Second Law Optimization of Water-to-Water Heat Pump System

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This paper presents a thermodynamic analysis of heat pump system using water as a heat source and heat sink. The primary object in this study is the optimization of exergetic efficiency. Two different systems, 2-stream and 1-stream system, are analyzed in detail. Mass flow ratio (the ratio of mass flow rate of water through evaporator to that through condenser) is identified as the most important parameter to be optimized. It is shown that there exists an optimum mass flow ratio to maximize exergetic efficiency. The variation of optimum exergetic efficiency of 2-stream system is quite small and the value lies between 0.2~0.23 for the range of investigation in this study. However, far better performance can be obtained from 1-stream system. This means considerable irreversibilities are generated through condenser of the 2-stream system. The effects of adiabatic efficiency of compressor-motor unit on the overall system performance are also examined in the analysis.

Key Words : COPH, Exergetic Efficiency, Exergy, Water-to-Water Heat Pump

Nomenclature

$\dot{A}$: Exergy rate (kW)
a : Dimensionless exergy
C : Specific heat of water (kJ/kg·K)
h : Enthalpy (kJ/kg)
$\dot{m}$: Mass flow rate of water (kg/s)
T : Temperature (K)
$\dot{W}$: Power (kW)
γ : Mass flow ratio
θ : Dimensionless temperature
Φ : Exergetic efficiency

Subscripts

c : Condenser
d : Dead state
e : Evaporator
el : Electricity
i : Inlet
G : Saturated vapor state of refrigerant in the condenser
mc : Motor-compressor combination
o : Outlet
opt : Optimum
R : Resources
s : Isentropic

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1. Introduction

Heat pump is an appropriate device for low temperature applications and can show substantial energy saving effects under suitable conditions. However, it is difficult to apply heat pump in the area where outside temperature is very low due to defrosting problems. Moreover, high electricity cost makes its application more difficult.

The water-to-water heat pump system uses water as a heat source and heat sink. There is no defrosting problem in this system and the required power can be considerably lowered if source water with relatively high temperature can be supplied sufficiently.

The conventional performance parameter of heat pump system is COPH (Coefficient of Performance in Heating). However, this concept is not suitable for water source heat pumps because the energy value of source water is not reflected in this parameter. The exergy analysis is an appropriate method for this system. The concept of exergy and its application were well established in the thermodynamic text (Moran, 1989 ; Moran and Shapiro, 2000). Also this concept is widely being applied in the energy system analysis recently (Al-Ragom, et al., 1997 ; Nikolaidis and Probert, 1998 ; Smith and Few, 2001).

Several authors found that in the analysis of water-to-water heat pump system second law analysis was very useful. Akau and Scheonhals (1980) have shown several values of second law efficiencies of water-to-water heat pumps as functions of the constraints given to the system. Liang and Kuehn (1991) have performed detailed irreversibility analysis on every part of the experimental water-to-water heat pump system.

The main interest of present work is the second law optimization of overall system performance. The mass flow ratio defined in Eq. (3) is one of the most important parameter to be optimized to maximize the exergetic efficiency. Such an optimal operation is also regarded as the operation with minimum irreversibility generation.

It should be recognized that the results from thermodynamic optimization are not necessarily coincident with the results from economic optimization. There are several references (Reistad and Gaggioli, 1980 ; Gyftopoulos and Widmer, 1980 ; Moran and Shapiro, 2000) for the relationship between both analyses. The authors believe that substantial guidance on the design of heat pump system may be achieved through knowledge of the optimal thermodynamic performances.

2. Assumptions

Figure 1 shows a schematic diagram of the

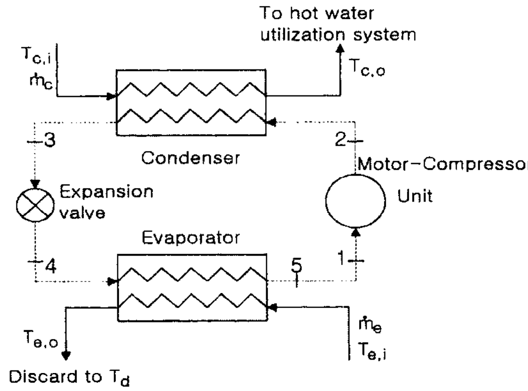

Fig. 1 Schematic diagram of water-to-water heat pump system

system using R-22 as a working fluid. In this figure, two water streams, one being heated through condenser and the other being cooled through evaporator are shown. Usually the water stream through evaporator is waste warm water. Although this arrangement is typical for most water-to-water heat pump applications, one water stream case is also analyzed in this study.

