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International Journal of Thermal Sciences 47 (2008) 479–485

International Journal of Thermal Sciences

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Performance optimization for an irreversible four-temperature-level absorption heat pump

Yuewu Huang [a,*], Dexing Sun [b], Yanming Kang [a]

[a] *College of Environmental Science and Engineering, Donghua University, Shanghai 200051, PR China*
[b] *School of Municipal and Environmental Engineering, Harbin Institute of Technology, Harbin 150090, PR China*

Received 22 August 2006; received in revised form 27 February 2007; accepted 23 March 2007

Available online 11 May 2007

Abstract

The optimal performance of an absorption heat pump operating between four temperature levels with the losses of heat resistance and internal irreversibility is analyzed first by taking the total heat-transfer area of heat exchangers as an objective function. The minimum total heat-transfer area is described in terms of the rate of the entropy changes of four heat reservoirs, and the generally optimal relation among the heating load, the coefficient of performance and total heat-transfer area is achieved. Then, an ecological optimization criterion is proposed for the best mode of operation of the absorption heat pump. We investigate the ecological optimization performance and derive the optimal heating load, coefficient of performance and entropy production rate at the maximum ecological criterion for the absorption heat pump. Effects of thermal reservoir temperature and the internal irreversibility on the ecological function have been discussed.
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Keywords: Finite-time thermodynamics; Absorption heat pump; Minimum total heat-transfer area; Generally optimal relation; Ecological optimization

1. Introduction

Absorption heat pumps can employ the available waste heat from power plants and industrial processes as driving energy to make waste heat recovery economically attractive, thus improving the total energy utilization and reducing environment pollution [1,2]. In recent years, finite-time thermodynamics [3–12] was applied to the performance study of absorption heat pumps [1,13–22], and a lot of results obtained are different from those according to the theory of classical thermodynamics. A four-temperature-level absorption heat-pump cycle model is closer to a real absorption heat pump when the absorption heat pump is required to supply heat simultaneously to two spaces at different temperatures. Chen [17] established a four-temperature-level absorption heat-pump cycle model, considering the effects of heat resistance and internal irreversibility, and studied its performance with a linear (Newtonian) heat-transfer law. Chen [21] further analyzed a four-temperature-level absorption heat-pump cycle model including heat leak between the heated space and the environmental reservoir. Qin et al. [22] analyzed the performance of an endoreversible four-temperature-level absorption heat pump with a generalized heat-transfer law.

Some efforts have been made to investigate the influence of multi-irreversibilities on the performance of a class of heat pumps driven by heat energy. But these results arose mostly by focusing one's attention on the working substance. The optimal temperatures of the working substance under maximum coefficient of performance at constant total heat-transfer surface area and heating load condition were derived [17]. We should pay attention both to the heat reservoirs and to working substance in isolated system affected by the irreversibility of finite-rate heat-transfer and the internal irreversibilities of the working substance. When total heat-transfer area of the four heat exchangers is assumed to be constant, the optimal relation only between the coefficient of performance and the heating load is not generally optimal relation in these studies. So that it is necessary to develop the new theory of four-temperature-level absorption heat pumps further. Angulo-Brown [23] firstly established the ecological criterion of a Carnot heat-engine and Yan [24] modified it. Yan and Lin [25] established an eco-

* Corresponding author. Tel.: +86 21 65553637; fax: +86 21 65554206.
E-mail address: huangyaowu@sina.com (Y. Huang).

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doi:10.1016/j.ijthermalsci.2007.03.013

Nomenclature

A	total heat-transfer surface area	m^2
A_i $(i = g, a, c, e)$	heat-transfer surface area of the ith heat exchanger	m^2
COP	coefficient of performance	
E	ecological criterion function	kW
I	internal irreversibility factor	
L	Lagrangian function	
n	distribution of the total rate of heat-reject between the absorber and the condenser	
q	heating load	kW
$q_i(g, q, c, e)$	rate of heat-transfer of ith heat reservoir	kW
T_i $(i = g, a, c, e)$	temperature of ith heat reservoir	K
T_i $(i = 1, 2, 3, 4)$	working substance temperature in ith heat exchanger	K
U_i $(i = g, a, c, e)$	heat-transfer coefficient of the ith heat exchanger	$kW\,m^{-2}\,K^{-1}$

Greek symbols

λ	Lagrangian multiplier	
μ	dissipation coefficient of the heating load	
σ	entropy-production rate during the whole cycle	$kW\,K^{-1}$
σ_i $(i = g, a, c, e)$	rate of entropy changes of the ith heat reservoir	$kW\,K^{-1}$

Subscripts

E	at maximum ecological function point
$0, I$	at zero ecological function points
max	maximum
min	minimum
r	reversible cycle

logical criterion and analyzed the ecological optimal performance of a three-temperature-level refrigerator by assuming that the overall heat-transfer coefficients of the heat exchangers are the same. Huang [19] has analyzed the optimal ecological performance of an irreversible four-temperature-level absorption heat-transformer. Qin et al. [20] investigated the optimal ecological performance of endoreversible absorption heat pumps. This paper introduces an ecological criterion for the best mode of operation of the four-temperature-level absorption heat-pump cycle, and analyzes the ecological optimal performance of an irreversible absorption heat pump assuming a linear (Newtonian) heat-transfer law.

