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Optimization study on the heat transfer area of the sewage source heat pump system based on year-round coefficient of performance

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Abstract

Based on simulation model of the sewage source heat pump system (SSHPS), weigh the influence of coefficient of performance under different work conditions, the ACOP was defined and as the optimization function, with the heat exchange areas of the evaporator, condenser and the sewage-midwater heat exchanger as the optimization variables. The optimization mathematical models for the on-off control scheme were set up to calculate the ACOP to study the optimal design and areas matching between the evaporator, condenser and sewage-midwater heat exchanger. Comparison with the initial design parameters and different optimization designs with HSPF, SEER and EER, the ACOP optimization is nearer to the objective energy conservation effect.

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Keywords: Sewage source heat pump system (SSHPS); Simulation; Annual coefficient of performance (ACOP); Optimization

1. Introduction

Urban sewage-source heat pump (USSHP) is a kind of heat pump air conditioning unit using secondary urban original waste water or sewage treatment plant as a direct or indirect cooling and heat sources, which has significant performance advantages and energy saving effect, and has got rapid development and application in our country and

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around the world. Many re-search scholars has optimized research on the system or part of the components, and obtained some useful results [1].

At present, the research on optimization about the sewage-source heat pump system is mostly under the rated conditions or under certain conditions, inlet water temperature and quantity of the heat pump refrigeration (hot) are set to a fixed value [2]. In fact, as the change of the outside meteorological parameters, cold and heat source temperature change constantly, at the same time, the change of indoor heat source led to the cold and hot load of building changing. As a result, with constant change of the operating mode of heat pump, at the same time building load demand system control mode has a good capacity to meet the actual demand. According to the current design habits, heat pump system is running under partial load at the most of the time [3].Components under different working conditions, due to the characteristics and the influence of such factors as cold heat source temperature changes, or standard working conditions at the single condition to get the optimal structure of system performance, cannot guarantee full-year operating range has higher efficiency. And regarding heat pump system efficiency throughout the year as the highest optimal objective function evaluation standard not only balanced the effects of the annual operating efficiency under different working conditions, also conforms to the principle of energy conservation. On the basis of the establishment of refrigeration and heating heat pump system simulation model is, by using the direct search method with constraints, researching the optimal allocation of heat exchange area the sewage-source heat pump system, to achieve the optimal design of the annual operating efficiency of the whole system.

2. Simulation model of SSHPS

According to whether the sewage enter directly into the evaporator and condenser heat pump units, the sewage-source heat pump system can be divided into direct and indirect system, so far, there is no new type of high-efficient direct sewage heat pump units, and adopt indirect while system designs. The sewage-source heat pump principle of indirect system is shown in figure 1, the sewage prevention machine model is determined by the flow of water[4]; Sewage-intermediate water heat exchanger tube and shell heat exchanger, water tube side, the shell side of the intermediate water go, its model was determined by the design of change heat quantity; By the principle of loop independence, secondary sewage pump, inter-mediate water circulation pump and circulating pump are determined by the flow rate and resistance of the loop. While heat pump unit absorbs heat in winter and release cool in summer to the waste water through the sewage heat exchanger. Through open or close different valves to realize cooling and heating switch. From figure 1, the mathematical model of this system includes six sub-model, model of evaporator, condenser, compressor model, thermal expansion valve model, water and sewage pump model - the mediation model of the heat exchanger. Refrigerants is R134a.

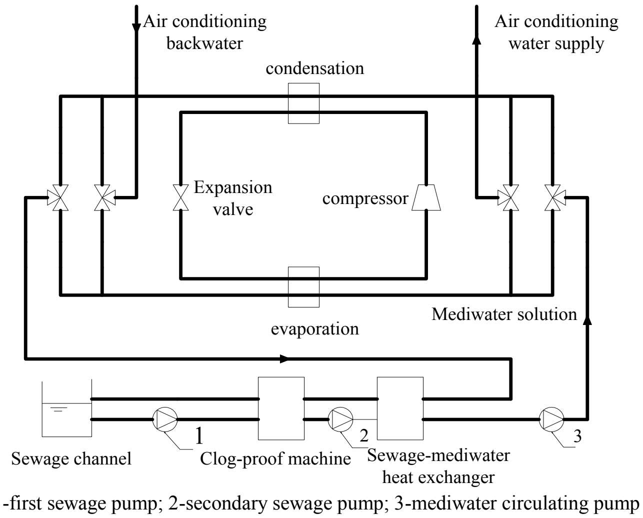

Fig. 1. Schematic of sewage source heat pump system

2.1. Evaporator model

$$Q_e = G_R(h_{e,2} - h_{e,1}) = G_e c_e (t_{e,1} - t_{e,2}) \tag{1}$$

The heat transfer equation:

