ANALYSIS OF HEAT TRANSFER AND IRREVERSIBILITY OF ORGANIC RANKINE CYCLE EVAPORATOR FOR SELECTING WORKING FLUID AND OPERATING CONDITIONS

by

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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 difference 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 fluid and the heat source fluid. The equivalent thermal resistance, deducing from the concept of entransy, to measure the irreversability during the heat transfer process is used. Then, the parameter κ_r, the ratio between latent heat and sensible heat of working fluid is defined. 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*

Introduction

Organic Rankine cycle 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 conditions 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 en-

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ergy 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 system. Yu *et al.* [7] classified heat transfer process in evaporator into preheating pinch point (PPP) and vaporizing pinch point (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 conditions simultaneously basing on the analysis of the pinch point position. The aforementioned 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 aforementioned 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.

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]. Figure 1 shows the different locations of pinch point at *T-H* diagram. Figure 1(a) shows the PPP, and fig. 1(b) shows the VPP.

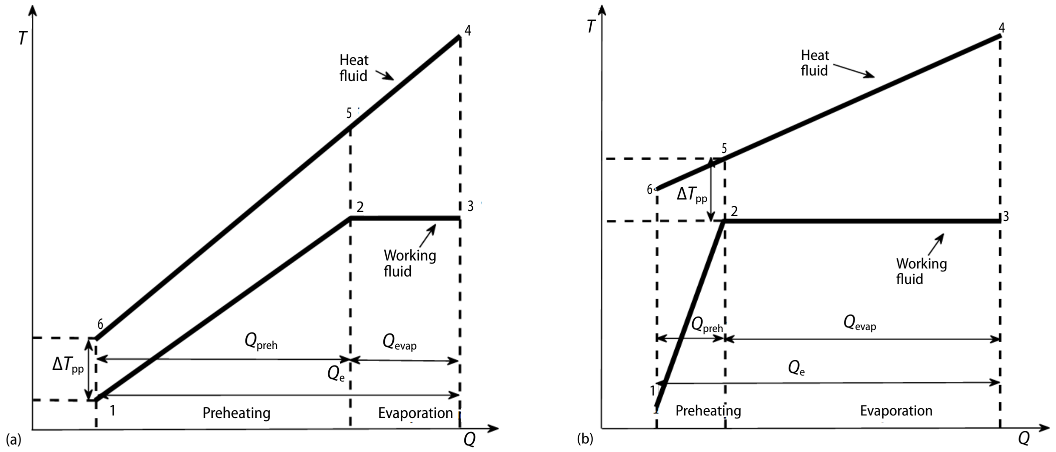

Figure 1. The *T-H* diagram of different position of pinch point; (a) PPP and (b) VPP

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.

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

$$CP = \frac{\mathrm{d}H}{\mathrm{d}T} \tag{1}$$

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:

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

Heat balance analysis

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

- Heat capacity flow rate of heat fluid CP_{h} is determined.
- Heat source fluid inlet temperature $T_{\mathrm{h,in}}$ and working fluid inlet temperature $T_{\mathrm{w,in}}$ are determined.
- Pinch point temperature difference ΔT_{pp} is determined.
- Working fluid is saturated vapour at exist of evaporator, in other words, vaporizing temperature $T_{\mathrm{w,evap}}$ equals to the outlet temperature of working fluid $T_{\mathrm{w,out}}$.

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

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

$$Q_{\mathrm{e}} = CP_{\mathrm{h}}\left(T_{\mathrm{h,in}} - T_{\mathrm{h,out}}\right) = \dot{m}_{\mathrm{w}}\left(S + L\right) \tag{4}$$

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:

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

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

$$Q_\mathrm{preh} = CP_\mathrm{w}\left(T_\mathrm{w,evap} - T_\mathrm{w,in}\right) = \dot{m}_\mathrm{w} S = CP_\mathrm{h}\left(T_\mathrm{h,evap} - T_\mathrm{h,out}\right) \tag{6}$$

$$Q_\mathrm{evap} = \dot{m}_\mathrm{w} L = CP_\mathrm{h}\left(T_\mathrm{h,in} - T_\mathrm{h,evap}\right) \tag{7}$$

$$Q_\mathrm{e} = \Delta Q_\mathrm{preh} + \Delta Q_\mathrm{evap} = CP_\mathrm{w}\left(T_\mathrm{w,evap} - T_\mathrm{w,in}\right) + \dot{m}_\mathrm{w} L = CP_\mathrm{h}\left(T_\mathrm{h,in} - T_\mathrm{h,out}\right) \tag{8}$$

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

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

We define the temperature difference between heat source and working fluid at the starting point of preheating as $\Delta T'$, and define the temperature difference at the starting point of vaporization as $\Delta T''$, respectively:

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

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

When pinch point is PPP, as showed in fig.1(a), the exit temperature of heat source fluid can be calculated from eq. (11):

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

$$Q_\mathrm{e} = CP_\mathrm{h}\left(T_\mathrm{h,in} - T_\mathrm{h,out}\right) = CP_\mathrm{h}\left[T_\mathrm{h,in} - \left(T_\mathrm{w,in} + \Delta T_\mathrm{pp}\right)\right] \tag{12}$$

From fig.1(a), by comparing eqs. (9) and (10), we have:

$$\Delta T_\mathrm{pp} = \Delta T' \le \Delta T'' \tag{13}$$

And we also have eq. (14) by eqs. (5), (9), (10), and (13):