The system is assumed to be operated under the following assumptions.

1) All the pressure drops are negligible except those through expansion device and compressor suction line. The pressure loss in the compressor suction line is 30 kPa.

2) Compressor is operated adiabatically with the adiabatic compression efficiency of compressor-motor combination, η_{mc}. Heat losses from other parts of the system are also negligible.

3) The working fluid, R-22, is saturated at the condenser outlet (point 3 of Fig. 2) and 5℃ superheated at the evaporator outlet (point 5 of Fig. 2).

Thus Fig. 2 shows the temperature profiles of water and refrigerant through condenser and evaporator, respectively. The numerical letters in Fig. 2 refer to the points of Fig. 1. Point G in Fig. 2 refers to the saturated vapor state at condensing temperature of refrigerant, T_{con}. The inlet and exit water temperatures of condenser and evaporator are designated as $T_{c,i}$, $T_{c,o}$, $T_{e,i}$ and $T_{e,o}$ respectively. The mass flow rates, $\dot{m}_c$ and $\dot{m}_e$, of this figure are those of water through

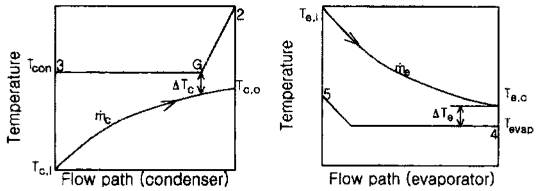

Fig. 2 Temperature profiles in the condenser and evaporator

condenser and evaporator respectively. The minimum temperature differences, ΔT_c and ΔT_e, shown in Fig. 2 are functions of heat transfer characteristics of condenser and evaporator respectively and can approach to zero as the heat transfer performances increase. Thus another assumption about these temperature differences is added as follows.

4) In this study $\Delta T_c = \Delta T_e = 8$℃.

The exergetic efficiency Φ of the system, defined as the ratio of exergy exiting to the exergy entering (Moran, 1989), can be expressed as

$$\Phi = \frac{\dot{A}_{c,o}}{\dot{A}_{c,i} + \dot{A}_{e,i} + \dot{W}_{el}} \tag{1}$$

where $\dot{A}_{c,i} = \dot{m}_c C\left(T_{c,i} - T_d - T_d \ln \frac{T_{c,i}}{T_d} \right)$

$$\dot{A}_{e,i} = \dot{m}_e C\left(T_{e,i} - T_d - T_d \ln \frac{T_{e,i}}{T_d} \right)$$

$$\dot{A}_{c,o} = \dot{m}_c C\left(T_{c,o} - T_d - T_d \ln \frac{T_{c,o}}{T_d} \right)$$

In Eq. (1) T_d is the dead state temperature, C the specific heat of water considered as constant and $\dot{W}_{el}$ is the required electrical power to the compressor.

The energy balance of the system is given by

$$\dot{m}_e C(T_{e,i} - T_{e,o}) + \dot{W}_{el} = \dot{m}_c C(T_{c,o} - T_{c,i}) \tag{2}$$

Eq. (2) can be easily reduced to

$$\gamma(T_{e,i} - T_{e,o}) = (T_{c,o} - T_{c,i})\left(1 - \frac{1}{COPH}\right) \tag{3}$$

where, $\gamma = \dot{m}_e / \dot{m}_c$ (mass flow ratio),

$$COPH = \dot{m}_c C(T_{c,o} - T_{c,i}) / \dot{W}_{el}$$

Under the given conditions of $T_{c,i}$, $T_{c,o}$ and $T_{e,i}$, each state point of refrigerant on Fig. 1 (1, 2, 3, 4, and 5) can be calculated as follows.

