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New Defrosting System for Residential Heat Pumps

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NEW DEFROSTING SYSTEM FOR RESIDENTIAL HEAT PUMPS

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1. INTRODUCTION

Generally, when the heat pump is operated under low outdoor temperature conditions, the outdoor coil is gradually frosted to reduce the heating capacity and this necessitates defrosting the outdoor coil from time to time. However, a heat pump of the reverse cycle defrost type which has been widely used has the disadvantage that room heating is suspended during defrosting operation with a consequent decrease in indoor comfort.

Therefore, to obviate the above disadvantage, we explored a variety of defrosting methods and eventually developed the following new defrosting system.

Thus, a heat pump embodying the new defrosting system retains a heating capacity equal to about one-sixth of the capacity available during the regular heating operation, without compromising the designed compressor dependability, even during the defrosting operation.

2. THE CONVENTIONAL HEAT PUMP

The heat pump used in the present study was a split-type inverter-driven heat pump with a rated output of 1 HP. The principal components of the heat pump are an inverter-driven compressor, a reversing valve, an indoor coil, an outdoor coil, and an expansion valve.

In the regular heating operation, the refrigerant discharged from the compressor flows through the reversing valve to the indoor coil, the expansion valve and the outdoor coil in that order and is finally returned to the compressor. When the temperature sensor comprising a thermistor installed in the outdoor coil detects a temperature below a predetermined level, a controller having a built-in microcomputer judges that the outdoor coil has been frosted and transmits a signal directing a start of defrosting operation.

To start the defrosting operation, the reversing valve is reversed. As a result, the refrigerant discharged from the compressor flows into the outdoor coil to thaw the frost and, then, flows through the expansion valve into the indoor coil, where it absorbs some heat from the room, and is finally returned to the compressor. Since the heat in the room is thus absorbed, though in a limited degree, indoor comfort is adversely affected.

3. THE REFRIGERANT CIRCUIT OF THE HEAT PUMP EMBODYING THE NEW DEFROSTING SYSTEM

Fig. 1 shows the refrigerant circuit of a heat pump embodying our new defrosting system. The refrigerant circuit according to this new system is characterized by the use of an electromotive expansion valve driven by a stepping motor and the provision of a bypass circuit.

During the regular heating operation, the solenoid valve in the bypass circuit remains closed and accordingly the refrigerant from the compressor flows to the indoor coil, the electromotive expansion valve and the outdoor coil in that order and is finally returned to the compressor. And defrosting is started when the temperature sensor detects that the outdoor coil has been frosted.

Fig. 2 shows the control mode for defrosting operation.

Fig. 3 is a pressure-enthalpy chart of the refrigeration cycle after some interval from start of the defrosting operation.

A portion of the refrigerant compressed ($\underline{a}\rightarrow\underline{b}$) by the compressor flows into the bypass circuit, where it is decompressed ($\underline{b}\rightarrow\underline{f}$). The remainder of the refrigerant flows to the indoor coil, where it is utilized for heating and is condensed. This refrigerant is decom-

