

Experimental Study on a Novel Defrost Cycle for Air Source Heat Pump

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ABSTRACT

On the basis of the hot-gas bypass defrost cycle, a new defrost method for air-source heat pump is proposed. In the new defrost cycle the outdoor heat exchanger is divided into two parts which are connected by a capillary tube. The front and rear parts of the heat exchanger are used as the evaporator and condenser respectively during defrosting, and can be defrosted orderly by using the four-way valve. The defrosting performance of the new cycle is investigated experimentally and the results are compared with that of the reverse-cycle defrost system. The experimental results indicate that the energy is used more efficiently in the new defrost cycle, therefore, the defrosting duration and losses are less than these of the reverse-cycle defrost system. Moreover, it does not extract heat from the indoor space during defrosting in terms of the new defrost cycle. Compared with the reverse-cycle defrost system, it is found that the switching times of the four-way valve in the new defrost system are the same, and the fluctuation of the discharge and suction pressures during the defrosting is much less than that in the reverse-cycle defrosting, hence, the mechanical impact on the system is much less.

Keywords: Defrost, New defrost method, Defrosting characteristics; Air source heat pump.

INTRODUCTION

Air-source heat pump (ASHP) is widely used with profit of its advantage in energy saving and environmental conservation. However, when the outdoor temperature is around 0°C, the frost formation on the heat exchanger becomes seriously, and the heating capacity and the average coefficient of performance (COP) decrease significantly [1]. Therefore, it is necessary to remove the frost periodically, and the defrost energy loss is usually more than 10%. So it is of great significance to improve the defrosting efficiency.

At the present time, the widely used defrosting methods are the reverse-cycle defrost (RCD) and the hot-gas bypass defrost (HGBD) methods. The

defrosting duration for the RCD method is less than that for the HGBD method, but defrosting by RCD method compromises the indoor thermal comfort because of the extracted heat from the indoor space [2]. The major problem with HGBD cycle is that an excessive amount of the liquid refrigerant may enter the compressor, resulting in compressor failure. To eliminate the problem Krakow [3] presented a hot-gas bypass defrost system incorporating a vaporizer. In the system a heat exchanger, termed a “vaporizer”, is incorporated to evaporate liquid refrigerant before it enters the compressor. The energy used to evaporate the liquid refrigerant is obtained from refrigerant leaving the compressor before it enters the condenser. The experimental results indicated that the defrosting system may operate in a steady state mode indefinitely without adverse effects on equipment. Mei [4] proposed a new defrosting technology that does not rely on heat taken from the conditioning space of the house by adding heat to the liquid refrigerant in the accumulator. The accumulator became an evaporator, providing heat for coil defrosting. An obvious advantage is that there is not a cold blow phenomenon during the defrosting period. Liang [5] presented a new sensible heat defrost method which uses the sensible heat of the hot refrigerant vapor from the compressor to defrost. The comparative experiment between the sensible heat defrost cycle and conventional reverse-cycle defrost shows that the sensible heat defrost performed well and shortened the defrosting duration. It overcame fluctuating working condition of reverse-cycle defrost and solved the problem of large amount of oil rushing out from compressor when defrosting. These defrost methods above mentioned offer a new ideas of improving the defrosting efficiency, but the more investigation is needed to find the possible problem.

On the basis of the hot-gas bypass defrost cycle, a new defrost method for ASHP is proposed in this paper. The defrosting energy is used more efficiently in the new defrost cycle and the problem of liquid refrigerant flood back to the compressor is avoided. The pressure fluctuation range during defrosting is reduced significantly and the reliability of the ASHP is increased.

THE NEW DEFROST CYCLE AND EXPERIMENTAL PROTOTYPE

The new defrost cycle is based on the conventional hot-gas bypass defrost scheme, and the outdoor coil is divided into two parts which are connected by a capillary tube and solenoid valve in parallel. The schematic diagram of the new defrost cycle is shown in Figure 1. When the heat pump unit operates in the normal cooling/heating mode, the bypass valve is closed and the solenoid valve opened synchronously. While the heat pump operates in defrosting mode, the bypass valve is opened and the solenoid valve closed synchronously. The defrosting period is divided into two stages. In the first stage, the hot refrigerant vapor leaving the compressor flows through the bypass pipe firstly into the front part of the outdoor heat exchanger. It is condensed in the coil, and then throttled by the capillary tube and evaporated in the rear part of the outdoor heat exchanger. The front and rear parts of the outdoor coil act as a condenser and an evaporator, respectively. In the second defrosting stage, the four-way valve is switched and the hot refrigerant vapor from the compressor flows firstly into the rear part of the outdoor coil, and the rear pipes act as a condenser, while the front pipes act as an evaporator. By the way the frost on the front and rear parts of the outdoor coils can be melted orderly. The new defrost method is termed grouping-throttle defrost (GTD) cycle provisionally. The defrosting performance of the ASHP with GTD cycle is investigated experimentally in this paper.

