

Optimization of WSHP System Utilizing Waste Heat in Power Plant

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Abstract: The reutilization of industry waste heat and low quality heat in power plant by WSHP system has aroused peoples' widely recognitions recently. The WSHP system has proved to be effective energy economization equipment. There are many heat sources in power plant, most of which are lost because of the two reasons below: 1) the heat in emission gas. It is medium heat resource, about 160°C, and easy to recycle. 2) the heat in recycle water system and water-cool tower. These heat sources' temperature is low, about 35°C. People often do not pay attention to the utilization of this energy because of the low quality, and vent them into the atmosphere. In fact, this is a waste of energy and brings about the heat pollution to environment.

Keywords: waste heat recovery; refrigerants; water source heat pump (WSHP); optimization design

NOMENCLATURE

E_p energy cost of water pump
 H water system resistance
 t_{ch} condensate temperature in heat production
 t_{cr} condensate temperature in heat production
 F heat transfer areas
 COP_r performance factor in refrigeration production
 q_0 refrigeration load
 q_m refrigeration flow
 Q_0 heat load of vaporization
 V_{tr} practical transmissions
 W_0 theoretical work
 N_i indicated work of compressor
 N_L motor power

N_p pump power
 T_a service days a year
 t_{ah} evaporator temperature in refrigeration production
 t_{er} evaporator temperature in refrigeration production
 COP_h performance factor in heat production
 C_i running cost
 q_k heat load
 V_{tr} transmission of compressor
 Q_k heat load of condensate
 π_i inner pressure proportion
 W_i indicated rate of compressor
 N_a axle power of compressor
 COP practical performance factor

1. INTRODUCTION

This paper is based on the study of large low temperature waste heat occurring in turbine circle water located in certain Power Plant, a WSHP system is proposed to recover low temperature waste heat and supply in winter and cool in summer for plant office.

2. THE DESIGN OF WSHP SYSTEM

A certain plant has four boilers and three steam turbines. The emission gas from the 3rd condensing steam turbine exchange heat with water in the steam trap, and the heat in it vent into the atmosphere. If we can take the circle water as low temperature heat source, we can supply the office building heat in winter and cool in summer by heat pump technology.

2.1 The combination of the circle water system and heat pump

This system takes the heat of the 3rd steam turbine circle water as the low temperature heat source of WSHP. According to the arrangement of circulating water system and the character of WSHP, circulating water is drawn into the 2nd heat exchanger directly. If

the pressure is high enough, the 7th pump is not necessary. We can call it parallel connection circulating water system. The sketch diagram is as below:

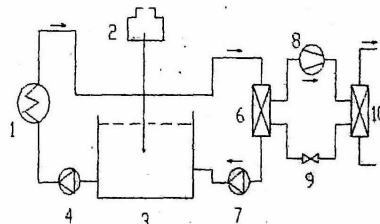


Figure 1 The water source heat pump system diagram using circling water

1—steam trap, 2—cooling tower, 3—circulating water pool, 4—circulating pump, 6—2nd heat exchanger, 7—circulating pump, 8—compressor, 9—expansion valve, 10—3rd heat exchanger

2.2 The arrangement of system terminal equipment

According to the character of WSHP above and the actual conditions, we adopt the combination of a

big WSHP with fan coil cooler, the reasons are as follows:

a) This kind of WSHP and relate equipments are rather mature in China, have more guarantees of the after service.

b) Considered that noise pollution to the office room, the terminal equipment without compressor is more feasible.

c) Because of the cold weather in Dalian in winter, the WSHP system has a temperature request—the new air must be heated before get into the equipment.

