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Research Article

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Performance and empirical models of a heat pump water heater system with heat recovery of waste water

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ABSTRACT

In order to save energy, the author developed a type of air-wastewater source heat pump (AWSHP) water heater system. What presented here is about the experimental performance and empirical models of the AWSHP system. Experiments and analysis showed that when the temperatures of waste water and city water varied in the ranges of 16.9~35.1 °C and 18.3~32.6 °C, respectively, the AWSHP system could produce 41.2~53.9 °C hot water with flow rate of 3.53×10^{-5}~9.31×10^{-5} m^3/s. The coefficient of heating performance (COP_h) varied from 3.89 to 5.35. On the basis of energy balance, synthetic degree of grey incidence (SDGI) and grey-box methods, it was found that the COP_h and hot water flow rate depended on the ratio of inlet and outlet temperature differences of waste water and hot water, i.e. $\Delta T_w/\Delta T_h$. The input power of compressor was proportional to the outlet temperature of hot water. Thus, the empirical correlation equations were obtained.

Keywords: heat recovery; heat pump; waste water; empirical model; energy saving

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INTRODUCTION

In families and other fields such as artificial warm springs, sauna rooms and HVAC engineering, hot water is generally required above 40 °C. But in these fields, especially in families and sauna rooms, the waste hot water generated during shower bath or sauna is directly drained into the environment. This causes both energy-wasting and thermal pollution to the environment. Measurements showed that the temperature of waste water generated by shower bath was in the range of 30~36 °C, which is still valuable for utilizing. In order to recover the heat of shower waste water and provide local air-conditioning, on the basis of research results of [1–4], the author developed a type of air-water source heat pump (AWSHP). It uses air and waste water, especially shower waste water, as heat sources. The AWSHP could produce enough hot water for bathing by recovering the heat from air and shower waste water, and showed satisfactory performance of energy saving. What presented in this paper is about the experimental performance and the data-based empirical models of the AWSHP water heater system, which is a further study on the performance of AWSHP presented by [5].

FLOW CHART AND EXPERIMENTAL SET OF THE AWSHP

The basic flow chart diagram of the AWSHP water heater is shown in Figure 1. It can be installed in bathroom or elsewhere. There is no need of hot water tank and this makes it more compact. Its basic principle is the same as common water-source heat pump water heater. In this type of AWSHP, the air source evaporator (AE) and water source evaporator (WE) are arranged in serial instead of parallel to maintain the different evaporating temperatures.

The AWSHP can efficiently recover heat from waste water and air to save energy and avoid the heat pollution caused by directly drained hot water.

In the experimental set, the outlet temperature of hot water is regulated by a condenser-pressure-dependent valve which also controls the condenser pressure and temperature by regulating cooling water flowing through the condenser.

The throttling part of the AWSHP is a set of capillary tubes. The air source evaporator is a self-made heat exchanger with a fan. A plate-type heat exchanger and a shell-tube heat exchanger are used as the condenser and waste water evaporator, respectively. The refrigerant is R22 and the input power of waste water pump is 25W.

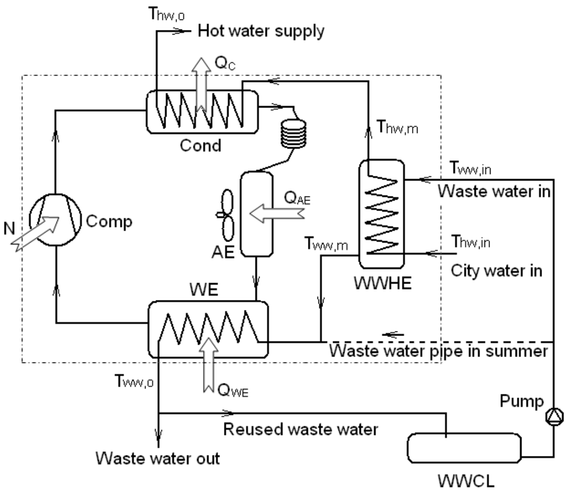

Figure 1. Diagram of flow chart and energy-exchanging of AWSHP water heater.
Comp—compressor; Cond—condenser; Cap—capillary; AE—air source evaporator; WE—water evaporator; WWHE—water-water heat exchanger; WWCL—waste water collector.

