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Performance analysis and optimization for an endoreversible Carnot heat pump cycle with finite speed of the piston

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Abstract
Performance of an endoreversible Carnot heat pump cycle with finite speed of the piston is investigated by using finite time thermodynamics. The analytical formulae between the optimal heating load and the coefficient of performance (COP), as well as between the optimal heating load and speed ratio of the piston are derived. It is found that the heating load versus COP characteristics are parabolic-like, and there exist a maximum heating load and the corresponding COP. These are different from the monotonically decreasing characteristic of the endoreversible Carnot heat pump without consideration of the finite speed of the piston. At the same time, the effects of reservoir temperature ratio on the optimal relations are analyzed by numerical examples. In the analysis and optimization, two cases with and without limit of cycle period are included.

Keywords: Finite time thermodynamics; Reciprocating endoreversible Carnot heat pump cycle; Heating load; COP; Finite speed of the piston.

1. Introduction

Since the 1970s, finite time thermodynamics was applied to study performance optimization problems of thermodynamic cycles by many researchers and many practical significant results were obtained [1-5]. In the research field of heat pump cycles, Blanchard [6] was the first to derive the COP bounds for the fixed heating load for an endoreversible Carnot heat pump with Newtown's heat transfer law. Goth and Feidt [7] also obtained the similar results. Chen *et al.* [8, 9] established the performance holographic spectrum and derived the optimization criteria of the endoreversible Carnot heat pump [8], and analyzed the effect of heat transfer law on the performance of the endoreversible Carnot heat pump. Chen *et al.* [10] investigated the specific heating load optimization and the COP optimization of the endoreversible Carnot heat pump, derived the bounds of specific heating load and COP as well as the optimal relation between the optimal specific heating load and COP. Chen *et al.* [11] studied the effects of heat resistance and internal irreversibility on the characteristics of air heat pump cycles with constant and variable temperature thermal reservoirs. Bi *et al.* [12, 13] derived the analytical formulae between the dimensionless heating load and pressure ratio, between the COP and pressure ratio, as well as between the dimensionless heating load density and pressure ratio of the endoreversible air heat pump with constant and variable temperature thermal reservoirs, and optimized the heating load and the heating load density by searching the optimal distribution of heat conductance of the cycle. Recently, Agrawal and Menon [14] and Agrawal [15] established a new cycle model of finite speed of the piston based on the assumption that the finite speeds of the piston on the four thermodynamic branches are equal for studying performance of classical reversible Carnot engine [14] and endoreversible Carnot engine [15]. On the basis mentioned above, this paper will establish a cycle model of endoreversible Carnot heat pump considering the characteristics of finite time and finite speed of the piston with the Newtown's heat transfer law, and derive the relations among the heating load, the COP, and the speed ratio of the piston of the cycle. In the analysis and optimization, two cases with and without limit of cycle period are considered based on the assumption that the finite speeds of the piston on the four thermodynamic processes are unequal.

2. Cycle description

The *P-V* diagram and *T-s* diagram of the endoreversible Carnot heat pump cycle are shown in Figure 1 and Figure 2. The process (1-2) is isothermal absobing heat branch of the working fluid from heat source T_L, the process (3-4) is isothermal releasing heat branch of the working fluid to heat sink T_H, the processes (2-3, 4-1) are reversible adiabatic compression branch and reversible adiabatic expansion branch of the working fluid. The following assumption are made. The mass of the working fluid per cycle is m. The ratio of the specific heats of the working fluid is γ. The absorbing and realeasing temperatures of working fluid are T_{LC} and T_{HC}. the volumes of the gas at the four states are $V_i (i = 1, 2, 3, 4)$. The gas constant is R_g. The speeds of the piston on the four thermodynamic processes are $v_i (i = 1, 2, 3, 4)$. They are defined as the passed volume of the piston per second. The temperature ratio of the heat reservoirs is $\tau = T_H / T_L$. The cycle period is t_s. The times spent on the four branches are $t_i (i = 1, 2, 3, 4)$.