$$Q_e = K_e F_e (LMTD)_e \tag{2}$$

$$(LMTD)_e = (t_{e,1} - t_{e,2}) / \ln \frac{t_{e,1} - t_e}{t_{e,2} - t_e} \tag{3}$$

In those equations, Q_e is the evaporator heat transfer (kW), G_R is the quality of the refrigerant flow rate (kg/s), $h_{e,2}$ is the gas enthalpy of refrigerant evaporator outlet(kJ/kg), $h_{e,1}$ is the liquid enthalpy of refrigerant evaporator entrance (kJ/kg), G_e is the mass flow rate of liquid passed in and out of the evaporator (kg/s), c_e is the specific heat of liquid passed in and out of the evaporator (kJ/(kg ℃)), $t_{e,1}$ and $t_{e,2}$ are the liquid temperature which passes in and out of the evaporator (℃), K_e is the heat transfer coefficient of the evaporator (kw/(m^2.℃)) , F_e is the heat exchange area of the evaporator(m2), $(LMTD)_e$ is the logarithmic mean temperature difference of evaporator (℃), t_e is the evaporator evaporation temperature (℃).

2.2. Condenser model

Unit generally adopts horizontal shell and tube type water-cooling condenser (refrigerant goes shell side). High temperature and high pressure refrigerant vapor from compressor, enters into the top of the condenser, and condensates on horizontal tube cluster fixed in the tube plate. Refrigerant liquid flows from the bottom of the shell liquid pipe.

Energy equation:

$$Q_c = G_R(h_{c,1} - h_{c,2}) = G_c c_c (t_{c,1} - t_{c,2}) \tag{4}$$

The heat transfer equation:

$$Q_c = K_c F_c (LMTD)_c \tag{5}$$

The calculation of the Logarithmic mean temperature difference of accords to the saturated vapor condensation, as

$$(LMTD)_c = (t_{c,2} - t_{c,1}) / \ln \frac{t_c - t_{c,1}}{t_c - t_{c,2}} \tag{6}$$

In those equations , Q_c is the heat transfer of condenser(kW), G_R is the mass flow rate of the refrigerant(kg/s), $h_{c,1}$ and $h_{c,2}$ are the gaseous enthalpy of refrigerant in the condenser entrance and outlet (kJ/kg), G_c is t the mass flow rate of liquid flows in and out of the condenser tube side (kg/s), c_c is the specific heat of liquid flows in and out of the condenser tube side (kJ/(kg) ℃)), $t_{c,1}$ and $t_{c,2}$ are the temperature of fluid flows in and out of the condenser tube side (℃), K_c is the heat transfer coefficient of the condenser (. ℃)), F_e is the heat exchange area of the condenser(m2), $(LMTD)_c$ is the logarithmic mean temperature difference of condenser , t_c is the condensation temperature (℃).

2.3. Sewage–intermediate water heat exchanger model

The thermal balance equation of heat exchanger is as follows：

$$Q_w = G_w c_w \left(t_{w1} - t_{w2}\right) \tag{7}$$

$$Q_z = G_z c_z \left(t_{z2} - t_{z1}\right) \tag{8}$$

$$Q_w = Q_z = K_h F_h (LMTD)_h \varphi \tag{9}$$

$$(LMTD)_h = (t_{w1} - t_{z2}) / \ln \frac{t_{w1} - t_{z2}}{t_{w2} - t_{z1}} \tag{10}$$

In which Q_w, Q_z are known as the capacity of heat transfer about sewage and intermediate water (kw); G_w, G_z as the mass flow of sewage and intermediate water(kg/s), t_{w1}, t_{w2} as the temperature of sewage passing through the exchanger(℃), t_{z1}, t_{z2} as the temperature of intermediate water passing though the exchanger(℃); K_h as the transfer coefficient of exchanger（kw/(m^2.℃））; F_h as the heat transfer area of exchanger（m^2）; $(LMTD)_h$ as the average temperature difference of these exchangers (℃); φ as the coefficient factor of temperature difference as φ=0.9.