A data acquisition system based on LabView virtual instrument technology is equipped to the experimental set. The temperature sensors are T-type thermocouples. The flow rates of hot water and waste water are measured by intelligent electromagnetic flow meters with accuracy of ±1.0%. The pressures are manually measured by refrigerant pressure gauges. The input power of the heat pump system is measured on line by a Fluke 43B power quality analyzer. The humidity of air flowing through the air source evaporator is measured by digital temperature-humidity meters with accuracy of ±3.0% (for relative humidity), and the velocity of air flow in the air source evaporator is measured by an anemometer with accuracy of ±2.0%.

CALCULATION BASIS

In the AWSHP water heater, driven by input power of compressor, the refrigerant only exchanges energy with air, city water and waste water, as shown in Figure 1. From the view of energy balance, the performance of the AWSHP should depend on input power of compressor (N) and heat capacities of condenser (Q_C), air source evaporator (Q_{AE}) and waste water source evaporator (Q_{WE}).

The coefficient of heating performance of the AWSHP is defined by Eq.1:

$$COP_h = \frac{Q_C}{N} \tag{1}$$

where, N can be measured by the Fluke 43B power quality analyzer. Q_C is also the heat transferred to hot water. If the heat loss of condenser is neglected, Q_C can be calculated from the parameters of hot water as Eq.2:

$$Q_C = \frac{\rho c V_{hw}(T_{hw,o} - T_{hw,m})}{3600} \tag{2}$$

where $T_{hw,m}$ is the temperature of hot water leaving off the WWHE, °C. $T_{hw,m}$ is determined by the heat exchanging efficiency of WWHE and the initial temperature of city water named as $T_{hw,in}$ (as shown in Figure.1).

Based on the energy balance of the system, if the heat loss of the system is neglected, there is Eq.3:

$$Q_C = N + Q_{AE} + Q_{WE} \tag{3}$$

Where,

$$Q_{WE} = \frac{\rho_w c V_{ww}(T_{ww,m} - T_{ww,o})}{3600} \tag{4}$$

$$Q_{AE} = \rho_A V_A (h_{A,in} - h_{A,o}) \tag{5}$$

In Eqs.4 and 5, $T_{ww,m}$ is the temperature of waste water leaving off the WWHE, °C (as shown in Figure 1). $T_{ww,m}$ is determined by the heat exchanging efficiency of WWHE and the initial temperature of waste water named as $T_{ww,in}$. V_A is the volume flow rate of air flowing through the AE, m^3/s. $V_A = v \cdot F$, v and F are the velocity of air flow (m/s) and area (m^2) of the air supplying section of the air source evaporator, respectively. Eq.3 also can be used to judge the heat loss of the system.

PERFORMANCE AND EMPIRICAL MODELS OF THE AWSHP

There are many theoretical models based on mathematic equations of compressor, condenser, throttle valve (or capillary tube) and evaporator to simulate the performance of heat pump system, such as [2, 6–9]. But all of the theoretical models need many parameters and sophisticated computer calculation. They are suitable for simulating and designing the system but not convenient for engineering evaluation on the fields. Yu (1995) [10] established an empirical COP model solely depending on the temperature difference of condensing and evaporating temperature, but it is found not suitable for evaluating the performance of the AWSHP shown in Figure 1.

In order to investigate the performance and establish the empirical models of the AWSHP, experiments are carried out at various conditions such as different inlet temperatures of waste water and city water, with or without water-water heat exchanger, different outlet temperature of hot water, different volume of charged refrigerant (R22), and so on.

Experimental performance

Experiments and analysis show when the temperatures of waste water and city water are in the range of 16.9~35.1 °C and 18.3~32.6 °C, respectively, the AWSHP can produce hot water with temperature of 41.2~53.9 °C, and the volume flow rate of hot water varies in the range of 0.13~0.34 m^3/h (3.53×10^{-5} ~ $9.31\times10^{-5} m^3/s$). The COP_h varies in the range of 3.89~5.35. When the input power of pump and fan are taken into account, the overall energy efficiency ratio (EER) varies in 3.74~5.10. It is also found that the WWHE deteriorates the COP_h, but it can efficiently increase the hot water flow rate. Therefore, the WWHE is necessary to provide enough hot water when the city water temperature is lower such as in winter.

Experiments show the temperature and relative humidity of air at outlet of the AE vary in the range of 12.3~20.8 °C and 83.1~93.4%, respectively. But the refrigeration capacity of the AE is just 5.2~13.2% of that produced by the WE. Therefore, the influence of the AE on the performance of AWSHP can be neglected in the present analysis. It also indicates that the AE should be redesigned to supply enough cold air for local air-conditioning.

