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Application of Exergy Analysis to Chilled Water Circuit and Heat Pump System

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Abstract

To elucidate the energy-saving effect of pumps along with the heat-transfer performance of terminal units in heat pump systems, a comprehensive analysis of an assumed chilled-water circuit at two supply water temperatures under four variable-flow control modes was carried out from the viewpoint of available energy, i.e., exergy. Subsequently, based on the operating data, the exergy analysis of a heat pump system was carried out to verify the energy-saving effect after variable-frequency transformation of the chilled water pump. The corresponding results based on the operating data of an actual heat pump system are consistent with those obtained through theoretical analysis.

Keywords - exergy analysis; exergy input; exergy consumption; variable-flow control mode; supply water temperature

1. Introduction

Reducing the amount of energy consumed by pumping systems in an HVAC system is essential. The variable-flow control mode of a chilled water circuit and chilled water temperature are important factors that influence energy savings in heat pump systems. Most previous studies on these two factors were carried out mainly from the viewpoint of the amount of energy being used, and not the quality of that energy [1, 2]. In thermodynamics, energy can be divided into available and unavailable energy. Exergy is energy that is available for use. Unlike conventional energy analysis, exergy analysis enables us to determine where and how much exergy is consumed [3]. Therefore, in this study, the effects of different variable-flow control modes and supply water temperatures on the exergy budget of an assumed chilled water circuit were analyzed theoretically. Subsequently, a follow-up analysis based on actual operating data of a heat pump system was carried out to verify the results of the theoretical analysis.

2. Exergy analysis of an assumed chilled water circuit

2.1. Assumed system and its corresponding exergy budget

The assumed system is shown in Fig. 1. For simplicity, it was assumed that there is only one FCU (Fan Coil Unit) on the demand side and one pump for supplying the chilled water. The system consists of three subsystems: the air channel inside the FCU, the tube wall between the air and water inside the FCU, and the chilled water channel inside the tube. The flow of exergy through the three subsystems is shown in Fig. 2.

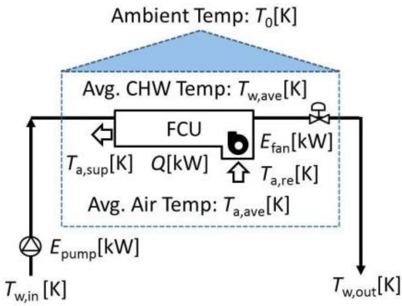


Fig. 1 Assumed chilled water circuit

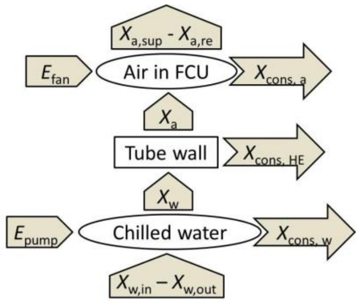


Fig. 2 Flow of exergy through the system

The derivations of the system exergy budget are shown as follows. In the following equations, the mean air temperature $T_{a,ave}$ [K] and mean chilled water temperature $T_{w,ave}$ [K] in the FCU are approximated by the mean values of the supply and return air temperatures, $T_{a,sup}$ [K] and $T_{a,re}$ [K], respectively, and the mean values of the inlet and outlet temperatures of the chilled water circuit, $T_{w,in}$ [K] and $T_{w,out}$ [K], respectively. c_a [kJ/(kg·K)] and c_w [kJ/(kg·K)], ρ_a [kg/m^3] and ρ_w [kg/m^3], F_a [m^3/h] and F_w [m^3/h] are the specific heat capacities, densities, and flow rates of the air and chilled water, respectively. Q [kW] is the thermal-energy transfer rate in the FCU. $S_{g,a}$ [kW/K], $S_{g,HE}$ [kW/K], and $S_{g,w}$ [kW/K] are the rates of entropy generation in the air within the FCU, in the tube wall between air and water, and in the water within the chilled water circuit, respectively.

