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Simulation and Analysis of a Novel Cascade Double Cooling Source Temperature and Humidity Independent Control Air-conditioning System

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Simulation and Analysis of a Novel Cascade Double Cooling Source Temperature and Humidity Independent Control Air-conditioning System

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Abstract: In view of the climate characteristics of hot summer and cold winter area, a novel cascade double cooling source air- conditioning CDCS system is proposed in this paper. The system can achieve heat-humidity independent control in hot and humid summer and high efficiency heating in cold winter. In this paper, theoretical analysis and Aspen Plus simulations are carried out to study the refrigeration of the system under various operating parameters and climatic conditions, meanwhile the comparison between the proposed system and two conventional systems is performed. Results demonstrate that the refrigerating efficiency of the novel system are 52.7% and 22.7% higher than that of the conventional single-cold air-conditioning and dual-cold air-conditioning, under the optimal design parameters.

1. Introduction

In recent years, with the development and deepening of the independent control theory of temperature and humidity, the energy-saving problems of centralized cooling and heating in hot summer and cold winter areas have attracted wide attention at home and abroad. In hot and humid summer, the air conditioning system generally adopts the traditional condensation dehumidification method (using 7℃ chilled water) to achieve the cooling and dehumidification of air. In order to avoid too low temperature of air supply, the air after treatment often needs to be reheated, which results in a great waste of energy utilization grade. In addition, condensation dehumidification will seriously affect the quality of indoor air [1]. For heating in winter, excessive high-low pressure difference of heat pump will lead to difficulties in starting, heat attenuation, system performance degradation, frequent defrosting and other phenomena. In order to solve these problems effectively, a series of new devices and technologies based on the theory of heat and moisture control have been applied and studied extensively [2-6].

Academician Jiang Yi of Tsinghua University proposed centralized dual-cold-source air conditioning unit [7-8]and solution dehumidification air conditioning unit [9]. Both of them raise the temperature of cooling water under sensible heat load from7°Cto18°C, compared with the conventional air conditioning system, so that the performance coefficient of the whole machine has been improved obviously. However, the operation condition of the low temperature stage of the double cooling source air conditioning unit are still the same as those of traditional air-conditioning unit, so there is room to improve its performance coefficient, while the solution dehumidification air conditioning unit is limited by the need of providing additional high temperature heat source to regenerate the solution. In winter heating, the optimum design parameters and reasonable operation control scheme of the two-stage coupled heat pump need to be further improved [10-15]. Based on the above researches, a novel cascade double cooling source air-conditioning the CDCS system designed in this paper. The system not only increases the evaporation temperature of high temperature stage, but also decreases the condensation temperature of low temperature stage in summer. Under the condition of heating in winter, it can also realize the graded heating from low temperature to high temperature.