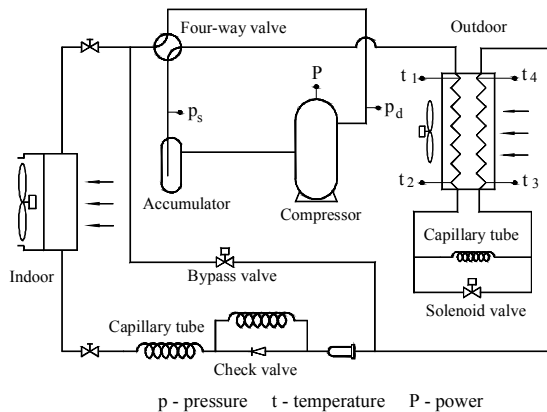


Figure 1
Schematic diagram of new defrost cycle

The experimental prototype is reconstructed from a split type air conditioner. The rated input power of the rotary compressor is 710W. The air flow rate through the outdoor heat exchanger is 610m³/h under the frost-free condition. The maximum air flow rate of the 3-speed indoor fan is 500m³/h. The outdoor and indoor coils are the wavy finned-tube heat exchangers, and the parameters of the outdoor heat exchanger are shown in the table 1.

Table 1
Parameters of outdoor heat exchanger

Pipe Diameter (mm)	Pipe Pitch (mm)	Fin Spacing (mm)	Fin Thickness (mm)	Pipe Length (mm)	Pipe row/column number
9.52	25.4	1.56	0.1	760	20 / 2

EXPERIMENTAL SET-UP AND PROCESS

Experimental apparatus: The experiments were conducted in the two psychrometric rooms, in which the air temperature and relative humidity are adjusted by the two sets of air handling units (AHU). Each AHU contains a cooling coil, an electrical heater, and a steam humidifier, which are controlled by a computer program using PID control algorithm to keep the ambient air parameters at the desired values. The experimental system is shown in Figure 2.

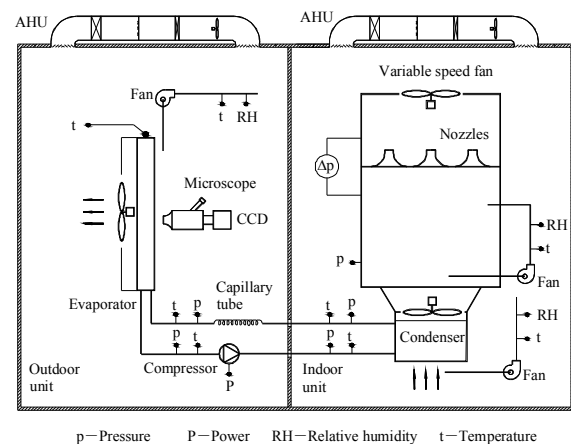


Figure 2
Schematic diagram of the experimental set-up

The indoor unit is located in the indoor psychrometric room and its discharge air is fed into a chamber, and then is exhausted into the room through the nozzles by a variable speed fan. The air flow rate through the indoor exchanger is measured by the multiple nozzles, across which the pressure drop is detected in terms of a precision difference pressure transducer ($\pm 0.08\%$). The outdoor unit is located in the outdoor psychrometric room. To obtain the air enthalpy and relative humidity, the air dry/wet bulb temperatures in both of the psychrometric rooms and in the outlet of the indoor heat exchanger are measured in terms of Pt 100 Ω RTD sensors ($\pm 0.1^\circ\text{C}$) in the sampling ducts, in which the velocity is about 5 m/s. The power consumption of the test heat pump is measured in terms of a precision power transducer ($\pm 0.5\%$). The pipe wall temperatures of the outdoor heat exchanger are measured by using the pre-calibrated T-type copper-constantan thermocouples ($\pm 0.1^\circ\text{C}$) which are stuck on the copper elbows of the evaporator. The refrigerant-side pressures and temperatures are measured in terms of precision pressure transducers

($\pm 0.08\%$) and Pt 100 Ω RTD sensors ($\pm 0.1^\circ\text{C}$) respectively. The frost thickness was obtained from processing the images captured with the CCD camera through a microscope with a magnification of 90 times on a front view of the outdoor heat exchanger. The frost growth and melting on the front and rear fins/tubes are observed and recorded by two CCD cameras, respectively. The defrosting time is recorded by the computer clock.

