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Performance analysis of air-water dual source heat pump water heater with heat recovery

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A new air-water dual source heat pump water heater with heat recovery is proposed. The heat pump system can heat water by using a single air source, a single water source, or air-water dual sources. The water is first pre-heated by waste hot water, then heated by the heat pump. Waste heat is recovered by first preheating the cold water and as water source of the heat pump. According to the correlated formulas of the coefficient of performance of air-source heat pump and water-source heat pump, and the gain coefficient of heat recovery-preheater, the formulas for the coefficient of performance of heat pump in six operating modes are obtained by using the dimensionless correspondence analysis method. The system characteristics of heat absorption and release associated with the heat recovery-preheater are analyzed at different working conditions. The developed approaches can provide reference for the optimization of the operating modes and parameters. The results of analysis and experiments show that the coefficient of performance of the device can reach 4–5.5 in winter, twice as much as air source heat pump water heater. The utilization of waste heat in the proposed system is higher than that in the system which only uses waste water to preheating or as heat source. Thus, the effect of energy saving of the new system is obvious. On the other hand, the dimensionless correspondence analysis method is introduced to performance analysis of the heat pump, which also has theoretical significance and practical value.

heat pump water heater, waste heat recovery, dual source heat pump, correspondence analysis method, performance analysis

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1 Introduction

Energy saving and emissions reduction are the inevitable requests of the sustainable development of human society. The energy grades of electrical energy, chemical energy of fuel, mechanical energy are higher than that of thermal energy, and the energy grade of high-temperature thermal energy is higher than that of low-temperature thermal energy [1]. Therefore, we should focus on not only saving energy in quantity but also the effective use of energy in quality. Heat pump water heater (HPWH) is an important unit for energy saving. It consumes high grade electrical energy and through the refrigeration of heat pump circulation, absorbs low grade thermal energy from the surrounding air or water to produce 50°C–55°C domestic hot water. The coefficient of performance (*COP*) of a HPWH is generally about 2–4.5 in a year, because of the irreversible losses in a real cycle. Hepbasli and Kalinci [2] have given a detailed review of HPWH systems. Air-source heat pump is suitable for a wide range of applications, but has a very low *COP* in winter.

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Water-source heat pump can recover heat from waste hot water in winter, and therefore has high *COP*, but it will not be able to supply hot water if waste water runs out. Air-water dual source HPWH takes advantage of both high efficiency as water-source heat pump and wide range of applications as air-source heat pump. It is suitable for the place requiring hot water such as bathhouses and hotels. Air-water dual source HPWH can take waste heat from waste bathing water with temperature of 31°C–36°C to product 50°C–55°C hot water by a refrigeration cycle. Ito and Miura [3] investigated a heat pump using water and air heat sources in parallel, the results showed that it has a higher *COP* than that of a single air-source heat pump. Chen [4] proposed a water heater for bath hot water suppling by using waste bathing water to preheat cold water and dual source heat pump in 2001. Zhu [5] developed a device for bath waste water recovery. Zheng [6] and Huang [7] also proposed similar devices. Zheng [6] analyzed the heat pump system with bath waste water heat recovery, in which the copper coil heat exchanger was dipped in the waste water pool for heat recovery to preheat cold water, and the preheated water was then heated by a heat pump. The maximum achievable *COP* of this system is 5.2, but it is not involved in the discussion of dual source heat pump and the two-stage waste heat recovery. These patents and research work show that the heat pump system with bath waste water heat recovery has great potential in energy saving and a good market prospect. It is necessary to carry out further research.

2 System process and working principle of new air-water dual source HPWH with waste heat cascade recovery

Figure 1 is the principle diagram of the new air-water dual source heat pump water heater with waste heat cascade recovery. In Figure 1, #1 is the compressor, #2 is the first four-way directional valve, #3 is the hot water heat exchanger, #4 is the liquid storage tank, #5 is the filter, J is the throttle device, #6 is the second four-way directional valve, #7 is the water source heat exchanger, #8 is the air source heat exchanger, #8f is the fan, #9 is the low pressure gas-liquid separator, #10 is the hot water circulating pump, #11 is the waste heat recovery-preheater, #12 is the waste hot water pump, DF1 is the inlet water electromagnetic valve, ZF1 is the inlet water flow regulator, TF1 is the outlet water flow regulator, D1 and D2 are waterway check valves, F1 is the inlet water bypass-valve , and #13 is the storage water tank.

The device has three refrigerant cycles as follows.

1) Air-source cycle. The refrigerant cycle is: The compressor, the first four-way directional valve, the hot water heat exchanger (where refrigerant releases heat to hot water and is condensed into liquid), the liquid storage tank, the throttling device, the second four-way directional valve, the air source heat exchanger, the first four-way directional valve, the low pressure gas liquid separator, and the compressor. In this cycle, the system absorbs heat from the air to produce hot water.

