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A generalized model of a combined refrigeration cycle and its performance

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Abstract — The building of a steady flow combined refrigeration cycle model with heat resistance, heat leakage and internal irreversibility is described in this paper. The optimal performance of the model is studied. The relation between optimal cooling load and coefficient of performance (COP), as well as the maximum COP and the corresponding cooling load are derived. © Elsevier, Paris.

combined refrigeration cycle / performance / optimization / cooling load / coefficient of performance

Résumé — **Modélisation des performances en temps finis des cycles de machines frigorifiques étagées.** On présente un modèle de détermination des performances de cycles étagés de machines frigorifiques fonctionnant en régime permanent, prenant en compte les imperfections de la machine : résistances thermiques aux échangeurs, fuites thermiques et irréversibilités internes. Le modèle permet de déterminer les tailles relatives des échangeurs pour obtenir les performances optimales à compte tenu de certaines contraintes. Les expressions donnant le COP en fonction de la chaleur soutirée à la source froide et leurs optima sont données. L'analyse, partant d'un cycle à deux étages, est étendue au cas à *n* étages. © Elsevier, Paris.

réfrigération étagée / performance / optimisation / charge frigorifique / coefficient de performance

Nomenclature

A	function defined in equation (42)	
B	function defined in equation (31)	
B_1	function defined in equation (43)	
B_2	function defined in equation (48)	
B_3	function defined in equation (49)	
C_i	internal heat conductance	$W \cdot K^{-1}$
F_T	total heat transfer surface area	m^2
F_1	heat transfer surface area of high-temperature side heat exchanger	m^2
F_2	heat transfer surface area of intermediate-temperature heat exchanger	m^2
F_3	heat transfer surface area of low-temperature side heat exchanger	m^2
K_i	overall heat transfer coefficient of ith stage heat exchanger	$W \cdot m^{-2} \cdot K^{-1}$
K_1	overall heat transfer coefficient of high-temperature heat exchanger	$W \cdot m^{-2} \cdot K^{-1}$
K_2	overall heat transfer coefficient of intermediate-temperature heat exchanger	$W \cdot m^{-2} \cdot K^{-1}$
K_3	overall heat transfer coefficient of low-temperature side heat exchanger	$W \cdot m^{-2} \cdot K^{-1}$
M	function defined in equation (27)	
M_{min}	minimum value of M	
n	counting number defined in equation (43)	
P	total power input	W
P_1	power input to topping cycle	W
P_2	power input to bottom cycle	W
Q_H	rate of heat transfer to heat sink	W
q	rate of heat leak	W
Q_L	rate of heat removed from heat source (cooling load)	W

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Q_1	rate of heat transfer from topping cycle to heat sink in the high-temperature side heat exchanger...	W
Q'_1	rate of heat transfer from top endoreversible cycle to heat sink in the high-temperature side heat exchanger........................	W
Q_2	rate of heat transfer from bottom cycle to top cycle.................	W
Q'_2	rate of heat transfer from bottom endoreversible cycle to top cycle...	W
Q_3	rate of heat transfer from heat source to bottom cycle in the low-temperature side heat exchanger...	W
R	cooling load......................	W
R_m	cooling load at the maximum COP point...........................	W
R_{max}	maximum cooling load............	W
T_H	temperature of heat sink..........	K
T_L	temperature of heat source........	K
T_1	temperature of working fluid in the warm-side of topping cycle........	K
T_2	temperature of working fluid in the cold-side of topping cycle.........	K
T_3	temperature of working fluid in the warm-side of bottom cycle........	K
T_4	temperature of working fluid in the cold-side of bottom cycle..........	K
U_T	total heat exchanger inventory.....	$W{\cdot}K^{-1}$
U_1	heat exchanger inventory of high-temperature side heat exchanger...	$W{\cdot}K^{-1}$
U_2	heat exchanger inventory of intermediate-temperature heat exchanger.........................	$W{\cdot}K^{-1}$
U_3	heat exchanger inventory of low-temperature side heat exchanger...	$W{\cdot}K^{-1}$
x	heat transfer surface area ratio defined in equation (25)	
x_{opt}	optimum value of x	
y	heat transfer surface area ratio defined in equation (25)	
y_{opt}	optimum value of y	
z	heat transfer surface area ratio defined in equation (25)	
z_{opt}	optimum value of z	

