ANALYSIS OF HEAT TRANSFER AND IRREVERSIBILITY OF ORC EVAPORATOR FOR SELECTING WORKING FLUID AND OPERATING CONDITIONS

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Abstract - Organic Rankine Cycle (ORC) is suitable. to converting the normally hard to utilize low-temperature thermal energies, such as geothermal energy, solar energy, and industrial waste heat, to electricity through utilizing low boiling organic working fluids. The performance of ORC system is dramatically affected by the selections of working fluid and working conditions. As a key component of waste heat recovery, the irreversible loss of evaporator also has great influence on the performance of ORC system. In this paper, we study the heat transfer performance in evaporator under the condition that the heat source parameters and pinch point temperature are identified. It is found that the heat transfer performance is affected by C_r, the ratio of heat capacity flow rates between the working fluids and the heat source fluid. We use the equivalent thermal resistance, deducing from the concept of entransy, to measure the irreversibility during the heat transfer process. Then, we define a parameter κr, the ratio between latent heat and sensible heat of working fluid, With the parameters C_r and κ_r, we investigate the relationship between the heat transfer and irreversible loss, and deduce the condition that maximum heat transfer and minimum equivalent thermal resistance occurs. Finally, a calculation method is established to choose the optimum working fluid and the evaporation condition..

Key words*: evaporator; heat recovery; entransy dissipation; equivalent heat resistance; simulation*

1. Introduction

Organic Rankine Cycle (ORC) use the organic fluid, which has low boiling point, as working fluid to produce high-grade energy (electricity) from the low or medium grade thermal energy (solar energy, geothermal energy, biomass, and industrial waste heat etc.) [1]. The working fluid and operating condition have important influence on ORC performance, which include heat recovery efficiency and irreversible loss [2].

Hung [3] divided organic fluids into dry, wet and isentropic fluid. He pointed out that wet fluid is unsuitable for ORC. Drescher et al. [4] determined the best working fluid from 700 organic fluids by simulating the performance of ORC system, which is applied to biomass energy and thermal plant. Desai et al. [5] found thermal efficiency of ORC is proportional to evaporation temperature, when condensation temperature is fixed. However, because the amount of heat recovery is limited by the pinch point temperature difference of evaporator, higher evaporation temperature may result in lower output power, even with higher thermal efficiency of system. Therefore, the amount of heat recovery in the system depends on the heat transfer between the working fluid and the heat source. Li et al. [6] did a research to find out the influence of pinch point temperature difference and evaporating temperature on the performance of ORC cycle. Yu et al. [7] classified heat transfer process in evaporator into PPP and VPP process according to the position of the pinch point. They concluded that maximum heat recovery occurs at PPP region, when the heat source condition is determined. With the target of maximum heat recovery, they proposed a method of optimizing working fluid and working condition simultaneously basing on the analysis of the pinch point position. The above mentioned literatures only account for the problem of the heat transfer capacity between the waste heat carrier and the working fluid, but do not clarify the influence of irreversible loss in the heat transfer process.

As the heat recovery equipment, evaporator has the maximum irreversible loss and the highest price in ORC system [8]. Hung et al. [9] explained that ORC with a lower irreversibility has a better performance. Larjola [10] further pointed out when the working fluid "matches" the heat source fluid better, it will reduce the irreversibility between the heat source and the refrigerant during the heat transfer process, which results in higher power output. Yan et al. [11] divided the irreversible loss of heat transfer between the heat source fluid and the working fluid into effective and ineffective irreversible loss. They explained that the ineffective irreversible loss can be reduced by increasing the "match" of the heat source fluid and the working fluid. Reddy et al. [12] deduced the dimensionless entropy production for the waste heat recovery process in evaporator. Walraven et al. [13] compared the exergy efficiency of the ORC using different type of evaporators and then pointed out the plate evaporator is more suitable for ORC than the tube and shell evaporator. The above studies used entropy production or the exergy efficiency as an index to evaluate the irreversibility of the evaporator.