2. An irreversible cycle model

Fig. 1 shows a schematic diagram of a four-temperature-level absorption heat pump that consists of a generator, an absorber, a condenser and an evaporator without the solution heat exchanger [17]. The flow of the working substance in the cycle is stable and the working substance exchanges heat with the heat reservoirs at temperatures T_g, T_α, T_c and T_e in the generator, absorber, condenser and evaporator, respectively. T_e is the environment temperature of the cycle. There exists thermal resistance between the working substance and the external heat reservoirs. The corresponding temperatures of the working substance in the generator, absorber, condenser and evaporator are T_1, T_2, T_3 and T_4, respectively. The heat-transfer between the working substance and the external heat-reservoir in the heat exchanger is carried out under a finite temperature-difference. Thus, the heat-transfer equations in the generator, absorber, condenser and evaporator may be written as

$$q_g = U_g A_g (T_g - T_1) = -\sigma_g T_g \quad (1)$$

$$q_a = U_a A_a (T_2 - T_a) = \sigma_a T_a \quad (2)$$

$$q_c = U_c A_c (T_3 - T_c) = \sigma_c T_c \quad (3)$$

$$q_e = U_e A_e (T_e - T_4) = -\sigma_e T_e \quad (4)$$

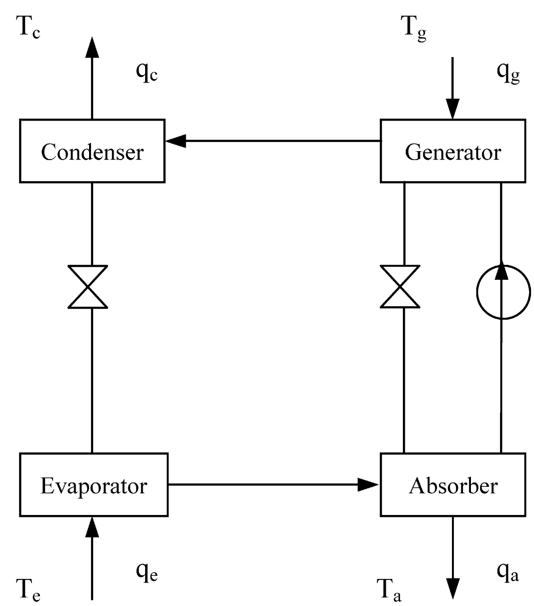

Fig. 1. A schematic diagram of a four-temperature-level absorption heat pump.

where σ_g, σ_a, σ_c and σ_e are corresponding the rate of entropy changes of four heat reservoirs, U_g, U_a, U_c and U_e are the overall heat-transfer coefficients, and A_g, A_a, A_c and A_e are the heat-transfer surface areas of the generator, absorber, condenser and evaporator, respectively. The total heat-transfer area between the cycle system and the external heat reservoirs is

$$A = A_g + A_a + A_c + A_e \quad (5)$$

From the first law of thermodynamics,

$$q_g + q_e - q_a - q_c = \sigma_g T_g + \sigma_e T_e + \sigma_a T_a + \sigma_c T_c = 0 \quad (6)$$

According to the second law of thermodynamics, an irreversibility factor [17] is introduced to describe the irreversibility due to the internal dissipation of the working substance.

$$I = (q_a/T_2 + q_c/T_3)/(q_g/T_1 + q_e/T_4) \geqslant 1 \quad (7)$$

Eq. (7) shows clearly that when the cycle of the working substance is endoreversible, $I = 1$; when the cycle of the working substance is internally irreversible, $I > 1$.

Using Eqs. (1)–(7), the coefficient of performance COP and the heating load q of an absorption heat pump are given by

$$COP = \frac{q_a + q_c}{q_g} = \frac{I(1+n)(T_1 - T_4)T_3T_2}{T_1[IT_3T_2(1+n) - T_4(nT_3 + T_2)]} \tag{8}$$

and

$$q = q_a + q_c = A\left[\frac{1}{COP\,U_g(T_g - T_1)} + \frac{COP - 1}{COP\,U_e(T_e - T_4)} + \frac{1}{(1+n)U_c(T_3 - T_c)} + \frac{n}{(1+n)U_a(T_2 - T_a)}\right]^{-1} \tag{9}$$

where $n = q_a/q_c$ denotes the distribution ratio of the total heat-reject quantity between the absorber and the condenser. The cycle model mentioned above is more realistic and useful than those adopted in classical thermodynamics and a generalization of the endoreversible cycle models.