The references [5] and [6] gave the models of Compressor, thermal expansion valve and water pump model. The sewage source heat pump system model is combined by evaporator, condenser, compressor, thermostatic expansion valve, pump and sewage-intermediate water heat exchanger.

2.4. Model verification

The simulated results will be compared with the experimental results of sewage source heat pump system in order to verify the correctness of the model. Pictures and performances of the experiment equipments are shown in Fig.2 and Table.1. Fig.3 and Fig.4 is the comparative results of heating capacity, refrigerating capacity, COP and ERE separately. As can be seen from the Figures, the analog results coincide with the experimental results and over 80.6% errors are within 10%, which prove that model has high precision and can be used in system simulation and optimization.

Fig. 2. pictures of heat pump and fan coil in the Lab

Table 1. Experiment equipments and their performance.

Heat pump unit	Type	NGHP-0078WA	
	Refrigerant	R22	
	Scroll compressor	Input power for heating	19 kW
	Double-pipe evaporator	Heat areas	5.5 m^2
	Double-pipe Condenser		8.0 m^2
	Heating conditions 15/7℃—40/45℃	Heating outputs	78 kW
Sewage heat excanger	Heat pipe diameter(mm) /areas(m^2)	20/10	
Sewage water pump	flux（m^3/h）/ head（mH_2O）	20/12	
Midwater pump		20/12	
Terminal cycling pump		20/12	

Fan coil	Air volume(m^3/h) /unit number	650/10

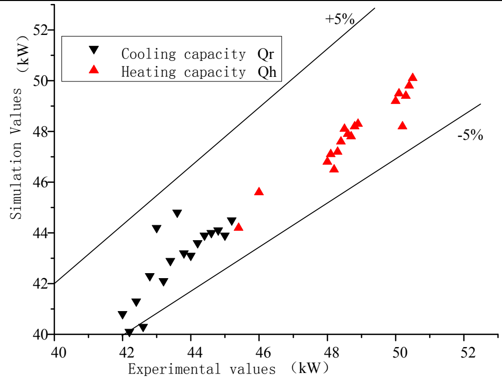

Fig. 3. comparison between the simulation and experimental of the Q_h, Q_r

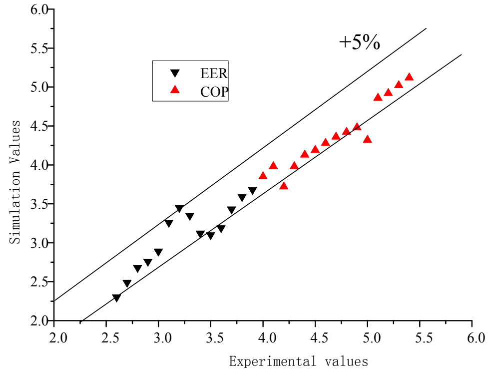

Fig. 4. comparison between the simulation and experimental of the COP, EER

3. Performance optimization for the whole year

System optimization design aimed to achieve highest efficiency of heat pump, is accomplished by setting the parameters of the components in the system first, and then adjusting the parameters according to the simulation model of satisfied requirements of refrigerating or heating capacity. The operating characteristics of this heat pump system is more match with the load demand for the whole year, and it has the real objective comfort and energy saving effect.