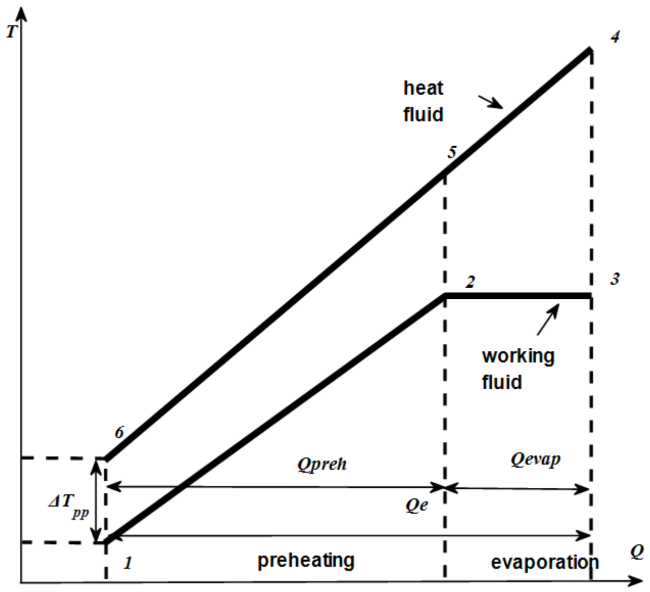
The researches of ORC focus on two main objects: one is to recover as much as possible thermal energy from heat source; the other is to reduce the irreversible loss during heat exchanging. In this paper, we choose dry organic fluids as candidate working fluids, and presume that the working fluid is in a saturated vapour state at the exit of evaporator. We firstly investigate the parameters which affect the heat transfer performance of evaporator. Then we use the equivalent thermal resistance, deducing from the concept of entransy, to measure the irreversibility during the heat transfer process. Finally, we establish a calculation method to iteratively search the optimum working fluid and evaporating temperature of evaporator by comparing the amount of heat transfer and equivalent thermal resistance.

2. Thermodynamic Model

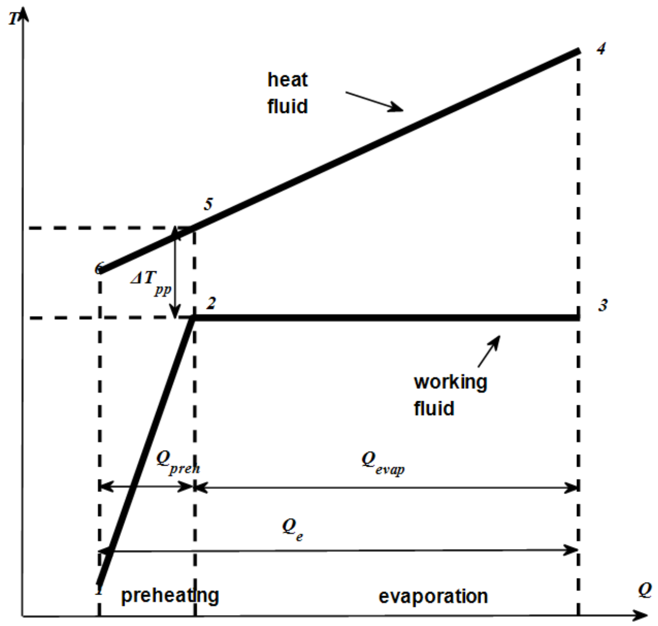
The heat transfer process in evaporator can be divided into two successive processes: a preheating process that the working fluid is heated up to a saturated liquid state (process 1-2 in Fig.1

(a)); and an evaporation process that the state of working fluid is changed from saturated liquid to saturated vapour while its temperature remains unchanged (process 2-3 in Fig.1 (b)). Pinch point is the position that minimum temperature difference occurs between two fluids in evaporator. According to the heat exchanging characteristics, the pinch point may locate at either the starting point of preheating or the starting point of vaporization [7]. Fig. 1 shows the different locations of pinch point at T-H diagram. Figure 1 (a) shows the preheating pinch point (PPP), and Figure 1 (b) shows the vaporizing pinch point(VPP).