3. The minimum total heat-transfer area

It turns out that focusing on the working substance leaves much of the picture hidden. There are several additional inequalities governing the absorption heat pump operating finite thermal conductance subject to both thermal resistance losses and the internal irreversibility of the working substance. These inequalities emerge from a more reservoir-oriented viewpoint on the problem. We explore the set of possible absorption heat pump operations in as complete a fashion as possible; in particular, we will show exactly which processes are indeed feasible.

Using Eqs. (1)–(4), Eq. (7) may be rewritten as

$$\phi(A_g, A_a, A_c, A_e) = \frac{IU_gA_g\sigma_g}{U_gA_g + \sigma_g} + \frac{IU_eA_e\sigma_e}{U_eA_e + \sigma_e} + \frac{U_cA_c\sigma_c}{U_cA_c + \sigma_c} + \frac{U_aA_a\sigma_a}{U_aA_a + \sigma_a} = 0 \tag{10}$$

To minimize the total heat-transfer area for a given constraint condition, i.e. Eq. (10), the Lagrangian is introduced

$$L = A_g + A_a + A_c + A_e + \lambda\phi(A_g, A_a, A_c, A_e) \tag{11}$$

For given values of σ_g, σ_a, σ_c and σ_e which satisfy the first law of thermodynamics, i.e. Eq. (6), from the Euler–Lagrange equations

$$\delta L/\delta A_i = 0 \quad (i = g, a, c, e) \tag{12}$$

and Eq. (10), we can prove, when the heat-transfer area of the generator, absorber, condenser and evaporator are given by

$$A_g = -\sigma_g\big(\theta + \sqrt{IU_g}\big)(\theta U_g)^{-1} \tag{13}$$

$$A_a = \sigma_a\big(-\theta + \sqrt{U_a}\big)(\theta U_a)^{-1} \tag{14}$$

$$A_c = \sigma_c\big(-\theta + \sqrt{U_c}\big)(\theta U_c)^{-1} \tag{15}$$

$$A_e = -\sigma_e\big(\theta + \sqrt{IU_e}\big)(\theta U_e)^{-1} \tag{16}$$

respectively, where $\theta = [I(\sigma_g + \sigma_e) + \sigma_a + \sigma_c][\sigma_a/\sqrt{U_a}.. + \sigma_c/\sqrt{U_c} - \sqrt{I}(\sigma_g/\sqrt{U_g} + \sigma_e/\sqrt{U_e})]^{-1}$.

The total heat-transfer area attains its minimum, i.e.

$$A_{\min} = \theta^{-1}\left[\frac{\sigma_a(-\theta + \sqrt{U_a})}{U_a} + \frac{\sigma_c(-\theta + \sqrt{U_c})}{U_c} - \frac{\sigma_e(\theta + \sqrt{IU_e})}{U_e} - \frac{\sigma_g(\theta + \sqrt{IU_g})}{U_g}\right] \tag{17}$$

The inequality $A \geqslant A_{\min}$ holds for the absorption heat pump described, where all heat-reservoir temperatures are constant. For comparison with the second law of thermodynamics, it is more useful to express Eq. (17) in the slightly different form

$$\begin{aligned}\sigma_g + \sigma_a + \sigma_c + \sigma_e &\geqslant (1 - I)(\sigma_g + \sigma_e) + \frac{1}{A}\left(\frac{\sigma_a}{\sqrt{U_a}} + \frac{\sigma_c}{\sqrt{U_c}} - \frac{\sqrt{I}\sigma_g}{\sqrt{U_g}} - \frac{\sqrt{I}\sigma_e}{\sqrt{U_e}}\right)\\ &\times\left[\frac{\sigma_a(-\theta + \sqrt{U_a})}{U_a} + \frac{\sigma_c(-\theta + \sqrt{U_c})}{U_c} - \frac{\sigma_e(\theta + \sqrt{IU_e})}{U_e} - \frac{\sigma_g(\theta + \sqrt{IU_g})}{U_g}\right]\end{aligned} \tag{18}$$

This inequality (18) can be expressed as an inequality relating the total heat-transfer area A, the net entropy changes σ_g, σ_a, σ_c and σ_e of the reservoirs and an irreversibility factor I. It sets a positive definite lower bound on the total entropy production rate of the universe resulting from the whole cycle. The equality holds if Eqs. (13)–(16) are satisfied.