2) Water-source and air-source cycle in series. The refrigerant cycle is: the compressor, the first four-way directional valve, the hot water heat exchanger, the liquid storage tank, the throttling device, the second four-way directional valve, the water source heat exchanger, the second four-way directional valve, the air source heat exchanger, the first four-way directional valve, the low pressure gas liquid separator, and the compressor. In this cycle, the heat pump can absorb heat both from the air and the waste water to produce hot water when the waste hot water pump and the fan are both running; and it can be a water source heat pump (WSHP) for water heating when the fan is stopped.

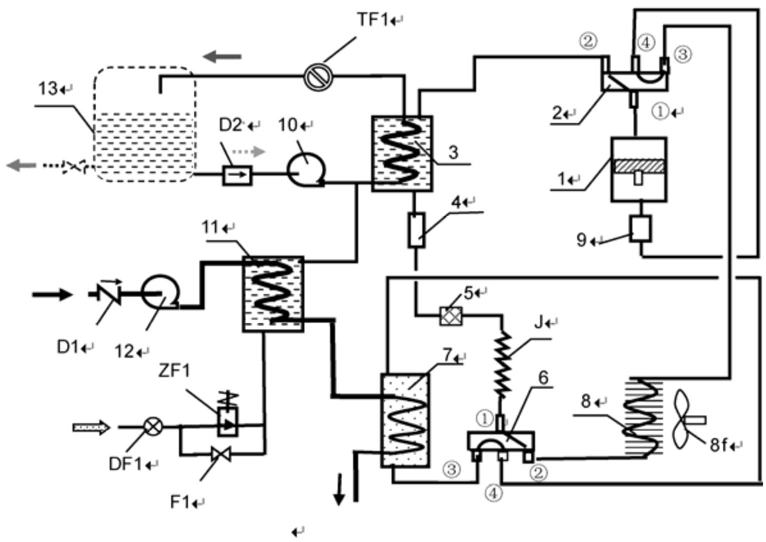

Figure 1 Principle diagram of the new air-water dual source heat pump water heater with waste heat cascade recovery.

3) Defrost cycle. The refrigerant flow is in the direction opposite to the air source cycle through the actions of the first and the second four-way directional valves. The fan of the air source heat exchanger is stopped. The high temperature refrigerant vapor will release heat to defrost the the air source heat exchanger. The hot water circulating pump keeps running. In this cycle, the system absorbs heat from the hot water in the storage water tank to defrost the air source heat exchanger.

Figure 2 is the heat transfer process between the clear water and waste water in the proposed system. The clear water of temperature T_1 and flow m_w first exchanges heat with the waste water of temperature T_4 and flow m_{pw} through the heat recovery-preheater, where it absorbs heat Q_w and its temperature is heated to T_2. After that, the preheated clear water flows into the hot water heat exchanger of the heat pump, where it absorbs heat Q_h and its temperature is heated to T_3. At the same time, the waste water releases heat Q_w in the heat recovery-preheater, and is cooled to T_5, then it flows into the water heat exchanger of the heat pump for the second heat recovery and is cooled to T_6 by releasing heat Q_e. Finally, it drains into the trench.

3 System characteristic analysis

The *COP* and heating capacity of a heat pump vary with the air temperature, the inlet water temperature, the temperature of the water, the water temperature of user requirement, the operation modes, the operation parameters, etc. The operation modes can be optimized through the characteristic analysis.

3.1 Operation modes of the device

The operation modes of this device are:

a) Air source heat pump (ASHP) mode;

b) ASHP with heat recovery-preheating mode;

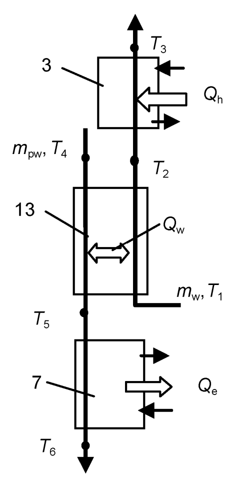

Figure 2 Heat transfer process between the clear water and waste water.

c) WSHP with heat recovery-preheating mode;

d) Combination mode 1: running WSHP with heat recovery-preheating mode when there is enough waste water, and running single ASHP mode when waste water runs out;

e) Combination mode 2: running ASHP with heat recovery-preheating mode when there is enough waste water, and running single ASHP mode when waste water runs out;

f) Combination mode 3: running air-water dual source heat pump with heat recovery-preheating mode when there is enough waste water, and running single ASHP mode when waste water runs out.

3.2 Basic thermodynamic relations

The heat transfer equation for the waste heat recovery- preheater is

$$Q_w = m_{pw} c_w (T_4 - T_5) = m_w c_w (T_2 - T_1) = K_w A \Delta T_w, \quad (1)$$

where K_w, A and ΔT_w are the heat transfer coefficient, the heat transfer area and the temperature difference of the waste heat recovery-preheater, respectively. A is a constant for a given device. K_w will be decreased because of the fouling, thus the waste water pipe of the waste heat recovery-preheater needs to be regularly cleaned. ΔT_w is generally about 3°C–5°C.

