,u}$) at optimal distributions of heat conductance (u_{Hopt} and u_{Lopt}). As mentioned previously, there is a maximum profit rate for the curve of the dimensionless profit rate versus pressure ratio. Hence, there is a double maximum value for the dimensionless profit rate with the distributions of heat conductance (u_H and u_L) and pressure ratio (π) as variables. The double maximum dimensionless profit rate can be searched out using the Matlab optimization toolbox. When $k = 1.4$, $C_{wf} = 0.8 kW/K$, $U_T = 5 kW/K$, $\psi_1 / \psi_2 = 20$, $\eta_c = \eta_t = 0.95$, $\tau_1 = 1.25$, $\tau_2 = 1$ and $D = 0.96$ are set, the double maximum dimensionless profit rate ($\overline{M}_{\max,\max}$) is 0.0217 with the optimal distributions of heat conductance (u_{Hopt} and u_{Lopt}) equaling to 0.4737 and 0.3339, respectively, and the optimal pressure ratio (π_{opt}) being 7.8316. In Figure 10, the influences of the total heat exchanger inventory (U_T) on the double maximum dimensionless profit rate ($\overline{M}_{\max,\max}$) and its corresponding optimal distributions of heat conductance (u_{Hopt} and u_{Lopt}) can be observed. When increasing the total heat exchanger inventory (U_T), the double maximum dimensionless profit rate ($\overline{M}_{\max,\max}$) increases, the hot-side distribution of heat conductance (u_{Hopt}) decreases and the cold-side distribution of heat conductance (u_{Lopt}) first increases and then decreases.

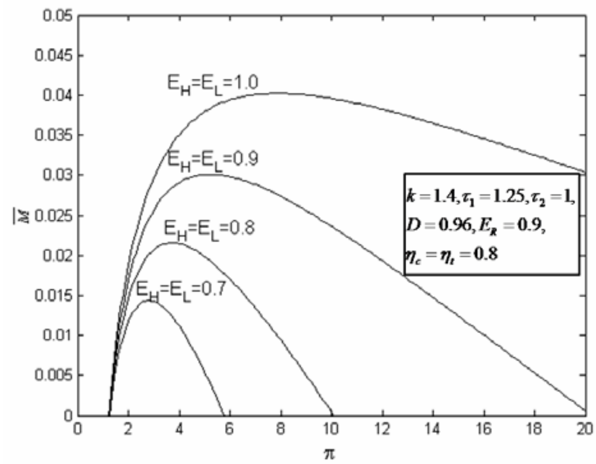

Figure 3. The effects of the effectivenesses of the hot- and cold- side heat exchangers on the profit rate versus the pressure ratio characteristic

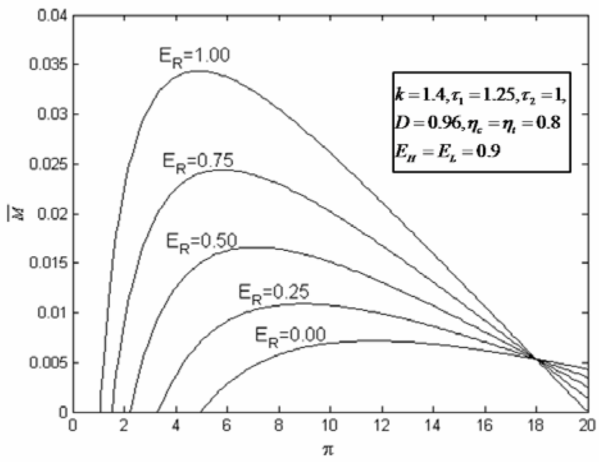

Figure 4. The effect of the effectiveness of the regenerator on the profit rate versus the pressure ratio characteristic

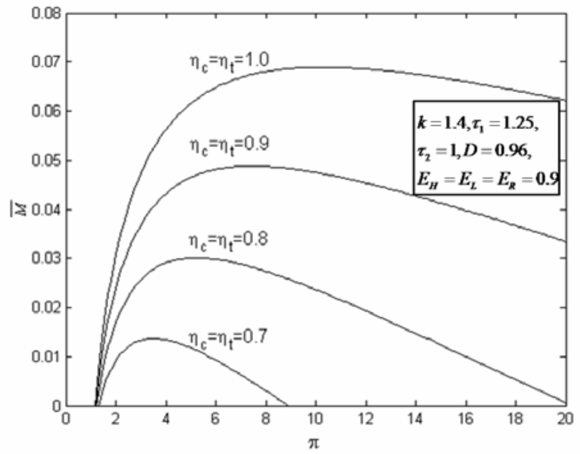

Figure 5. The effects of the efficiencies of the compressor and expander on the profit rate versus the pressure ratio characteristic