When the overall heat-transfer coefficients of the heat exchangers are the same, i.e. $U_g = U_e = U_a = U_c$, the inequality (18) may be simply expressed as

$$\sigma_g + \sigma_a + \sigma_c + \sigma_e \geqslant (1 - I)(\sigma_g + \sigma_e) - \frac{(1 + \sqrt{I})^2(\sigma_g + \sigma_e)(\sigma_a + \sigma_c)}{AU_e} \tag{19}$$

When A approaches infinite, and the cycle is internally reversible, i.e., $I = 1$, the inequality (19) may be expressed as

$$\sigma_g + \sigma_a + \sigma_c + \sigma_e \geqslant 0 \tag{20}$$

Inequality (19) reduces to the familiar form of the second law as the process approaches the limit of infinite total heat-transfer area and internal reversibleness. The inequality (19) defines the set of physically feasible outcomes of a four-temperature-level absorption heat-pump cycle of the type described.

Using Eqs. (1)–(4), (6)–(8) and (13)–(16), we can prove that when the temperature of the working substance in the generator, absorber, condenser and evaporator are given by

$$T_1 = T_g\sqrt{IU_g}/\big(\theta_1 + \sqrt{IU_g}\big) \tag{21}$$

$$T_2 = -T_a\sqrt{U_a}/\big(\theta_1 - \sqrt{U_a}\big) \tag{22}$$

$$T_3 = -T_c\sqrt{U_c}/\big(\theta_1 - \sqrt{U_c}\big) \tag{23}$$

$$T_4 = T_e\sqrt{IU_e}/\big(\theta_1 + \sqrt{IU_e}\big) \tag{24}$$

respectively, where

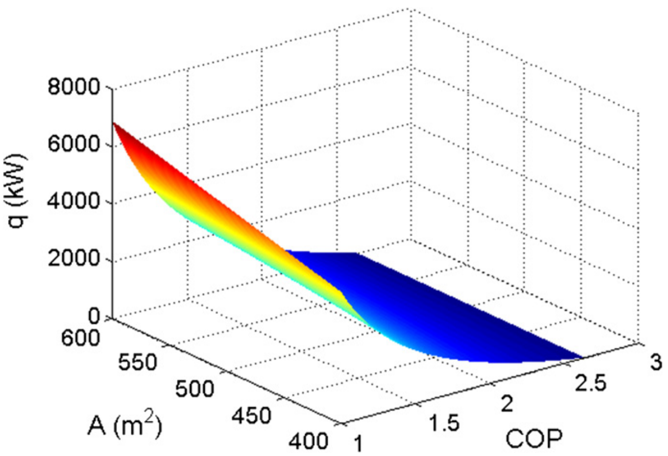

Fig. 2. The optimal heating load q versus the total heat-transfer area A and the coefficient of performance COP ($T_g = 413$ K, $T_e = 283$ K, $T_a = 313$ K, $T_c = 333$ K, $n = 1.3$, and $I = 1$ by assuming $U_g = U_a = U_c = U_e = 0.5$ kW/(m^2 K)).

$$\theta_1 = \left[\frac{1}{(1+n)}\left(\frac{n}{T_a}+\frac{1}{T_c}\right) - \frac{I(COP-1)}{COP\,T_e} - \frac{I}{COP\,T_g}\right] \times \left[\frac{1}{(1+n)} \times \left(\frac{n}{T_a\sqrt{U_a}}+\frac{1}{T_c\sqrt{U_c}}\right) + \frac{\sqrt{I}(COP-1)}{COP\,T_e\sqrt{U_e}} + \frac{\sqrt{I}}{COP\,T_g\sqrt{U_g}}\right]^{-1}$$

The total heat-transfer area attains minimum. Substituting Eqs. (21)–(24) into (9), we can obtain a generally fundamental optimum relation of the absorption heat pump

$$q = A\theta_1\left[\frac{n(\sqrt{U_a}-\theta_1)}{(1+n)T_aU_a} + \frac{(\sqrt{U_c}-\theta_1)}{(1+n)T_cU_c} + \frac{(COP-1)(\sqrt{IU_e}+\theta_1)}{COP\,T_eU_e} + \frac{(\sqrt{IU_g}+\theta_1)}{COP\,T_gU_g}\right]^{-1} \tag{25}$$