The energy equation of a heat pump is

$$Q_h = Q_e + W, \quad (2)$$

where Q_e is the heat absorbed in the heat pump evaporator, which can be one of the evaporators in the cycles of single air source, single water source and air-water dual source. The absorbed heat by the evaporator in a WSHP is

$$Q_{e,w} = m_{pw} c_w (T_5 - T_6) = K_{e,w} A_{e,w} \Delta T_{e,w}, \quad (3)$$

where Q_h is the output heat of the heat pump. If the clear water passes by the heat recovery-preheater, Q_h can be calculated by the following equations:

$$Q_h = m_w c_w (T_3 - T_1), \quad (4a)$$

$$Q_h = m_w c_w (T_3 - T_2), \quad (4b)$$

where T_2 can be obtained by eq. (1).

3.3 Dimensionless correspondence analysis method

The main methods to know the performance of a heat pump are experiment and modeling analysis method [8–11], which are both limited in some degree. Dimensionless correspondence analysis method is a generalized method to obtain the basic relationships of the research objects, because similar things usually have a certain degree of similarity, such as the corresponding state principle for the thermodynamic properties, similarity analysis for heat transfer and fluid mechanics. Characteristics of vapor compression refrigeration/heat pump cycle also have a certain similarity. Therefore, the dimensionless correspondence analysis method is introduced to analysis of the performance of these devices.

1) Dimensionless correspondence analysis method

Dimensionless corresponding parameters can be defined as the ratio of the variable parameters at a certain working conditions to the same parameters at the standard working conditions. For example, the dimensionless corresponding theoretical coefficient of performance $\tilde{COP}_{\rm i}$ can be defined as

$$\tilde{COP}_{\rm i} = COP_{\rm i} / COP_{\rm i,0}, \tag{5}$$

where subscript "0" indicates that the parameter is at the standard working conditions, and subscript "i" indicates that the parameter is in the theoretical cycle. The temperatures of air source, water source, inlet water, and outlet water of a heat pump have been defined in the national standards for the standard working conditions. The deviations of the real performances from the theoretical performances for a heat pump are different for different manufacturers. It can be modified by the equipment coefficient η, which is defined as the ratio of the real COP to the theoretical COP_i:

$$\eta = COP / COP_{\rm i}. \tag{6}$$

Equipment coefficient at the standard working conditions is η_0. Heat pumps in the same model would have the similar heat exchanger area and system structure, and therefore the same equipment coefficients. Thus, the equipment coefficient can be eliminated through the comparison of the actual COP at certain working conditions and the COP_0 at the standard working conditions, and the actual dimensionless corresponding $\tilde{COP}$ can be obtained as

$$\tilde{COP} = COP / COP_0. \tag{7}$$

The equation of $\tilde{COP}$ can be obtain by modifying the curve of $\tilde{\tau}_{\rm pw} = \tau_{\rm pw} / \tau_{\rm T} = 1/J$ according to the actual performance of a heat pump. The actual COP is

$$COP = \tilde{COP} \cdot COP_0 = \eta_0 \cdot \tilde{COP} \cdot COP_{0,\rm i}. \tag{8}$$

$\tilde{COP}$ has generality, and η_0 reflects the difference of the performance of different heat pumps. Therefore, the estimation of the performance of a heat pump at different working conditions becomes simple by using the correspondence analysis method.

2) Dimensionless working time of waste water $\tilde{\tau}_{\rm pw}$

In bathing place, the amount of waste water $M_{\rm pw}$ equals the amount of produced hot water $M_{\rm w}$ in a day. But the production of waste water is not stable. If a working time is defined by the average flow of clear water as $\tau_{\rm T} = M_{\rm w} / m_{\rm w}$, then the working time of waste water flowing through the heat recovery-preheater and the WSHP should be

$$\tau_{\rm pw} = \frac{M_{\rm pw}}{m_{\rm pw}} = \frac{M_{\rm w}}{J m_{\rm w}} = \frac{\tau_{\rm T}}{J}. \tag{9}$$

The dimensionless working time with waste water is

$$\tilde{\tau}_{\rm pw} = \tau_{\rm pw} / \tau_{\rm T} = 1/J. \tag{10}$$

The dimensionless working time without waste water is

$$\tilde{\tau}_{\rm a} = 1 - 1/J, \tag{11}$$

$$J = m_{\rm pw} / m_{\rm w}. \tag{12}$$

3) Corresponding total production of hot water $\tilde{M}_{\rm w}$ and COP of the system