Experimental operating conditions and process:

The research work of Guo [1] indicated that the frost growth rate is maximized, therefore, the performance of the ASHP unit degrades rapidly when the outdoor air temperature is about 0°C . So in the experiments the outdoor air temperature and the relative humidity are selected in the ranges from -7 to 2°C and 60% to 90%, respectively. In order to compare the characteristics of the two defrosting cycles, the defrosting experiments for RCD and GTD were conducted under the same conditions, respectively.

Before performing the defrosting experiments, the frosting experiments were conducted. When the distance between the frost surfaces on the two neighboring fins was about 0.2mm, the defrosting experiments were started by switching the four-way valve from heating to cooling mode for RCD cycle, and by opening the bypass valve and closing the solenoid valve synchronously for GTD cycle. In the defrosting mode, the outdoor fan was stopped and the indoor fan operated in low speed mode. For the GTD cycle, after the frost on the front part of the outdoor heat exchanger was melted completely, the four-way valve was switched from heating to cooling mode to melt the frost on the rear pipes and fins. The defrosting process for RCD and GTD cycles was added about 2 minutes to drain water on the fins and tubes after the frost was melted completely. The defrosting cycle was terminated by switching the four-way valve from cooling to heating mode, in addition, closing the bypass valve and opening solenoid valve for GTD cycle, so the next heating cycle was started.

The experimental results indicated that at the ends of frosting periods, the frost accumulation mass on the outdoor coil for the two defrosting cycles, which was measured according to the reference [1], was nearly equivalent under the same conditions.

EXPERIMENTAL RESULTS AND DISCUSSION

The defrosting performances for the RCD and GTD cycle under the typical conditions, i.e., 2°C outdoor air temperature and 80% relative humidity are compared in the following aspects.

Comparison of the suction and discharge pressures of the compressor:

The dynamical suction and

discharge pressures of the compressor during the defrosting period with the two defrosting methods under the typical conditions are shown in Figure 3. It can be seen that the defrosting duration with the RCD cycle is about 440 seconds, and 430 seconds with the GTD cycle, which is slightly shorter than that for RCD method. For the GTD cycle, the defrosting time for the front part of the outdoor coil is about 330 seconds, and 100 seconds for the rear part. In fact, the defrosting mode for RCD cycle was terminated some earlier in the experiment, resulting from the judgment error in the melting of frost on the coil. Hence, the time for draining water during defrosting with RCD cycle is less than 120 seconds. This problem will be further discussed later combining with the frost images on the coil.