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

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 $T_\mathrm{h,in}$ to the pinch point temperature $T_\mathrm{h,evap}$. So the mass flowrate of working fluid can be calculated by eq. (15):

$$\dot{m}_\mathrm{w} = \frac{CP_\mathrm{h}\left(T_\mathrm{h,in} - T_\mathrm{h,evap}\right)}{L} = \frac{CP_\mathrm{h}\left[T_\mathrm{h,in} - \left(T_\mathrm{w,evap} + \Delta T_\mathrm{pp}\right)\right]}{L} \tag{15}$$

The mass-flow rate of working fluid can also be calculated by eq. (16):

$$\dot{m}_\mathrm{w} = \frac{CP_\mathrm{w}\left(T_\mathrm{w.eavp} - T_\mathrm{w,in}\right)}{S} \tag{16}$$

According to eqs. (5), (15), and (16), C_r then can be deduced:

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

According to eqs. (8), (15), and (16), the exit temperature of heat source fluid can be calculated from eq. (18):

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

The amount of heat recovered at evaporator is given by eq. (19):

$$Q_{\text{e}} = \frac{CP_{\text{h}}\left[T_{\text{h,in}} - \left(T_{\text{w,evap}} + \Delta T_{\text{pp}}\right)\right](S+L)}{L} \tag{19}$$

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 eqs. (18)-(20), the amount of heat recovered at evaporator is rewritten:

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

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

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

And we also have eq. (23) by eqs. (5), (9), (10), and (22):

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

Comparing eqs. (12) and (21), we can see that the the amount of heat exchanged is changed with the position of pinch point, and $Q_{\text{e,PPP}}$ when the position of pinch point is at the start point of preheating (PPP), $\ge Q_{\text{e,VPP}}$ 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 ($C_{\text{r}} \ge 1$). The amount of heat exchanged does not change with the increasing of C_{r} after $C_{\text{r}} = 1$.

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.

The definition equation of entransy is shown:

$$G = \frac{1}{2}Q_{vh}T = \frac{1}{2}\dot{m}cT^2 \tag{24}$$

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

Electrical conduction process			Heat transfer process		
Parameters	Symbol	Unit	Parameters	Symbol	Unit
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	$\dot{q}_\mathrm{e}$	A/m^2	Heat flow density	$\dot{q}_\mathrm{h}$	W/m^2
Capacitance	$C_e = \dfrac{Q_{ve}}{U}$	C/V	Heat capacity	$C_h = \dfrac{Q_{vh}}{T}$	J/K
Ohm's law	$\dot{q}_\mathrm{e} = -k_\mathrm{e}\dfrac{\mathrm{d}U}{\mathrm{d}n}$		Fourier law	$\dot{q}_\mathrm{h} = -k_h\dfrac{\mathrm{d}T}{\mathrm{d}n}$	
Electrical potential energy in a capacitor	$E_\mathrm{e} = \dfrac{1}{2}Q_{ve}U$	CV	Entransy	$G = \dfrac{1}{2}Q_{vh}T$	JK

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 [14]:

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

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 ($C_\mathrm{r} \geq 1$), the vaporization temperature of working fluid $T_\mathrm{w,evap}$, and the temperature of heat source fluid $T_\mathrm{h,evap}$ at the starting point of vaporization of working fluid can be calculated from:

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

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

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

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

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

The entransy dissipation can be calculated:

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

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:

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

From eqs. (30) and (31), the equivalent thermal resistance in evaporator R_{e} is calculated from:

$$R_e = \frac{\dot{G}_{\text{diss}}}{Q^2} = \frac{1 - \dfrac{1+2\kappa_r}{C_r \left(1+\kappa_r\right)^2}}{2CP_{\text{h}}} + \frac{\Delta T_{\text{pp}}}{Q_{\text{e}}} \tag{32}$$

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 $C_{\text{r}} \geq 1$ and $\kappa_{\text{r}} \geq 0$ in this paper. When $\kappa_{\text{r}} = 0$, which means a single-phase heat transfer status, then, the entransy dissipation $\dot{G}_{\text{diss}}$ 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 $C_{\text{r}} = 1$.

If $\kappa_{\text{r}} \neq 0$, 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:

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

Equation (32) can be rewritten:

$$R_e = \frac{\dot{G}_{\text{diss}}}{\Delta H^2} = \frac{1-X}{2CP_{\text{h}}} + \frac{\Delta T_{\text{pp}}}{Q_{\text{e}}} \tag{34}$$

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:

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

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. 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.

A method to optimize working fluid and operating condition

Basing on previous discussion, 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:

- Calculating the maximum heat transferred Q_e, and the minimum equivalent thermal resistance R_e, limit from eqs. (12), (32), respectively.
- In the ±20% range of the minimum equivalent thermal resistance R_e, limit, calculating the value range of X by eq. (33).
- 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.
- Comparing the X value of each working fluid, the optimum working fluid is the one with maximum X value. The optimum evaporation temperature can be calculated at the condition that X reaches maximum.

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

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. The $Q_{e,opt}$ 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 [7] as the best working fluid and 423.15 K is calculated as the optimum evaporating temperature. Table 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

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

Working fluid	$T_{\mathrm{w,crit}}$ [K]	$T_{\mathrm{evap,opt}}$ [K]	$\dot{m}_{\mathrm{w,opt}}$ [kgs^{-1}]	$Q_{\mathrm{e,opt}}$ [W]	$R_{\mathrm{e,opt}}$ [KkW^{-1}]	$W_{\mathrm{T,opt}}$ [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

method proposed in this paper on choosing working fluid and operating conditions is more reasonable.

Conclusions

In this paper, we study the heat transfer performance in evaporator under the condition that the heat source parameters and pinch point temperature difference 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 $C_r \geq 1$, 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 κ_r, 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 [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 [7].

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Nomenclature

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

Subscripts

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

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