Entropy **2004**, *6*(3), 316-326

Entropy

ISSN 1099-4300

www.mdpi.org/entropy/

Optimal Cooling Load and COP Relationship of a Four-Heat-Reservoir Endoreversible Absorption Refrigeration Cycle

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Received: 5 *April* 2004 / *Accepted:* 14 *June* 2004 / *Published:* 15 *June* 2004

Abstract: On the basis of a four-heat-reservoir endoreversible absorption refrigeration cycle model, another linear heat transfer law [i.e., the heat-flux $Q \propto \Delta(T^{-1})$] is adopted, the fundamental optimal relation between the coefficient of performance (COP) and the cooling load, as well as the maximum cooling load and the corresponding COP of the cycle coupled to constant-temperature heat reservoirs are derived by using finite-time thermodynamics or thermodynamic optimization. The optimal distribution of the heat-transfer surface areas is also obtained. Moreover, the effects of the cycle parameters on the COP and the cooling load of the cycle are studied by detailed numerical examples. The results obtained herein are of importance to the optimal design and performance improvement of an absorption refrigeration cycle.

Keywords: Finite Time Thermodynamics, Thermodynamic Optimization, Four-Heat-Reservoir Endoreversible Absorption Refrigeration Cycle, Optimal Performance

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Introduction

The absorption refrigerators can be driven by ‘low-grade’ heat energy such as waste heat in industries, solar energy and geothermal energy, and have a large potential for reducing the heat pollution for the environment. Thus, absorption refrigerators for industrial and domestic use are generating renewed interest throughout the world. In the last years, finite-time thermodynamics (or endoreversible thermodynamics, or entropy generation minimization, or thermodynamic optimization) [1-4] was applied to the performance study of absorption refrigerators, and a lot of results, which are different from those by using the classical thermodynamics, have been obtained. Yan *et al.* [6], Wijeysundera [7, 8], Goktun [9] and Chen *et al.* [10] analyzed the performance of the three-heat-reservoir endoreversible [6-8] and irreversible [9,10] absorption refrigeration cycles with Newton's heat transfer law. Chen *et al.* [11-13] studied the performance of the three-heat-reservoir endoreversible [11, 12] and irreversible [13] absorption refrigeration cycles with another linear heat transfer law, i.e., linear phenomenological law, $Q \propto \Delta(T^{-1})$. A three-heat-reservoir absorption refrigerator is a simplified model that the temperature of a condenser is equal to that of an absorber, but a real absorption refrigerator is not. Therefore, a four-heat-reservoir absorption refrigeration cycle model is closer to a real absorption refrigerator. The performance of the four-heat-reservoir absorption refrigeration cycle with Newton's heat transfer law was studied by Chen [14], Shi *et al.* [15] and Zheng *et al.* [16]. Chen [14] deduced the maximum cooling load limit and the corresponding coefficient of performance (COP) of the endoreversible four-heat-reservoir absorption refrigeration cycle with the sole irreversibility of heat transfer, Shi *et al.* [15] deduced the fundamental optimal relation between the cooling load and the COP of the endoreversible four-heat-reservoir absorption refrigeration cycle with the sole irreversibility of heat transfer, and Zheng *et al.* [16] deduced the optimal heat transfer surface areas of the four heat exchangers the endoreversible four-heat-reservoir absorption refrigeration cycle with the sole irreversibility of heat transfer. On the basis of these research work, a four-heat-reservoir endoreversible absorption refrigeration cycle with linear phenomenological heat transfer law is established in this paper. The fundamental optimal relation between the COP and the cooling load, as well as the maximum cooling load and the corresponding COP of the cycle are derived. The results can provide the theoretical bases for the optimal design and operation of real absorption refrigerator operating between four temperature levels. The present work is different from a recent work of the authors [17]. In Ref. [17], an endoreversible four-heat-reservoir absorption heat-transformer with Newton's heat transfer law was established, and the fundamental optimal relation between the COP and the heating load, as well as the maximum heating load and the corresponding COP of the cycle were derived.