Equations (1) and (2) show the energy and entropy budgets of the air subsystem. By combining these two equations with the ambient temperature, T_0 [K], the exergy-budget equation can be developed, as shown in (3) [4].

$$E_{fan} + c_a\rho_a F_a\left(T_{a,re} - T_0\right)/3600 = c_a\rho_a F_a\left(T_{a,\sup} - T_0\right)/3600 + Q \tag{1}$$

$$c_a\rho_a F_a \cdot \ln\left(T_{a,re}/T_0\right)/3600 + S_{g,a} = c_a\rho_a F_a \cdot \ln\left(T_{a,\sup}/T_0\right)/3600 + Q/T_{a,ave} \tag{2}$$

$$E_{fan} + X_a - X_{cons,a} = X_{a,\sup} - X_{a,re} \tag{3}$$

In the same manner, the exergy budgets of the tube-wall subsystem and chilled-water subsystem are given by (6) using (4) and (5) along with T_0 and by (9) using (7) and (8) along with T_0, respectively.

$$Q = Q \tag{4}$$

$$Q/T_{a,ave} + S_{g,HE} = Q/T_{w,ave} \tag{5}$$

$$X_w - X_{cons,HE} = X_a \tag{6}$$

$$E_{pump} + c_w \rho_w F_w \left(T_{w,in} - T_0\right)/3600 + Q = c_w \rho_w F_w \left(T_{w,out} - T_0\right)/3600 \tag{7}$$

$$c_w \rho_w F_w \cdot ln\left(T_{w,in}/T_0\right)/3600 + Q/T_{w,ave} + S_{g,w} = c_w \rho_w F_w \cdot ln\left(T_{w,out}/T_0\right)/3600 \tag{8}$$

$$E_{pump} + X_{w,in} - X_{cons,w} = X_w + X_{w,out} \tag{9}$$

By combining (3), (6), and (9), the exergy-budget equation for the entire system can be deduced, as given by (10).

$$\left(E_{fan} + E_{pump}\right) + \left(X_{w,in} - X_{w,out}\right) - \left(X_{cons,a} + X_{cons,HE} + X_{cons,w}\right) = X_{a,sup} - X_{a,re} \tag{10}$$

The rate of overall exergy inputs is the sum of the fan power E_{fan} [kW], pump power E_{pump} [kW], and the rate of net exergy input from the chiller to the chilled water circuit, expressed as the difference between $X_{\mathrm{w,in}}$ [kW] and $X_{\mathrm{w,out}}$ [kW]. $X_{\mathrm{w,in}}$ [kW] and $X_{\mathrm{w,out}}$ [kW] represent the rate of exergy carried by the supply water leaving the chiller and that carried by the return water coming from the FCU and going towards the chiller, respectively, as shown in (11) and (12).

$$X_{w,in} = \left(c_w \rho_w F_w \left(T_{w,in} - T_0\right) - c_w \rho_w F_w T_0 \cdot ln\left(T_{w,in}/T_0\right)\right)/3600 \tag{11}$$

$$X_{w,out} = \left(c_w \rho_w F_w \left(T_{w,out} - T_0\right) - c_w \rho_w F_w T_0 \cdot ln\left(T_{w,out}/T_o\right)\right)/3600 \tag{12}$$

The rate of overall exergy consumptions is the sum of $X_{\mathrm{cons,a}}$ [kW], $X_{\mathrm{cons,HE}}$ [kW], and $X_{\mathrm{cons,w}}$ [kW], which are the rates of exergy consumed in the air channel within the FCU, in the tube wall between the air and water, and in the water within the chilled water circuit, respectively, as shown in (13), (14), and (15).

$$X_{cons,a} = S_{g,a} T_0 \tag{13}$$

$$X_{cons,HE} = S_{g,HE} T_0 \tag{14}$$

$$X_{cons,w} = S_{g,w} T_0 \tag{15}$$

The rate of exergy output is the rate of net exergy output from the FCU to the indoor space, expressed as the difference between $X_{\mathrm{a,sup}}$ [kW] and $X_{\mathrm{a,re}}$ [kW], which are the rates of exergy carried by supply air leaving the FCU and that carried by the return air coming into the FCU, respectively, as shown in (16) and (17).