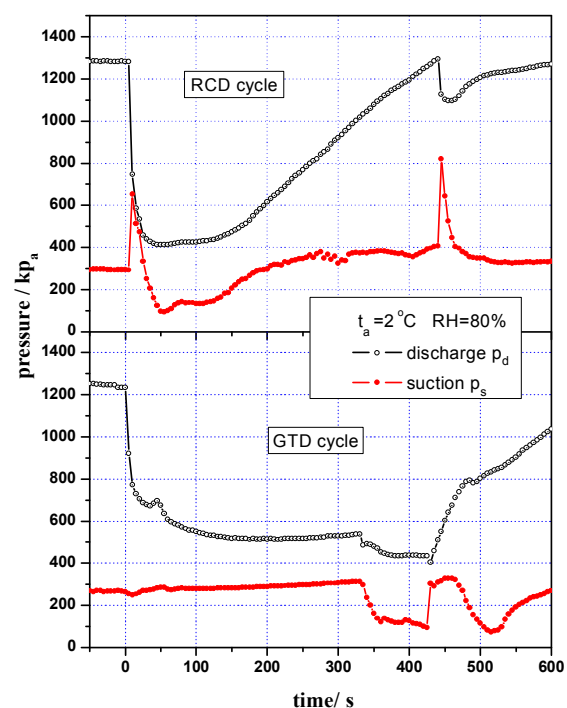


Figure 3
Dynamic suction and discharge pressures

For the RCD cycle, the suction and discharge pressures are balanced quickly after the four-way valve is switched from the heating mode to the cooling mode, and then the suction pressure quickly dropped. After about 50 seconds from the beginning of defrosting, the suction and discharge pressures begin to rise slowly with the time, and the rising rate of the pressures becomes more and more. In the end of the RCD cycle, the four-way valve is switched from the cooling mode to the heating mode, which results in the large fluctuation of the suction and discharge pressures, especially of the suction pressure. For the GTD cycle, after defrosting is started by closing the solenoid valve and opening the bypass valve synchronously, the discharge pressure dropped quickly to a steady state, while the suction pressure is nearly invariable. Both of the suction and discharge pressure

fluctuation are very small. After the frost on the front part of the outdoor coil is melted completely, about 330 seconds, the four-way valve is switched to melt the frost on the rear part. The suction and discharge pressures drop quickly, then decrease gradually with time. During the switching process of the four-way valve, pressure fluctuation does not occur. In the end of the defrosting period with GTD cycle, the four-way valve is switched from the cooling mode to the heating mode, the discharge pressure increased rapidly and the suction pressure fluctuated considerably. It is noted from Figure 3 that the dynamic pressures during defrosting with GTD cycle are considerably distinguished from that with RCD cycle. Firstly, for the RCD cycle, the suction pressure of the compressor fluctuates by a big margin both at the beginning and the ending of the defrosting cycle and the fluctuation range reached at 555kpa and 490kpa, respectively. Moreover, the discharge pressure of the compressor fluctuated only at the ending of the defrosting cycle and the pressure fluctuation range is about 200kpa. For the GTD cycle, the suction pressure fluctuated only at the ending of the defrosting cycle, and the fluctuation range is 252kpa, which is much less than that for the RCD cycle. It is favorable for reducing the mechanical impact to the compressor during the defrosting process. Secondly, for the GTD cycle, the suction pressure is relatively stable in the first defrosting stage. The difference between the suction and discharge pressures are less than that for the RCD cycle, especially at the end of the defrosting cycle, the pressure difference for RCD cycle is 2-2.8 times than that for GTD cycle. This would have a considerable effect on the compressor power consumption.

Comparison of power consumption of defrosting:

Figure 4 shows the comparison between the power consumptions during defrosting with the RCD and GTD cycles under the typical conditions. The power consumption curves under other conditions present a similar trend with that under the typical condition.

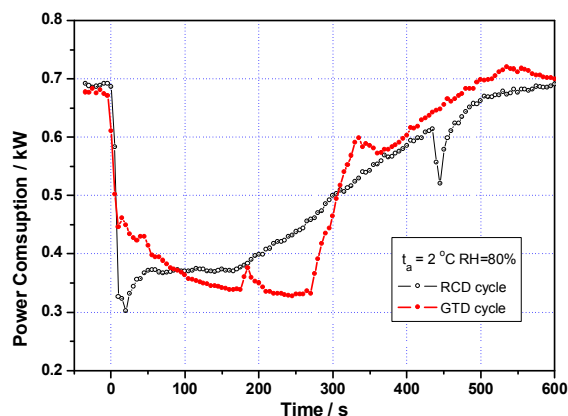


Figure 4
Power consumption during defrosting

It can be seen in Figure 4 that in the first period, from

beginning of defrosting to about 100 seconds, the power consumption for GTD cycle is greater than that for the RCD cycle. In the second period, from 100 seconds to about 300 seconds, the power consumption for GTD cycle is less than that for RCD cycle. This is because that at the initial stage, the pressure ratio of the compressor for the GTD cycle is greater than that for the RCD cycle, but is less than that for the RCD cycle in the second period, as shown in Figure 3. The total power consumptions during defrosting for the two defrosting cycles are 200.7kJ and 191.1kJ, respectively, the power consumption with GTD cycle is decreased about 5%.

Comparison of heating performance: The comparison between the heating capacities of the ASHP unit during defrosting with the two cycles under various outdoor air temperature conditions is shown in Figure 5.