As Larjola [10] mentioned, due to the difference of heat capacity flow rate, the amount of heat recovered and the outlet temperature of heat fluid are rather different for these two cases.



（a）Preheating Pinch Point(PPP)



（b）Vaporization Pinch Point(VPP)

Fig.1 T-H diagram of different position of pinch point

Here, heat capacity flow rate is the amount of heat exchanged for each 1K temperature rise. The differential form can be expressed as:

$$CP = \frac{dH}{dT} \tag{1}$$

If a fluid has a constant heat capacity flow rate, it means the amount of heat exchanged is proportional to the temperature difference. Apparently, the heat capacity flow rate is the reciprocal of the slope for a heat-transfer fluid in T-H diagram. So the amount of heat transfer can be calculated as:

$$\Delta H = m \cdot \Delta h = CP \cdot \Delta T \tag{2}$$

2.1. Heat Balance Analysis

In order to simplify the analyzation, some assumptions are adopted in this paper:

（1）Heat capacity flow rate of heat fluid CP_h is determined;

（2）Heat source fluid inlet temperature $T_{h,in}$ and working fluid inlet temperature $T_{w,in}$ are determined;

（3）Pinch point temperature difference ΔT_{pp} is determined;

（4）Working fluid is saturated vapour at exist of evaporator, in other words, vaporizing temperature $T_{w,evap}$ equals to the outlet temperature of working fluid $T_{w,out}$.

$$T_{w,out} = T_{w,evap} \tag{3}$$

Based on the assumptions above, the heat exchanged between the heat source fluid and the working fluid can be expressed as:

$$Q_e = CP_h \left(T_{h,in} - T_{h,out}\right) = m_w(S + L) \tag{4}$$

Pinch point is determined by the relative position of working fluid line and heat source line in the T-H diagram. We define C_r as the ratio of heat capacity flow rate between working fluid and the heat fluid at preheating process:

$$C_r = \frac{CP_w}{CP_h} = \frac{T_{h,in} - T_{h,out}}{T_{w,evap} - T_{w,in}} \tag{5}$$

According to Eq. (2), (4), heat exchanging amount of preheating, vaporizing at evaporator are given by Eq. (6) to Eq. (8), respectively,

$$Q_{preh} = CP_w \left(T_{w,evap} - T_{w,in}\right) = m_w S = CP_h\left(T_{h,evap} - T_{h,out}\right) \tag{6}$$

$$Q_{evap} = m_w L = CP_h\left(T_{h,in} - T_{h,evap}\right) \tag{7}$$

$$Q_e = \Delta Q_{preh} + \Delta Q_{evap} = CP_w\left(T_{w,evap} - T_{w,in}\right) + m_w L = CP_h\left(T_{h,in} - T_{h,out}\right) \tag{8}$$

where S and L are the sensible heat and the latent heat of working fluid, respectively.

Eq. (8) points out that heat exchanging amount of evaporator Q_e increases with the decreasing of outlet temperature of the heat source fluid $T_{h,out}$ when CP_h and $T_{h,in}$ are determined. In other words, a minimum $T_{h,out}$ means a maximum heat exchanging occurs.

We define the temperature difference between heat source and working fluid at the starting point of preheating as ΔT', and define the temperature difference at the starting point of evaporation as ΔT'', respectively:

$$\Delta T' = T_{h,out} - T_{w,in} \tag{9}$$

$$\Delta T'' = T_{h,evap} - T_{w,evap} \tag{10}$$

When pinch point is PPP, as showed in Fig.1 (a), the exit temperature of heat source fluid can be calculated from Eq. (9).

$$T_{h,out} = T_{w,in} + \Delta T_{pp} \tag{11}$$

$$Q_e = CP_h\left[T_{h,in} - T_{h,out}\right] = CP_h\left[T_{h,in} - \left(T_{w,in} + \Delta T_{pp}\right)\right] \tag{12}$$