$$X_{a,sup} = \left(c_a \rho_a F_a \left(T_{a,sup} - T_0\right) - c_a \rho_a F_a T_0 \cdot ln\left(T_{a,sup}/T_0\right)\right)/3600 \tag{16}$$

$$X_{a,re} = \left(c_a \rho_a F_a \left(T_{a,re} - T_0\right) - c_a \rho_a F_a T_0 \cdot ln\left(T_{a,re}/T_0\right)\right)/3600 \tag{17}$$

X_a [kW] and X_w [kW] in Equations (3), (6), and (9) are the rate of exergy absorbed by air owing to heat release and that discharged from the chilled water owing to heat absorption, respectively, as shown in (18) and (19).

$$X_a = \left(1 - T_0/T_{a,ave}\right)(-Q) \tag{18}$$

$$X_w = \left(1 - T_0/T_{w,ave}\right)(-Q) \tag{19}$$

2.2. Four variable-flow control modes

Four methods of variable-flow control for the chilled water circuit were assumed, as shown in Figs. 3–6.

In the case of throttle-valve control (TV control), the water flow rate is adjusted only by the throttle valve, with no change in the pump frequency. The other three methods use variable-frequency control.

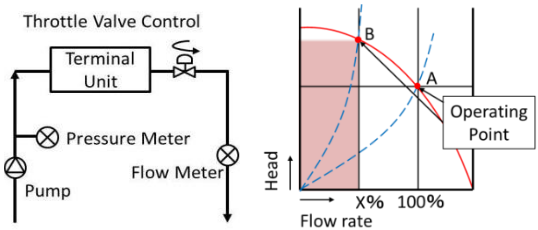


Fig. 3 Throttle-valve (TV) control

Under constant-pressure control (CP control), the pump maintains a constant discharge pressure on the chilled-water circuit.

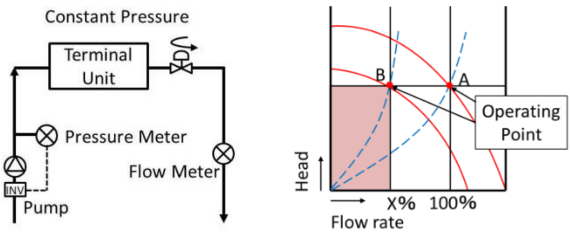


Fig. 4 Constant-pressure (CP) control

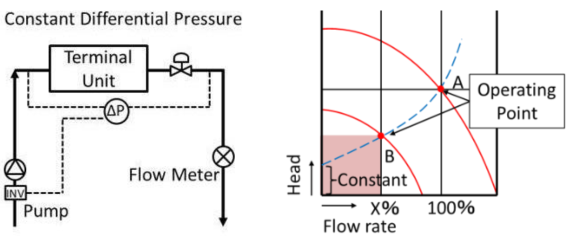


Fig. 5 Constant-differential-pressure (CDP) control

Under constant-differential-pressure control (CDP control), the pump maintains a constant pressure difference at a specific interval for the chilled water system. Under predictive-system-curve control (PSC control), the discharge pressure of the pump is determined from a predictive system curve, which describes the relationship between the pump discharge pressure and the water flow rate demand of the chilled water circuit.

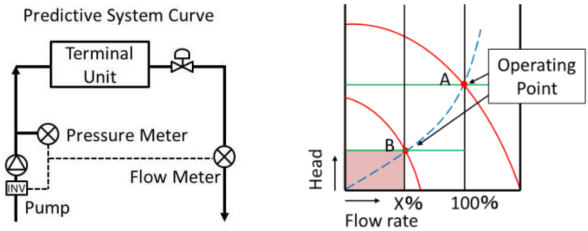


Fig. 6 Predictive-system-curve (PSC) control

2.3. Heat-transfer characteristics of FCU

Dehumidification was not taken into account in this study. The heat-transfer characteristic of the FCU can be represented by a semi-empirical formula with respect to the relationship between the relative heat-transfer rate Q/Q^* and the relative flow rate F/F^*, as given by (20) [5]. The characteristic coefficient of the FCU α [-] is defined by (21), in which $T^*_{w,in}$ [K] and $T^*_{w,out}$[K] represent the inlet and outlet temperatures of the chilled water circuit, respectively, and $T^*_{a,re}$ [K] is the return air temperature for the design condition.