Greek symbols

β	coefficient of performance (COP)
β_{max}	maximum COP
Φ_i	degree of internal dissipation of ith stage refrigeration cycle
Φ_T	total degree of internal dissipation of combined cycle
Φ_1	degree of internal dissipation of topping cycle
Φ_2	degree of internal dissipation of bottom cycle

1. INTRODUCTION

According to classical thermodynamics, the performance of a combined refrigeration cycle is identical to that of a single-cycle operating in the same temperature range. For example, when a reversible combined cycle refrigerator operates between hot and cold reservoirs at temperatures T_H and T_L, respectively, its COP is equal to the COP of a reversible Carnot refrigerator operating in the same temperature range. However, the use of reversible processes as standards of performance in industry is not desirable because a reversible process must be carried out at an infinitesimally slow pace. The high performance criterion of an ideal reversible refrigerator is seldom approached.

The consequence of incorporating finite-time processes into an otherwise ideal thermodynamic cycle was elegantly demonstrated by Novikov (1957) [1], Chambadal (1957) [2] and Curzon and Ahlborn (1975) [3]. They considered the case of finite rates of heat transfer to and from a Carnot heat engine. After maximizing the power output, they derived a simple expression for the efficiency that was different from the well-known Carnot efficiency. Since then finite-time thermodynamics or entropy generation minimization has advanced, and many authors have studied the effect of irreversibilities on the performance of thermodynamic processes and cycles. Some detailed literature surreys were given in several books by Andresen [4], Tsirlin [5], Feidt [6], Sieniutycz and Salamon [7], De Vos [8], Radcenco [9], Bejan [10], and Bejan, Tsatsaronis and Moran [11] and some review articles by Andresen et al. [12–14], Orlov et al. [15], Wu [16, 17], Chen et al. [18, 19], Sieniutycz and Shiner [20] and Bejan [21–24]. Several researchers including Chen and Yan [25], Chen et al. [26–28], Chen and Wu [29, 30] and Goktun [31] have investigated effects of the finite rate of heat transfer, together with other major irreversibilities on the performance of combined refrigeration cycles using endoreversible (heat resistance) model [25, 26, 27, 29, 30], heat resistance and heat leak model [28] and heat resistance and internal irreversibility model [31]. There exist different characteristics for various irreversibilities effects on the performance of single-cycle refrigerators [32-40] and combined-cycle refrigerators [28, 31]. More recently, Chen et al. [41,42] presented a generalized irreversible Carnot refrigerator model by taking additional account of several internal irreversibilities of the refrigerators. This was done by means of a constant parameter and a constant coefficient, together with loss of heat resistance. It is the purpose of this paper to present a more complete, combined refrigerator model, including both internal and external irreversibilities of the combined plant by extending Chen's work [41, 42] to combined-cycle refrigerators, and to analyze the optimal performance of a combined refrigerator system affected by the irreversibility of finite rate heat transfer, internal dissipation of the working fluid and the bypass heat leakage.