From Fig.1 (a), by comparing Eq. (9) and (10), we have:

$$\Delta \mathrm{T}_{pp} = \Delta T' \leq \Delta T'' \tag{13}$$

And we also have Eq. (14) by Eq. (5), (9), (10) and (13)

$$C_r \geq 1 \tag{14}$$

When pinch point is VPP, as showed in Fig.1 (b), the exchanged heat, which evaporates the working fluid from saturated liquid to saturated vapour, is equal to the heat released by the heat source fluid from the inlet temperature $T_{h,in}$ to the pinch point temperature $T_{h,evap}$. So the mass flowrate of working fluid can be calculated by Eq. (11)

$$m_w = \frac{CP_h\left[T_{h,in} - T_{h,evap}\right]}{L} = \frac{CP_h\left[T_{h,in} - \left(T_{w,evap} + \Delta T_{pp}\right)\right]}{L} \tag{15}$$

The mass flowrate of working fluid can also be calculated by Eq. (12)

$$m_w = \frac{CP_w\left[T_{w.eavp} - T_{w,in}\right]}{S} \tag{16}$$

According to Eq. (5), (11), (12), C_r then can be deduced as:

$$C_r = \frac{[T_{h,in} - (T_{w,evap} + \Delta T_{pp})]S}{[T_{w.eavp} - T_{w,in}]L} \tag{17}$$

According to Eq. (8), (11),(12), the exit temperature of heat source fluid can be calculated from Eq. (13).

$$T_{h,out} = T_{h,in} - \frac{[T_{h,in} - (T_{w,evap} + \Delta T_{pp})](S + L)}{L} \tag{18}$$

The amount of heat recovered at evaporator is given by Eq. (14):

$$Q_e = \frac{CP_h[T_{h,in} - (T_{w,evap} + \Delta T_{pp})](S + L)}{L} \tag{19}$$

We introduce a parameter κ_r to indicate the ratio of the latent heat to the sensible heat, which is absorbed by working fluid:

$$\kappa_r = \frac{L}{S} \tag{20}$$

According to Eq. (13), (14), (15), the amount of heat recovered at evaporator is rewritten as:

$$Q_e = CP_h[T_{h,in} - (T_{w,in} + \Delta T_{pp})]\frac{C_r\kappa_r + C_r}{C_r\kappa_r + 1} \tag{21}$$

From Fig.1 (b), when the position of pinch point is at the start point of evaporation, by comparing Eq. (9) and (10), we have:

$$\Delta T_{pp} = \Delta T'' \leq \Delta T' \tag{22}$$

And we also have Eq. (23) by Eq. (5), (9), (10) and (22)

$$C_r \leq 1 \tag{23}$$

Comparing Eq. (12) and Eq. (21), we can see that the the amount of heat exchanged is changed with the position of pinch point, and $Q_{e,ppp}$ (When the position of pinch point is at the start point of preheating (PPP))$\geqslant Q_{e,vpp}$ (When the position of pinch point is at the start point of vaporization (VPP)), which means Q_e, reaches maximum only when the position of pinch point is at the start point of preheating ($C_r \geqslant 1$) . And, the amount of heat exchanged does not change with the increasing of C_r after C_r=1.

2.2. Entransy Analysis

Guo et al. [14] compared the heat transfer process with the electrical conduction process and gave the analogies between the parameters for the two processes, shown as Tab. 1. Then they proposed a new physical quantity, termed entransy to correspond to Electrical potential energy in a capacitor, also shown in Tab. 1.

Tab.1 Analogies between electrical and thermal parameters [14]

Electrical conduction process			Heat transfer process		
parameters	Symbol	SI	parameters	Symbol	SI
Electrical charge stored in capacity	Q_{ve}	C	Heat stored in an object	Q_{vh}	J
Electrical current	I	A	Heat flow	$\dot{Q}$	W
Electrical potential	U	V	Thermal potential (temperature)	T	K
Electrical current density	$\dot{q_e}$	$\mathrm{A/m^2}$	Heat flow desity	$\dot{q_h}$	$\mathrm{W/m^2}$
Capacitance	$C_e = \frac{Q_{ve}}{U}$	C/V	Heat capacity	$C_h = \frac{Q_{vh}}{T}$	J/K
Ohm's law	$\dot{q_e} = -k_e \frac{dU}{dn}$		Fourier law	$\dot{q_h} = -k_h \frac{dT}{dn}$	
Electrical potential energy in a capacitor	$E_e = \frac{1}{2} Q_{ve} U$	C · V	Entransy	$G = \frac{1}{2} Q_{vh} T$	J · K