$$Q/Q^* = 1\Big/\left[1 + \alpha\left(1\big/\left(F/F^*\right) - 1\right)\right] \tag{20}$$

$$\alpha = 0.6\left(T^*_{w,in} - T^*_{w,out}\right)\Big/\left(T^*_{w,in} - T^*_{a,re}\right) \tag{21}$$

2.4. Calculation parameters

The ambient temperature and return air temperature were assumed to be constant at 32°C and 27°C, respectively. The temperatures of the supply air and supply water were respectively set to 17°C and 7°C in Case 1 and 12°C and 22°C in Case 2. The return water temperature was assumed to be 5°C higher than that of the supply water temperature under full-load conditions, but it can vary with the heat-transfer rate in the FCU and with the variable-flow control method under a partial-load condition. The design heat-transfer rate was assumed to be 15 kW. The flow rate, pumping head, and pump power under full-load conditions were assumed to be 2.7 m^3/h, 44 m, and 0.53 kW, respectively. The fan power in Cases 1 and 2 under full-load was simply assumed to be 0.2 kW and 0.4 kW, respectively. Both the system and pump performance were characterized by respective quadratic curves. For TV control, when the flow rate was zero, the pumping head was assumed to be 1.4 times the value of the rated pumping head. The pumping head, based on a constant pressure difference in CDP control, was assumed to be 15 m.

2.5. Results of calculations

The rate of exergy inputs and outputs under full-load (15 kW) and partial-load (12 kW) conditions for Cases 1 and 2 are shown in Figs. 7 and 8, respectively.

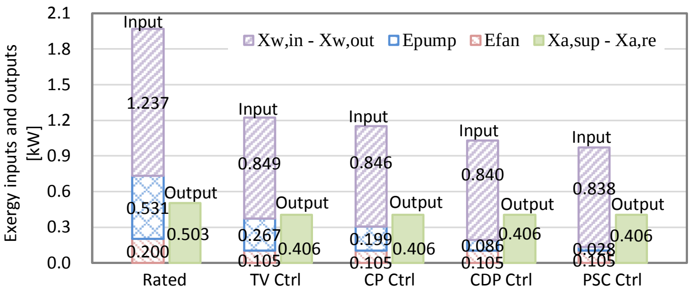


Fig. 7 Rate of exergy inputs and exergy outputs (Case 1)

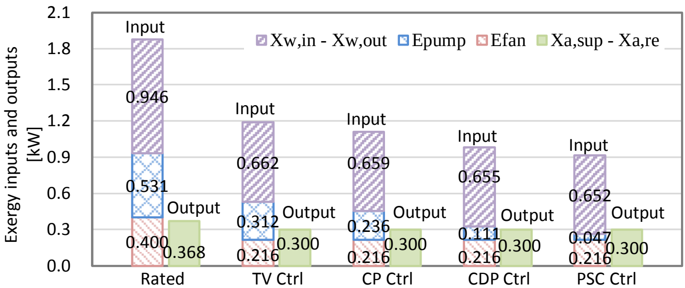


Fig. 8 Rate of exergy inputs and exergy outputs (Case 2)

The differences between the rate of exergy inputs and outputs in the two cases are the rate of exergy consumptions, as shown in Fig. 9. In these figures, the vertical bars from left to right represent full-load mode, and the TV, CP, CDP, and PSC control modes.

It can be seen that the rate of exergy inputs and consumptions under any partial-load condition are significantly smaller than those under a full-load condition, and those under any variable-frequency control mode are smaller than those under throttle-valve control. In the variable-frequency control mode, the rate of exergy inputs decrease in the following order: constant-pressure control, constant-differential-pressure control, and predictive-system-curve control. As for the effect of the supply water temperature, the rate of exergy inputs is smaller at 12°C than at 7°C.