2. CDCS system introduction

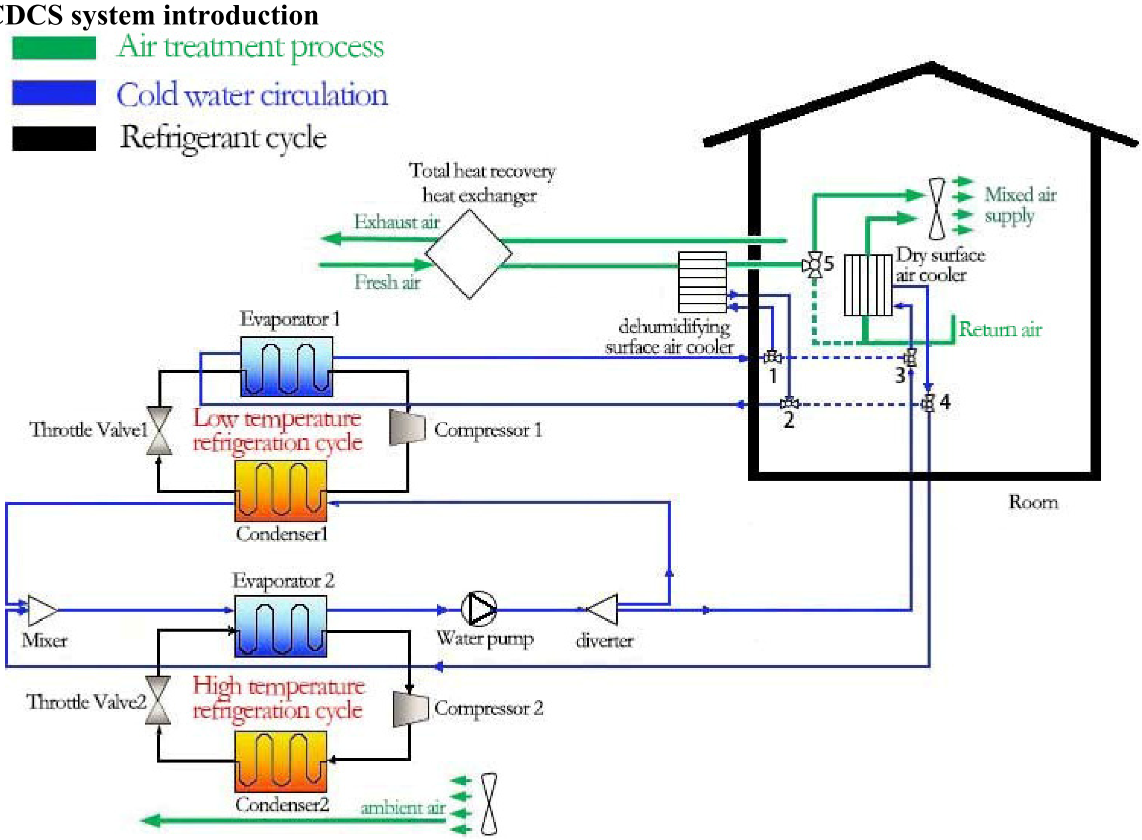

Fig. 1 Schematic diagram of the CDCS system

As shown in Fig. 1, the CDCS system consists of two single-stage refrigeration cycles and an intermediate water loop.

2.1. Under the summer refrigeration condition, the low temperature stage produces (7-12℃) chilled water to take away the latent heat load in the room, while the high temperature stage produces (16-21℃) Cooling water to bear the Indoor sensible heat load and the condensation load of the low temperature stage, thus increasing the COP of the two chillers and realizing Independent control and regulation of temperature and humidity. The air treatment process can be seen in Fig. 2: W→W' and W'→L are the cooling and dehumidifying processes of fresh air in full heat recovery unit and dehumidifying surface air cooler respectively; N→P is the cooling process of return air in dry surface air cooler; P→O←L is the process of mixing fresh air and return air to supply point O.

2.2. Under the heating condition in winter, through the switching of four-way valve and distributor, the medium temperature hot water (17-21 ℃) produced by the first stage (air/water) heat pump unit is supplied to the intermediate water loop as the low-temperature heat source of the second stage (water/water) heat pump to prepare high temperature hot water (45-55℃). Then, through switching the water valve 1-4 in water system and the reversing valve 5 in the end of the fresh air system (shown in the dotted line part of Fig. 1), the high-temperature hot water is supplied to the indoor fan coil unit, so that the fresh air and the return air are mixed and then reheated. The air treatment process can be seen in Figure 3: W→W' is the heating and humidifying process of fresh air in the total heat recovery unit; W'→L is the isothermal humidifying process of fresh air; N→P←L is the mixing process of fresh air and return air; P→O is the process of the mixed air being heated to the supply point O [16].