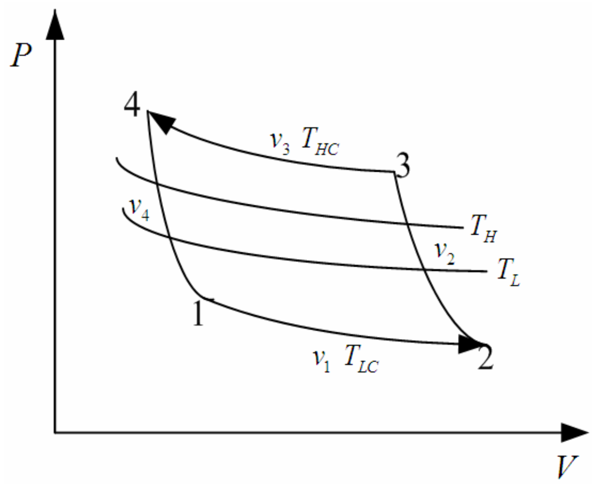

Figure 1. *P-V* diagram for cycle model

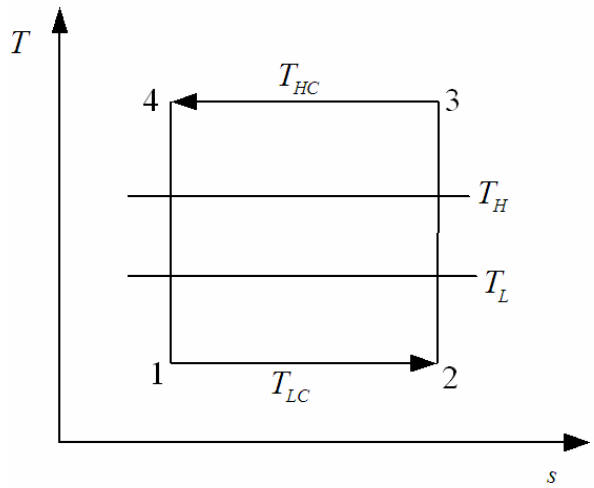

Figure 2. *T-s* diagram for cycle model

3. Performance analysis of the heat pump cycle

Consider that the heat transfer between the heat reservoirs and working fluid obeys Newtown's law. According to the properties of heat transfer, the quantity of heat transfer (Q_{LC}) supplied by the heat source and the quantity of heat transfer (Q_{HC}) released to the heat sink are, respectively, given by

$$Q_{LC} = K_1 F_1 (T_L - T_{LC}) t_1 \tag{1}$$

$$Q_{HC} = K_2 F_2 (T_{HC} - T_H) t_3 \tag{2}$$

where K_1 and K_2 are the heat transfer coefficients between the working fluid and heat reservoirs, F_1 and F_2 are the heat transfer surface areas of the heat exchangers.

For an endoreversible cycle, the entropy change of the working fluid is zero, i.e., $\oint ds = 0$, therefore, one has

$$Q_{LC} / Q_{HC} = T_{LC} / T_{HC} = \mathrm{a} \tag{3}$$

According the state equations of idea gas and the equations of adiabatic process, one has

$$V_3/V_2 = V_4/V_1 = (T_{LC}/T_{HC})^{1/(\gamma-1)} = \mathrm{a}^{1/(\gamma-1)} \tag{4}$$

$$V_2/V_1 = V_3/V_4 = V^* \tag{5}$$

Assuming that the relations of speeds of the piston on the four branches are as follows:

$$\mathrm{x} = v_3/v_1 \text{ , } v_2 = v_4 \text{ , } \mathrm{y} = v_2/v_1 \tag{6}$$

From equations (4)-(6), the times of the four branches can be obtained as follows:

$$t_1 = \frac{V_2 - V_1}{v_1} = \frac{V_1}{v_1}(\frac{V_2}{V_1} - 1) = \frac{V_2}{v_1 V^*}(V^* - 1) \tag{7}$$

$$t_2 = \frac{V_2 - V_3}{v_2} = \frac{V_1}{v_2}(\frac{V_2}{V_1} - \frac{V_3}{V_1}) = \frac{V_2}{\mathrm{y} v_1}(1 - \mathrm{a}^{1/(\gamma-1)}) \tag{8}$$

$$t_3 = \frac{V_3 - V_4}{v_3} = \frac{V_1}{v_3}(\frac{V_3}{V_1} - \frac{V_4}{V_1}) = \frac{V_2}{\mathrm{x} v_1 V^*}(V^* - 1)\mathrm{a}^{1/(\gamma-1)} \tag{9}$$

$$t_4 = \frac{V_1 - V_4}{v_4} = \frac{V_1}{v_4}(1 - \frac{V_4}{V_1}) = \frac{V_2}{\mathrm{y} v_1 V^*}(1 - \mathrm{a}^{1/(\gamma-1)}) \tag{10}$$