The definition equation of entransy is shown as Eq. (24).

$$G = \frac{1}{2} Q_{vh} T = \frac{1}{2} mcT^2 \tag{24}$$

The physical meaning of entransy can be expressed as the "potential energy" which possesses both the nature of "energy" and the transfer ability [14].

Practical heat transfer process is an irreversible, non-equilibrium process due to the temperature difference. Though total thermal energy of system is always conservative through heat transfer process, to optimize the process, it is needed to measure the irreversibility of heat transfer process. Chen et al. [15] classified various heat transfer processes into two categories: one is for heat-work conversion and the other is directly using thermal energy for heating or cooling only. For the latter, they said the entransy dissipation rate is the best measure of irreversibility, because the so-called "entropy generation paradox" occurs if entropy generation is used as the irreversibility measurement during a pure heat transfer. In this paper, we use the concept of entransy, to measure the irreversibility during the heat transfer process. In a heat transfer process, the difference between total entransy flow in and total entransy flow out represents the "potential energy" dissipation through heat transfer, even though the energy is conserved. This loss was proposed as entransy dissipation flow and adopted to quantize irreversibility of heat transfer, which can be calculated from Eq. (25) [14].

$$\dot{G}_{diss} = \dot{G}_{in,total} - \dot{G}_{out,total} \tag{25}$$

Chen et al. [16] expressed that in the T-Q property diagram, which shows the heat transfer performance of a two-flow heat exchanger, the area between the hot fluid line and cold fluid line stands for the entransy dissipation due to the heat flow transferred from the hot fluid to the cold fluid.

When the position of pinch point is at the start point of preheating ($C_r \geqslant 1$), the vaporization temperature of working fluid $T_{w,evap}$, and the temperature of heat source fluid $T_{h,evap}$ at the starting point of vaporization of working fluid can be calculated from Eq. (26), (27):

$$T_{w,evap} = \frac{T_{h,in} - \left(T_{w,in} + \Delta T_{pp}\right)}{C_r(1+\kappa_r)} + T_{w,in} \tag{26}$$

$$T_{h,evap} = \frac{1}{1+\kappa_r} T_{h,in} + \frac{\kappa_r}{1+\kappa_r} T_{w,in} + \frac{\kappa_r}{1+\kappa_r} \Delta T_{pp} \tag{27}$$

According to Eq. (6), (7), (11), (26), (27), the heat exchanged amount of preheating, vaporization at evaporator can be calculated by Eq. (28), (29), respectively:

$$Q_{e,preh} = CP_h\left[T_{h,in} - \left(T_{w,in} + \Delta T_{pp}\right)\right]\frac{1}{1+\kappa_r} \tag{28}$$

$$Q_{e,evap} = CP_h\left[T_{h,in} - \left(T_{w,in} + \Delta T_{pp}\right)\right]\frac{\kappa_r}{1+\kappa_r} \tag{29}$$

The entransy dissipation can be calculated as (30):

$$\begin{aligned}\dot{G}_{diss} &= \frac{1}{2} Q_{e,preh}\left(T_{h,evap} + T_{h,out} - T_{w,in} - T_{w,evap}\right)\\ &+ \frac{1}{2} Q_{e,evap}\left(T_{h,in} + T_{h,evap} - 2T_{w,evap}\right)\\ &= \frac{1}{2} Q_e \left(1 - \frac{1+2\kappa_r}{C_r(1+\kappa_r)^2}\right) T_{h,in} - \left(1 - \frac{1+2\kappa_r}{C_r(1+\kappa_r)^2}\right) T_{w,in}\\ &+ \left(1 - \frac{1+2\kappa_r}{C_r(1+\kappa_r)^2}\right)\Delta T_{pp}\end{aligned} \tag{30}$$