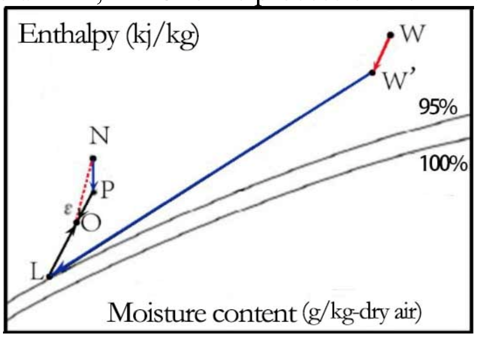

Fig. 2 air treatment process of the CDCS system in Summer

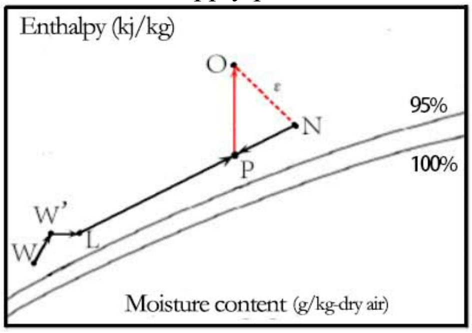

Fig. 3 air treatment process of the CDCS system in Winter

3. System theory analysis

In order to obtain high temperature cold water at 18°C, the evaporation temperature t_{02} of the high temperature stage can be set to 16°C. Because there are two stages of heat transfer between condensation temperature t_{K1} of the low-temperature stage and evaporation temperature t_{02} of the high temperature stage, the temperature difference between them can be set to 10 °C. The preliminary theoretical working conditions are as follows: low temperature stage: t01=5°C, tK1=26°C, high temperature stage: t02=16°C, tK2=50°C. Assuming that the load ratio of high and low temperature stage is 6:4, the theoretical refrigeration COP of the CDCS system can be calculated: COP=12.2×0.4+7.17×0.6=9.182, which is 1.84 times of that of conventional air conditioning (COP=5) under the same working condition.

4. **Simulation of refrigeration performance of the CDCS system in summer**

Taking an office building in Nanjing as an example, the building has 12 floors, and the total air conditioning area of the building is 900 ㎡. In summer, indoor design temperature T_n = 26°C, moisture content d_n = 12g/kg, personnel density is 0.125 person/m^2. According to the standard of 30 m^3/h per person, the fresh air volume of the whole building is calculated as follows: 30 m3/ (person• h) × 0.125 person/m^2×900m^2= 3375m^3/h. Summer cooling load Q=45 kw, wet load W=11800 kg/h, latent heat recovery efficiency of total heat recovery exchanger 55%, sensible heat recovery efficiency 60%, the overall cooling efficiency of the CDCS system is defined as follows:

$$COP_{CDCS}=\frac{Q_{O1}+Q_{O2}}{W_1+W_2+W_{water}}$$

In the upper model, $Q_{01}$and Q_{02} are summer cooling capacity and winter heating production of high and low temperature stage, respectively. W_1, W_2 and W_{water} are the compression power of high temperature stage and low temperature stage and the total power consumption of pumps, respectively.

4.1. Calculation of Initial conditions for simulation

1. Heat-humidity ratio： $\varepsilon=\frac{Q}{W}=\frac{45000}{3.28}=13700$, temperature difference between supply air and return air $\Delta t=t_n-t_o=8$℃, O points can be determined；
2. w' point can be determined by $t_{w'}=t_w-(t_w-t_n)60\%$ and $h_{w'}=h_w-(h_w-h_n)55\%$;
3. In order to make fresh air take away the residual humidity in the room, it is necessary to treat fresh air to L point：

$d_L=d_n-\frac{W}{\rho Gw_{new\text{-}air}}$=9.1g/kg，且$\varphi_L$=95%，L points can be determined；

4. In the enthalpy-wet diagram, the isohumid line crossing N points intersects p points with the extension lines of O and L points；

5、Total air volume ：$Gw_o=\frac{45}{C_P\times(t_N-t_O)}$=16800 m^3/h，Where C_P is the constant pressure specific heat capacity of air；

6、Return air volume：$Gw_{return\text{-}air}=Gw_o-Gw_{new\text{-}air}=13425\ m^3/h$；

7、Indoor partial cooling load borne by fresh air ：$Q_X=C_P\times(t_N-t_L)\times Gw_{new\text{-}air}$=14.4KW；

8、Cold Load of Dry surface air cooler : $Q_{dry}=Q-Q_X$=30.6KW；

From the above calculation, the states of each points in Fig. 2 are shown in Table 1.

Table 1 State parameters of each points in Fig. 2

State point	Dry bulb temperature (℃)	Wet bulb temperature (℃)	Relative humidity (%)	Moisture content	Enthalpy (KJ/Kg)
W	34.8℃	28.05	60.28	21.30	90
W’	29.5	23.91	63.13	16.41	71.85
N	26	19.88	57.10	12	57
L	13.3	12.95	95	9.1	36.5
P	19.9	18.04	83.75	12	51
O	18	16.66	87.63	11.3	46.8

Set the following parameters: t_{01}=5℃, t_{k2}=50℃, the temperature difference between t_{k1} and t_{02} is 10 ℃, the temperature difference between high temperature cooling water supply and return water is 3℃, and the temperature of low temperature chilled water supply and return water is 7 and 12℃, The design temperature of t_{k1} and t_{02} can be 26/16, 25/25, 24/14, 23/13, 22/12, 21/11, 20/10 (℃), respectively. The energy consumption and efficiency of the whole machine under the above 7 design parameters are studied below.