If the hot water production M_0 by consuming electric power W with efficiency of 1 is taken as the reference parameter, then the corresponding total hot water production in the dimensionless unit time (which is also the $COP_{\rm s}$ of the system) is

$$\tilde{M}_{\rm w} = M_{\rm w} / M_0 = COP_{\rm s}, \tag{13}$$

where M_0 can be obtained by the following equation:

$$M_0 = \frac{W}{c_{\rm w}(T_3 - T_1)} \quad \text{or} \quad M_0 c_{\rm w}(T_3 - T_1) = W. \tag{14}$$

3.4 Coefficient of performances for six operating modes

1) $COP_{\rm a}$ of ASHP

The corresponding theoretical coefficient of performance of ASHP is defined as

$$\tilde{COP}_{\rm a,i} = COP_{\rm a,i} / COP_{\rm a,i,0}, \tag{15}$$

where subscript "a" indicates the ASHP cycle, $COP_{\rm a,i}$ is the coefficient of performance of ASHP in theoretical cycle, $COP_{\rm a,i,0}$ is the theoretical coefficient of performance of ASHP at standard working conditions. Taking refrigerant R22 and the theoretical cycle at the standard working conditions (evaporating temperature $T_{\rm e}$ is 15°C and condensing temperature is 55°C) for example, the $COP_{\rm a,i,0}$ is 6.87. But the actual coefficient of performance $COP_{\rm a,0}$ of an ASHP accorded with national standard is 3.8. Here we define an equipment coefficient of ASHP $\eta_{\rm a,0}$ as

$$\eta_{\rm a,0} = COP_{\rm a,0} / COP_{\rm a,i,0}. \tag{16}$$

In this case, $\eta_{\rm a,0}$ is 0.553. Thus, in spite of working under the same standard working conditions, the actual coefficient of performance has a great difference to the theoretical coefficient of performance. By comparing the actual $COP_{\rm a}$ to the $COP_{a,0}$ at certain working conditions, we can obtain the dimensionless corresponding actual coefficient of performance of ASHP as

$$\tilde{COP}_a = COP_a / COP_{a,0}. \tag{17}$$

Figure 3 is the relationship between $\tilde{COP}_a$ and T_e of ASHP. The curve of $\tilde{COP}_a$ is very close to the curve of $\tilde{COP}_{a,i}$ because the $\tilde{COP}_a$ can eliminate the influence of the structural difference of the same model heat pump by eliminating the equipment coefficient of heat pump $\eta_{a,0}$.

If a heat pump running at high-temperature working conditions, the increasing rate $\tilde{COP}_a$ would decrease gradually because the growth rate of the evaporator's heat transfer capacity is lower than that of the compressor suction-density. The fitting formula of $\tilde{COP}_a$ based on $\tilde{COP}_{a,i}$ is

$$\tilde{COP}_a = \tilde{T}_e^8, \tag{18}$$

where $\tilde{T}_e$ is the corresponding evaporation temperature using the evaporation temperature 288 K in the standard working conditions as the reference parameter:

$$\tilde{T}_e = T_e/288 = (T_a - \Delta T_e)/288, \tag{19}$$

where the unit of T_a and T_e is K, ΔT_e is the average temperature difference of the evaporator, which is about 5°C–7°C.

The experimental point data and the curve of $\tilde{COP}_a$ calculated by eq. (18) are shown in Figure 3. The solid square points are the experimental data of the latest prototype. In this case, the heating capacity is measured as 40.5 kW with the input power of 9.2 kW at the working condition of the air temperature 23.4°C, and thus COP_a is 4.5. At the standard working conditions, $COP_{a,0}$ is 3.85. The solid round points are the experimental results of an ASHP manufactured by a factory in Guangdong. Its COP_a is 4.3 at the air temperature 29°C, and $COP_{a,0}$ is 3.4 at the standard working conditions.

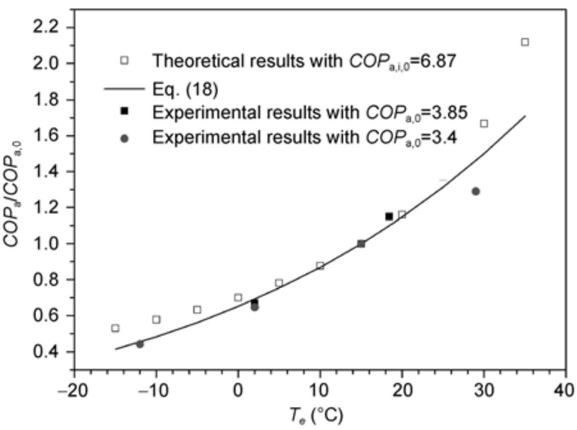

Figure 3 Relationship between $\tilde{COP}_a$ and T_e of ASHP.