Empirical models

The empirical models are based on energy-exchanging and the affecting factors of the AWSHP. It can be deduced From Eqs.1–3 that the COP_h of the AWSHP is the function of N, Q_{AE} and Q_{WE}. As Q_{AE} and Q_{WE} depend on the temperatures and flow rates of air flow and waste water flow, the COP_h can be expressed by the f_1 function of the parameters as Eq.6:

$$COP_h = f_1(N, h_{A,in}, h_{A,o}, V_A, T_{ww,in}, T_{ww,o}, V_{ww}) \tag{6}$$

Because the experiments show that the Q_{AE} is very small and it takes little effect on COP_h, the parameters of the AE can be omitted by adjusting other coefficients, e.g. $T_{hw,in}$. It is also found that the N is closely related to the outlet temperature of hot water ($T_{hw,o}$). Thus, when the waste water flow rate V_{ww} is kept constant, Eq.6 can be simplified as Eq.7:

$$COP_h = f_2(T_{hw,o}, T_{hw,in}, T_{ww,in}, T_{ww,o}) \tag{7}$$

Further analysis on experimental data shows that COP_h of the AWSHP closely depends on the ratio of temperature difference $\Delta T_w/\Delta T_h$ when V_{ww} is constant. $\Delta T_w/\Delta T_h$ can be calculated as Eq.8:

$$\frac{\Delta T_w}{\Delta T_h} = \frac{T_{ww,in} - T_{ww,o}}{T_{hw,o} - T_{hw,in}} \tag{8}$$

Therefore, Eq.7 can be further simplified as Eq.9:

$$COP_h = f_3(\Delta T_w/\Delta T_h) \tag{9}$$

Similar to COP_h, the influence factors of the performances of the AWSHP can be summarized as $T_{ww,in}$, $T_{ww,o}$, $T_{hw,in}$, $T_{hw,o}$, ΔT_w, ΔT_h and $\Delta T_w/\Delta T_h$. Based on the analysis method of synthetic degree of grey incidence (SDGI) proposed by [11], the most important factors of the performances of the AWSHP can be determined as $\Delta T_w/\Delta T_h$ and $T_{hw,o}$. Thus, the empirical models can be proposed by choosing $\Delta T_w/\Delta T_h$ or $T_{hw,o}$ as the variables.

If the grey-box method is used to determine the empirical models, when the waste water flow rate is kept constant, the empirical regression equations of COP_h, V_{hw} and N can be established based on the experiment data and principle of the least square method as Eqs.10–12:

$$COP_h = A \cdot \left(\frac{\Delta T_w}{\Delta T_h}\right)^2 + B \cdot \frac{\Delta T_w}{\Delta T_h} + C \tag{10}$$

$$V_{hw} = D \cdot \frac{\Delta T_w}{\Delta T_h} + E \tag{11}$$

$$N = m \cdot T_{hw,o} + n \tag{12}$$

where A, B, C, D, E, m and n are constant coefficients determined by regression data. The experiment data of COP_h, V_{hw} and N are shown in Figures 2–4. The corresponding empirical regression equations of COP_h, V_{hw} and N in Figures 2–4 are Eqs.13–15:

$$COP_h = -2.5182\left(\frac{\Delta T_w}{\Delta T_h}\right)^2 + 5.2704\frac{\Delta T_w}{\Delta T_h} + 3.1159 \tag{13}$$