4.2. Summer refrigeration simulation based on Aspen Plus

Using Aspen Plus to establish the simulation process is shown in Figure4. The solid line in the figure represents the mass flow and the dotted line is the energy flow. The nomenclature of each component is expressed by its first letters. Dry surface cooler is represented by counter-flow heat exchanger, dehumidifying surface air cooler is represented by counter-flow heat exchanger and gas-liquid separator. Resistance loss in water system is equivalent replaced by throttle valve. Physical property method is PENG-ROB [17]. Based on the above calculation results, the specific parameters and initial logistics parameters of each equipment are obtained. The process is finally converged iteratively in Aspen by sequential modular method. The simulation results are shown in Table 2.

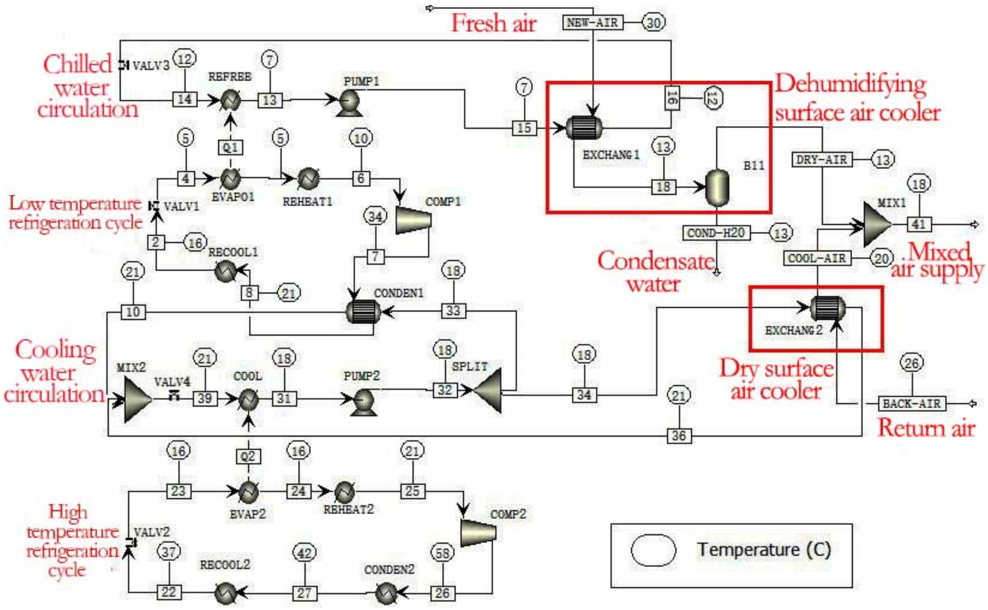

Fig.4 Flow chart of simulation of refrigeration process for the CDCS system in summer

Table 2 Simulation results of refrigeration process for the CDCS system in summer

T01	°C				5			
Tk1	°C	26	25	24	23	22	21	20
To2	°C	16	15	14	13	12	11	10
Tk2	°C				50			
$GmR22_{low}$	kg/s	0.2282	0.2266	0.2260	0.2250	0.222	0.2205	0.2191
$GmR22_{high}$	kg/s	0.479	0.478	0.478	0.478	0.477	0.477	0.478
G chilled water	kg/s	1.9	1.9	1.9	1.9	1.9	1.9	1.9
G cooling water	Kg/s	5.788	5.766	5.756	5.746	5.725	5.694	5.803
$Q0_1$	kw	41.7348	41.7328	41.7322	41.7320	41.7318	41.7312	41.7309
$Q0_2$	kw	73.6218	73.602	73.425	73.2456	73.065	72.73	72.696
W_1	kw	3.655	3.478	3.32882	3.1426	2.9079	2.724	2.592
W_2	kw	11.17	11.578	11.9652	12.3562	12.751	13.1212	13.55
W_{water}	kw	2.9051	2.8948	2.889	2.8832	2.8734	2.8598	2.8612
Area of Desiccant Meter Cooler	M^2	5.13105	5.0750	5.1320	5.2257	5.6200	5.3030	5.0800
Area of dry surface cooler	M^2	9.8960	7.8600	6.0393	5.0500	4.2300	3.6050	3.3900
Area of Condenser 1	M^2	7.8780	7.5551	7.5460	7.5328	7.6600	7.3470	7.4900
COP_{CDCS}	1	6.5062	6.4250	6.3332	6.2549	6.1944	6.1192	6.0214
Air supply temperature	°C	18.2	18.22	18.3	18.2	18.1	18.2	18.1
Air supply humidity	g/kg	10.9	10.9	11.2	10.9	10.7	10.9	10.7