2. MODEL OF THE COMBINED REFRIGERATION PLANT

The real combined refrigeration cycle formed by two steady irreversible cycles and its surrounding heat reservoirs are shown in the *figure*. The combined cycle operates between two reservoirs at temperatures T_H and T_L ($T_H > T_L$). The temperatures of the working fluid in the top irreversible cycle are T_1 and T_2, when heat leaves and enters the working fluid. In the bottom irreversible cycle, they are T_3 and T_4 .The following temperature relation hold:

$$T_1 > T_H > T_3 > T_2 > T_L > T_4 \tag{1}$$

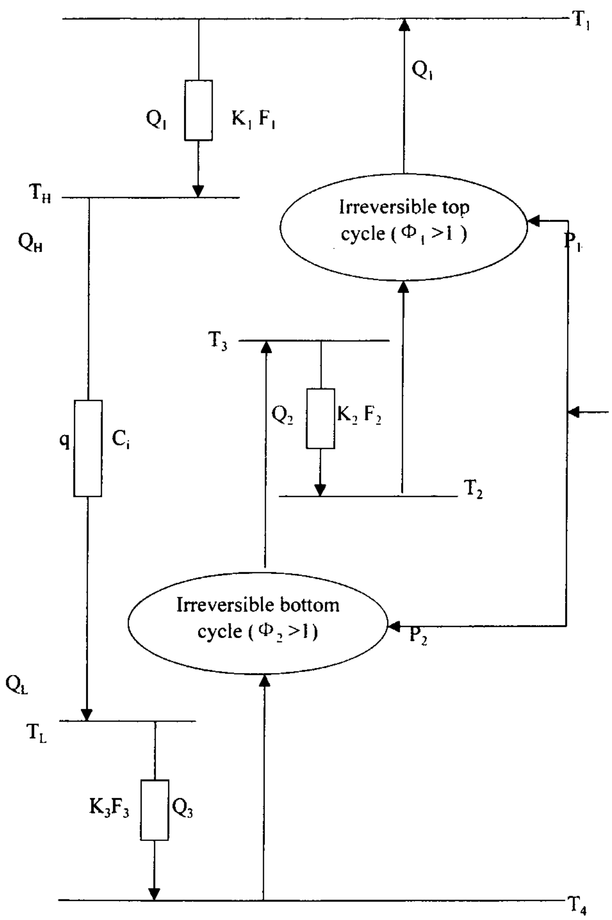

Figure 1. Generalized combined refrigeration cycle with all irreversibilities.

Heat exchange between the two irreversible cycles is carried out directly through the same heat transfer surface area so that the quantity of heat absorbed by the working fluid in the top cycle is equal to that rejected by the working fluid in the bottom cycle. The irreversibilities are the constant rate of bypass heat leak q which flows from the heat sink at temperature T_H to heat source at temperature T_L, the internal dissipation of the working fluids in the top and bottom cycles, and the three heat transfers caused by the three finite temperature differences $(T_1 - T_H)$, $(T_3 - T_2)$ and $(T_L - T_4)$ required by the heat rejection and absorption rates.

The first law of thermodynamics requires

$$Q_3 = Q_L + q = R + q \tag{2}$$

$$Q_1 = Q_H + q \tag{3}$$

where Q_1 and Q_3 are the rates of heat flow from the top cycle to the heat sink in the high-temperature side heat exchanger and from the heat source to the bottom cycle in the low-temperature side heat exchanger of the combined cycle respectively; Q_H and Q_L are the heat transfer rate to the heat sink and the heat removed rate from the heat source (i.e. cooling load, R for convenience), respectively.

The internal dissipation including miscellaneous factors such as friction, turbulence and non-equilibrium activities inside the two cycles are characterized by two constant coefficients [41, 42]:

$$\Phi_1 = \frac{Q_1}{Q_1'} \geq 1 \tag{4}$$

$$\Phi_2 = \frac{Q_2}{Q_2'} \geq 1 \tag{5}$$

where Q_2 is the rate of heat exchange between the top and bottom irreversible cycles in the intermediate heat exchanger, Q_2' is the rate of heat absorbed from the bottom endoreversible cycle to the top irreversible cycle in the intermediate heat exchanger, and Q_1' is the rate of heat rejected from the top endoreversible cycle to the heat sink in the high-temperature side heat exchanger. Because the power input required by the real refrigerator is larger than that of the endoreversible one with the same cooling load, the rate of heat rejected from the warm working fluid to the heat sink for the real cycle is larger than that for the endoreversible one. Thus, $Q_1 > Q_1'$ and $Q_2 > Q_2'$ hold. For the top endoreversible refrigerator, the second law of thermodynamics requires that (no matter whether the bottom refrigerator is endoreversible)