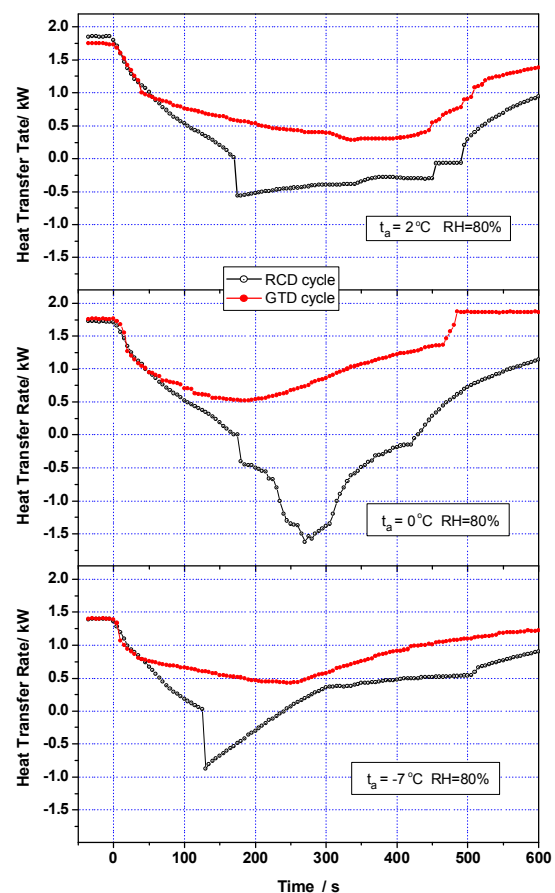


Figure 5
Heat transfer rate of indoor exchanger during defrosting

For the RCD method, the total heat extracted from the conditioning space during the defrosting process under the typical conditions is 112.9 kJ. The extracted heat should be provided to the conditioning space in the next heating operation, hence, it can be considered as an additional defrosting loss. Moreover, the extracted heat from the indoor space compromises the indoor thermal comfort. For the GTD method, the

heating capacity remains positive, i.e. there is no heat extracted from the indoor space, therefore, no additional defrosting loss, and the temperature fluctuation in the conditioning space is smaller. It is found from Figure 5 that for the RCD cycle, the extracted heat from indoor room during defrosting is the greatest, about 187.9 kJ, when the outdoor air temperature is about 0°C. The reason is that the frost mass accumulated on the surface of the outdoor heat exchanger reaches maximum under the conditions of 0°C outdoor air temperature [1].

It is noted that the heating capacity of the ASHP unit rise more rapidly with the time for the GTD cycle than for the RCD cycle in the beginning of next heating period after defrosting. This is because that for the RCD cycle, the indoor coil is acted as the evaporator during the defrosting cycle, so the temperatures of the outdoor coil and fins are relatively lower, a part of the heat is stored in the metal materials in the initial stage of the next heating cycle, and the heat providing to the indoor room is reduced. For the GTD method, the refrigerant hardly flows through the indoor coil during defrosting, and the pipe wall temperatures is nearly invariable in the defrosting period, hence the ASHP unit can provide heat more quickly to the indoor room after defrosting.

The comparison of tube wall temperatures and defrosting effects: The experimental results showed that the dynamic wall temperature curves on the different elbows of the front pipes almost completely coincide with each other. The trend of the dynamic temperature on the different elbows of the rear pipes is similar, too. Therefore, the data close to the inlet/outlet of the heat exchanger are selected to represent the wall temperature on the front and rear row pipes, respectively, t_1 and t_4 as shown in Figure 1. The dynamic wall temperatures on the front and rear rows are shown in Figure 6.

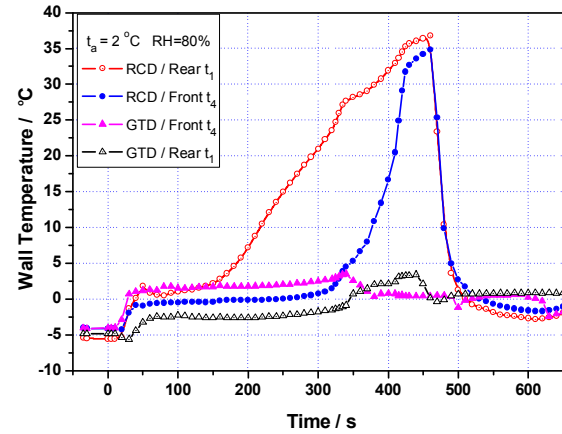


Figure 6
Dynamic wall temperature of outdoor heat exchanger

For the RCD cycle, the high temperature refrigerant vapor firstly flows through the rear pipes of the outdoor coil, on which the frost layer is very thin. So the frost is melted very quickly, and after about 150 seconds, the wall temperature on the rear pipes begin to rise quickly, while the wall temperatures of the front pipes almost are invariable until about 330 seconds because of the thicker frost layer on it. For the GTD method, the hot refrigerant vapor flows firstly through the front pipes, as shown in Figure 6, the wall temperature on the front pipes keeps steady during the defrosting process and no superheating occurs. In the second defrosting stage, the hot refrigerant vapor firstly flows through the rear pipes of the outdoor coil, so the wall temperature on the rear pipes rise quickly. Because of the short defrosting time in the second stage, the wall of the rear pipes is not superheated.