Since $(\partial A/\partial T_i)_{q,T_j\neq T_i} = -\frac{(\partial q/\partial T_i)_{A,T_j\neq T_i}}{(\partial q/\partial A)_{T_1,T_2,T_3,T_4}}$ $(i, j = 1,2,3,4)$, The condition $(\partial A/\partial T_i)_{q,T_j\neq T_i} = 0$ corresponds to $(\partial q/\partial T_i)_{A,T_j\neq T_i} = 0$ under the circumstances $(\partial q/\partial A)_{T_1,T_2,T_3,T_4} \neq 0$. Therefore, Eq. (25) is a concise statement of the three analytical formulations that not only determines the minimum total heat-transfer area for given values of q and COP, but also ascertains the maximum heating load for given values of A and COP, and makes sure optimum coefficient of performance for given values of A and q. Eq. (25) can be used to discuss the irreversible absorption heat pump main performance characteristics. From Eq. (25), the q–A–COP characteristics curved surface can be generated as shown in Fig. 2.

It is seen from Fig. 2 that the heating load is a monotonically increasing function of the total heat-transfer area for a given value of the coefficient of performance, and that the heating load is a monotonically decreasing function of the coefficient of performance for a given value of total heat-transfer area. Both q and COP are contradictory to each other and consideration must be given to both simultaneously. Thus the ecological optimization criterion is introduced to investigate the best mode of operation of the irreversible absorption heat pump further.

4. Ecological optimization criterion

The entropy-production rate σ during the whole cycle mentioned above is

$$\sigma = q_a/T_a + q_c/T_c - q_g/T_g - q_e/T_e = q(1/T_e - 1/T_g)\left(COP^{-1} - COP_r^{-1}\right) \tag{26}$$

where $COP_r = \frac{T_aT_c(1+n)(T_g-T_e)}{T_g\{T_a[T_c(1+n)-T_e]-nT_eT_c\}}$ is the coefficient of performance for a reversible absorption heat pump. According to the definition of the general ecological criterion function [20, 23–27], the ecological criterion function E of an absorption heat pump may be written as

$$E = q - \mu T_e\sigma = q - \frac{\bar{T}_cT_e\sigma}{\bar{T}_c - T_e} = q\left(2 - COP^{-1}COP_r\right) \tag{27}$$

where $\bar{T}_c = (1+n)(1/T_c + n/T_a)^{-1}$ is the entropic mean temperature of the total heat rejection [28], and μ is the dissipation coefficient of the heating load, whose the physical meaning is that, in theory, if the rate of availability $T_e\sigma$ were not lost, it would produce a heating rate $\mu T_e\sigma$ through a reversible Carnot heat pump operating between T_e and $\bar{T}_c$. This shows that μ is equal to the coefficient of performance of the reversible Carnot heat pump, i.e. $\mu = \bar{T}_c/(\bar{T}_c - T_e)$. Thus, $\mu T_e\sigma$ may be called the loss of heating load, and using the ecological optimization criterion, one can attain the best compromise between the heating load and the loss of heating load. Moreover, we want to mention that we do not discuss part-load but different design possibilities [1].

Respectively, substituting Eq. (25) into Eqs. (27) and (26) yields the optimal relation among the ecological criterion, the coefficient of performance and total heat-transfer area

$$E = A\theta_1\left(2 - COP^{-1}COP_r\right)\theta_2, \tag{28}$$

and yields the optimal relation among the entropy-production rate of the cycle, the coefficient of performance and total heat-transfer area

$$\sigma = A\theta_1(1/T_e - 1/T_g)\left(COP^{-1} - COP_r^{-1}\right)\theta_2 \tag{29}$$

where

$$\theta_2 = \left\{\frac{1}{(1+n)}\left[\frac{n(\sqrt{U_a}-\theta_1)}{T_aU_a} + \frac{(\sqrt{U_c}-\theta_1)}{T_cU_c}\right] + \frac{(COP-1)(\sqrt{IU_e}+\theta_1)}{COP\,T_eU_e} + \frac{(\sqrt{IU_g}+\theta_1)}{COP\,T_gU_g}\right\}^{-1}$$

There are two zero points in Eq. (28). One point is at

$$COP_0 = COP_r/2 \tag{30}$$

and another point is at

$$COP_I = \frac{IT_aT_c(1+n)(T_g-T_e)}{T_g\{T_a[IT_c(1+n)-T_e]-nT_eT_c\}} \tag{31}$$

It is worthy of note that COP_0 must be less than COP_I, otherwise E cannot be positive and the best compromise between the heating rate and its loss cannot be attained. From this, one can deduce that I must be less than $\frac{COP_r(1/T_c+n/T_a)}{(n+1)[(COP_r-2)/T_e+2/T_g]}$.