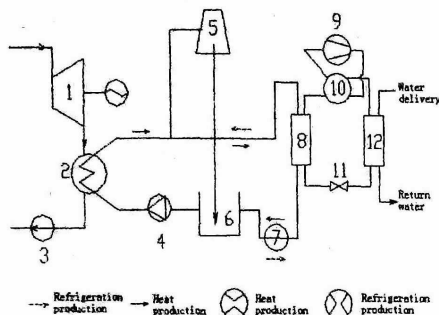


Figure 2 The water source heat pump system diagram using circling water

1-turbine 2-steam trap 3-condensate extraction pump 4-circulating water pump 5-cold tower 6-water jet aspirator 7-circulating water pump 8-heat exchanger 9-compressor 10-cross valve 11-expansion valve 12-heat exchanger

2.3 The calculation of the heat pump

The project is for air-condition in office building, the heating area is about 20 thousand square meters. The working medium is R22, the liquor supercool temperature is 5K; the gas superheat temperature is 10K. Figure 3 represents the pressure-enthalpy diagram. The working temperature of the heat production circulation can be calculated by next formulas:

$$T_k = \frac{T_{cw2} - T_{cw1}}{\exp\left(\frac{T_{cw2} - T_{cw1}}{LMTD}\right) - 1} + T_{cw2}$$

$$T_o = T_{chw2} - \frac{T_{chw1} - T_{chw2}}{\exp\left(\frac{T_{chw1} - T_{chw2}}{LMTD}\right) - 1}$$

The evaporation temperature in heat production (t_{eh}) is 22°C, the condensate temperature in heat production (t_{eh}) is 57°C; the evaporation temperature in refrigeration production (t_{er}) is 3°C, the condensate temperature in refrigeration production (t_{er}) is 41°C.

The parameters of heat production and cool production are listed in table 1 and table 2. The thermodynamic results are listed in the table 3.

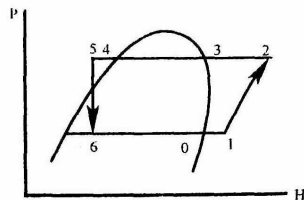


Figure 3 The P-H scheme of heat pump thermal circulation

The figure is the pressure-enthalpy of the practical circulation, which is composed by four parts: 1) compress 1-2; 2) condensation 2-3-4-5; 3) restricted flow; 4) vaporization 6-7-1.

Table 1 The thermal parameter of each point for heating

Phase Points	P(kPa)	t(°C)	h(kJ/kg)	v (m³/kg)
0	961.9	22	412.5	0.0259
1	961.9	32	420.6	0.0261
2	2272.8	—	448.3	
2a	2272.8	81	—	
3	2272.8	57	418.0	
4	2272.8	57	266.0	
5	2272.8	52	259.6	
6	961.87	22	259.6	

Table 2 The thermal parameter of each point for cooling

Phase Points	P(kPa)	t(°C)	h(kJ/kg)	v (m³/kg)
0	548.05	3	406.5	0.0429
1	548.05	13	413.8	0.0452
2	1424.30	—	442.9	
2a	1424.30	63	438.6	
3	1424.30	37	415.9	
4	1424.30	37	239.2	
5	1424.30	32	232.8	
6	548.05	3	232.8	

Table 3 The thermal calculation of system circulation

Order	Item	Heat production	Refrigeration production
1	q_o (kJ/kg)	161.0	180.9
2	q_k (kJ/kg)	188.7	210.1
3	q_m (kg/s)	6.4	7.18
4	V_r (m³/h)	602.2	1169.3
5	QO (kW)	1032.7	—
5	Qk	—	1509.2

5	V_{v1}	661.7	1284.9
(volumetric proportion, $\eta_v=0.91$)			
5	π_1	2.36	2.60
6	$W_0(\text{kJ/kg})$	22.2	24.8
7	W_1	27.7	30.9
11	N_1	177.8	222.3
(indicated efficiency, $\eta_i=0.8$)			
12	N_e	197.6	247.0
(mechanical efficiency, $\eta=0.9$)			
13	N_L	219.5	274.4
(motor efficiency, $\eta_c=0.9$)			
14	COP	5.47	4.74

The heat removed by circulation water when refrigeration produces is about 1511 KW, the needed water is about 91.8t/h, the power of compressor is about 275 KW; when heat produces, 1033 KW heat is necessary, the needed water is 58.2t/h, the power of the compressor is about 220KW.