Table 2 shows that the air supply parameters meet the design requirements. Moreover, the exhaust temperature of high and low compressors in the new system is only 40.4℃and 69.5℃, which is much lower than the exhaust temperature of compressors in conventional air-conditioning units (74.3℃). Therefore, compared with conventional air-conditioning units, the new system can operate more stably.

4.3. Refrigeration process simulation results in summer analysis

From Fig. 5 and Fig. 6, it can be seen that with the decrease of design temperature of T_{02} and t_{k1}, Q_{01} in low temperature stage has light microwave motion, but W_1 shows a downward trend; Q_{02} in high temperature stage slowly decreases while W_2 rapidly rises, and w-water consumption of water pump slightly increases, which eventually leads to the rapid increase of COP_1 in low temperature stage, while COP_2 in high temperature stage slowly decreases, because the load ratio of Q_{01} to Q_{02} is 0. 32:0.68, the load ratio of high temperature stage is significant, so the COP_{CDCS} of the whole machine shows a downward trend. In summary, the COP_{CDCS} of the whole machine reaches its maximum value 6.5062 in summer, when T_{K1} = 26℃ and T_{02} = 16℃. Because the COP of the cryogenic refrigeration cycle is much larger than that of the high temperature refrigeration cycle, it is necessary to increase the load ratio of the cryogenic refrigeration source as much as possible in order to improve the COP_{CDCS} of the whole machine. Therefore, this system is especially suitable for summer climate where low temperature cold source is used to dehumidify air.

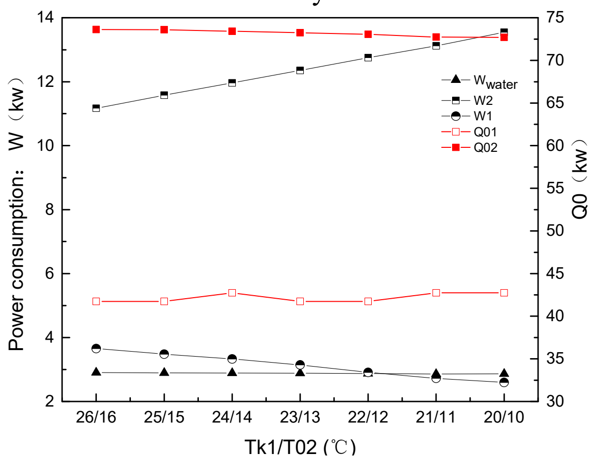

Fig. 5 Changes of refrigeration capacity and power consumption under different T_{02} and $_{tk1}$

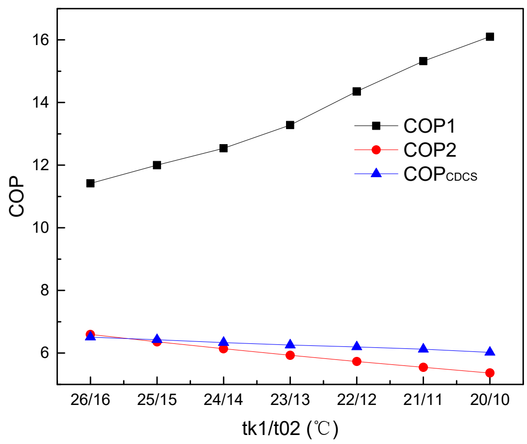

Fig. 6 Changes of COP under different T_{02} and T_{K1}

4.4. Operation characteristics and energy saving effect of CDCS system under refrigeration conditions

1) Under the above optimal design parameters, this paper continues to simulate the operating characteristics of the new system under variable operating conditions. From figs.7 and 8, it can be seen that with the increase of outdoor air temperature, the refrigeration capacity Q_{02} decreases rapidly, resulting in the rise of intermediate cooling water temperature, which leads to the decrease of refrigeration capacity Q_{01} in the low temperature stage, and ultimately leads to the rise of the temperature of the chilled water. COP_1, COP_2 and COP_{CDCS} of the whole machine decrease with the increase of outdoor temperature, and the increase of outdoor temperature has more severe influence on the high temperature stage of the CDCS system.