According to the properties of working fluid, the quantities of heat transfer (Q_{LC} , Q_{HC})can also be given by

$$Q_{LC} = \int_1^2 PdV = mR_g T_{LC} \ln V^* \tag{11}$$

$$Q_{HC} = -\int_3^4 PdV = mR_g T_{HC} \ln V^* \tag{12}$$

Combining equations (1), (2), (3), (7) with (9) gives the releasing heat temperature of working fluid

$$T_{HC} = \frac{T_L[K_1F_1\mathrm{x} + K_2F_2\mathrm{a}^{\gamma/(\gamma-1)}\tau]}{K_1F_1\mathrm{xa} + K_2F_2\mathrm{a}^{\gamma/(\gamma-1)}} \tag{13}$$

Combining equations (1), (7), (11) with (13) gives

$$\frac{V^* \ln V^*}{V^* - 1} = \frac{V_2 K_1 K_2 F_1 F_2 \mathrm{a}^{1/(\gamma-1)}(1 - \mathrm{a}\tau)}{v_1(K_1F_1\mathrm{x} + K_2F_2\mathrm{a}^{\gamma/(\gamma-1)}\tau)mR_g} \tag{14}$$

Combining equations (2), (9), (12) with (13) also gives

$$\frac{V^* \ln V^*}{V^* - 1} = \frac{V_2 K_1 K_2 F_1 F_2 \mathrm{a}^{1/(\gamma-1)}(1 - \mathrm{a}\tau)}{v_1(K_1F_1\mathrm{x} + K_2F_2\mathrm{a}^{\gamma/(\gamma-1)}\tau)mR_g} \tag{15}$$

From equations (7)-(10), the cycle period can be obtained as follows:

$$t_s = \frac{V_2}{V^* v_1}\left[\frac{(V^* - 1)(\mathrm{x} + \mathrm{a}^{1/(\gamma-1)})}{\mathrm{x}} + \frac{(1 - \mathrm{a}^{1/(\gamma-1)})(V^* + 1)}{\mathrm{y}}\right] \tag{16}$$

The COP (β) and heating load (π) of the cycle can be, respectively, obtained as follows:

$$\beta = \frac{Q_{HC}}{Q_{HC} - Q_{LC}} = \frac{1}{1 - \mathrm{a}} \tag{17}$$

$$\pi = \frac{Q_{HC}}{t_s} = \frac{K_1K_2F_1F_2T_L[1 - (1 - \beta^{-1})\tau](V^* - 1)\mathrm{xy}}{\{(V^* - 1)[\mathrm{x} + (1 - \beta^{-1})^{1/(\gamma-1)}]\mathrm{y} + (V^* + 1)[1 - (1 - \beta^{-1})^{1/(\gamma-1)}]\mathrm{x}\} \\ [K_1F_1\mathrm{x}(1 - \beta^{-1})^{(\gamma-2)/(\gamma-1)} + K_2F_2(1 - \beta^{-1})]} \tag{18}$$

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4. Discussions

4.1 Without the limit of cycle period

From equations (17) and (18), one can see that the COP (β) increases monotonically with the increase of temperature ratio (a) of working fluid, the heating load (π) is the function of the speed ratios (x, y) of the piston and temperature ratio (a) of working fluid, and π increases monotonically with the increase of y. According to the extremum condition $d\pi/d\mathrm{a}=0$, when $\mathrm{a}=\mathrm{a}_{opt}$ satisfies the following equation