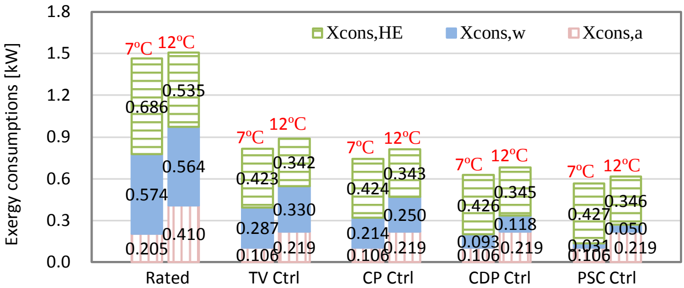
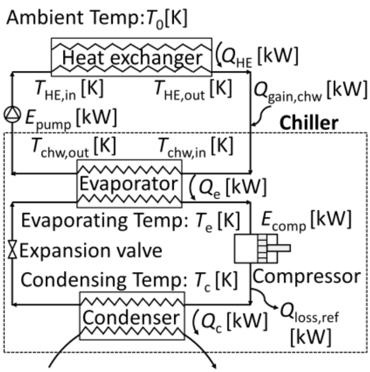


Fig. 9 Rate of exergy consumptions of the two cases

3. Exergy analysis of an actual heat pump system

3.1. System to be analyzed and its corresponding exergy budget

A part of a heat pump system in a teaching building at the University of Tokyo, as shown in Fig. 10, was analyzed. The corresponding flow of exergy through the system is shown in Fig. 11.



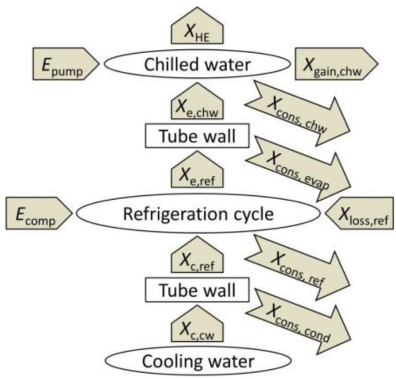


Fig. 10 Analysis object in the heat pump system   Fig. 11. Flow of exergy through the system

The system consists of four subsystems: the tube wall between the cooling water and refrigerant, the refrigeration cycle, the tube wall between the refrigerant and chilled water, and the chilled water circuit. The energy budget, entropy budget, and exergy budget of the four subsystems are shown in (22) to (33). In these equations, the mean cooling water temperature $T_{cw,ave}$ [K], the mean refrigerant temperature $T_{ref,ave}$ [K], the mean chilled temperature $T_{chw,ave}$ [K] in the evaporator, and the mean chilled water temperature $T_{HE,ave}$ [K] in the heat exchanger are approximated by the mean value of the inlet temperature $T_{\mathrm{cw,in}}$ [K] and outlet temperature $T_{\mathrm{cw,out}}$ [K] of the cooling water, that of the evaporating temperature T_{e} [K] and condensing temperature T_{c} [K], that of the inlet temperature $T_{\mathrm{chw,in}}$ [K] and outlet temperature $T_{\mathrm{chw,out}}$ [K] of chilled water in the evaporator, and that of the inlet temperature $T_{\mathrm{HE,in}}$ [K] and outlet temperature $T_{\mathrm{HE,out}}$ [K] of chilled water in the heat exchanger, respectively. $S_{\mathrm{g,cond}}$ [kW/K], $S_{\mathrm{g,ref}}$ [kW/K], $S_{\mathrm{g,evap}}$ [kW/K], and $S_{\mathrm{g,chw}}$ [kW/K] are the rate of entropy generation in the tube wall between the cooling water and refrigerant, in the refrigeration cycle, in the tube wall between the refrigerant and chilled water, and in the chilled water circuit, respectively.