$$\frac{Q_2}{Q_1'} = \frac{T_2}{T_1} \tag{6}$$

For the bottom endoreversible refrigerator, the second law of thermodynamics requires that (no matter whether the top refrigerator is endoreversible)

$$\frac{Q_3}{Q_2'} = \frac{T_4}{T_3} \tag{7}$$

Combining equations (4), (5), (6) and (7) gives:

$$\frac{Q_2}{Q_1} = \frac{T_2}{\Phi_1 T_1} \tag{8}$$

$$\frac{Q_3}{Q_2} = \frac{T_4}{\Phi_2 T_3} \tag{9}$$

The power input to the combined refrigerator is:

$$P = P_1 + P_2 = Q_H - Q_L = Q_1 - Q_3 \tag{10}$$

where P_1, P_2 and P are the power input to the top and bottom cycles and the total power input to the combined refrigerator.

$$\beta = \frac{Q_L}{P} = \frac{R}{P} = \frac{1 - q/Q_3}{Q_1/Q_3 - 1} \tag{11}$$

Assuming that the rates of heat flow in the heat exchangers follow Newton's law, we have:

$$Q_1 = (T_1 - T_H)\, K_1\, F_1 \tag{12}$$

$$Q_2 = (T_3 - T_2)\, K_2\, F_2 \tag{13}$$

$$Q_3 = (T_L - T_4)\, K_3\, F_3 \tag{14}$$

where K_1 is the overall heat transfer coefficient and F_1 is the heat transfer surface area of the high-temperature (hot) side heat exchanger between the top cycle and the heat sink, K_2 is the overall heat transfer coefficient and F_2 is the heat transfer surface area of the intermediate heat exchanger between the top and bottom cycle, and K_3 is the overall heat transfer coefficient and F_3 is the heat transfer surface area of the low-temperature (cold) side heat exchanger between the bottom cycle and the heat source.

The total heat transfer surface area (F_T) of the combined cycle is assumed to be constant

$$F_T = F_1 + F_2 + F_3 \tag{15}$$

3. ANALYSIS AND OPTIMIZATION

From equation (14), we have

$$T_4 = T_L - \frac{Q_3}{K_3 F_3} \tag{16}$$

Substituting equations (13) and (16) into equation (9) yields:

$$T_3 = \frac{T_2\,[T_L - Q_3/(K_3 F_3)]}{T_L - Q_3\,[(K_3 F_3)^{-1} + (K_2 F_2/\Phi_2)^{-1}]} \tag{17}$$

Combining equations (12), (13) and (8) gives

$$T_1 = \frac{K_1 F_1 T_H T_2}{(K_1 F_1 + \Phi_1 K_2 F_2)\, T_2 - \Phi_1 K_2 F_2 T_3} \tag{18}$$

Substituting equation (17) into equation (18) yields:

$$T_1 = \frac{T_H\,\{T_L - Q_3\,[(K_3 F_3)^{-1} + (K_2 F_2/\Phi_2)^{-1}]\}}{T_L - Q_3\,[(K_3 F_3)^{-1} + (K_2 F_2/\Phi_2)^{-1} + (K_1 F_1/\Phi_T)^{-1}]} \tag{19}$$

where Φ_T is the total degree of internal dissipation in the combined system, only for convenience.

$$\Phi_T = \Phi_1 \Phi_2 \tag{20}$$

From equation (12), we have:

$$T_1 = T_H + \frac{Q}{K_1 F_1} \tag{21}$$

Combining equations (19) and (21) gives:

$$\frac{Q_3}{Q_1} = \frac{T_L - Q_3\,[(K_3 F_3)^{-1} + (K_2 F_2/\Phi_2)^{-1} + (K_1 F_1/\Phi_T)^{-1}]}{\Phi_T T_H} \tag{22}$$