$$\begin{aligned}&K_1K_2F_1F_2T_L(K_1F_1\mathrm{x}\mathrm{a}_{opt}{}^{(r-2)/(r-1)}+K_2F_2\mathrm{a}_{opt})(r-1)[(1-\mathrm{a}_{opt}\tau)V_{\mathrm{a}}^*-\tau(V^*-1)]\\&[(V^*-1)(\mathrm{x}+\mathrm{a}_{opt}{}^{1/(r-1)})\mathrm{y}+(V^*+1)(1-\mathrm{a}_{opt}{}^{1/(r-1)})\mathrm{x}]\\&=(1-\mathrm{a}_{opt}\tau)(V^*-1)(K_1F_1\mathrm{x}\mathrm{a}_{opt}{}^{(r-2)/(r-1)}+K_2F_2\mathrm{a}_{opt})\{[(\mathrm{y}+1)\mathrm{x}+(\mathrm{y}-\mathrm{x})\mathrm{a}_{opt}{}^{1/(r-1)}]V_{\mathrm{a}}^*(r-1)\\&+\mathrm{a}_{opt}{}^{(2-r)/(r-1)}[(\mathrm{y}-\mathrm{x})V^*-(\mathrm{x}+\mathrm{y})]\}+(1-\mathrm{a}_{opt}\tau)(V^*-1)[(V^*-1)(\mathrm{x}+\mathrm{a}_{opt}{}^{1/(r-1)})\mathrm{y}\\&+(V^*+1)(1-\mathrm{a}_{opt}{}^{1/(r-1)})\mathrm{x}][K_1F_1\mathrm{x}(r-2)\mathrm{a}_{opt}{}^{-1/(r-1)}+K_2F_2(r-1)]\end{aligned}\tag{19}$$

The relation between the optimal heating load $\pi_{opt-\mathrm{x}}$ and x can be obtained as the following equation:

$$\pi_{opt-\mathrm{x}}=\frac{K_1K_2F_1F_2T_L(1-\mathrm{a}_{opt}\tau)(V^*-1)\mathrm{x}\mathrm{y}}{[(V^*-1)(\mathrm{x}+\mathrm{a}_{opt}{}^{1/(\gamma-1)})\mathrm{y}+(V^*+1)(1-\mathrm{a}_{opt}{}^{1/(\gamma-1)})\mathrm{x}](K_1F_1\mathrm{x}\mathrm{a}_{opt}{}^{(\gamma-2)/(\gamma-1)}+K_2F_2\mathrm{a}_{opt})}\tag{20}$$

where

$$V_{\mathrm{a}}^*=\frac{dV^*}{d\mathrm{a}_{opt}}=\frac{\begin{aligned}&V_2K_1K_2F_1F_2(V^*-1)^2[(\mathrm{a}_{opt}{}^{(2-r)/(r-1)}-r\tau\mathrm{a}_{opt}{}^{1/(r-1)})(K_1F_1\mathrm{x}+K_2F_2\mathrm{a}_{opt}{}^{\gamma/(\gamma-1)}\tau)\\&-(\mathrm{a}_{opt}{}^{1/(r-1)}-\mathrm{a}_{opt}{}^{\gamma/(\gamma-1)}\tau)K_2F_2r\tau\mathrm{a}_{opt}{}^{1/(\gamma-1)}]\end{aligned}}{v_1mR_g(V^*-\ln V^*-1)(r-1)(K_1F_1\mathrm{x}+K_2F_2\mathrm{a}_{opt}{}^{\gamma/(\gamma-1)}\tau)^2}\tag{21}$$

According to the extremum condition $d\pi/d\mathrm{x}=0$, when $\mathrm{x}=\mathrm{x}_{opt}$ satisfies the following equation,

$$\begin{aligned}&K_1K_2F_1F_2T_L\mathrm{y}(\mathrm{x}_{opt}V_{\mathrm{x}}^*+V^*)(K_1F_1\mathrm{x}_{opt}\mathrm{a}^{(\gamma-2)/(\gamma-1)}+K_2F_2\mathrm{a})\\&[(V^*-1)(\mathrm{x}_{opt}+\mathrm{a}^{1/(\gamma-1)})\mathrm{y}+(V^*+1)(1-\mathrm{a}^{1/(\gamma-1)})\mathrm{x}_{opt}]\\&=\mathrm{x}_{opt}(K_1F_1\mathrm{x}_{opt}\mathrm{a}^{(\gamma-2)/(\gamma-1)}+K_2F_2\mathrm{a})(V^*-1)\{[(\mathrm{y}+1)\mathrm{x}_{opt}+(\mathrm{y}-\mathrm{x}_{opt})\mathrm{a}^{1/(\gamma-1)}]V_{\mathrm{x}}^*\\&+(\mathrm{y}+1-\mathrm{a}^{1/(\gamma-1)})V^*-\mathrm{y}-\mathrm{a}^{1/(\gamma-1)}+1\}+\mathrm{x}_{opt}(V^*-1)[(V^*-1)(\mathrm{x}_{opt}+\mathrm{a}^{1/(\gamma-1)})\mathrm{y}\\&+(V^*+1)(1-\mathrm{a}^{1/(\gamma-1)})\mathrm{x}_{opt}]K_1F_1\mathrm{a}^{(\gamma-2)/(\gamma-1)}\end{aligned}\tag{22}$$