By assuming T_{evap} with assumptions 1) and 3), states 5 and 1 can be determined and $T_{e,o}$ can be calculated with assumption 4). Then T_{con} is assumed. The enthalpy increase of refrigerant $h_2 - h_1$ can be calculated by

$$h_2 - h_1 = \frac{1}{\eta_{mc}}(h_{2s} - h_1) \tag{4}$$

In Eq. (4) h_{2s} means the enthalpy of refrigerant resulted from isentropic compression. Then the results must be checked with following energy balance.

$$\frac{(T_{con} - \Delta T_c) - T_{c,i}}{T_{c,o} - (T_{con} - \Delta T_c)} = \frac{h_G - h_3}{h_2 - h_G} \tag{5}$$

If this equation is not satisfied, new T_{con} must be assumed until the above energy balance is satisfied.

The COPH can be obtained by

$$COPH = \frac{h_2 - h_3}{h_2 - h_1} \tag{6}$$

Then γ can be calculated from Eq. (3).

By introducing dimensionless temperature θ which is defined as

$$\theta = T / T_d \tag{7}$$

Eq. (1) becomes as

$$\Phi = \frac{a_{c,o}}{a_{c,i} + \gamma a_{e,i} + (\theta_{c,o} - \theta_{c,i}) / COPH} \tag{8}$$

where $a_{c,i} = \theta_{c,i} - 1 - \ln \theta_{c,i}$

$$a_{e,i} = \theta_{e,i} - 1 - \ln \theta_{e,i}$$

$$a_{c,o} = \theta_{c,o} - 1 - \ln \theta_{c,o}$$

Eq. (2) can be also reduced to

$$\gamma(\theta_{e,i} - \theta_{e,o}) = (\theta_{c,o} - \theta_{c,i})\left(1 - \frac{1}{COPH}\right) \tag{9}$$

In this study, we will investigate the thermodynamic performances of two different systems, which are called 2-stream system and 1-stream system, respectively. The performance will be calculated under the condition of $T_d = 290$ K and $\eta_{mc} = 0.55$.

3. 2-Stream System

Two separate water streams are used in this

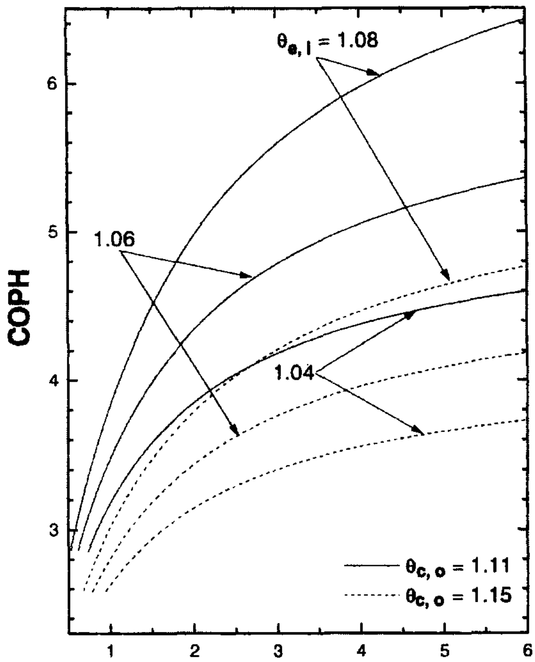

Fig. 3 COPH as a function of γ of 2-stream system

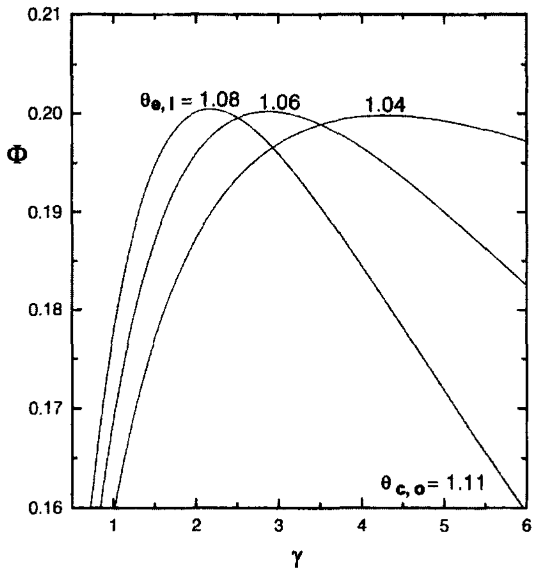

Fig. 4 Exergetic efficiencies of 2-stream system

system. One stream through evaporator is warm waste water and the other stream through condenser is cleaned cold water at the dead state temperature. Thus

$$\theta_{c,i}=1 \tag{10}$$

This system is typical for most water-to-water heat pump system. With the condition above, Eq. (8) can be reduced to

$$\Phi=\frac{a_{c,o}}{\gamma a_{e,i}+(\theta_{c,o}-1)/COPH} \tag{11}$$

Eq. (11) will be solved with the following constraint from Eq. (9),

$$\gamma(\theta_{e,i}-\theta_{e,o})=(\theta_{c,o}-1)\left(1-\frac{1}{COPH}\right) \tag{12}$$