Substituting equation (22) into equations (11) yields

$$\beta = \frac{R}{R+q} \cdot \left\{ \frac{\Phi_T T_H}{T_L - (R+q)\,[(K_3 F_3)^{-1} + (K_2 F_2/\Phi_2)^{-1} + (K_1 F_1/\Phi_T)^{-1}]} \right. \tag{23}$$

Equation (23) provides a general relation between the COP and cooling load of the combined plant. For given values of T_H, T_L, Φ_1, Φ_2, q, K_1, K_2, K_3, F_T and R, β is a function of the distribution among F_1, F_2 and F_3. Equation (23) indicates that the maximization of β is equivalent to the minimization of M, as defined by equation (24):

$$M = (K_3 F_3)^{-1} + (K_2 F_2/\Phi_2)^{-1} + (K_1 F_1/\Phi_T)^{-1} \tag{24}$$

We define three heat transfer surface area ratios x, y and z,

$$x = \frac{F_1}{F_T},\ y = \frac{F_2}{F_T},\ z = \frac{F_3}{T_T} \tag{25}$$

thus

$$F_1 = x F_T,\ F_2 = y F_T,\ F_3 = z F_T,\ z = 1 - x - y \tag{26}$$

and

$$M = \frac{\Phi_T}{x K_1 F_T} + \frac{\Phi_2}{y K_2 F_T} + \frac{1}{(1-x-y) K_3 F_T} \tag{27}$$

To find the minimum M from equation (27), taking the partial derivative of M with respect to x and y and setting them equal to zero ($\partial M/\partial x = 0$ and $\partial M/\partial y = 0$ gives the optimal heat transfer surface area distribution

(x_{opt}, y_{opt} and z_{opt}) and the minimum M (M_{min}) as follows:

$$x_{opt} = \left(\frac{F_1}{F_T}\right)_{opt} = \left(\frac{\Phi_T}{K_1}\right)^{\frac{1}{2}} \left[\frac{1}{(K_3)^{\frac{1}{2}}} + \left(\frac{\Phi_2}{K_2}\right)^{\frac{1}{2}} + \left(\frac{\Phi_T}{K_1}\right)^{\frac{1}{2}}\right]^{-1} \tag{28}$$

$$y_{opt} = \left(\frac{F_2}{F_T}\right)_{opt} = \left(\frac{\Phi_2}{K_2}\right)^{\frac{1}{2}} \left[\frac{1}{(K_3)^{\frac{1}{2}}} + \left(\frac{\Phi_2}{K_2}\right) + \left(\frac{\Phi_T}{K_1}\right)^{\frac{1}{2}}\right]^{-1} \tag{29}$$

$$z_{opt} = \left(\frac{F_3}{F_T}\right)_{opt} = \left(\frac{1}{K_3}\right)^{\frac{1}{2}} \left[\frac{1}{(K_3)^{\frac{1}{2}}} + \left(\frac{\Phi_2}{K_2}\right)^{\frac{1}{2}} + \left(\frac{\Phi_T}{K_1}\right)^{\frac{1}{2}}\right]^{-1} \tag{30}$$

and

$$M_{min} = \left[\left(\frac{1}{K_3}\right)^{\frac{1}{2}} + \left(\frac{\Phi_2}{K_2}\right)^{\frac{1}{2}} + \left(\frac{\Phi_T}{K_1}\right)^{\frac{1}{2}}\right]^2 \frac{1}{F_T} = B \tag{31}$$

Substituting equation (31) into equation (23) yields

$$\beta_{max} = \frac{(T_L - q\,B)\,R - B\,R^2}{B\,R^2 + (\Phi_T\,T_H - T_L + 2\,q\,B)\,R + (\Phi_T\,T_H - T_L + q\,B)\,q} \tag{32}$$