2) COP_{a+r} of ASHP with heat recovery-preheating

The coefficient of performance of ASHP with heat recovery-preheating is

$$COP_{a+r} = \frac{T_3 - T_1}{T_3 - T_2} COP_a = \left(1 + \frac{T_2 - T_1}{T_3 - T_2}\right) COP_a. \tag{20}$$

Define the gain coefficient of heat recovery-preheater as

$$h = COP_{a+r} / COP_a = (T_3 - T_1)/(T_3 - T_2). \tag{21}$$

3) COP_{w+r} of WSHP with heat recovery-preheating

If there is enough waste water, waste water should flow through the heat recovery-preheater first, then flow into the evaporator of WSHP. The coefficient of performance of WSHP with heat recovery-preheating is

$$COP_{w+r} = \frac{T_3 - T_1}{T_3 - T_2} COP_w = \left(1 + \frac{T_2 - T_1}{T_3 - T_2}\right) COP_w, \tag{22}$$

where COP_w is the coefficient of performance of WSHP.

One of the differences between WSHP and ASHP is that the temperature of water would be continuously decreasing when flowing through the evaporator of WSHP, and the evaporation temperature of refrigerant is lower than the outlet water temperature T_6 of evaporator. Therefore, COP_w is mainly related to T_6. For the water source direct-heating heat pump, the standard working conditions give the inlet and outlet water temperatures of condenser and the inlet water temperature of evaporator as 15°C, 55°C and 15°C, respectively, but the outlet water temperature and water flow are not given for evaporator. Thus, the water flow of evaporator could be very large to improve the performance of WSHP, and the temperature difference between inlet and outlet water is therefore only 4°C–5°C. The ratio of evaporator water flow to condenser water flow is also up to 8. Thus, the corresponding coefficient of performance of water source direct-heating heat pump $\tilde{COP}_w$ can be correlated with T_6 or the average of inlet and outlet water temperature T_m. From eq. (18), the fitting equation of $\tilde{COP}_w$ is

$$\tilde{COP}_w = \tilde{T}_{e,w}^8, \tag{23}$$

$$\tilde{T}_{e,w} = (T_6 - \Delta T_{e,w})/288. \tag{24}$$

The coefficient of performance of a direct-heating WSHP is normally lower than that of a circulation heating WSHP. The reason is that the condensation pressure of a direct-heating WSHP is always about the saturation pressure of the hot water temperature 55°C, while the compression ratio is lower than that of a direct-heating WSHP in most of the time for a circulation heating WSHP because the water temperature is gradually rising in its condenser. At the standard working conditions, the *COP* is higher than 4.0 for a circulation heating WSHP, while lower than 3.5 for a direct-heating WSHP.

In order to utilize the rest heat of waste water flowing out of the heat recovery-preheater, the COP_{w} of this system running in the WSHP mode is restrained by the flow ratio of clear water and waste water *J*. According to eq. (2), the COP_{w} of WSHP can be derived as

$$COP_{\mathrm{w}} = \frac{Q_{\mathrm{h}}}{W} = 1 + \frac{Q_{\mathrm{e,w}}}{Q_{\mathrm{h}}} \cdot \frac{Q_{\mathrm{h}}}{W} = 1 + \frac{J\left(T_5 - T_6\right)}{T_3 - T_2} COP_{\mathrm{w}},$$

$$COP_{\mathrm{w}} = 1 \Big/ \left(1 - \frac{T_5 - T_6}{T_3 - T_2} \cdot J\right), \tag{25}$$

where T_2, T_5 and T_6 are the unknown parameters. According to eqs. (1) and (8), there is

$$T_5 = T_4 - (T_2 - T_1) / J. \tag{26}$$

Combining the number of transfer units (*NTU* [12]) and eq. (3) for heat recovery-preheater, we can obtain

$$NTU = \frac{kA}{m_{\mathrm{w}} c_{\mathrm{w}}} = \frac{T_2 - T_1}{\Delta T_{\mathrm{w}}}, \tag{27}$$

where

$$\Delta T_{\mathrm{w}} = \frac{\left(T_5 - T_1\right) - (T_4 - T_2)}{\ln \dfrac{(T_5 - T_1)}{(T_4 - T_1)}}. \tag{28}$$

When $J = 1$, $T_5 - T_1 = T_4 - T_2 = \Delta T_{\mathrm{w1}}$. Thus, the value of *NTU* can be determined with a given ΔT_{w1}. T_2, T_5, T_6 and COP_{w} can be obtained by eqs. (25)–(28) with iteration method.

4) COP_{S1} of combination mode 1

When the waste water is supplied in the time of $\tilde{\tau} = 1/J$, WSHP system is working, while ASHP works in the rest of time. The coefficient of performance is

$$COP_{\mathrm{S1}} = \tilde{M}_{\mathrm{w}} = \left(\frac{T_3 - T_1}{T_3 - T_2} \cdot COP_{\mathrm{w}}\right) / J + (1 - 1/J) COP_{\mathrm{a}}. \tag{29}$$