2) Under different outdoor temperature and meteorological parameters, this paper compares the CDCS system with traditional air conditioning and dual-cooling source air conditioning [18]. As shown in Fig.8, under 6 different outdoor conditions, the COP_{CDCS} is 52.7% and 22.7% higher than that of the traditional air conditioning and the dual-cooling source air conditioning, respectively. This is because the evaporation temperature T_{02} of the high temperature stage rises from conventional 5℃ to 16℃, and the condensation temperature T_{k1} of the low temperature stage decreases from conventional 50 ℃ to 26 ℃. With the increase of outdoor temperature, the COP of the three systems will decrease, but the

COP_{CDCS} of the new system will decrease relatively quickly. This is because the outdoor temperature rise will not only affect the efficiency of high temperature stage, but also affect the efficiency of low temperature stage.

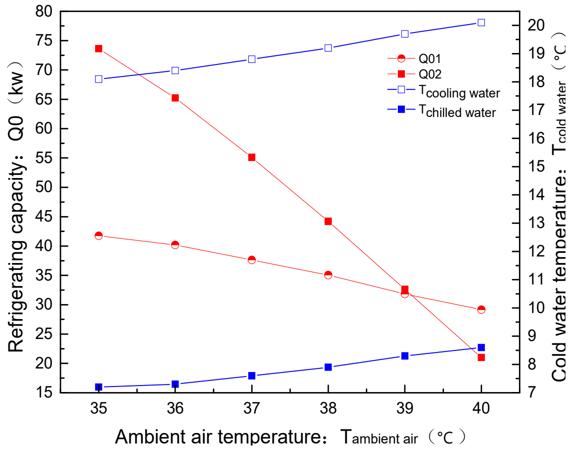

Fig.7 Changes of Q_0 and T cold water at different ambient air temperature

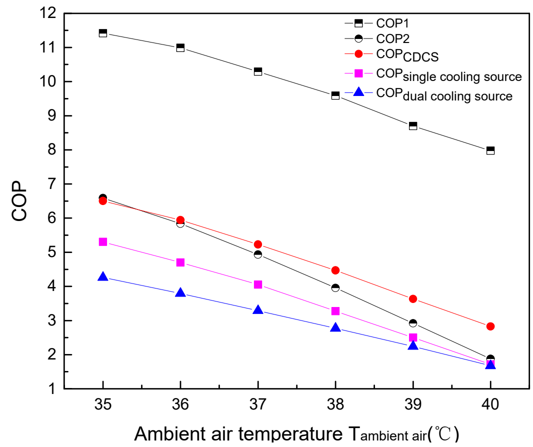

Fig.8 COP comparison between CDCS system and conventional system

5. Conclusion

A Novel Cascade Double Cooling Source Temperature and Humidity Independent Control Air-conditioning the CDCS System is proposed in this paper. The following conclusions are drawn from theoretical analysis and Aspen Plus simulation:

(1) Under the summer refrigeration condition in Nanjing, the optimum design parameters of the new system are: low temperature stage t_{01}=5°C, t_{K1}=26°C, high temperature stage t_{02}=16°C, t_{K2}=50°C. The average COP_{CDCS} of refrigeration is as high as 5.8， which are 52.7% and 22.7% higher than that of the COP of the conventional single-cold air-conditioning and dual-cold air-conditioning, respectively.

(2) Compared with traditional air conditioning and dual-cooling source air conditioning system, the pressure ratio of the CDCS system at all levels is smaller, and the exhaust temperature of compressor is lower. However, the fluctuation of outdoor temperature has a more severe impact on it, so it is necessary to rationally distribute refrigerant flow at high and low temperatures stage to ensure the stable operation of the unit under variable operating conditions.

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