3.1. Optimization of the objective function

It is well known that we can evaluate the heat pump system with seasonal by the parameters of SEER (Seasonal Energy Efficiency Ratio) and HSPF (Heating Seasonal Performance Factor) [7]. SEER is the ratio of total refrigerating capacity to the total energy consumption of the system in air conditioning season, as:

$$SEER = \frac{\sum Q_{e,i} \cdot T_{r,i}}{\sum W_{r,i}} = \frac{\sum Q_{e,i} \cdot T_{r,i}}{\sum (N_{e,i} + N_{p,i}) T_{r,i}} \tag{11}$$

HSPF is the ratio of the total heat to the total energy consumption of the system in heating season, as:

$$HSPF = \frac{\sum Q_{c,j} T_{h,j}}{\sum W_{h,j}} = \frac{\sum Q_{c,j} \cdot T_{h,j}}{\sum (N_{e,j} + N_{p,j}) T_{h,j}} \tag{12}$$

In which $Q_{e,i}, Q_{c,j}$ is the instantaneous cold (hot) capacity proved by heat pump system (kW); $T_{r,i}, T_{h,j}$ is the running time of the cooling and heating system (1 h); W is the instantaneous power consumption in heat pump system (kW.h), including the compressor power consumption W_c and water pump power consumption W_p; $N_{p,i}, N_{p,j}$ is instantaneous power of the compressor and pump (kW).

Considering that system optimization used only with the SEER and HSPF as target function cannot comprehensively evaluate performance of heat pump system throughout the year, so we choose ACOP (annual coefficient of performance) as objective function, ACOP is the ratio of the sum of output total cooling and heating capacity for all year to the sum of the system total energy consumption (including all the compressor and pump), as:

$$ACOP = \frac{\sum Q_{e,i} \cdot T_{r,i} + \sum Q_{c,j} T_{h,j}}{\sum W_{r,i} + \sum W_{h,j}} = \frac{\sum Q_{e,i} \cdot T_{r,i} + \sum Q_{c,j} T_{h,j}}{\sum (N_{e,i} + N_{p,i}) T_{r,i} + \sum (N_{e,j} + N_{p,j}) T_{h,j}} \tag{13}$$

In conclusion, we choose ACOP as the optimization objective function of this system.

3.2. Optimization variables and constraints

Under the premise of determining the sewage-source heat pump system forms, through the reasonable design the area of evaporator, condenser and water–sewage exchanger, we can make evaporator and condenser better to match the compressor. Sewage-mediation is not only can exchange heat in sewage water but also can match the evaporator and condenser of the heat pump, which can improve the whole performance of the system. So we select the unit evaporator heat transfer area of the Fe, condenser heat transfer area of the Fc, and sewage -intermediate water heat exchanger area of F_h as optimization variables.

Constraint conditions, (1) the evaporator outlet temperature>2℃. (2) the operation of the heating, the condenser outlet temperature is 45℃. Operation of the refrigeration, outlet temperature of the evaporator is 7℃. (3) the condensation temperature is greater than the evaporation temperature, $t_{cr} > t_{er}$, $t_{ch} > t_{eh}$. (4) in the condenser, the condensation temperature is greater than the water temperature, water temperature is greater than the water temperature， $t_{cr} > t_{cr2} > t_{cr1}$, $t_{ch} > t_{ch2} > t_{ch1}$. (5) in the evaporator, the evaporation temperature is less than the water temperature, water temperature is less than the water temperature, $t_{er} < t_{er2} < t_{er1}, t_{eh} < t_{eh2} < t_{eh1}$.

To simplify the calculation, we assume that a sewerage flow rate equal to the mediation flow at any time. we determine water flow rate at the end of the cycle according to the cooling(heating) load, import and export water temperature. In the operation of the heat pump sys-tem in the year, the demand of the building dynamic load is met by the compressor start-stop control.

4. Case analysis

In this paper, we use the former sewage source heat pump system as an example, which use screw compressor, and flooded refrigerant evaporator and condenser tubes using 10mm copper pipe, sewage-water heat exchanger using intermediary shell and tube heat exchangers, heat exchange tubes of 20mm cast iron pipe. table 1. is a start-stop control unit, different optimal objective function value at the appropriate system for each heat exchanger heat transfer area. here, EER is a standard refrigeration condition, when the biggest meet building cooling load, the heat pump cooling capacity and total system power consumption ratio amounts to optimize this objective function is to optimize a single condition. Optimized to ACOP, HSPF and SEER target function belongs to the variable condition performance optimization.