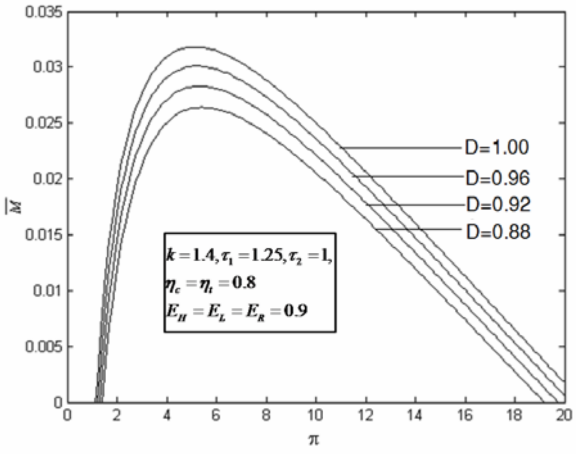

Figure 6. The effect of the pressure recovery coefficient on the profit rate versus the pressure ratio characteristic

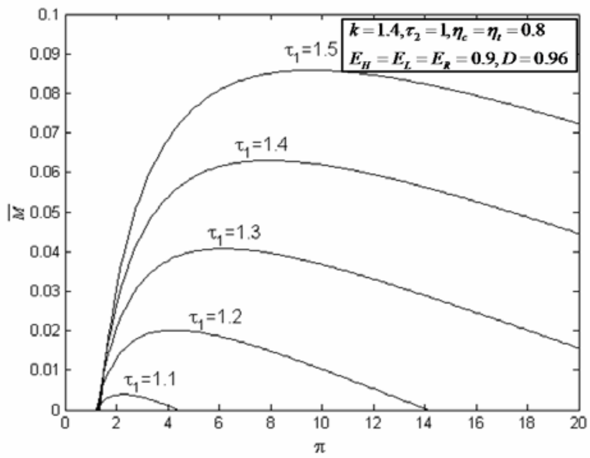

Figure 7. The effect of the heat reservoir temperature ratio on the profit rate versus the pressure ratio characteristic

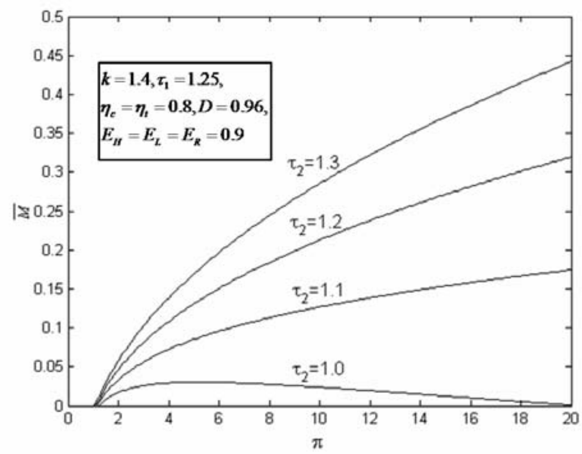

Figure 8. The effect of the ratio of hot-side heat reservoir temperature to the ambient temperature on the profit rate versus the pressure ratio characteristic

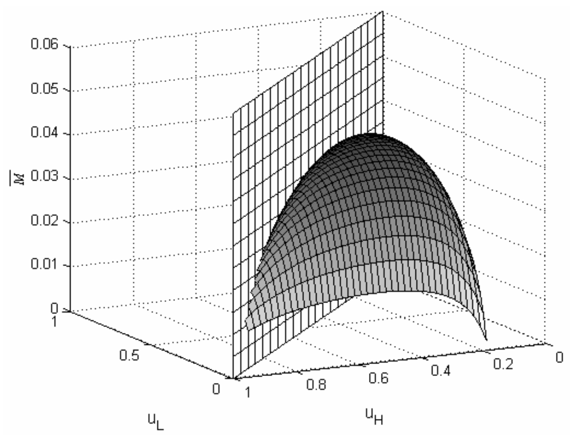

Figure 9. The profit rate versus the hot-side heat conductance distribution and the cold-side heat conductance distribution