Equation (32) is the optimal performance characteristics of the generalized model of real combined refrigeration plants with irreversibilities of heat resistance, heat leakage and internal dissipation. It determines the optimal COP for the given R. Equation (32) also indicates that $\beta_{max} = 0$ when $R = 0$ and $R = R_{max}$, where

$$R_{max} = T_L/B - q \tag{33}$$

There exists a double maximum COP point. To find the double maximum COP from equation (32), taking the derivative of β_{max} with respect to R and setting it equal to zero ($\mathrm{d}\beta_{max}/\mathrm{d}R = 0$) gives the double maximum COP ($\beta_{max,max}$) and the corresponding cooling load (R_0) as follows:

$$\beta_{max,max} = \left\{\frac{(\Phi_T\,T_H\,T_L)^{\frac{1}{2}} - [(\Phi_T\,T_H - T_L + q\,B)\,q\,B]^{\frac{1}{2}}}{(q\,B\,T_L)^{\frac{1}{2}} + [\Phi_T\,T_H\,(\Phi_T\,T_H - T_L + q\,B)]^{\frac{1}{2}}}\right\}^2 \tag{34}$$

and

$$R_m = \frac{[\Phi_T\,T_H\,T_L\,(\Phi_T\,T_H - T_L + q\,B)\,q\,B]^{\frac{1}{2}} - (\Phi_T\,T_H - T_L + q\,B)\,q\,B}{B\,(\Phi_T\,T_H + q\,B)} \tag{35}$$

Combining equations (11) and (32) gives the optimal cooling load and the maximum COP for a given total power input (P), and the minimum total power input for a given cooling load in the following forms:

$$R = \frac{\left\{[\Phi_T\,T_H - T_L + B\,(2\,q + P)]^2\right.}{2B} \frac{\left.-4\,B\,[\Phi_T\,T_H\,q - (T_L - q\,B)\,(q + P)]\right\}^{\frac{1}{2}}}{2B} - \frac{[\Phi_T\,T_H - T_L + B\,(2\,q + P)]}{2B} \tag{36}$$

$$\beta = \frac{\left\{[\Phi_T\,T_H - T_L + B\,(2\,q + P)]^2\right.}{2\,B\,P} \frac{\left.-4\,B\,[\Phi_T\,T_H\,q - (T_L - q\,B)\,(q + P)]\right\}^{\frac{1}{2}}}{3\,B\,P} - \frac{[\Phi_T\,T_H - T_L + B\,(2\,q + P)]}{2\,B\,P} \tag{37}$$

and

$$P = \frac{B\,R^2 + (\Phi_T\,T_H - T_L + 2\,q\,B)\,R + (\Phi_T\,T_H - T_L + q\,B)\,q}{T_L - q\,B - B\,R} \tag{38}$$

When the combined system obtains the optimal COP for a given R, the temperatures of the working fluid in the isothermal processes must satisfy some optimal relations. Substituting equations (28)–(31) into equations (19), (16) and (17) yields:

$$T_1 = T_H\,\frac{T_L - (R + q)\,[(\Phi_2/K_2)^{\frac{1}{2}} + (1/K_3)^{\frac{1}{2}}]\,A}{T_L - (R + \varepsilon)\,B} \tag{39}$$

$$T_4 = T_L - (R + q)(1/K_3)^{\frac{1}{2}}\,A \tag{40}$$

$$\frac{T_3}{T_2} = \frac{T_L - (R + q)(1/K_3)^{\frac{1}{2}}\,A}{T_L - (R + q)\,[(\Phi_2/K_2)^{\frac{1}{2}} + (1/K_3)^{\frac{1}{2}}]\,A} \tag{41}$$

where

$$A = \frac{1}{F_T}\left[\left(\frac{\Phi_T}{K_1}\right)^{\frac{1}{2}} + \left(\frac{\Phi_2}{K_2}\right)^{\frac{1}{2}} + \left(\frac{1}{K_3}\right)^{\frac{1}{2}}\right] \tag{42}$$

The COP versus cooling load of the generalized irreversible combined refrigeration plant is a parabolic one. This qualitative behavior of COP versus cooling load characteristics appears to be common for real refrigerators [36–38, 43, 44].