4.1. The optimal area (F_e 、 F_c 、 F_h)

It shows from the table 2 that the heat transfer area of each heat exchanger of system which corresponds with the performance coefficient peak of the different objective function is different. Each objective function values for different heat pump system corresponding to the evaporator. ACOP peak is between HSPF peak and SEER peak, and the changing trend of the heat transfer area of sewage-mediation water's heat exchanger which is consistent with each objective function peak is similar, while that is opposite for the unit condenser. When the ACOP is optimal, the heat transfer area of each heat exchanger is smaller than that was de-signed initially, with evaporator reducing 6.5%, condenser decreasing 13.6%, and sewage- mediation water's heat exchanger reducing 14.1%.

Table 2. the heat exchanger areas and E according to different objective function

	F_e (m^2)	F_c (m^2)	F_h (m^2)	E (kW.h/m^2.y)
initial design COP=4	2.78	2.06	13.5	54.29
$ACOP_{max} = 4.13$	2.6	1.78	11.6	51.67
$HSPF_{max} = 3.84$	2.71	1.69	12.5	52.05
$SEER_{max} = 4.36$	2.54	1.82	11.2	51.84
$EER_{max} = 4.23$	2.43	1.95	11.0	52.24

4.2. Annual energy consumption of unit building area (E)

The definition of the annual energy consumption of unit building area E of sewage-source heat pump system is, during one year's operation of heat pump system, the ratio between the system's total power consumption, including the power consumption of compressor and all pumps, and a total construction area. The first table shows that E value made with the ACOP as the optimization goal is 0.57 kW.h/m2.y smaller than that is made with the EER as the optimization goal, and is 2.62 kW.h/m2.y smaller than the system was designed initially. Mean-while, the E value is the least when it is made with the SEER and HSPF as the optimization goal, which proves that the optimal system made with the ACOP as the optimization goal has better energy saving effect.

Actually, for a certain annual load distribution, when the form of heat pump is fixed and the allocation of unit heat exchanger area changes, the changing of system's evaporation temperature and condensation temperature will lead to the changing of the coefficient of performance of heat pump unit. The optimal ACOP can weigh the system's coefficient under various operating conditions during the whole year, which makes, on most of the working conditions, the evaporation temperature higher and condensation temperature lower, as a result, annual performance coefficient reaches the highest. The objective function based on it can not only reflect the effects of the outside meteorological parameters, and make cooling and heating performance, fully describe the performance of heat pump, following energy conservation principle.

4.3. The relationship between climatic conditions and cooling, heating and annual performance coefficient

For buildings with the same construction area, because of the difference of climate condition, in different areas, there is a different annual load distribution, and the proportion of cold and hot load throughout the year is different. It shows from the type of 18 that the greater the cooling load is, the closer the ACOP value is to the SEER, and the greater the heat load ratio is, the closer to the HSPF values. It means that when the annual performance coefficient of system reaches optimal and the form of heat pump is fixed, In order to simplify the calculation, it can base on the geographical position of buildings to use the SEER and HSPF as optimal objective function to replace the ACOP. As for our country, with high heat load and low cooling load on the northeast area, the ACOP is closer to the HSPF.

On the contrary, for the southern area, ACOP is closer to the SEER, while cold and hot load is fairly equal in the north areas of China. In addition, for the heat pump system used in the same building area under different climate conditions or in different parts, it makes different optimal area. It shows that, in the process of development of heat pump product, we should base on different region or climate to design different products to achieve better energy saving and economic effect.

5. Conclusions

(1) Compared with the optimization results of operation mode, the annual energy consumption E value which was in initial design for per unit floor area is the largest, and single conditions optimization EER is in the middle, and the E value with the ACOP as the optimization goal is least. As a result, the design of the heat pump system should put priority on the optimization design variable working condition.
(2) In the design of variable condition, with heat pump annual performance coefficient ACOP optimization design, it can weigh the system's coefficient under various operating conditions during the whole year and make cooling and heating performance, fully describe the performance of heat pump, following energy conservation principle.
(3) For buildings with the same construction area, because of the difference of climate condition, in different areas, the proportion of cold and hot load throughout the year is different, which results in the difference on optimal structure of heat pump system. Therefore, in the optimal design of the heat pump products, it should consider the effect of different areas or different climate conditions, and adopt the SEER and HSPF to optimize instead of ACOP optimization.
(4) While the optimization example selected in paper is not completely conform to the actual condition, however, the conclusion has great reference value.

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