$$V_{hw} = 0.454\frac{\Delta T_w}{\Delta T_h} + 0.0551 \tag{14}$$

$$N = 0.0214T_{hw,o} + 0.2032 \tag{15}$$

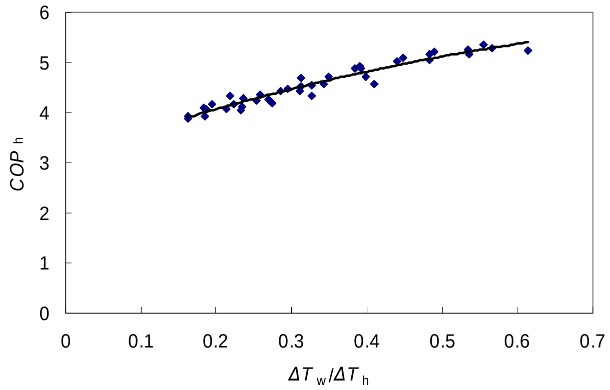

Figure 2. Relationship of COP_h and $\Delta T_w/\Delta T_h$

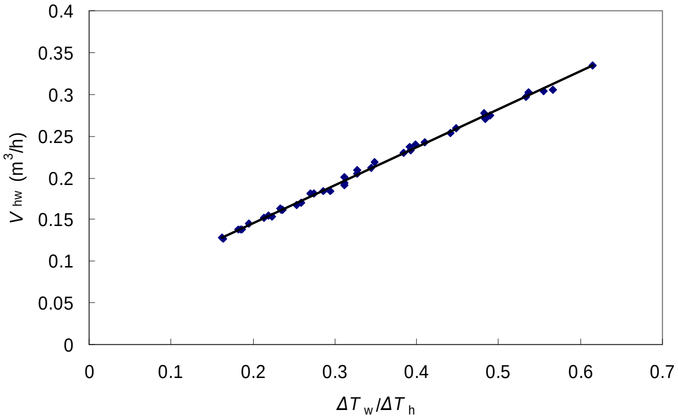

Figure 3. Relationship of V_{hw} and $\Delta T_w/\Delta T_h$

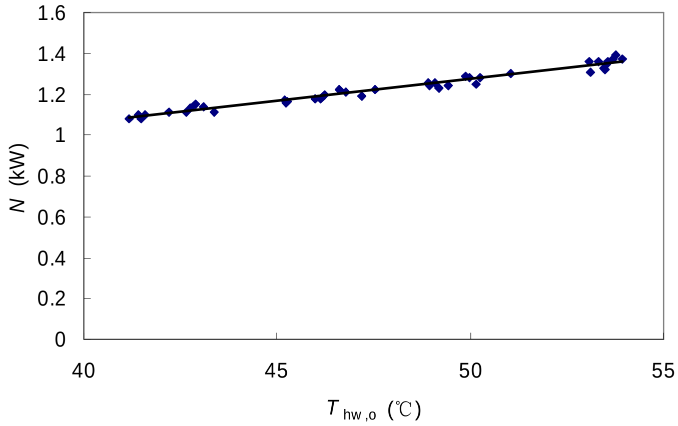

Figure 4. Relationship of N and $T_{hw,o}$

Validation and discussion of empirical models

In order to investigate the validity of models expressed by Eqs.10–12, experiments were carried out on the AWSHP with reduced charged refrigerant (R22). The results showed the perfect accordance of Eqs.10–12 and only the constant coefficients were different. For example, the COP_h at this condition is shown in Figure 5, and the corresponding empirical regression equation of COP_h is expressed by Eqs.16.

$$COP_h = -13.735\left(\frac{\Delta T_w}{\Delta T_h}\right)^2 + 11.895\frac{\Delta T_w}{\Delta T_h} + 2.3817 \quad (16)$$

It demonstrates the validity of empirical models expressed by Eqs.10–12. As the temperatures of hot water and waste water can be easily measured, these models can be conveniently used to evaluate and modulate the performances of water-source heat pump water heaters on the field.

It also indicates that the COP_h has a maximum or optimal value at certain $\Delta T_w/\Delta T_h$. This provides an engineering guide to improve the heating performance of the AWSHP. Thus, designers should keep in mind to set the $\Delta T_w/\Delta T_h$ of AWSHP in close proximity to its optimal value to obtain the maximum COP_h.

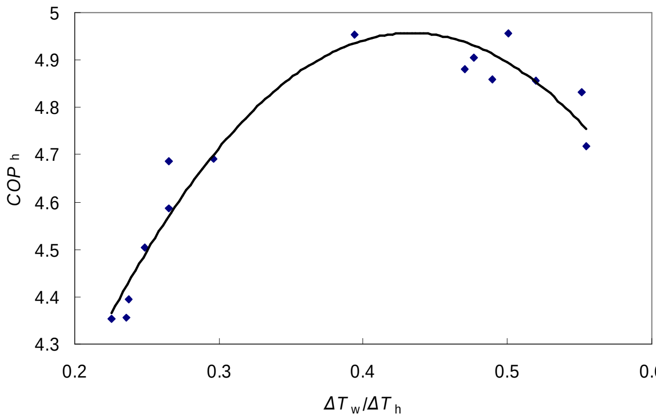

Figure 5. Relationship of COP_h and $\Delta T_w/\Delta T_h$ in the AWSHP with reduced charged R22.