Physical Model

A four-heat-reservoir endoreversible absorption refrigeration cycle that consists of a generator, an evaporator, an absorber and a condenser is shown in Fig. 1. The flow of the working fluid in the cycle system is stable and the different parts of the working fluid exchange heat with the heat reservoirs at temperature T_H, T_L, T_O and T_M during the full time (cycle period) τ, whereas there are thermal resistances between the working fluid and the external heat reservoirs. Therefore, the corresponding working fluid temperatures are T_1, T_2, T_3 and T_4, respectively. Work input required by the solution pump in the system is negligible relative to the energy input to the generator and is often neglected for the purpose of analysis [6-17]. It is assumed that the heat transfers between the working fluid in the heat exchangers and the external heat reservoirs are carried out under a finite temperature difference and obey linear phenomenological heat transfer law [i.e., the heat-flux $Q \propto \Delta(T^{-1})$], and these heat exchange processes are isothermal and the equations of heat transfer may be written as

$$Q_1 = U_1 A_1 (T_1^{-1} - T_H^{-1})\tau \quad (1)$$

$$Q_2 = U_2 A_2 (T_2^{-1} - T_L^{-1})\tau \quad (2)$$

$$Q_3 = U_3 A_3 (T_O^{-1} - T_3^{-1})\tau \quad (3)$$

$$Q_4 = U_4 A_4 (T_M^{-1} - T_4^{-1})\tau \quad (4)$$

where U_1, U_2, U_3 and U_4 are, respectively, the overall heat-transfer coefficients of the generator, evaporator, condenser and absorber; and A_1, A_2, A_3 and A_4 are, respectively, the heat-transfer surface areas of the generator, evaporator, condenser and absorber. The overall heat-transfer surface area A is

$$A = A_1 + A_2 + A_3 + A_4 \quad (5)$$

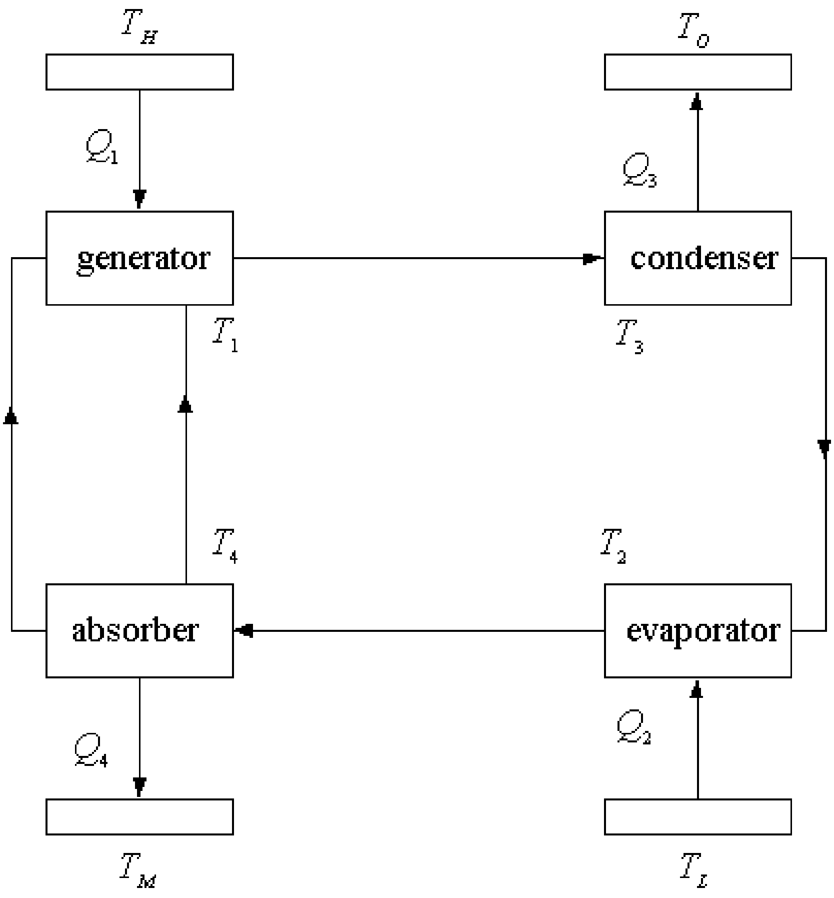

Fig.1 A four-heat-reservoir endoreversible absorption cycle model

Fundamental Optimal Relation

From the first law of thermodynamics, one has

$$Q_1 + Q_2 - Q_3 - Q_4 = 0 \quad (6)$$

From the second law of thermodynamics and the endoreversible property of the cycle, one has