the relation between the optimal heating load $\pi_{opt-\beta}$ and β can be obtained as the following equation:

$$\pi_{opt-\beta}=\frac{K_1K_2F_1F_2T_L[1-(1-\beta^{-1})\tau](V^*-1)\mathrm{x}_{opt}\mathrm{y}}{\begin{aligned}&\{\mathrm{y}(V^*-1)[\mathrm{x}_{opt}+(1-\beta^{-1})^{1/(\gamma-1)}]+\mathrm{x}_{opt}(V^*+1)[1-(1-\beta^{-1})^{1/(\gamma-1)}]\}\\&[K_1F_1\mathrm{x}_{opt}(1-\beta^{-1})^{(\gamma-2)/(\gamma-1)}+K_2F_2(1-\beta^{-1})]\end{aligned}}\tag{23}$$

where

$$V_{\mathrm{x}}^*=\frac{dV^*}{d\mathrm{x}_{opt}}=\frac{V_2K_1{}^2K_2F_1{}^2F_2(V^*-1)^2\mathrm{a}^{1/(\gamma-1)}(1-\mathrm{a}\tau)}{v_1mR_g(\ln V^*+1-V^*)(K_1F_1\mathrm{x}_{opt}+K_2F_2\mathrm{a}^{\gamma/(\gamma-1)}\tau)^2}\tag{24}$$

4.2 With the limit of cycle period

In the case with limit of cycle period (i. e. t_s is fixed), combining equations (2), (13) with (15), the heating load (π') of the cycle can be obtained as follows:

$$\pi'=\frac{Q_{HC}}{t_s}=\frac{T_LmR_g\ln V^*(K_1F_1\mathrm{x}+K_2F_2\tau\mathrm{a}^{\gamma/(\gamma-1)})}{t_s(K_1F_1\mathrm{x}\mathrm{a}+K_2F_2\mathrm{a}^{\gamma/(\gamma-1)})}\tag{25}$$

From equations (15) and (25), one can see that π' is the function of x and a. According to the extremum condition $d\pi'/da=0$, when $a=a'_{opt}$ satisfies the following equation

$$\begin{aligned} &T_L m R_g (K_1F_1 x a'_{opt} + K_2F_2 a'^{\gamma/(\gamma-1)}_{opt})[V_a^{*'}(r-1)(K_1F_1 x + K_2F_2\tau a'^{\gamma/(\gamma-1)}_{opt}) \\ &+ V^*\ln V^* K_2F_2\tau r a'^{1/(\gamma-1)}_{opt}] \\ &= t_s V^* \ln V^*(K_1F_1 x + K_2F_2\tau a'^{\gamma/(\gamma-1)}_{opt})[(r-1)K_1F_1 x + K_2F_2 r a'^{1/(\gamma-1)}_{opt}] \end{aligned} \quad (26)$$

The relation between the optimal heating load π'_{opt-x} and x can be obtained as the following equation:

$$\pi'_{opt-x} = \frac{T_L m R_g \ln V^*(K_1F_1 x + K_2F_2\tau a'^{\gamma/(\gamma-1)}_{opt})}{t_s(K_1F_1 x a'_{opt} + K_2F_2 a'^{\gamma/(\gamma-1)}_{opt})} \quad (27)$$

where $V_a^{*'}$ is determined by equation (21) by replacing a_{opt} with a'_{opt}.