The COPH calculated with the assumptions 1) through 4) is shown as a function of γ in Fig. 3. This figure shows a general tendency of COPH vs. γ relationship.

As γ increases, T_{evap} must be increased to meet the energy balance of the system, which results in reduction of compressor work and increased COPH.

It is not difficult to see that there exists Φ_{opt} from Eq. (11). As γ increases, exergy input of heat source water, $\gamma a_{e,i}$ of Eq. (11), increases while the exergy input from compressor work, $(\theta_{c,o}-1)$/COPH of Eq. (11), decreases. From these two phenomena, the maximum exergetic efficiency, Φ_{opt} (at the optimum mass flow ratio, γ_{opt}), can be determined.

However, exergy input due to high value of $\dot{m}_e$ on Φ is less significant as $\theta_{e,i}$ approaches dead state condition. It is seen from this figure that at low value of $\theta_{e,i}$ Φ is not so sharply maximized as the case of high value of $\theta_{e,i}$. Another point to be noted here is that Φ_{opt} is relatively constant irrespective of $\theta_{e,i}$, as shown in Fig. 4.

Figure 5 shows γ_{opt} as a function of $\theta_{e,i}$ for $\theta_{c,o}$ values of 1.11, 1.13 and 1.15. This figure is useful in determining the optimum mass flow ratio of water to keep the system performance maximized.

Figure 6 shows Φ_{opt} for 2-stream system. It is shown from this figure that the variation of Φ_{opt} is relatively small. For the values of $\theta_{c,o}$ and $\theta_{e,i}$ shown in this figure, the range of Φ_{opt} is 0.2~0.23. Since Φ means the degree of thermodynamic perfection, it should be noted from this numerical value that the thermodynamic per-

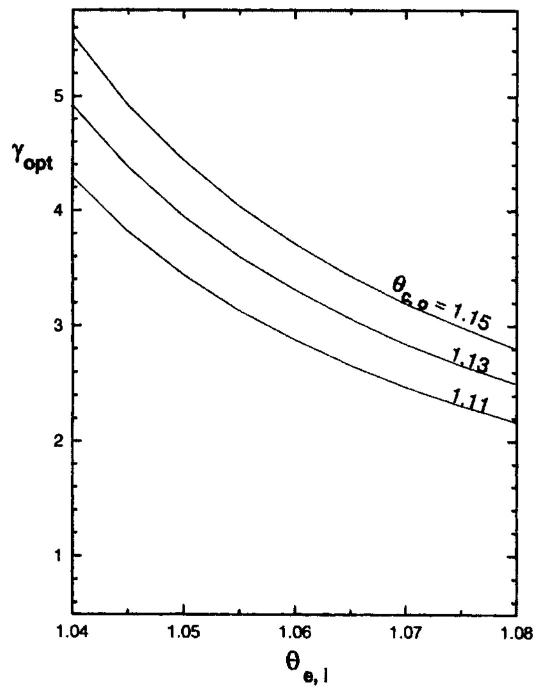

Fig. 5 Optimum mass flow ratio of 2-stream system

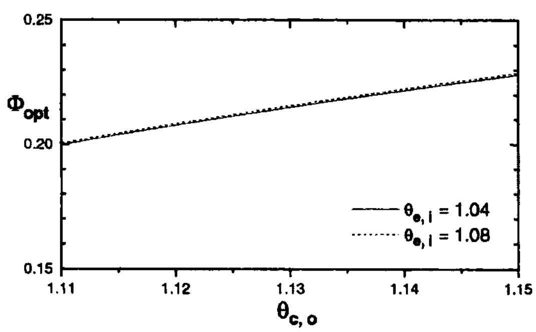

Fig. 6 Optimum exergetic efficiencies of 2-stream system

formance of 2-stream system is very poor, which means further study on the detailed irreversibility analysis of each part of the system is required to identify how much real losses are generated in each part.