$$\frac{Q_1}{T_1}+\frac{Q_2}{T_2}-\frac{Q_3}{T_3}-\frac{Q_4}{T_4}=0 \tag{7}$$

Defining the parameter a (the distribution of the total heat reject quantity between the condenser and the absorber)

$$a=Q_4/Q_3 \tag{8}$$

From equations (6) and (7), one can obtain the COP of an endoreversible absorption refrigerator

$$\varepsilon=\frac{Q_2}{Q_1}=\frac{T_3^{-1}+aT_4^{-1}-(1+a)T_1^{-1}}{(1+a)T_2^{-1}-T_3^{-1}-aT_4^{-1}} \tag{9}$$

Using equations (1) - (9), the cooling load of the refrigerator can be written as

$$R=\frac{Q_2}{\tau}=A[\frac{\varepsilon^{-1}}{U_1(T_1^{-1}-T_H^{-1})}+\frac{1+\varepsilon^{-1}}{(1+a)U_3(T_O^{-1}-T_3^{-1})}+\frac{a(1+\varepsilon^{-1})}{(1+a)U_4(T_M^{-1}-T_4^{-1})}+\frac{1}{U_2(B-T_L^{-1})}]^{-1} \tag{10}$$

where $B=\dfrac{(1+\varepsilon^{-1})(T_3^{-1}+aT_4^{-1})}{(1+a)}-\varepsilon^{-1}T_1^{-1}$.

Using equation (10) and the extremal conditions $\partial R/(\partial T_1^{-1})=0$, $\partial R/(\partial T_3^{-1})=0$ and $\partial R/(\partial T_4^{-1})=0$, one can derive the temperatures of the working fluid in the generator, absorber, condenser and evaporator, which correspond the optimal cooling load for the given COP. Substituting them into equation (10) yields the fundamental optimal relation between the cooling load and the COP of the four-heat-reservoir endoreversible absorption refrigeration cycle with linear phenomenological heat transfer law as follows

$$R=\frac{A(1+a)[(1+\varepsilon^{-1})(aT_M^{-1}+T_O^{-1})-(1+a)(T_L^{-1}+T_H^{-1}\varepsilon^{-1})]}{[(1+a)(U_2^{-\frac{1}{2}}+U_1^{-\frac{1}{2}}\varepsilon^{-1})+(1+\varepsilon^{-1})(U_3^{-\frac{1}{2}}+aU_4^{-\frac{1}{2}})]^2} \tag{11}$$

Equation (11) is the major results of this paper. It can reveal the $R-\varepsilon$ characteristics of a four-heat-reservoir endoreversible absorption refrigeration cycle affected by thermal resistance, and some significant results and new bounds may be derived from it.

Using equations (1)-(4), (11), and the temperatures of the working fluid in the generator, absorber, condenser and evaporator which are derived, one can obtain the optimal distribution relation of the heat-transfer surface areas as follows

$$\frac{A_1}{A_2}=\frac{U_2^{\frac{1}{2}}}{\varepsilon U_1^{\frac{1}{2}}} \tag{12}$$

$$\frac{A_3}{A_4}=\frac{U_4^{\frac{1}{2}}}{aU_3^{\frac{1}{2}}} \tag{13}$$

$$\frac{A_1+A_2}{A_3+A_4}=\frac{(U_1^{-\frac{1}{2}}+\varepsilon U_2^{-\frac{1}{2}})(1+a)}{(U_3^{-\frac{1}{2}}+U_4^{-\frac{1}{2}}a)(1+\varepsilon)} \tag{14}$$