Above analysis shows that the performance of the AWSHP water heater can be determined using the inlet and outlet temperatures of wastewater and hot water, especially the ratio of temperature differences of wastewater and hot water (i.e., $\Delta T_w/\Delta T_h$). The variable of $\Delta T_w/\Delta T_h$ makes the above empirical models different to the others presented in literatures in most of which the absolute, not the relative temperatures of fluids were selected as model variables. Meanwhile, the empirical models presented in this paper not only can be used to predict the COP_h, but also can be used to evaluate the hot water flow rate and the compressor power-consumption of the AWSHP water heater. Thus, with the above empirical models, the performance of AWSHP water heater can be investigated from different points of view, but not only from the COP_h.

The above empirical models also provide an easy way for engineers to analyze and evaluate the performance of the AWSHP water heater because measuring the temperatures of wastewater and hot water is easier than measuring refrigerant parameters. The empirical models are also helpful for evaluating and optimizing operational conditions.
Though Eqs.10–12 are obtained from experiment data of the AWSHP, they are also suitable for evaluating other water-source heat pump water heaters because the influence of air-source evaporator was neglected. It should be mentioned that the constant coefficients in Eqs.10–12 depend on operating conditions, such as charged volume of refrigerant, dimensions of capillary tube, with or without water-water heat exchanger, waste water flow rate and so on.

CONCLUSION

Experiments and analysis indicate that the AWSHP water heater has satisfactory energy saving performance. This system also provides a simple and efficient way to obtain sustainable use of waste water by recovering the low-grade heat from it. The main points of this paper can be summarized as following:

1) The coefficient of heating performance (COP_h) varies in the range of 3.89~5.35. Though the WWHE deteriorates the COP_h of AWSHP, it can efficiently increase the hot water flow rate. But the AE in the system should be redesigned to provide enough cooling capacity.

2) Based on the analysis of energy balance, SDGI and grey-box, empirical models are obtained and the models show that the COP_h, V_{hw} and N only depend on the temperatures of hot water and waster water, especially the ratio of temperature differences of wastewater and hot water (i.e., $\Delta T_w/\Delta T_h$).

3) The empirical models presented in this paper can be used to predict the COP_h, the hot water flow rate and the compressor power-consumption of the AWSHP water heater at the same time. It can be conveniently used to evaluate and modulate the performances of the AWSHP or other water-source heat pump water heaters on the field. Further research on the performances of the AWSHP with improved air-source evaporator will soon be carried out.

Nomenclature

c specific heat of water, kJ/(kg°C)
f_1, f_2, f_3 signs of functions
h specific enthalpy, kJ/(kg°C)
N input power of compressor, kW
Q heat capacity, kW
T temperature, °C
V volume flow rate, m^3/h

Greek symbols

ΔT temperature difference, °C
ρ density, kg/m^3

Superscripts and subscripts

A air
A,in inlet air
A,o outlet air
c condenser
h hot water
hw hot water
hw,in inlet of hot water
hw,m middle status of hot water
hw,o outlet of hot water
w water
WE waste source evaporator
ww wastewater
ww,in inlet of wastewater
ww,m middle status of wastewater
ww,o outlet of wastewater

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REFERENCES

[1] GS Chen; F Li; *Journal of Guangdong University of Technology*, **2002**, 19(4), 47–49.
[2] GX Kou; HQ Wang; XL Wang; XL Li; *HV&AC*, **2004**, 34(6), 43–45.
[3] HP Tan; Y Chen; *Fluid Machinery*, **2008**, 36(6), 83–85.
[4] HP Tan; *Fluid Machinery*, **2009**, 37(7), 65–68.
[5] HD Wang; *MACE2011 Proceedings*, IEEE Press, **2011**, 7, 5803–5806.
[6] J Ji; G Pei; W He; J Dong; WP Zhao; *HV&AC*, **2003**, 33(2), 19–23.
[7] H Tian; YT Ma; ZQ Wen; *Journal of Thermal Science and Technology*, **2008**, 7(4), 373–378.
[8] GY Xu; SH Li; XS Zhang; WB Wu; *Journal of Harbin Institute of Technology*, **2009**, 41(2), 211–213.
[9] J Li; S.M Jin; *Fluid Machinery*, **2010**, 38(8), 73–79.
[10] LQ Yu; *HV&AC*, **1995**, 25(1), 12–14.
[11] SF Liu; YG Dang; ZG Fang; *Grey system theory and application*, 3rd Edition, Science Press, Beijing, **2004**.