According to the extremum condition $d\pi'/dx=0$, when $x=x'_{opt}$ satisfies the following equation

$$\begin{aligned} &T_L m R_g (K_1F_1 x'_{opt} a + K_2F_2 a^{\gamma/(\gamma-1)})[V_x^{*'}(K_1F_1 x'_{opt} + K_2F_2\tau a^{\gamma/(\gamma-1)}) + V^*\ln V^* K_1F_1] \\ &= a t_s K_1F_1 V^* \ln V^*(K_1F_1 x'_{opt} + K_2F_2\tau a^{\gamma/(\gamma-1)}) \end{aligned} \quad (28)$$

The relation between the optimal heating load $\pi'_{opt-\beta}$ and β can be obtained as the following equation:

$$\pi'_{opt-\beta} = \frac{T_L m R_g \ln V^*[K_1F_1 x'_{opt} + K_2F_2\tau(1-\beta^{-1})^{\gamma/(\gamma-1)}]}{t_s[K_1F_1 x'_{opt}(1-\beta^{-1}) + K_2F_2(1-\beta^{-1})^{\gamma/(\gamma-1)}]} \quad (29)$$

where $V_x^{*'}$ is determined by equation (24) by replacing x_{opt} with x'_{opt}.

5. Numerical example

In the calculations, the parameters are set as follows: $m=0.5kg$, $R_g=287.1J/(kg\cdot K)$, $V_2/v_1=140ms$, $T_L=260K$, and $K_1F_1=K_2F_2=16kW/K$. When the cycle period is not limited, $y=4$; when the cycle period is limited, $t_s=160ms$. x and a are the variables with the ranges of $0.7\le x\le 1.1$ and $0.4\le a\le 0.8$. The characteristic curves between the optimal heating load and the speed ratio (x) of the piston, as well as between the optimal heating load and COP (β) in the two cases with and without limit of cycle period are shown in Figures 3-6.

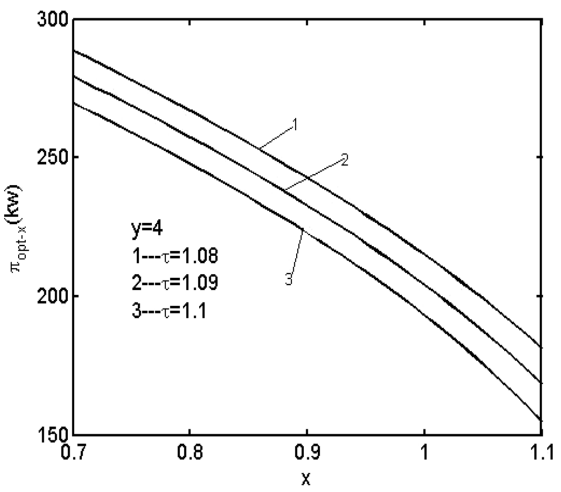

Figure 3. Relation between π_{opt-x} and x with different heat reservoir temperature ratio

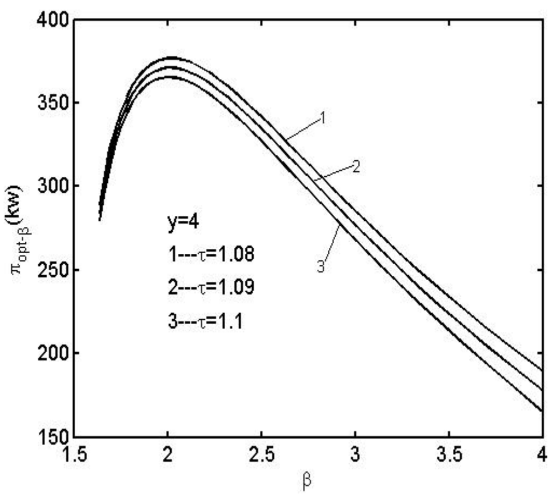

Figure 4. Relation between $\pi_{opt-\beta}$ and β with different heat reservoir temperature ratio

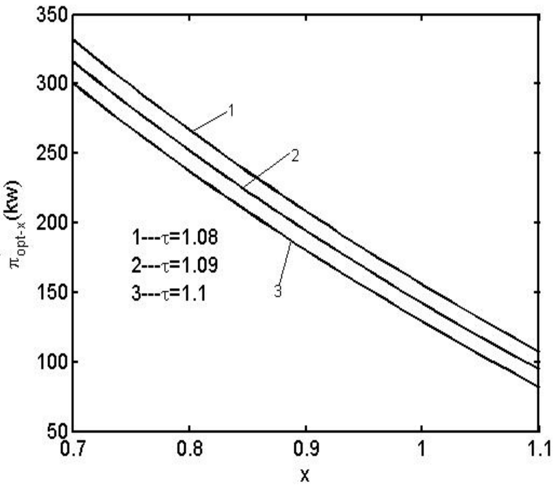

Figure 5. Relation between π'_{opt-x} and x with different heat reservoir temperature ratio