5) COP_{S2} of combination mode 2

When air temperature is higher than the temperature of waste water flowing out of the evaporator of WSHP, the system is running in this mode. ASHP is running with heat recovery-preheating during the time of $\tilde{\tau} = 1/J$ when the waste water is supplied. The coefficient of performance is

$$COP_{\mathrm{S2}} = \left(\frac{T_3 - T_1}{T_3 - T_2}\right) COP_{\mathrm{a}} / J + (1 - 1/J) COP_{\mathrm{a}}. \tag{30}$$

6) COP_{S3} of combination mode 3

In this mode, air-water dual source heat pump is running with heat recovery-preheating during the waste water is supplied in the time of $\tilde{\tau} = 1/J$. The coefficient of performance is

$$COP_{\mathrm{S3}} = \left(\frac{T_3 - T_1}{T_3 - T_2}\right) COP_{\mathrm{w+a}} / J + (1 - 1/J) COP_{\mathrm{a}}. \tag{31}$$

4 Cases and analysis

Case 1. T_1, T_3, T_4 and ΔT_{w1} are 10°C, 50°C, 31°C, and 3°C, respectively.

Case 2. T_1, T_3, T_4 and ΔT_{w1} are 10°C, 50°C, 36°C, and 5°C, respectively.

4.1 Relations between T_2, T_5, T_6 and J

Along with the value of *J* varying from 1 to 4 for the two cases, the results of T_2, T_5, and T_6 are calculated and shown in Figure 4. It shows that the value of T_2 is in a small range, and tends to T_4 quickly with the increase of *J*. T_5 and T_6 are affected by *J* seriously and varying in a large range. In the range of $J < 2$, T_5 and T_6 decrease quickly with the decrease of *J*.

4.2 Relations between COP_{w} of HPWH and the gain coefficient *h* with *J*

The relations of COP_{w} and the gain coefficient *h* varying with *J* in the two cases are shown in Figure 5. The COP_{w} of HPWH is increasing with the increasing of *J*. The reason is that the temperature of the evaporator of WSHP would be higher with a larger flow of waste water. However, the amount of waste water is limited, the increase of *J* makes the working time of WSHP decrease. Therefore, it's not exactly right that the results will be better if the value of *J* is bigger. The gain coefficient *h* will increase obviously with

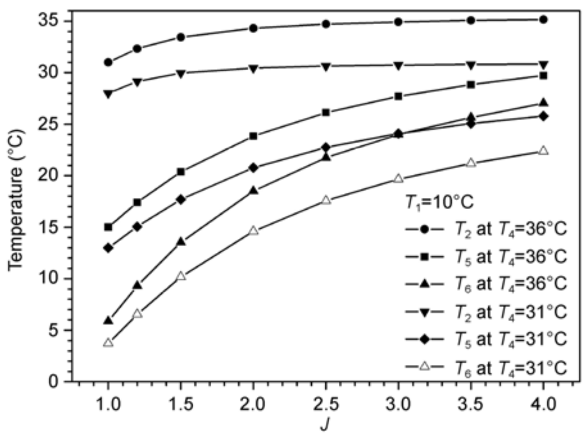

Figure 4 Relations between T_2, T_5, T_6 and J in the two cases.

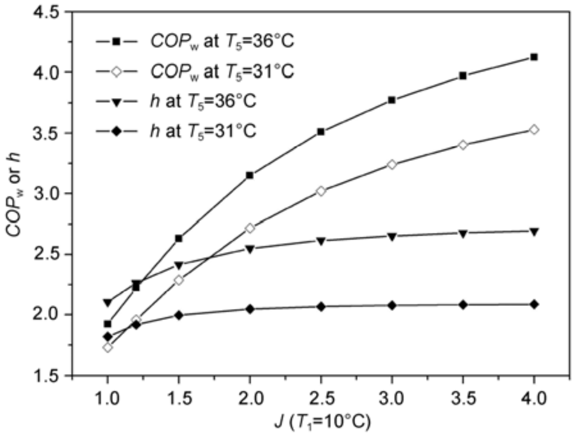

Figure 5 Relations between COP_w, h and J in the two cases.

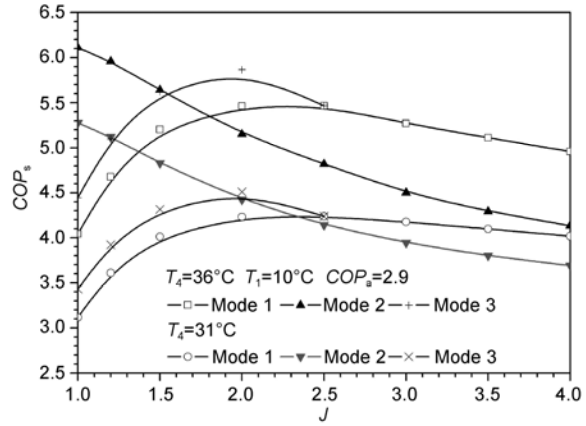

Figure 6 Relations between COP_s and J in the three combination modes (T_a=10°C).