From equations (5) and (12)-(14), one can find the relations between the heat-transfer surface areas of each heated exchanger and the total heat-transfer surface area A as follows

$$A_1=U_1^{-\frac{1}{2}}(1+a)A/[(U_1^{-\frac{1}{2}}+\varepsilon U_2^{-\frac{1}{2}})(1+a)+(U_3^{-\frac{1}{2}}+aU_4^{-\frac{1}{2}})(1+\varepsilon)] \tag{15}$$

$$A_2=\varepsilon U_2^{-\frac{1}{2}}(1+\varepsilon)A/[(U_1^{-\frac{1}{2}}+\varepsilon U_2^{-\frac{1}{2}})(1+a)+(U_3^{-\frac{1}{2}}+aU_4^{-\frac{1}{2}})(1+\varepsilon)] \tag{16}$$

$$A_3=U_3^{-\frac{1}{2}}(1+\varepsilon)A/[(U_1^{-\frac{1}{2}}+\varepsilon U_2^{-\frac{1}{2}})(1+a)+(U_3^{-\frac{1}{2}}+aU_4^{-\frac{1}{2}})(1+\varepsilon)] \tag{17}$$

$$A_4 = aU_4^{-\frac{1}{2}}(1+\varepsilon)A/[(U_1^{-\frac{1}{2}} + \varepsilon U_2^{-\frac{1}{2}})(1+a) + (U_3^{-\frac{1}{2}} + aU_4^{-\frac{1}{2}})(1+\varepsilon)] \tag{18}$$

Results and Discussion

1. When $R = 0$, one can obtain the reversible COP ε_r of the four-heat-reservoir absorption refrigeration cycle

$$\varepsilon_r = \frac{aT_M^{-1} + T_O^{-1} - (1+a)T_H^{-1}}{(1+a)T_L^{-1} - aT_M^{-1} - T_O^{-1}} \tag{19}$$

It can be seen that the optimal COP of the four-heat-reservoir endoreversible absorption refrigeration cycle can't exceed the reversible COP ε_r. This shows that the real absorption refrigerators must decrease the COP level if one wants to obtain some cooling load.

2. When $\varepsilon < \varepsilon_r$, there exists a maximum cooling load. Using equation (11) and the extremal condition $\partial R/(\partial \varepsilon) = 0$, one can obtain

$$\varepsilon_R = \frac{aT_M^{-1} + T_O^{-1} - (1+a)T_H^{-1}}{2(1+a)T_L^{-1} - (1+a)T_H^{-1} - aT_M^{-1} - T_O^{-1}} \tag{20}$$

and

$$R_m = \frac{A(T_L^{-1} - T_H^{-1})[aT_M^{-1} + T_O^{-1} - (1+a)T_H^{-1}]^2}{\{[2(1+a)U_1^{-\frac{1}{2}} + 2(U_3^{-\frac{1}{2}} + aU_4^{-\frac{1}{2}})]T_L^{-1} - [(1+a)(U_1^{-\frac{1}{2}} + U_2^{-\frac{1}{2}}) + 2(U_3^{-\frac{1}{2}} + aU_4^{-\frac{1}{2}})]T_H^{-1} + (U_2^{-\frac{1}{2}} - U_1^{-\frac{1}{2}})(aT_M^{-1} + T_O^{-1})\}^2} \tag{21}$$

where R_m is the maximum cooling load, and ε_R is the corresponding COP. R_m and ε_R are two important parameters of the four-heat-reservoir endoreversible absorption refrigeration cycle with linear phenomenological heat transfer law, because they determine the upper bound for the cooling load and the lower bound of the COP, and provide a finite-time thermodynamic criteria for the optimal design of real absorption refrigerators, i.e., the real absorption refrigerator design must match the condition $\varepsilon_r > \varepsilon \geq \varepsilon_R$ to make the refrigerator operates under the optimal conditions.