4. 1-Stream System

In this system single stream of warm water at temperature of T_R is used for both condenser and

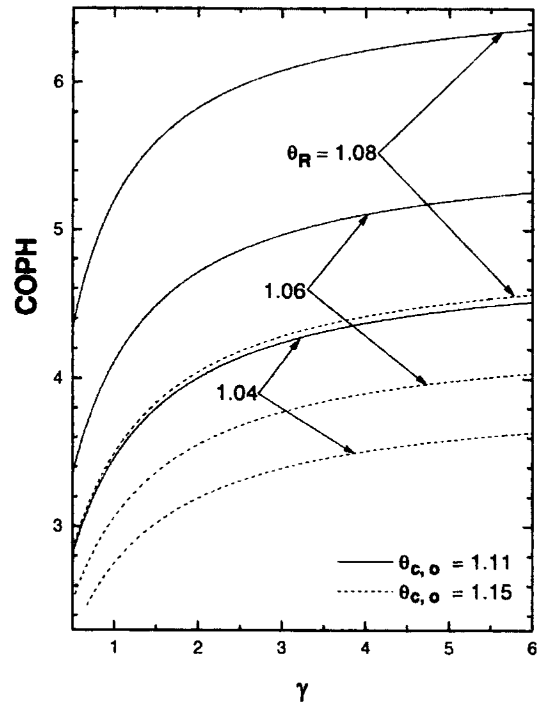

Fig. 7 COPH as a function of γ of 1-stream system

evaporator. Practically this case is possible when cleaned warm water is available. Because warm water flows through condenser, the irreversibilities generated in the condenser due to temperature difference is small and better performance than that of 2-stream system can be expected.

Since $\theta_{e,i}=\theta_{c,i}=\theta_R$ for this system Eqs. (8) and (9) become, respectively

$$\Phi=\frac{a_{c,o}}{(1+\gamma)\,a_R+(\theta_{c,o}-\theta_R)/COPH} \tag{13}$$

where $a_R=\theta_R-1-\ln\theta_R$

$$\gamma(\theta_R-\theta_{e,o})=(\theta_{c,o}-\theta_R)\left(1-\frac{1}{COPH}\right) \tag{14}$$

Figure 7 shows the COPH vs. γ results for the given values of θ_R and $\theta_{c,o}$ The characteristics of the curves plotted in this figure are similar to those of 2-stream system. As in the case of 2-stream system, there exists an optimum γ to maximize Φ. This is shown in Fig. 8 where the value of Φ is plotted as a function of γ. Each curve of this figure shows Φ_{opt} at a particular value of γ. These optimum values, Φ_{opt} and γ_{opt}, are plotted in Fig. 9.

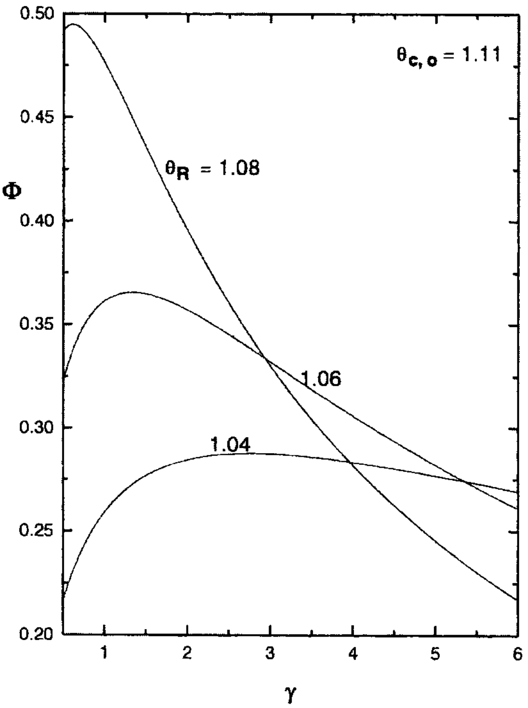

Fig. 8 Exergetic efficiencies of 1-stream system

As shown in Fig. 9, increase of θ_R results in increase of Φ_{opt} and decrease of γ_{opt}. This means irreversibilities generated through condenser due to temperature difference decreases as θ_R increases. Consequently substantial improvement of Φ_{opt} can be obtained.