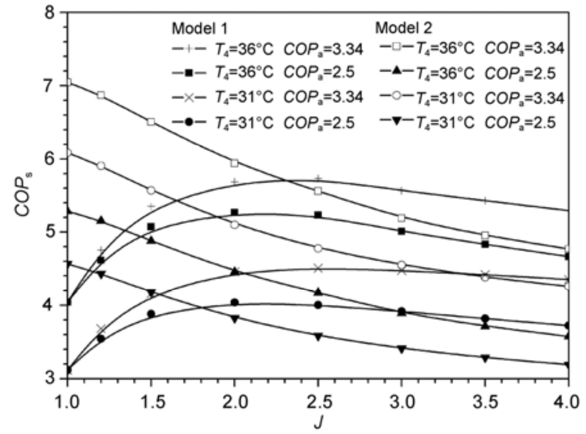

Figure 7 Relations between COP_s and J in combination modes 1 and 2.

the increase of J in the range of J<1.5, and graduallyincrease slowly in the other range. The theoretical coefficient of performance $hCOP_w$ of WSHP with waste heat recovery-preheater will be over 9. The real value of $hCOP_w$ will be smaller for the limitation of the capacity of hot water heat exchanger.

4.3 Relations between COP_S and J in the three combination modes

Figure 6 shows that the relations between COP_S and J in the three combination modes. COP_{S1} of the combination mode 1 increases first and decreases afterwards with the increase of J, and reaches the maximum value at J of 2. COP_{S2} of the combination mode 2 decreases with the increase of J. COP_{S3} of the combination mode 3 is only effective while $COP_w \leqslant COP_a$ and the evaporator temperature of WSHP is lower than the air temperature. If $COP_w > COP_a$, the heating capacity would not be increased in this mode. Considering the increase of COP_w with J, the system should be running in the combination mode 3 in the range of $J \leqslant 2$, and in the combination mode 1 in the range of J>2. Although COP_{S2} of the combination mode 2 is much higher in the case of J=1, the actual flow of waste water are usually unstable and the system could not run at J=1. The current experimental results indicate that the actual system has a better performance at J=1.5. It is consistent with the theoretical optimization result of J=2±0.5. When T_4=36°C, the maximum coefficients of performances of these 3 modes are COP_{S1}= 5.46(J =2), COP_{S2} = 6.10(J=1), COP_{S3}=5.86(J =2).

Figure 7 shows the influence of air temperature on the operating mode and parameters of the system. Except the calculation conditions of case 1 and case 2, further conditions for this calculation are as follows: T_a=5°C, COP_a=2.50 or T_a=15°C, and COP_a=3.34. Compared with Figure 6, the optimized values of J increase with the decrease of the air temperature, but are still in the range of J=2±0.5. Therefore, combination mode 1 is better at a lower air temperature, while mode 2 is better at a higher air temperature with a value of J as low as possible. The maximum of COP will be as follows: at low air temperature T_a=5°C and T_4=31°C, COP_{S1}=4.04 (J=2), COP_{S2}=4.56 (J=1), COP_{S2}=4.17 (J=1.5); and at higher air temperature T_a=15°C, T_4=36°C, COP_{S1}= 5.73 (J=2.5), COP_{S2}=7.04 (J=1), COP_{S2}=6.50 (J=1.5).

4.4 Test results of prototype

The prototype passed the achievement appraisal organized by the Science and Technology Bureau of Huai'an on January 13, 2012. The performance was tested by Jiangsu Institute of Supervision & Testing on Product Quality. The COP of the prototype was not less than 5.5 at the rated conditions (the initial temperature of waste water was 36°C, the temperature of inlet water of the condenser was 55°C, the air temperature was 20°C, the Wet-Bulb temperature was 15°C). Another testing was carried out on January 12, 2012. COP_{w+r} was measured as 4.407 at the conditions with air temperature of 8.12°C, J of 1.5, water temperature of 7.5°C, temperature of outlet water of 56.14°C, temperature of waste water of 38.18°C, and temperature of outlet waste water of 8.01°C. COP_{s3} was measured to be 5.232 at the conditions with air temperature of 7°C, water temperature of 7.31°C, temperature of outlet water of 52.46°C, temperature of waste water of 38.31°C, and temperature of outlet waste water of 8.01°C. In this case, COP_{s3}/COP_{w+r} was 1.18. The results indicate that when the air temperature is similar to the temperature of outlet waste water, the *COP* of the system will be effectively improved by using combination mode 3 with air-water dual sources.

4.5 Comparison of three wast water heat recovery methods

To further investigate the advantages of the wast water heat recovery in two stages, the following three methods were compared: waste water heat recovery in two stages of preheating and WSHP, wast water heat recovery for preheating, and wast water heat recovery for WSHP.