When $a = 1$, $T_M = T_O$ and $U_i (i = 1,2,3,4) = U$, the four-heat-reservoir endoreversible absorption refrigeration cycle with linear phenomenological heat transfer law becomes the three-heat-reservoir endoreversible absorption refrigeration cycle with linear phenomenological heat transfer law, and equations (20) and (21) become [11]

$$\varepsilon_R' = \frac{T_L(T_H - T_O)}{T_H(2T_O - T_L) - T_O T_L} \tag{22}$$

$$R_m' = \frac{UA}{16}\frac{T_L(T_H - T_O)^2}{T_O^2 T_H (T_H - T_L)} \tag{23}$$

3. Defining the ratio $x = T_O/T_L$ of the temperatures of the condenser to the evaporator, and the ratio $y = T_H/T_M$ of the temperatures of the generator to the absorption, one can analyze the performance numerically. In the calculation, $A = 1100m^2$, $T_M = 305K$, $T_L = 273K$, $U_1 = 458.167kW \cdot K/m^2$, $U_2 = 682.223kW \cdot K/m^2$, $U_3 = 1622.5kW \cdot K/m^2$ and $U_4 = 458.167kW \cdot K/m^2$ are set [5, 18].

The influence of a on the optimal cooling load R versus the COP ε of the four-heat-reservoir endoreversible absorption refrigeration cycle with $T_H = 403K$ and $T_O = 313K$ is shown in Fig. 2. The influence of x on the optimal cooling load R versus the COP ε with $T_H = 403K$ and $a = 1.5$ is shown in Fig. 3. The influence of y on the optimal cooling load R versus the COP ε with $T_O = 313K$ and $a = 1.5$ is shown in Fig. 4. Figures 2-4 show that the optimal cooling load R versus the COP ε is a parabolic curve, and there exists a maximum cooling load R_m and the corresponding COP ε_R. When $R < R_m$, there exist two different ε for a fixed R, one is larger than ε_R, and another is smaller than ε_R. When $\varepsilon < \varepsilon_R$, the COP decreases with the decrease of the cooling load, so the optimal operation range of the COP of the absorption refrigerators should be selected in $\varepsilon_r > \varepsilon \geq \varepsilon_R$.

The influences of x on the reversible COP ε_r, the maximum cooling load R_m and the corresponding COP ε_R versus a with $T_H = 403K$ and $y = 1.35$ are shown in Figs. 5-7. Figures 5-7 show that for a fixed a, ε_r, R_m and ε_R decrease with the increase of x. When a is larger than one in value, the influence of the ε_r, R_m and ε_R are less. There is a special point $x = 1.1172$ for the cycle. When $x < 1.1172$, $\varepsilon_r \geq 2.0747$, $\varepsilon_R \geq 0.5092$, both ε_r and ε_R decrease with the increase of a; and when $x > 1.1172$, $\varepsilon_r < 2.0747$, $\varepsilon_R < 0.5092$, both ε_r and ε_R increase with the increase of a. These