$$Q_c = Q_c \tag{22}$$

$$Q_c / T_c + S_{g,cond} = Q_c / T_{cw,ave} \tag{23}$$

$$X_{c,cw} - X_{cons,cond} = X_{c,ref} \tag{24}$$

$$Q_e + E_{comp} = Q_c + Q_{loss,ref} \tag{25}$$

$$Q_e / T_e + S_{g,ref} = Q_c / T_c + Q_{loss,ref} / T_{ref,ave} \tag{26}$$

$$E_{comp} + X_{c,ref} + X_{loss,ref} - X_{cons,ref} = X_{e,ref} \tag{27}$$

$$Q_e = Q_e \tag{28}$$

$$Q_e / T_{chw,ave} + S_{g,evap} = Q_e / T_e \tag{29}$$

$$X_{e,ref} - X_{cons,evap} = X_{e,chw} \tag{30}$$

$$Q_{HE} + E_{pump} + Q_{gain,chw} = Q_e \tag{31}$$

$$Q_{HE} / T_{HE,ave} + Q_{gain,chw} / T_{chw,ave} + S_{g,chw} = Q_e / T_{evap,ave} \tag{32}$$

$$E_{pump} + X_{e,chw} - X_{cons,chw} = X_{HE} + X_{gain,chw} \tag{33}$$

$X_{\mathrm{c,ref}}$ [kW], $X_{\mathrm{loss,ref}}$ [kW], $X_{\mathrm{e,ref}}$ [kW], $X_{\mathrm{e,chw}}$ [kW], and $X_{\mathrm{gain,chw}}$ [kW] in (24), (27), (30), and (33) are the rate of exergy absorbed by the refrigerant owing to heat release Q_{c} in the condenser, that absorbed by the refrigerant owing to net heat loss $Q_{\mathrm{loss,ref}}$ [kW] from the refrigeration cycle to the surroundings, that discharged from the refrigerant owing to heat absorption Q_{e} [kW] in the evaporator, that absorbed by the chilled water owing to heat release Q_{e} in the evaporator, and that discharged from the chilled water owing to heat invasion $Q_{\mathrm{gain,chw}}$ [kW] from the surroundings to the chilled water circuit, respectively, as shown in Equations (34)-(38).

$$X_{c,ref} = \left(1 - T_0 / T_c\right)\left(-Q_c\right) \tag{34}$$

$$X_{loss,ref} = \left(1 - T_0 / T_{ref,ave}\right)\left(-Q_{loss,ref}\right) \tag{35}$$

$$X_{e,ref} = \left(1 - T_0 / T_e\right)\left(-Q_e\right) \tag{36}$$

$$X_{e,chw} = \left(1 - T_0 / T_{chw,ave}\right)\left(-Q_e\right) \tag{37}$$

$$X_{gain,chw} = \left(1 - T_0 / T_{chw,ave}\right)\left(-Q_{gain,chw}\right) \tag{38}$$

By combining (24), (27), (30), and (33), the exergy-budget equation for the entire system can be deduced, as given by (39). The rate of overall exergy inputs is the sum of the compressor power E_{comp} [kW], pump power E_{pump} [kW], and the rate of exergy discharged from the cooling water owing to heat absorption Q_{c} [kW] in the condenser, expressed as $X_{\text{c,cw}}$ [kW] as shown in (40).

$$\begin{aligned}&\left(E_{comp}+E_{pump}+X_{c,cw}\right)-\\&\quad\left(X_{gain,chw}-X_{loss,ref}+X_{cons,cond}+X_{cons,ref}+X_{cons,evap}+X_{cons,chw}\right)=X_{HE}\end{aligned} \tag{39}$$

$$X_{c,cw}=\left(1-T_0/T_{cw,ave}\right)\left(-Q_c\right) \tag{40}$$

The rate of overall exergy consumptions is the rate of net exergy discharging from system to the environment ($X_{\text{gain,chw}} - X_{\text{loss,ref}}$) [kW], and the sum of $X_{\text{cons,cond}}$ [kW], $X_{\text{cons,ref}}$ [kW], $X_{\text{cons,evap}}$ [kW], and $X_{\text{cons,chw}}$ [kW], which is the rate of exergy consumed in the tube wall between the cooling water and refrigerant, in the refrigeration cycle, in the tube wall between the refrigerant and chilled water, and in the chilled water circuit, respectively, as shown in (41)-(44).