3. THE OPTIMIZATION OF WSHP

3.1 The optimization of thermodynamics efficiency

After the simulation of heat pump system on the basis of pinch analysis, the relationship between COP and heat transfer area can be represented by the figure 4 below. From this chart by MATLAB, we can easily find that: at beginning, the COP has a direct proportion with heat transfer areas, the bigger the area is, the higher the COP is; with the development of heat transfer areas, the curve line becomes flat and the increase of COP is limited. So, COP and heat transfer areas should both be taken into consideration.

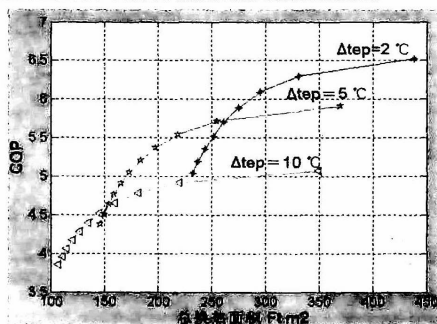


Figure 4 The relation curve of COP and heat transfer areas

According to the discussion above, the optimization aim is to increase COP under the condition of constant heat transfer areas.
 $COP_{total} = \eta_h COP_K + \eta_r COP_r$
 COP_{total} : the compositive performance factor;
 COP_K : the performance factor in heat production;

COP_r : the performance factor in refrigeration production;

η_h, η_r —the factor number, $\eta_h \gg 0, \eta_r \gg 0$,

$$\eta_h + \eta_r = 1$$

η_h and η_r depends on the importance of COP_r and COP_h , and should be affirmed by the running conditions and local weather conditions.

the objective function is:

$$f = 1 / COP_{total} = 1 / (\eta_h COP_K + \eta_r COP_r)$$

the final boundaries function is:

$$\min(f, \Delta tep, \Delta tcp)$$

$$\text{s.t. } F(\Delta tep, \Delta tcp) = C$$

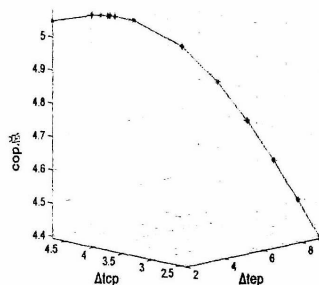


Figure 5 The relation curve of $\Delta tep, \Delta tcp$ and COP_{total} .

From this chart, the COP_{total} depends on the pinch pinch temperature contrast under the conditions of constant heat transfer areas.

3.2 The optimization of thermodynamics efficiency

The heat economic optimization for the WSHP should balance the investment fee and running fee. Its optimization object is minimizing the calculation cost of output charge per unit. That is:

$$\min(f) = (c_i + c_o + c_y) / (\beta_1 Q_h + \beta_2 Q_r)$$

c_i and c_o are the average investment fee and energy cost of the system per year. c_y represents the other cost. The whole cost per year C represents the summation of yearly cost c_i and yearly running cost c_o and c_y .

The yearly cost index is: $CRF(i, n) = i / [1 - (1+i)^{-n}]$

" i " represents yearly rate. " n " represents service year.

$$c_i = CRF(i, n) (Ceva + Ccon + Ccom + Cpum)$$

$Ccom, Ceva, Ccon, Cpum$ represents compressors invest, vaporization invest, condensates invest and water pump invest respectively.