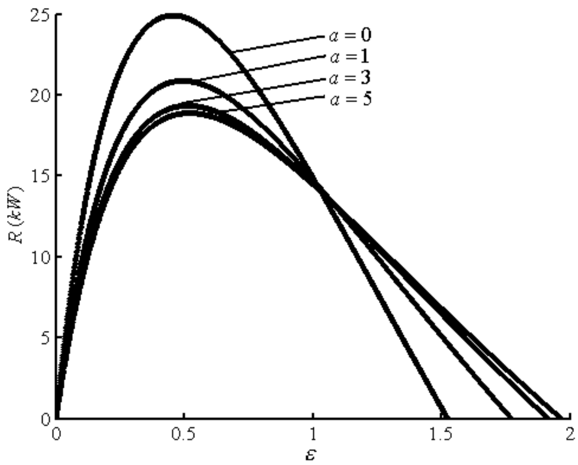

Fig.2 The influence of a on the optimal cooling load R versus the COP ε

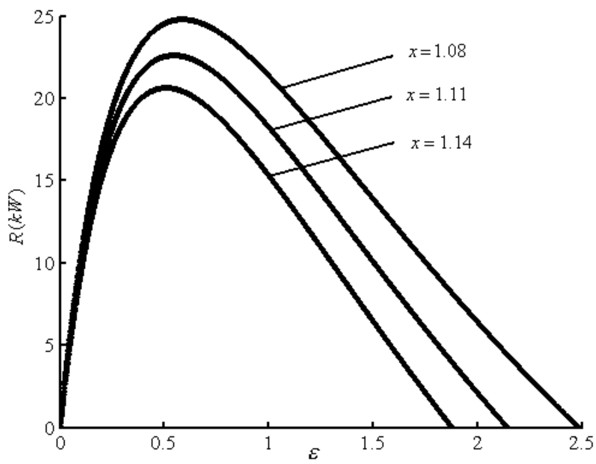

Fig.3 The influence of x on the optimal cooling load R versus the COP ε

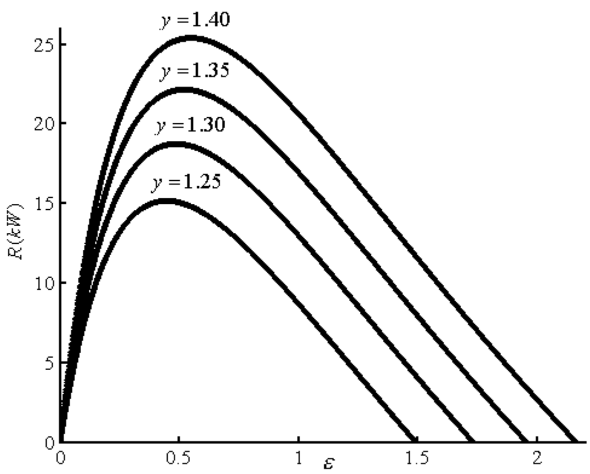

Fig.4 The influence of y on the optimal cooling load R versus the COP ε

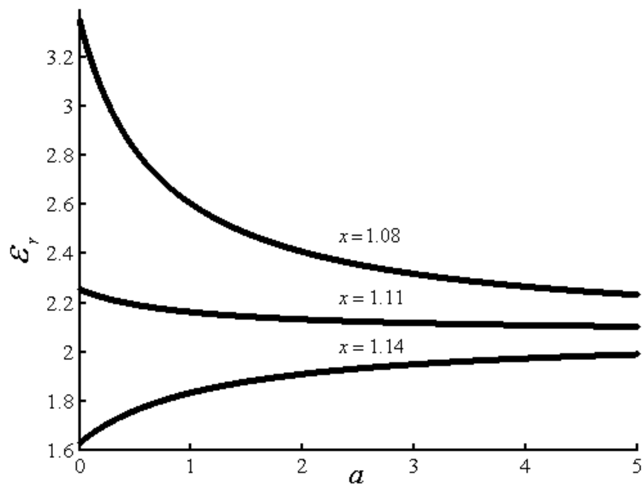

Fig.5 The influence of x on the COP ε_r versus a

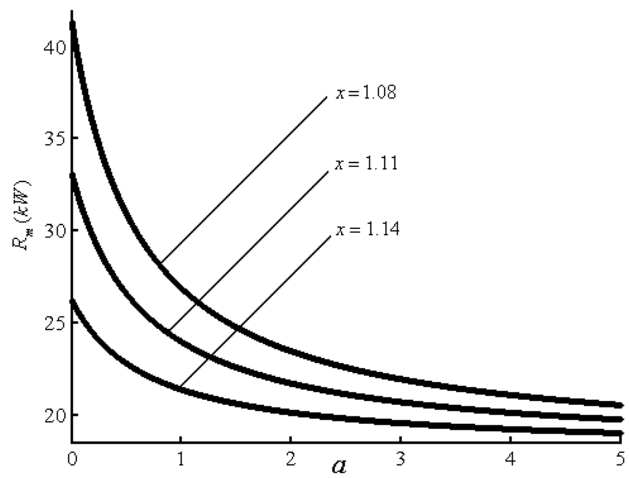

Fig.6 The influence of x on the maximum cooling load R_m versus a

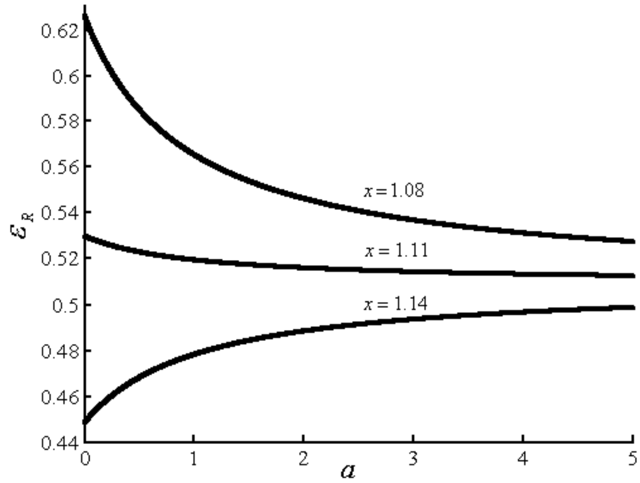

Fig.7 The influence of x on the COP ε_R versus a

imply that ε_r and ε_R will reach their asymptotic values, i.e., $\varepsilon_r \to 2.0747$ and $\varepsilon_R \to 0.5092$ when a tends to infinity. R_m decreases with the increase of a, and when a is larger than one in value, the influence of them are less significant.

The influence of y on the reversible COP ε_r, the maximum cooling load R_m and the corresponding COP ε_R versus a with $T_O = 313K$ and $x = 1.14$ are shown in Figs. 8-10. Figures 8-10 show that for a fixed a, ε_r, R_m and ε_R increase with the increase of y. Here $x > 1.1172$ holds, R_m decreases with the increase of a, ε_r and ε_R increase with the increase of a.