4. DISCUSSION

For a combined refrigeration cycle, which is formed by several irreversible refrigeration cycles, with bypass heat leakage, the optimal performance characteristics have the same expressions as equations (32)–(42), simply replace B by B_1 from $i = 1$ to $i = n + 1$

$$B_1 = \frac{1}{F_\mathrm{T}} \left\{ \sum_{i=1}^{n+1} \left(\prod_{j=i}^{n+1} \Phi_j / K_i \right)^{\frac{1}{2}} \right\}^2 \tag{43}$$

where n is the number of stages of the combined cycle, K_i is the heat transfer coefficient for the ith stage heat exchanger between working fluids or between working fluid and the reservoirs, and Φ_j is the internal dissipation for the jth stage refrigeration cycle and $\Phi_{n+1} = 1$ is taken.

For the combined cycle, the optimization problem can be extended by distributing the total heat exchanger inventory U_T[32,38] among heat exchanger sizes: $U_1 = K_1 F_1$, $U_2 = K_2 F_2$, and $U_3 = K_3 F_3$.

The total heat exchanger inventory (U_T) of the combined cycle is assumed to be a constant

$$U_\mathrm{T} = U_1 + U_2 + U_3 \tag{44}$$

Using the above constraint equation, the optimal distribution of the heat exchanger inventory is obtained in the following expressions:

$$\left(\frac{U_1}{U_\mathrm{T}}\right)_\mathrm{opt} = \frac{(\Phi_\mathrm{T})^{\frac{1}{2}}}{1 + (\Phi_2)^{\frac{1}{2}} + (\Phi_\mathrm{T})^{\frac{1}{2}}} \tag{45}$$

$$\left(\frac{U_2}{U_\mathrm{T}}\right)_\mathrm{opt} = \frac{(\Phi_2)^{\frac{1}{2}}}{1 + (\Phi_2)^{\frac{1}{2}} + (\Phi_\mathrm{T})^{\frac{1}{2}}} \tag{46}$$

and

$$\left(\frac{U_3}{U_\mathrm{T}}\right)_\mathrm{opt} = \frac{1}{1 + (\Phi_2)^{\frac{1}{2}} + (\Phi_\mathrm{T})^{\frac{1}{2}}} \tag{47}$$

The optimal performance characteristics have the same expressions as equations (32)–(42); again, simply replace B by B_2

$$B_2 = \frac{1}{U_\mathrm{T}} \left[1 + (\Phi_2)^{\frac{1}{2}} + (\Phi_\mathrm{T})^{\frac{1}{2}} \right]^2 \tag{48}$$

For a combined cycle formed by n stages of irreversible refrigeration cycles, B should be replaced by B_3:

$$B_3 = \frac{1}{U_\mathrm{T}} \left\{ \sum_{i=1}^{n+1} \left[\prod_{j=i}^{n+1} \Phi_j \right]^{\frac{1}{2}} \right\}^2 \tag{49}$$

5. CONCLUSION

In the design of combined refrigeration cycle systems, knowledge of the upper bound of the COP is highly desirable. This paper presents a generalized irreversible combined refrigeration cycle model by using heat transfers, heat leakage and internal dissipation to simulate the energy exchange and irreversibilities. An analytical investigation of the performance of the model is undertaken. The COP versus cooling characteristics, and the maximum COP and the corresponding cooling load are derived. The optimal distributions of the heat transfer surface area or the heat transfer inventory for the heat exchangers of the combined refrigeration plant are also obtained. It provides new theoretical limits for designing a real combined refrigeration plant and for performance comparison between existing combined plants.

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