Basing on the concept of entransy dissipation, Guo et al. [17-18] defined an equivalent thermal resistance in the heat transfer process and developed the principle of the minimum thermal resistance. It describes not only heat transfer resistance but also the additional resistance caused by non-counter flow and non-equilibrium heat transfer process. The equivalent thermal resistance is calculated from Eq. (31)

$$R_e = \frac{\dot{G}_{diss}}{Q_e^2} \tag{31}$$

From Eq. (30), (31), the equivalent thermal resistance in evaporator R_e is calculated from Eq. (32)

$$R_e = \frac{\dot{G}_{diss}}{Q^2} = \frac{1 - \dfrac{1+2\kappa_r}{C_r(1+\kappa_r)^2}}{2CP_h} + \frac{\Delta T_{pp}}{Q_e} \tag{32}$$

Since the purpose of ORC research is focusing on reducing the irreversible loss under the condition that maximum thermal energy may be recovered from heat source, we only consider the region that $C_r \geqslant 1$ and $\kappa_r \geqslant 0$ in this paper. When $\kappa_r = 0$, which means a single-phase heat transfer status, then, the entransy dissipation $\dot{G}_{diss}$ and the equivalent thermal resistance R_e are increased monotonically with the increment of C_r. A minimum equivalent thermal resistance R_e is only achieved at the endpoint C_r=1.

If $\kappa r \neq 0$, which means a two-phase heat transfer status, then, the equivalent thermal resistance R_e is simultaneously affected by both C_r and κ_r. In Eq. (32), if we let

$$X = \frac{1 + 2\kappa_r}{C_r(1 + \kappa_r)^2} \tag{33}$$

Eq. (32) can be rewritten as:

$$R_e = \frac{\dot{G}_{diss}}{\Delta H^2} = \frac{1 - X}{2CP_h} + \frac{\Delta T_{pp}}{Q_e} \tag{34}$$

We can see that the equivalent thermal resistance R_e decreases with X monotonically, which means, minimum R_e will be obtained when maximum X is achieved. Because $C_r \geq 1$, $\kappa_r \geq 0$, then there is an inequality (35)

$$X = \frac{1 + 2\kappa_r}{C_r(1 + \kappa_r)^2} \leq \frac{1 + 2\kappa_r}{1 + 2\kappa_{r+}\kappa_r^2} \leq 1 \tag{35}$$

Maximum value X=1 is achieved only when $C_r = 1$, $\kappa_r = 0$, which means that minimum irreversible loss can only be obtained when single-phase heat transfer process occurs in evaporator. Otherwise, the value of R_e varies with working fluid changes, because different working fluid has different C_r-κ_r relation. For the heat transfer process in the evaporator, if the heat source and the pinch point temperature difference are identified, the determined C_r and κ_r also means that the evaporation temperature of the working fluid is identified. That is for a certain working fluid, there may be an optimum evaporation temperature at which the minimum equivalent thermal resistance R_e can be obtained while the quantity of heat transferred is maximum. And for different working fluids, the quantity of heat transferred and the equivalent thermal resistance are different. By comparing the quantity of heat transferred and the equivalent thermal resistance of different working fluid, we develop a calculation method to select the optimal working fluid and the evaporation temperature for an ORC system.

2.3. A method to optimize working fluid and operating condition

Basing on discussion above, we can establish a calculation method to select working fluid and operating condition simultaneously. For a pre-set heat source condition and pinch point temperature difference:

（1） Calculating the maximum heat transferred Qe, and the minimum equivalent thermal resistance Re,limit from Eq.(12), (32), respectively;

(2) In the ±20% range of the minimum equivalent thermal resistance Re,limit, calculating the value range of X by Eq.(33);

(3) Alternately calculating the value of X for each working fluid, and selecting the working fluids as the candidate working fluid while its X value is in the calculated range.