1) Waste water both for preheating and WSHP

In this method, the cold water is heated by waste water recovery-preheater and heat pump, the COP of the system running in different modes can be expressed as the following:

$$COP_s = hCOP = \left(\frac{T_3 - T_1}{T_3 - T_2}\right) COP, \tag{32}$$

where *COP* can be one of COP_a, COP_w and COP_{w+a}. Therefore, the waste heat utilization ratio of the waste water in preheating process is

$$R_1 = h - 1 = \frac{T_2 - T_1}{T_3 - T_2} = \frac{J\left(T_4 - T_5\right)}{T_3 - T_2}. \tag{33}$$

If the waste water keeps on being the source of WSHP after flowing through the waste water recovery-preheater, then the secondary waste heat utilization ratio of the waste water is

$$R_2 = COP_w - 1 = \frac{J\left(T_5 - T_6\right)}{T_3 - T_2}. \tag{34}$$

The total waste heat utilization ratio is defined as the ratio of the waste heat utilization from waste water to the maximum waste heat of waste water $m_{pw}\left(T_4 - T_1\right)$. The total waste heat utilization ratio of waste water heat recovery in two stages is

$$R = \frac{T_4 - T_6}{T_4 - T_1}. \tag{35}$$

The testing parameters of the prototype are listed in Table 1, and the testing results are listed in Table 2.

2) Waste water only for preheating

The recovered heat in this method equals the recovered heat in the method of heat recovery in two stages of preheating and WSHP. However, the water could not reach the required temperature without a heat pump heating. In this method the total waste heat utilization ratio of waste water heat recovery is 0.663.

3) Waste water only for WSHP

The *COP* of WSHP would be improved for the increase of the water source temperature. If the amount of the waste water is unlimited, it could be in the range of 3 to 4.5, or approximately calculated by eq. (23). The testing results of WSHP with waste water heat recovery are listed in Table 3. No.1 and No.2 indicate two prototypes produced by two factories. The input power includes the power of water circulation pumps in WSHP and the power of compressor.

Table 4 lists the testing results of WSHP with waste water heat recovery in winter. When the inlet water tempera-

Table 1 Testing parameters of the prototype

T_1 (°C)	T_2 (°C)	T_3 (°C)	T_4 (°C)	T_5 (°C)	T_6 (°C)	M_w (kg/h)	W (kW)	Q (kW)
7.5	29.3	56.1	38.1	17.8	8.3	815	10.5	46.1

Table 2 Testing results of the waste heat utilization ratios of the prototype

COP_s	h	R_1	COP_w	R_2	R	J	R_1COP_w
4.41	1.81	0.81	2.43	1.43	0.973	1.07	1.97

Table 3 Testing results of WSHP with waste water heat recovery in the summer

No.	T_1 (°C)	T_3 (°C)	T_4 (°C)	T_6 (°C)	M_w (kg/h)	M_{pw} (kg/h)	J	W (kW)	Q (kW)	COP
1	29	46	31	23	2250	3660	1.63	10.8	44.63	4.13
2	29	44	31	24	1740	2810	1.61	10.25	30.45	2.97

Table4 Testing results of WSHP with waste water heat recovery in the winter

No.	T_1 (°C)	T_3 (°C)	T_4 (°C)	T_6 (°C)	M_w (kg/h)	M_{pw} (kg/h)	J	W (kW)	Q (kW)	COP
1	10	46	31	23	1066	3660	3.45	10.8	44.63	4.13
2	10	44	31	24	766.5	2810	3.64	10.25	30.45	2.97

ture is 10°C and other parameters are kept the same as those in Table 3, we have R=(31−23)/(31−10)=0.381, which is the lowest waste heat utilization ratio among the investigated cases. The *COP* of the No.1 prototype is 4.13 in Table 4, which is close to COP_s of 4.41 in Table 2. However, J is 3.45 for the No.1 prototype in Table 4, and it means that WSHP has to be stopped in a time fraction $\tau=(1-1/J)$ =0.71 because of no waste water supply.

5 Conclusions

1) A new air-water dual source heat pump water heater with waste water heat recovery was developed. The new system has three type cycles, namely air-source cycle, water-source cycle and air-water dual source cycle. The water could be heated in two stages: preheated in waste water heat recovery-preheater first, and then heated in heat pump. Waste water releases heat in preheater and heat pump, thus the cascade utilization of waste heat is fulfilled.

2) The characteristics of the dual-source heat pump with the waste water heat recovery-preheater were analyzed by using dimensionless correspondence analysis method. The dimensionless formula of the coefficients of performance of air-source heat pump and water-source heat pump were given. The heat transfer characteristic of waste heat recovery-preheater with variable flow, and the coupling problem with heat pump were solved. The *COP* formulas were derived for six modes of the dual-source heat pump with the waste water heat recovery-preheater. Analysis work was carried out with cases. The results indicated that the *COP* of the new system can be 4–5.5 if running in the combination mode 3 with J of 2±0.5 in winter with low air temperature and water temperature. When the air temperature is higher than 15°C, the system should be running in combination mode 2 with J<1.5, and the *COP* would be more than 5.5.

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