5. Effect of η_{mc}

The results discussed so far are based on the assumption of $\eta_{mc}=0.55$. However, it is natural to expect performance improve upon the improved value of η_{mc}. Figure 10 shows the effect of η_{mc} on the system performance for 1-stream system and 2-stream system. It is shown from this figure that considerable efficiency improvement can be obtained with η_{mc} improved.

The curves of this figure are plotted for $\theta_{c,o}=1.13$ and $\theta_{e,i}=1.06$. However, the increasing rate in this figure can be approximately applied to the system of other operating conditions.

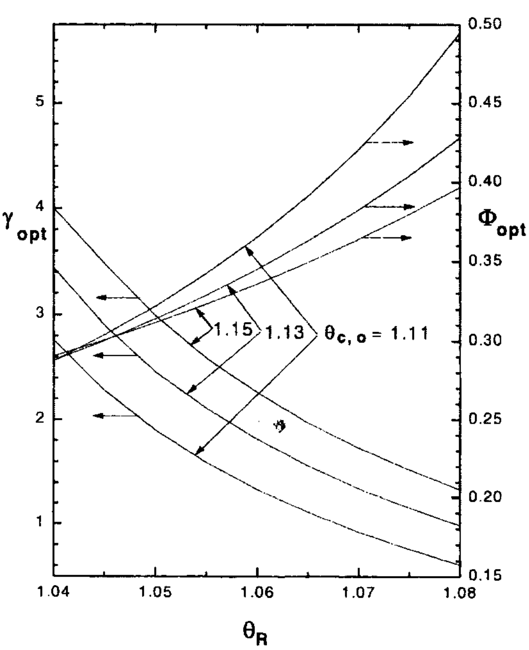

Fig. 9 Optimum mass flow ratios and exergetic efficiencies as a function of θ_R

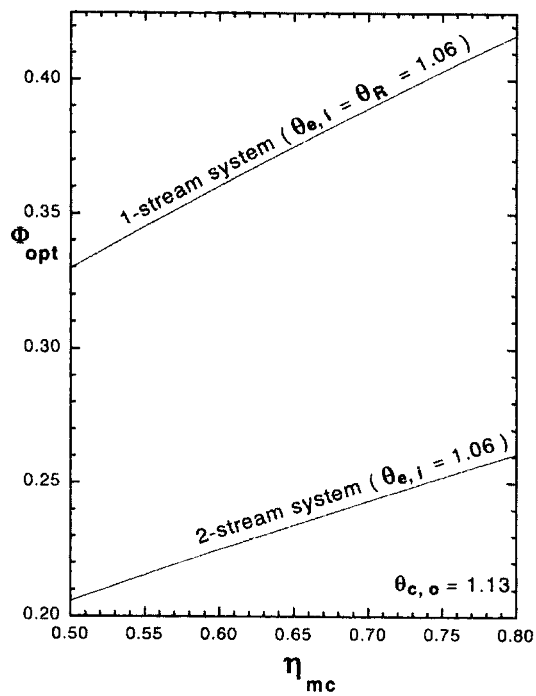

Fig. 10 Exergetic efficiencies of 1 and 2-stream systems as a function of η_{mc}

6. Conclusions

This paper presents a thermodynamic analysis of water-to-water heat pump system. The main interest here is the maximizing exergetic efficiency under given conditions, which also means minimizing irreversibilities generated. We have investigated two different systems, 1-stream system and 2-stream system. Mass flow ratio γ is identified as the most important parameter to be optimized.

The optimum exergetic efficiency, Φ_{opt}, is strongly affected by the heat source temperature for 1-stream system. However, such effect is small for 2-stream system.

The conventional performance parameter COPH increases with γ (Fig. 3 and Fig. 7). However, there exists optimum mass flow ratio, γ_{opt}, which results in maximum exergetic efficiency, Φ_{opt}.

The calculation of this analysis is based on the T_d=290 K. However, the results of this study can be closely valid to the cases of different dead state conditions. Reasonable ranges of T_d for application of this study would be about 270~290 K.

The thermodynamic performance of 2-stream system which is typical arrangement for most water-to-water heat pump applications is very poor, which means detailed irreversibility study of each component of the system may be inevitable for further study.

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