(4) Comparing the X value of each working fluid, the optimum working fluid is the one with maximum X value. And the optimum evaporation temperature can be calculated at the condition that X reaches maximum.

We use the example given in reference [7] to confirm the reliability of this method. The pre-set condition is: the temperature of the heat source $T_{h,in}$ is 463.15K, the heat capacity flow rate of heat fluid CP_h is 10 kWK^{-1}, the inlet temperature of the working fluid $T_{w,in}$ is 313.15K and the temperature difference of the pinch point ΔT_{pp} is 10K.

The calculated results are shown in Tab. 2.The optimum evaporation temperature $T_{evap,opt}$ is the temperature that the maximum heat recovery and the minimum equivalent thermal resistance are obtained for a selected working fluid. $Q_{e,opt}$ and $R_{e,opt}$ are the amount of heat transferred and the equivalent thermal resistance at optimum evaporation temperature, respectively. In our calculation, R245ca is the best working fluid, and 447.45 K is the best evaporating temperature. But R600 was chosen in Ref. [7] as the best working fluid and 423.15 K is calculated as the optimum evaporating temperature. Tab. 1 compares the heat recovery, equivalent thermal resistance of evaporator and output power of system for four different working fluids. According to Tab.1, selecting R245ca means the lower equivalent thermal resistance of evaporator and larger output power of system than that by selecting R600, even though the evaporation temperature and the amount of heat recovery are approximately same. Therefore, the method proposed in this paper on choosing working fluid and operating conditions is more reasonable.

Tab.2 Calculated results for the example case given in reference [7]

Working fluid	$T_{w,crit}$ [K]	$T_{evap,opt}$ [K]	$m_{w,opt}$ $[kg \cdot s^{-1}]$	$Q_{e,opt}$ [W]	$R_{e,opt}$ $[K \cdot kW^{-1}]$	$W_{T,opt}$ [kW]
R600	425.15	423.15	3.25	1400	0.0114	212.71
R245fa	427.15	423.25	6.10	1400	0.0096	210.56
R245ca	447.55	447.45	5.66	1400	0.0076	227.54
R601a	460.35	387.45	2.56	1149	0.0298	141.56

2.4. Conclusions

In this paper, we study the heat transfer performance in evaporator under the condition that the heat source parameters and pinch point temperature are identified.

It is found that the heat transfer performance is affected by C_r, the ratio of heat capacity flow rates between the working fluids and the heat source fluid. When the position of pinch point is at the start point of preheating (PPP), where $C_r \geqslant 1$, the maximum amount of heat transfer Q_e is achieved.

We use the equivalent thermal resistance to measure the irreversibility during the heat transfer process in evaporator. We also define another parameter κr, which is the ratio between latent heat and sensible heat of working fluid. We investigate the relationship between the irreversible loss and parameters C_r and κ_r, and deduce the condition under which the maximum heat transfer and minimum equivalent thermal resistance of each working fluid occur are deduced.

Finally, a calculation method is established by comparing the amount of heat transferred and the irreversibility of different working fluid. The method is used to choose the optimum working fluid and the evaporation temperature for a preset heat source condition. An example given in reference [7] is selected to confirm that the method provided in this paper can not only meet lower irreversible loss but also obtain larger output power for the system. Thus it is more reasonable than that in reference [7].

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Nomenclature

c fluid heat capacity rate [J kg^{-1} K^{-1}]
CP fluid heat capacity flow rate [W K^{-1}]
U internal energy [J]
G entransy [J K]
$\dot{G}_{diss}$ entransy dissipation flow [W K]
H enthalpy [J]
m mass flow rate [kg s^{-1}]
Q heat flow [W]
L latent heat [J kg^{-1}]
S sensible heat [J kg^{-1}]
R_e entransy based equivalent thermal resistance [K W^{-1}]
T temperature [K]
ΔT_{pp} Pinch point temperature difference [K]

Subscripts

e evaporator
evap evaporation
h heat source fluid
w working fluid
in inlet
out outlet
preh preheat
opt optimum
lim limitation

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