According to Eq. (28) and the extremal condition $\mathrm{d}E/\mathrm{d}COP = 0$, one can obtain the coefficient of performance COP_E to satisfy the maximum ecological criterion

$$COP_E = \left[2d_1 + (2\,COP_I + COP_r)\,d_2\right]^{-1}\Big\{COP_I\,COP_r\,d_2 - 2d_0 - \sqrt{\left(4d_0 + 2\,COP_r\,d_1 + COP_r^2\,d_2\right)} \times \sqrt{\left[d_0 + COP_I(d_1 + COP_I\,d_2)\right]}\Big\} \tag{32}$$

where

$$d_0 = -\left(1 - \sqrt{U_e/U_g}\right)^2$$

$$d_1 = \left(1 - \sqrt{U_e/U_g}\right)^2 + n(1+n)^{-1}\Big\{T_g\Big[\left(1 + \sqrt{IU_e/U_a}\right)^2/(IT_a) + \left(1 + \sqrt{IU_e/U_c}\right)^2/(nIT_c)\Big] - T_e\Big[\left(\sqrt{U_e/U_g} + \sqrt{IU_e/U_a}\right)^2/(IT_a) + \left(\sqrt{U_e/U_g} + \sqrt{IU_e/U_c}\right)^2/(nIT_c)\Big]\Big\}$$

$$d_2 = nT_g\Big\{\left(\sqrt{U_e/U_a} - \sqrt{U_e/U_c}\right)^2 T_e/\left[(1+n)IT_aT_c\right] - \Big[\left(1 + \sqrt{IU_e/U_a}\right)^2/(IT_a) + \left(1 + \sqrt{IU_e/U_c}\right)^2/(nIT_c)\Big]\Big\}(1+n)^{-1}$$

Substituting Eq. (32) into Eqs. (28), (25) and (29) yields the maximum ecological criterion $E_{\max}$, the corresponding heating load q_E and the entropy production rate σ_E as the maximum ecological criterion, respectively.

5. Results and discussion

From Eqs. (25), (28) and (29), at a given total heat-transfer area, the q–COP, σ–COP, and E–COP characteristic curves can be generated, as shown in Fig. 3. The comparison between the maximum ecological function point and the point where the coefficient of performance $COP_0 = COP_r/2$ shows that the ecological optimization makes the entropy generation rate decrease 76.8%, the coefficient of performance increase 33.6% and the heating load decrease 53.7%. These numerical results show that the ecological optimization criterion can make the best compromise between the heating rate and its loss and has a long-range goal in the sense that it is compatible with ecological objectives.

Fig. 3 illustrates that both the heating load q and entropy-production rate σ decrease as the coefficient of performance COP increase, and both the heating load q and entropy-production rate σ equal zero when the coefficient of performance approaches COP_I.

The effect of the internal irreversibility factor on the optimal performance is shown in Fig. 3. It is found that the maximum ecological function, the corresponding heating load and the entropy-production rate of the cycle decrease as the internal irreversibility factor increases. One also can see from Fig. 3 that the ecological optimal working region is located in $0 < E \leqslant E_{\max}$ and $COP_E \leqslant COP < COP_I$ that is less than that for the heating load criterion $0 < q < q_{COP=1}$ and $1 < COP < COP_I$.

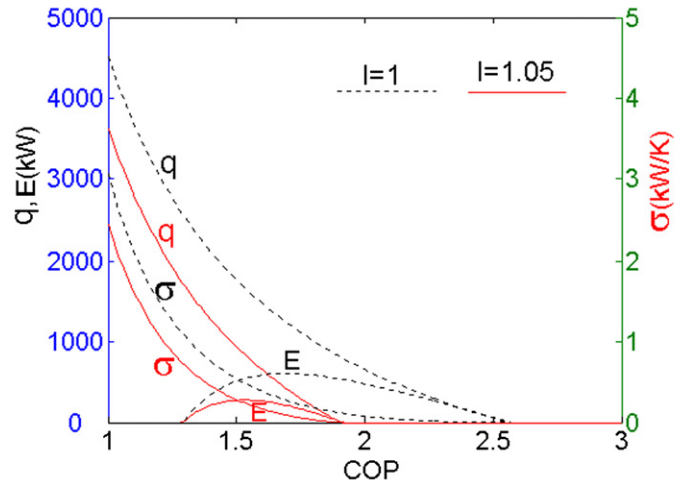

Fig. 3. The heating load q, the ecological criterion E and the entropy production rate σ versus the coefficient of performance COP for a given total heat-transfer area A at different internal irreversibility factor I ($T_g = 413$ K, $T_e = 283$ K, $T_a = 313$ K, $T_c = 333$ K, $n = 1.3$, and $A = 400\ \mathrm{m}^2$ by assuming $U_g = U_a = U_c = U_e = 0.5\ \mathrm{kW/(m^2\,K)}$).