β_1, β_2 are weighting factors, they depend on the importance of Q_h and Q_r .

$$\beta_1 = t_h / (t_h + t_c)$$

$$\beta_2 = t_c / (t_h + t_c)$$

The cost of vaporization and condensate is linearly related with its heat transfer areas respectively.

$$Ceva = a_1 Feva + b_1$$

$$Ccon = a_2 Fcon + b_2$$

The other formulas are as follows:

$$Ccom = c_1 \max(Ech, Ecr)$$

$$Cpum = d_1 \max(Gwh, Gwr)$$

$$c_o = c_t h(Ech + Eph) + c_r t_c(Ecr + Epr)$$

The energy cost of water pump can be calculated by the next formula:

$$E_p = N_p \times T_a = \frac{H \cdot G}{\eta} \times T_a$$

N_p represents the power of the pump, H represent the water system resistance, G represents water flow, T_a represent service time a year, η represents the efficiency of the water pump.

3.1 Given

1) Designing conditions: refrigerating capacity in summer, the temperatures of supply and recycle cold water, heat capacity in winter, the temperatures of supply and recycle hot water.

2) The state of the system can be calculated by the next 6 parameters: vaporizing temperature t_{er} and t_{eh} ; condensate temperature t_{cr} and t_{ch} ; the heat sources recycle water temperature t_{wr} and t_{wh} . And then we can figure out the heat transfer areas, theoretical transmission of compressor and electric motor efficiency.

3) The heat transfer areas in condensator and evaporator are constant.

3.2 Boundaries

1) The condensate temperature is higher than evaporation temperature

2) In condensator, the condensate temperature is higher than output temperature; the output temperature is higher than input temperature.

3) In evaporator, the evaporation temperature is lower than output temperature; the output temperature is lower than the input temperature.

3.3 The qualitative analysis of system optimization

Usually, the equipment running efficiency contradicts with economic economization. There are many influential factors of system optimization, and we should grasp the most influential factor.

Let us take the example that is mentioned in chapter 2 as the bases of the system optimization. The table 4 shows the differences between traditional method and optimization method.

Table 4 Comparison of the results between the optimization & tradition design methods

Units	tkh °C	teh °C	tkr °C	ter °C	Neh kw	Ner kw	Fa m ²	Fb m ²
Traditional method	57	22	37	2.8	273	206	110	118
Optimum method	56	23	36	4	258	190	137	161
Units	coph	copr	ci	Ee	Ce	Npr	Nph	f
			Yuan/y	Yuan/y	kw/h/y	kw	yuan/kw	
Traditional method	4.39	6.31	281227	140613	26.25	164.3		
Optimum method	4.64	6.84	264931	132465	26.45	141.3		

From the table below, we can easily find that compared to the traditional method, the heat transfer areas of heat exchanger A and B have increased. But, because of the reduction of running energy cost and the compressor cost, the calculation costs of output charge per unit nearly reduce 14%.

4. TECHNIQUE COST ANALYSIS OF WSHP

4.1 The comparison of running cost between WSHP and water cooling air-condition

The principle of WSHP is similar to water-cooling air-condition in summer. But the water cooling system in WSHP takes the circulating water in power plant as the heat source, so it saves the investment of cooling tower. Because of the high temperature of circulating water in summer, the COP of WSHP is similar to the water-cooling air-condition, and the effect of energy economization is not apparent.

4.2 The economic calculation of operating cost in winter

The steam that is used for office heating in winter is about 1270kg/h. If we can utilize WSHP to take the place of these steam, we will save steam 3962.4t in all, and sales profit can increased to 396240 yuan.

The evaporation lost can be reduced to 545t, and the related water fee is about 2000 yuan.

The reduction of electric profit is about 137280 yuan in all. And the profit in all is about 260879 yuan in the whole heating seasons (130 days).

5. CONCLUSION

According to analysis above, the conclusions obtained are as follows:

1) The practical performance factor does not have a direct ratio with heat transfer area. And it is not evident to improve COP by increasing heat transfer areas.

2) Compared to the ordinary air-condition system, the WSHP can save about 260 thousands yuan every winter. It is an ideal route for saving investment, resulting in reduction of groundwater and minimization of environmental pollution.

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