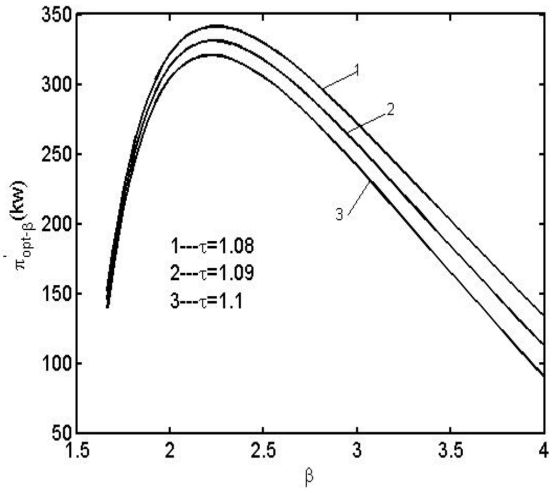

Figure 6. Relation between $\pi'_{opt-\beta}$ and β with different heat reservoir temperature ratio

From Figure 3 and Figure 5, one can see that the optimal heating loads (π_{opt-x} and π'_{opt-x}) decrease monotonically with the increase of x nearly linear-like in the two cases. The optimal heating loads (π_{opt-x} and π'_{opt-x}) decrease with the increase of heat reservoir temperature ratio (τ) for the same x. From Figure 4 and Figure 6, one can see that the optimal heating loads ($\pi_{opt-\beta}$ and $\pi'_{opt-\beta}$) versus β are parabolic-like ones, there exist maximum heating loads and the corresponding COPs. There are two COPs for a fixed heating load, obviously, the heat pump should operate at the point where the COP is larger. The COPs corresponding to the maximum heating loads for the three heat reservoir temperature ratios are nearly equal, the optimal heating loads ($\pi_{opt-\beta}$ and $\pi'_{opt-\beta}$) decrease with the increase of heat reservoir temperature ratio (τ) for the same β. When $\beta \to \beta_c = T_H/(T_H - T_L)$, there is $V^* \to 1$ (the volume expansion ratio of the gas), $\pi_{opt-\beta} \to 0$, and $\pi'_{opt-\beta} \to 0$. When $\beta \to 1$, there is also $V^* \to 1$, $\pi_{opt-\beta} \to 0$, and $\pi'_{opt-\beta} \to 0$. This is different from the performance characteristic of the endoreversible Carnot heat pump without consideration the finite speed of the piston. (the optimal heating load decrease monotonically with the increase of COP, when $\beta \to \beta_c = T_H/(T_H - T_L)$, there is $\pi_{opt} = 0$; when $\beta \to 1$, there is $\pi_{opt} = \pi_{\max}$ [6-8]).

6. Conclusion

Performance of an endoreversible Carnot heat pump cycle with finite speed of the piston is analyzed by using finite time thermodynamics. The two cases (with and without limit of cycle period) are discussed, the optimal formulae between the heating load and speed ratio of the piston, as well as between the heating load and COP are derived for the two cases. The characteristic curves of the heating load versus speed ratio of the piston, the heating load versus COP are obtained by numerical examples. The results show that, the optimal heating loads versus COP of the heat pump considering the characteristics of finite time and finite speed of the piston are parabolic-like one, and these are different from the monotonically decreasing characteristic of the endoreversible Carnot heat pump without consideration of the finite speed of the piston. Figs. 3-6 show that the temperature ratio has large effects on the relations of the optimal heating load versus speed ratio of the piston and the optimal heating load versus COP. The results of this paper can provide some theoretical guidelines for the operation estimation and parameter selection of practical heat pumps.

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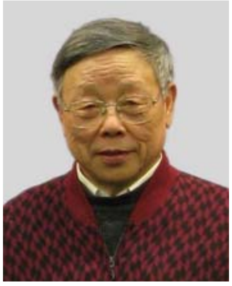

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