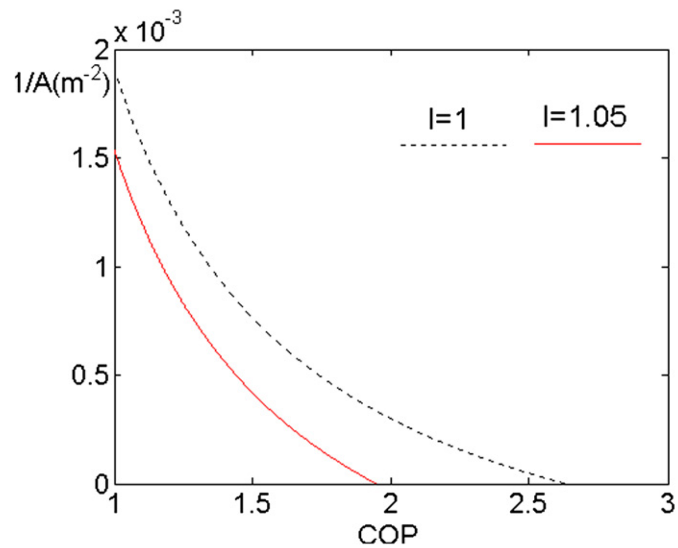

Fig. 4. The inverse of minimum total heat-transfer area A^{-1} versus the coefficient of performance COP for a given heating load q at different internal irreversibility factor I ($T_g = 413$ K, $T_e = 283$ K, $T_a = 313$ K, $T_c = 333$ K, $n = 1.3$, and $q = 6000$ kW by assuming $U_g = U_a = U_c = U_e = 0.5\ \mathrm{kW/(m^2\,K)}$).

Fig. 4 displays the inverse of minimum total heat-transfer area A^{-1} as a function of the coefficient of performance COP for a given heating load. It is noticeable that for a higher irreversibility factor, a larger the coefficient of performance will lead to a larger minimum total heat-transfer area. When the coefficient of performance approaches COP_I, the minimum total heat-transfer area has to be infinitely large in order to diminish the driving temperature differences [1]. Obviously, it is not significant to operate an absorption heat pump with a zero specific heating load ($q/A = 0$). This implies the fact that the coefficient of performance of a four-temperature-level is always smaller