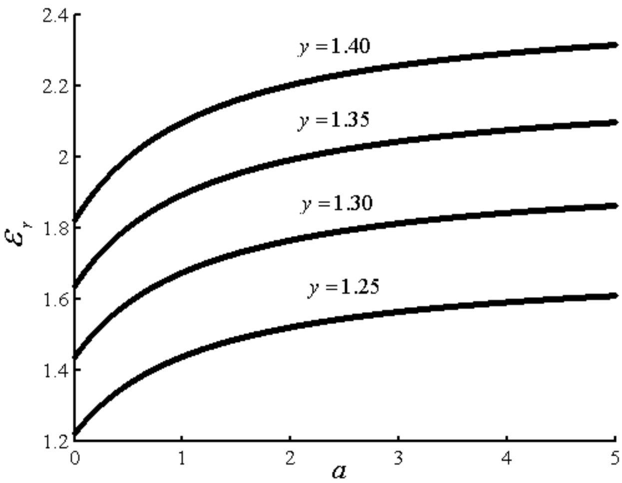

Fig.8 The influence of y on the COP ε_r versus a

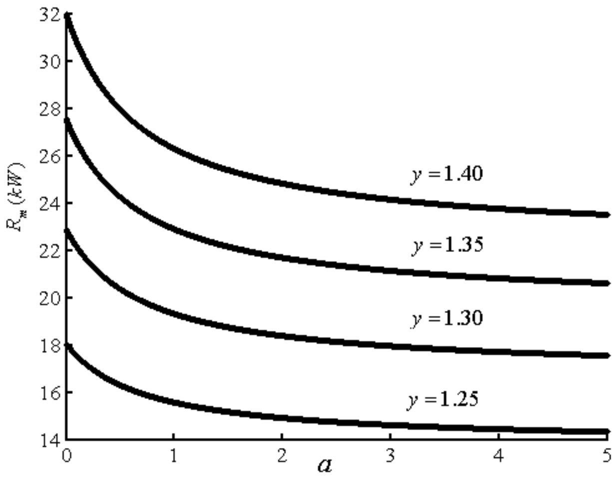

Fig.9 The influence of y on the maximum cooling load R_m versus a

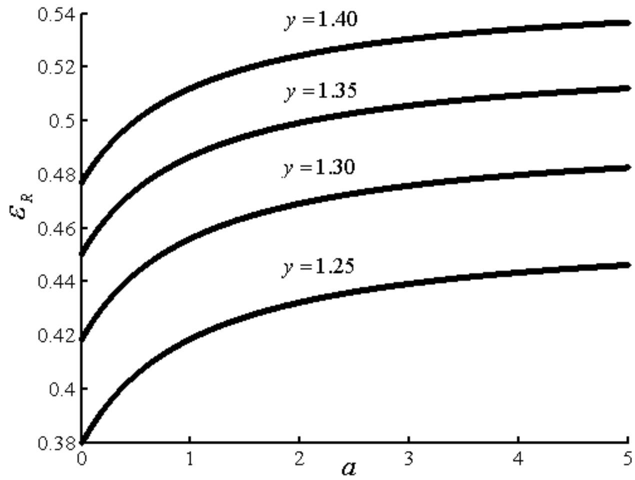

Fig.10 The influence of y on the COP ε_R versus a

4 The performance optimization can be carried out by optimizing the distribution of the heat exchanger total inventory [19, 20]. Using $UA = U_1A_1 + U_2A_2 + U_3A_3 + U_4A_4$ to replace equation (5), i.e. using the distribution of the heat conductances to replace the distribution of the heat-transfer surface areas, one can obtain the optimal distribution relation of the heat conductances as follows

$$\frac{U_1A_1}{U_2A_2} = \varepsilon^{-1} \tag{24}$$

$$\frac{U_3A_3}{U_4A_4} = a^{-1} \tag{25}$$

$$U_1A_1 + U_2A_2 = U_3A_3 + U_4A_4 \tag{26}$$

From equations (24)-(26), one can obtain

$$U_1A_1 = UA/2(1+\varepsilon) \tag{27}$$

$$U_2A_2 = \varepsilon UA/2(1+\varepsilon) \tag{28}$$

$$U_3A_3 = UA/2(1+a) \tag{29}$$

$$U_4A_4 = aUA/2(1+a) \tag{30}$$

The optimal relation between the cooling load and the COP in this case is as following

$$R = \frac{UA}{4}\left[\frac{aT_M^{-1} + T_O^{-1}}{(1+a)(1+\varepsilon^{-1})} - \frac{T_L^{-1} + T_H^{-1}\varepsilon^{-1}}{(1+\varepsilon^{-1})^2}\right] \tag{31}$$

Using equation (31) and the extremal condition $\partial R/(\partial \varepsilon) = 0$ yields the maximum cooling load R_m and the corresponding COP ε_R as follows

$$\varepsilon_R = \frac{aT_M^{-1} + T_O^{-1} - (1+a)T_H^{-1}}{2(1+a)T_L^{-1} - (1+a)T_H^{-1} - aT_M^{-1} - T_O^{-1}} \tag{32}$$

$$R_m = \frac{UA[aT_M^{-1} + T_O^{-1} - (1+a)T_H^{-1}]^2}{16(1+a)^2(T_L^{-1} - T_H^{-1})} \tag{33}$$

Conclusion

The performance of the four-heat-reservoir endoreversible absorption refrigeration cycle with linear phenomenological heat transfer law is analyzed and optimized by using finite-time thermodynamics in this paper. Moreover, the effects of the cycle parameters on the COP and the cooling load of the cycle are studied by detailed numerical examples. The selection range for the practice parameters of the four-heat-reservoir endoreversible absorption refrigeration cycle with another linear heat transfer law are derived. The results obtain herein have realistic significance and may provide some new theoretical guidance for the optimal design and performance improvement of real absorption refrigerators.

Acknowledgements

This paper is supported by The Foundation for the Author of National Excellent Doctoral Dissertation of P. R. China (Project No. 200136) and The National Key Basic Research and Development Program of P. R. China (Project No. G2000026301).

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