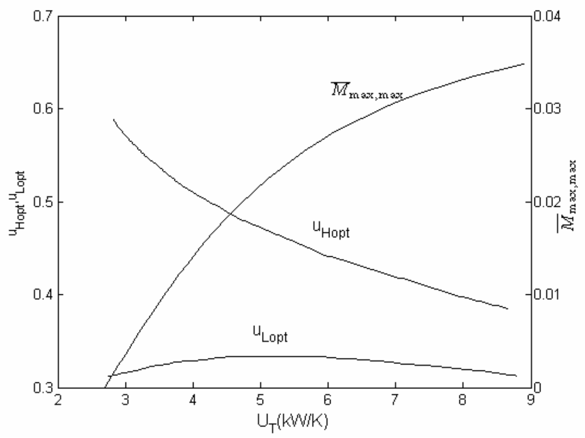

Figure 10. The double maximum profit rate and the corresponding optimum heat conductance distributions versus the total heat exchanger inventory

4.3 Influence of the price ratio

As the price of exergy output rate becomes very large compared with that of power input, i.e., $\psi_1/\psi_2 \to \infty$, the function of the dimensionless profit rate becomes

$$\overline{M} = M/(C_{wf}T_L\psi_1)$$
$$= \frac{\begin{array}{l} E_L\left(\tau_1/\tau_2-1\right)\{\eta_c-(x+\eta_c-1)(1-E_H)E_R-(D^{-m}x^{-1}\eta_t-\eta_t+1)\{[E_R\eta_c \\ +(x+\eta_c-1)(1-E_H)(1-2E_R)]+(1-E_R)E_H\tau_1\eta_c\}\} \\ -E_H(1/\tau_2-1)\{(x+\eta_c-1)(1-E_R)E_L+[(x+\eta_c-1)E_R-\eta_c]\tau_1 \\ +(D^{-m}x^{-1}\eta_t-\eta_t+1)(1-E_L)[(x+\eta_c-1)(1-2E_R)+E_R\eta_c]\tau_1\} \end{array}}{\begin{array}{l} \eta_c-(x+\eta_c-1)(1-E_H)E_R-(D^{-m}x^{-1}\eta_t-\eta_t+1)(1 \\ -E_L)[E_R\eta_c+(x+\eta_c-1)(1-E_H)(1-2E_R)] \end{array}} \tag{16}$$

If the hot reservoir temperature equals the ambient temperature ($\tau_2 \to 1$), the function of the dimensionless profit rate becomes

$$\overline{M} = M/(C_{wf}T_L\psi_1)$$
$$= \frac{\begin{array}{l} E_L\left(\tau_1/\tau_2-1\right)\{\eta_c-(x+\eta_c-1)(1-E_H)E_R-(D^{-m}x^{-1}\eta_t-\eta_t+1)\{[E_R\eta_c \\ +(x+\eta_c-1)(1-E_H)(1-2E_R)]+(1-E_R)E_H\tau_1\eta_c\}\} \end{array}}{\begin{array}{l} \eta_c-(x+\eta_c-1)(1-E_H)E_R-(D^{-m}x^{-1}\eta_t-\eta_t+1)(1 \\ -E_L)[E_R\eta_c+(x+\eta_c-1)(1-E_H)(1-2E_R)] \end{array}} = \frac{(T_0/T_L-1)R}{C_{wf}T_L} \tag{17}$$

The optimization of the profit rate also leads to the maximization of the cooling load (R).

On the other hand, with the price of power input approaching the price of the exergy output rate, i.e. $\psi_2/\psi_1 \to 1$, the function of the profit rate becomes

$$\overline{M} = M/(C_{wf}T_L\psi_1)$$
$$= \frac{\begin{array}{l} E_L\tau_1/\tau_2\{\eta_c-(x+\eta_c-1)(1-E_H)E_R-(D^{-m}x^{-1}\eta_t-\eta_t+1)\{[E_R\eta_c \\ +(x+\eta_c-1)(1-E_H)(1-2E_R)]+(1-E_R)E_H\tau_1\eta_c\}\} \\ -E_H/\tau_2\{(x+\eta_c-1)(1-E_R)E_L+[(x+\eta_c-1)E_R-\eta_c]\tau_1 \\ +(D^{-m}x^{-1}\eta_t-\eta_t+1)(1-E_L)[(x+\eta_c-1)(1-2E_R)+E_R\eta_c]\tau_1\} \end{array}}{\begin{array}{l} \eta_c-(x+\eta_c-1)(1-E_H)E_R-(D^{-m}x^{-1}\eta_t-\eta_t+1)(1 \\ -E_L)[E_R\eta_c+(x+\eta_c-1)(1-E_H)(1-2E_R)] \end{array}} = -\frac{T_0\sigma}{C_{wf}T_L} \tag{18}$$

where σ is the rate of entropy production of the regenerative air refrigeration cycle. When maximizing the profit rate under this condition, minimization of the entropy generation (σ) can be achieved.