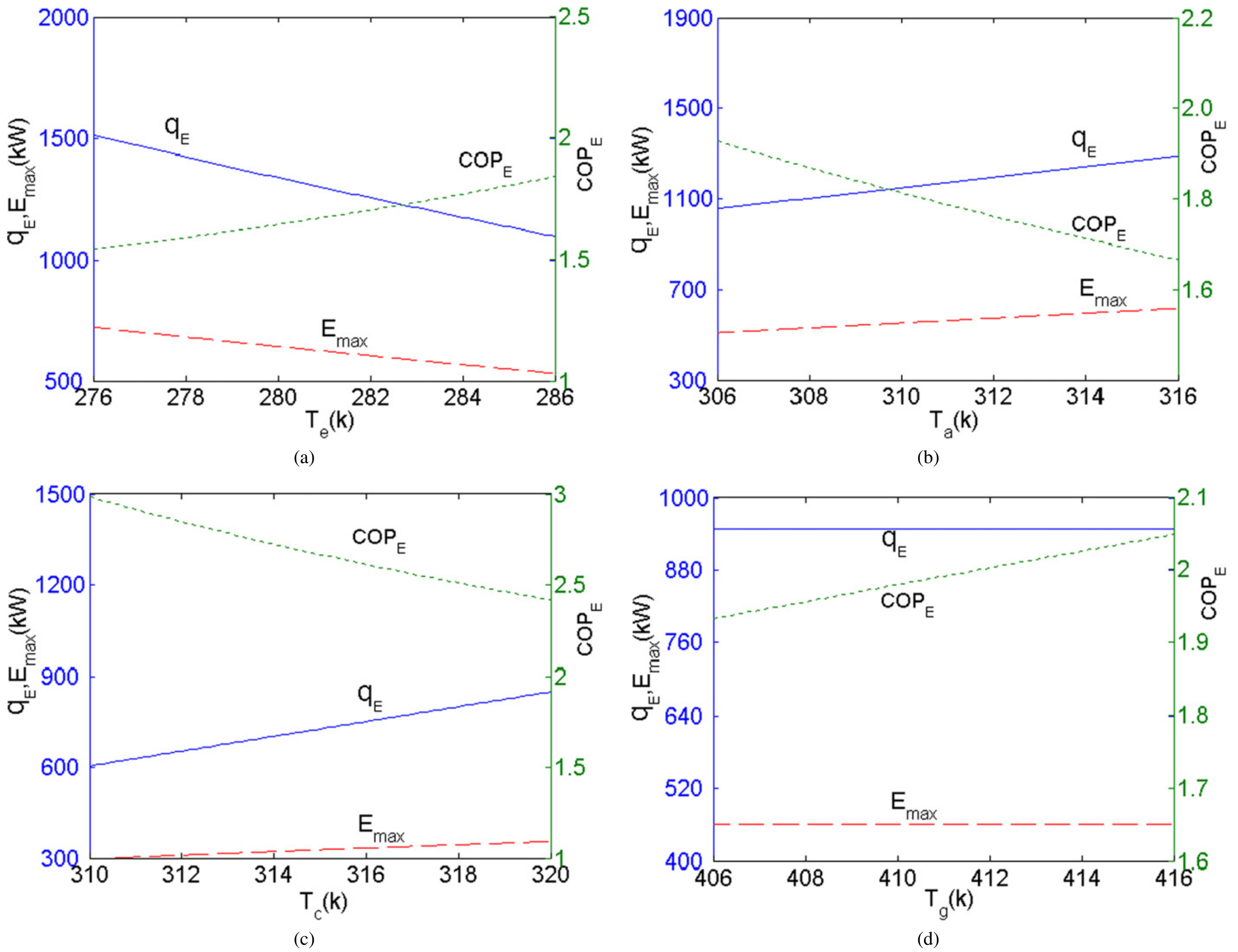

Fig. 5. The maximum ecological criterion $E_{\max}$, the corresponding heating load q_E and the entropy production rate σ_E versus the temperature of the heat reservoir (a) T_e; (b) T_a; (c) T_c and (d) T_g. ($T_g = 413$ K, $T_e = 283$ K, $T_a = 313$ K, $T_c = 333$ K, $n = 1.3$, and $A = 400\ \text{m}^2$ by assuming $U_g = U_a = U_c = U_e = 0.5\ \text{kW}/(\text{m}^2\,\text{K})$).

than COP_I because there always exist internal irreversibilities of the working substance.

Fig. 5(a) shows the maximum ecological function, the corresponding optimal coefficient of performance and the heating load as a function of environmental reservoir temperature T_e. It is seen that $E_{\max}$ and q_E decrease as environmental reservoir temperature increases, while COP_E increases as environmental reservoir temperature increases.

Figs. 5(b) and (c) respectively show the maximum ecological function, the corresponding optimal coefficient of performance and the heating load plotted against heated-space temperatures at T_a and T_c. It is observed $E_{\max}$ and COP_E all increase as heated-space temperature increases, while COP_E decreases as heated-space temperature increases.

Fig. 5(d) illustrates the maximum ecological function, the corresponding optimal coefficient of performance and the heating load plotted against heat source temperature T_g. It is observed COP_E increases as heat source temperature increases, while $E_{\max}$ and q_E all remain constant as heat source temperature varies. $E_{\max}$ and q_E are all independent of heat source temperature T_g, When the overall heat-transfer coefficients of the heat exchangers are the same. Under such circumstances, the coefficient of performance COP_E, the maximum ecological criterion $E_{\max}$ and the corresponding q_E may be simplified as

$$COP_E = \frac{I\bar{T}_c(T_g - T_e)}{T_g(I\bar{T}_c - \sqrt{IT_e(2T_e - \bar{T}_c)})} \tag{33}$$

$$E_{\max} = \frac{AU_e}{(1+\sqrt{I})^2}\frac{T_c(\sqrt{T_e} - \sqrt{I(2T_e - \bar{T}_c)})^2}{\bar{T}_c - T_e} \tag{34}$$

$$q_E = \frac{AI\bar{T}_c U_e}{(1+\sqrt{I})^2}\left(\frac{T_e}{\sqrt{IT_e(2T_e - \bar{T}_c)}} - 1\right) \tag{35}$$

6. Conclusion

The optimal performance of a four-temperature-level absorption heat pump affected by the irreversibility of finite-rate heat-transfer and the internal irreversibility of the working substance is analyzed based on a more reservoir-oriented viewpoint. An inequality relating the total heat-transfer area, the net entropy changes of the reservoirs and an irreversibility factor is derived that delimits feasible net effects of cyclic processes. The generally optimal relation among the heating load, the coefficient of performance and total heat-transfer area is achieved. The optimal performances for ecological function with respect to reservoir temperature and the internal irreversibility factor are investigated. The optimization of the ecological function makes the entropy production rate of cycle decrease greatly and the coefficient of performance increase with the cost of an appropriate decrease in the heating load. It is expected that these results may lay a foundation for the deeper investigation of real four-temperature-level absorption heat pumps.

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