$$X_{cons,cond}=S_{g,cond}T_0 \tag{41}$$

$$X_{cons,ref}=S_{g,ref}T_0 \tag{42}$$

$$X_{cons,evap}=S_{g,evap}T_0 \tag{43}$$

$$X_{cons,chw}=S_{g,chw}T_0 \tag{44}$$

The rate of exergy output is the rate of exergy discharged from the chilled water owing to heat absorption Q_{HE} [kW] in the heat exchanger, expressed as X_{HE} [kW] as shown in (45).

$$X_{HE}=\left(1-T_0/T_{HE,ave}\right)\left(-Q_{HE}\right) \tag{45}$$

The exergy efficiency β [-] and the rate of net exergy consumption per unit rate of exergy output η [-] are shown in (46) and (47), respectively.

$$\beta=X_{HE}\Big/\left(E_{comp}+E_{pump}+X_{c,cw}\right) \tag{46}$$

$$\eta=\left(E_{comp}+E_{pump}-X_{HE}\right)\Big/X_{HE} \tag{47}$$

3.2. Operating data and exergy analysis

The operating data and exergy budget before and after the variable-frequency transformation of the chilled water pump are compared in Figs. 12 and 13, respectively.

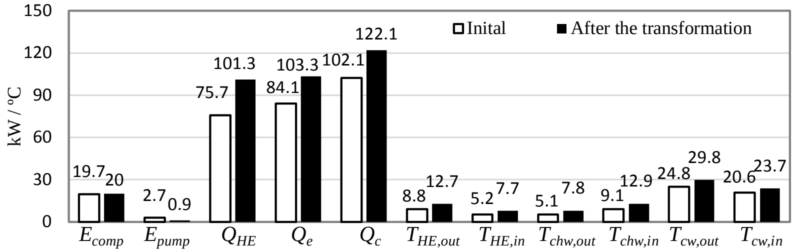


Fig. 12 Operating data before and after transformation

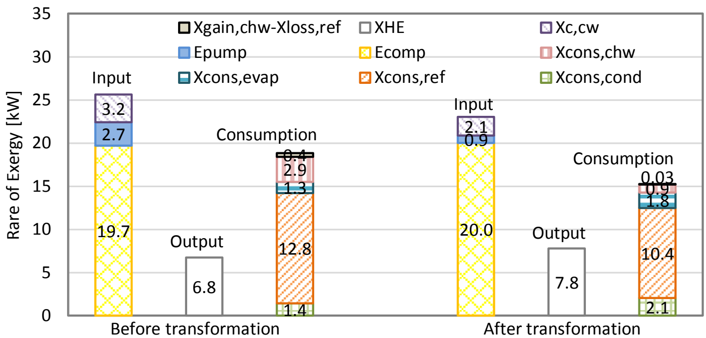


Fig. 13 Exergy budget before and after transformation

It was observed that after the rise in temperature of the chilled water and the variable-frequency transformation of the chilled water pump, the rate of exergy consumed in the chilled water circuit was reduced considerably from 2.9 kW to 0.9 kW. The exergy efficiency β was improved from 26.4% to 33.8%. The rate of net exergy consumption per unit rate of exergy output η was also reduced 27.6% from 2.32 to 1.68.

4. Conclusion

In this study, the exergy analysis of an assumed chilled water circuit at two supply water temperatures under four variable-flow control modes was carried out. Subsequently, the exergy budget of an actual chilled water circuit before and after the variable-frequency transformation of the chilled water pump were compared. The results of the calculations indicate that the use of variable-frequency control and a higher chilled water temperature can effectively reduce the exergy consumption rate and improve the exergy efficiency.

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