For the case when $\psi_1/\psi_2 \to 2$ is satisfied, the function of the profit rate becomes

$$\overline{M} = M/(C_{wf}T_L\psi_1)$$
$$= \frac{\begin{array}{l} E_L\left[\tau_1/\tau_2-1/2\right]\{\eta_c-(x+\eta_c-1)(1-E_H)E_R-(D^{-m}x^{-1}\eta_t-\eta_t+1)\{[E_R\eta_c \\ +(x+\eta_c-1)(1-E_H)(1-2E_R)]+(1-E_R)E_H\tau_1\eta_c\}\} \\ -E_H\left[1/\tau_2-1/2\right]\{(x+\eta_c-1)(1-E_R)E_L+[(x+\eta_c-1)E_R-\eta_c]\tau_1 \\ +(D^{-m}x^{-1}\eta_t-\eta_t+1)(1-E_L)[(x+\eta_c-1)(1-2E_R)+E_R\eta_c]\tau_1\} \end{array}}{\begin{array}{l} \eta_c-(x+\eta_c-1)(1-E_H)E_R-(D^{-m}x^{-1}\eta_t-\eta_t+1)(1 \\ -E_L)[E_R\eta_c+(x+\eta_c-1)(1-E_H)(1-2E_R)] \end{array}} = \frac{E}{2C_{wf}T_L} \tag{19}$$

where E is the ecological objective of the regenerative air refrigeration cycle. The optimization of the profit rate also leads to the maximization of the ecological function (E) objective.
Figure 11 shows the effect of the price ratio on the profit rate versus COP characteristic, with the COP function derived by Chen *et al*. [33]. From Figure 11, it can be observed that the optimal COP is larger when $\psi_1/\psi_2 = 20$ is satisfied, compared to the other two cases which corresponding to the cases of ecological objective ($\psi_1/\psi_2 \to 2$) and cooling load objective ($\psi_1/\psi_2 \to \infty$).

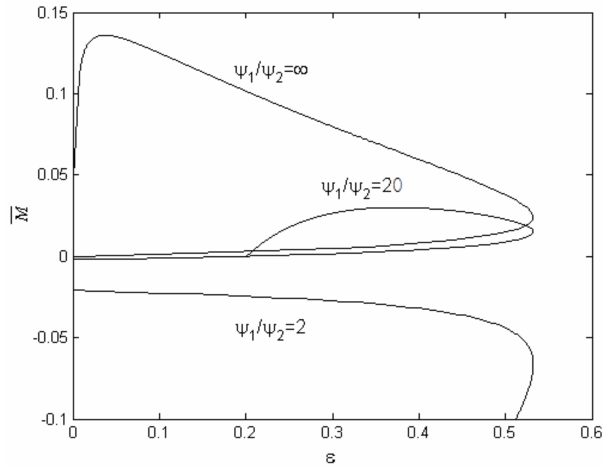

Figure 11. The profit rate versus COP with different price ratios

5. Conclusion

The analytical expression of the profit rate for the irreversible regenerated air refrigeration cycle coupled to constant heat reservoirs is derived based on the theoretical model. Numerical calculations are carried out to optimize the pressure ratio and the distributions of heat conductance between heat exchangers and regenerator with considering the influences of the internal irreversibilities (η_c, η_t and D), the heat reservoir temperature ratio (τ_1) and the ratio of hot-side heat reservoir temperature to the ambient temperature(τ_2). The relationship between the profit rate objective and other objectives is investigated. The comparison of these objectives has also been discussed.
There exists an optimal pressure ratio and a pair of optimal distributions of heat conductance corresponding to the double-maximum profit rate. Optimizing the air refrigeration cycle based on the profit rate objective can achieve higher COP compared to the cases using ecological function and cooling load as objectives.

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