The photos of melting frost layer on the front part of the outdoor coil with the two defrosting methods under the typical conditions, i.e. the outdoor air temperature 2°C and relative humidity 80%, are shown in Table 2.

Table 2
Images of frost layer on fins of front row under outdoor conditions of temperature 2°C and RH80%

Time (s)	0	30	60	90	120	150	180	240	300	360	420
RCD cycle											
GTD cycle											

It is found from table 2 that for the new defrosting method, the frost layer on the front part of the outdoor coil is melted completely in the period of about 180 seconds. Then it comes into the stage of draining water on the surface. At the 330th second the four-way valve is switched, the hot refrigerant vapor flows firstly through the rear pipes of the outdoor coil for melting frost, and the front pipes act as an evaporator. The re-frosting on the front pipes is not invisible in the second frosting stage. For the RCD cycle, the melting of the frost layer on the front pipes and fins is inconspicuous until 120 seconds. The frost layer is melted completely at about 420th second. Then it comes into the stage of draining melted frost.

As mentioned above, the defrosting with RCD cycle is terminated at 440th second. Therefore the duration for draining water is about 20 seconds, which is less than that for GTD cycle, about 120 seconds. In other words, if the time for draining water was kept the same, about 120 seconds, for the RCD and GTD cycles, the defrosting duration for the RCD cycle would be 540 seconds. So the power consumption and the heat extracted from indoor space during defrosting with RCD cycle would be greater than the values which are presented in the Figure 4 and Figure 5, respectively. The reason resulting in the error can be explained as follows, i.e. it is hard to determine accurately whether the frost on the coil is melted completely.

For the RCD cycle, the energy melted the frost layer is provided by input power of the compressor and the heat extracted from the indoor space, while for the GTD cycle, it is mainly comes from the input power of the compressor. Generally, the defrosting energy for the RCD cycle is greater than that for the GTD cycle. But the experimental data indicate that the defrosting duration for the GTD cycle is less than that for the RCD cycle. The reason can be explained as follows. For the RCD cycle, the hot refrigerant vapor discharged from the compressor firstly flows through the rear pipes of the outdoor heat exchanger, on which the frost layer is very thin. The energy is used to heat the pipes and fins, as a result, the rear pipes are highly superheated, and the frost on the front pipes and fins are melted very slowly. In another words, the defrosting energy is wasted. For the GTD cycle, the defrosting energy is used more efficiently. The hot refrigerant vapor firstly flows through the front pipes, on which the frost formation is relatively serious, hence, the pipe wall is not superheated. The results can be seen in Figure 6 and Table 2 clearly.

CONCLUSION

On the basis of the hot gas by-pass defrosting cycle, a new defrosting method, termed grouping-throttle

defrosting (GTD), for air source heat pump is proposed in this paper. The defrosting performance of the new cycle is investigated experimentally and the results are compared with that of the reverse-cycle defrosting (RCD) system. The conclusions obtained in this study are summarized as follows:

- (1) The defrosting duration and power consumption for the new GTD cycle are less than that for the traditional RCD method. Furthermore, the recovering rate of the heating capacity after defrosting for the GTD cycle is faster than that for the RCD cycle. During defrosting with GTD cycle no heat is extracted from the indoor room, therefore, the temperature fluctuation in the indoor room is relative less.
- (2) For both of the RCD and GTD cycles the four-way valve is switched two times during the defrosting process, but the pressure fluctuation during defrosting for the GTD cycle is far less than that for the RCD cycle, therefore, the mechanical impact on the system is much smaller.
- (3) The experimental results indicate that the defrosting energy is used more efficiently for the GTD cycle, and the pipes of the outdoor heat exchanger are not superheated during the defrosting process. While for the RCD cycle, the rear pipes of the outdoor coil are relatively superheated during defrosting, therefore, the part of the defrosting energy is wasted.

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