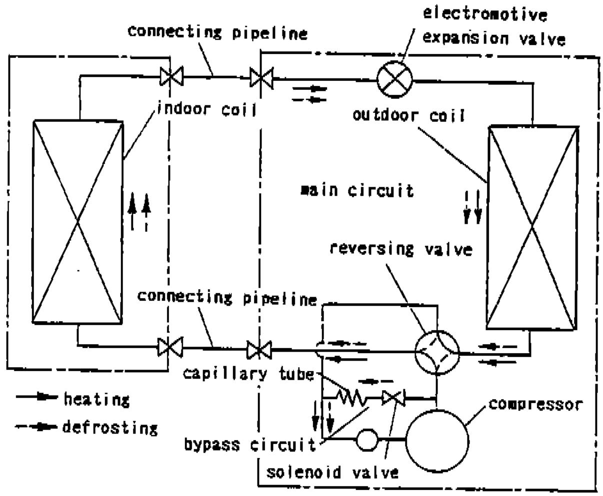

Fig. 1 - Refrigerant circuit of the heat pump embodying new defrosting system

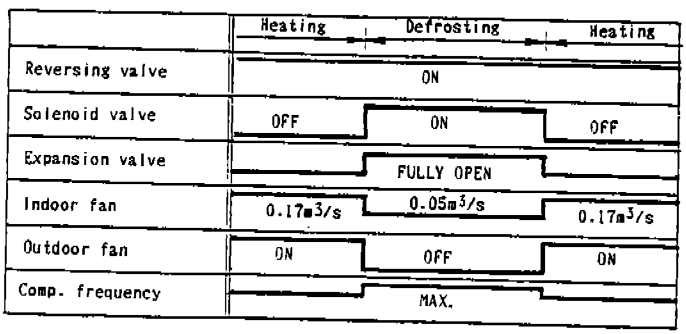

Fig. 2 - Control mode for defrosting operation

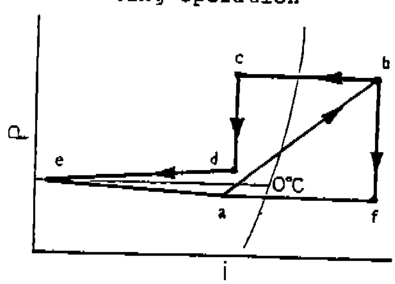

Fig. 3 - Pressure-enthalpy chart of the refrigeration cycle

pressed as it flows through the electromotive expansion valve and flows into the outdoor coil, where it is utilized for defrosting and is condensed (d→e). Immediately before returning to the compressor, this liquid-rich refrigerant (e) converges with the high-enthalpy refrigerant (f) coming through the bypass circuit. As a consequence, the quality of the refrigerant is enhanced (e→a) and enters the compressor in the state of a. Since the quality of the refrigerant is thus enhanced just before its return to the compressor, the dependability of the compressor is not compromised as will be explained below.

4. METHODS FOR PARAMETER MEASUREMENT AND CONTROL OF THE HEAT PUMP USED IN THE EXPERIMENT

The methods for parameter measurement and control of the heat pump embodying the new defrosting system, which was used in the experiment, are summarized in Fig. 4.

The refrigerant and air temperatures were measured with a T

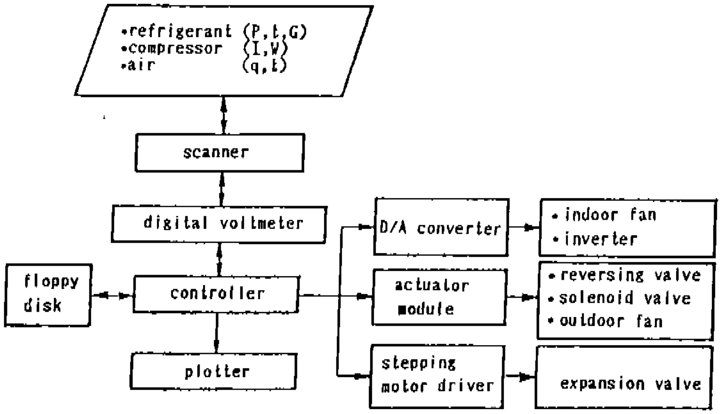

Fig. 4 - Methods for parameter measurement and control of the heat pump

thermocouple, the refrigerant pressure with a piezoelectric transducer, and the refrigerant flow with a turbine type flowmeter.

The scanner, digital voltmeter and controller (desk-top computer) are respectively connected by GP-IB interfacing. Measured data are taken into the controller at timed intervals and recorded on a floppy disk. Table I shows the defrosting conditions specified in the Japanese Industrial Standards (JIS).

All the test runs were carried out under the JIS defrosting condition in a room type psychrometric calorimeter.

5. THE NEW DEFROSTING SYSTEM VERSUS THE CONVENTIONAL (REVERSE CYCLE) DEFROSTING SYSTEM

Using the results of measurements obtained with the instrumentation described above, comparison was made between the new defrosting system and the conventional defrosting system.

Fig. 5 shows the changes in the compressor input $\bar{W}$, the pressure (Psuc) of the refrigerant drawn into the compressor, and the pressure (Pdis) of the refrigerant discharged from the compressor in the new defrosting system and the conventional defrosting system, respectively. Since, in the conventional defrosting system,

TABLE I
Defrosting condition of JIS

	indoor	outdoor
Dry bulb	21.0°C	1.5°C
Wet bulb	15.5°C	0.5°C

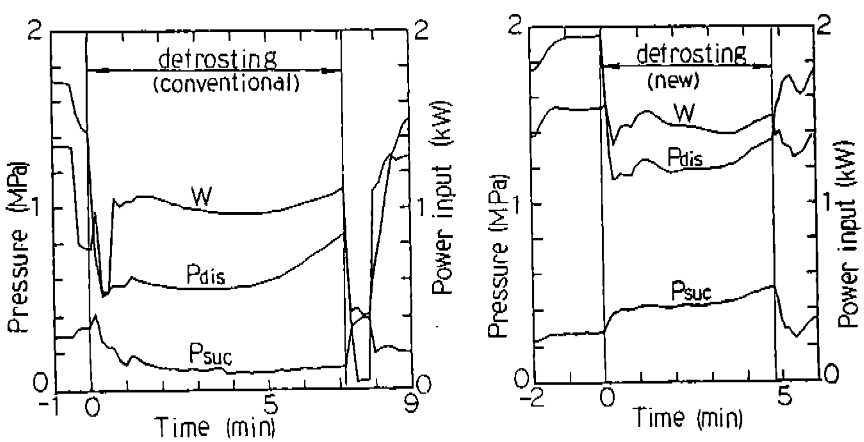

Fig. 5 - Power input and pressure changes during defrosting

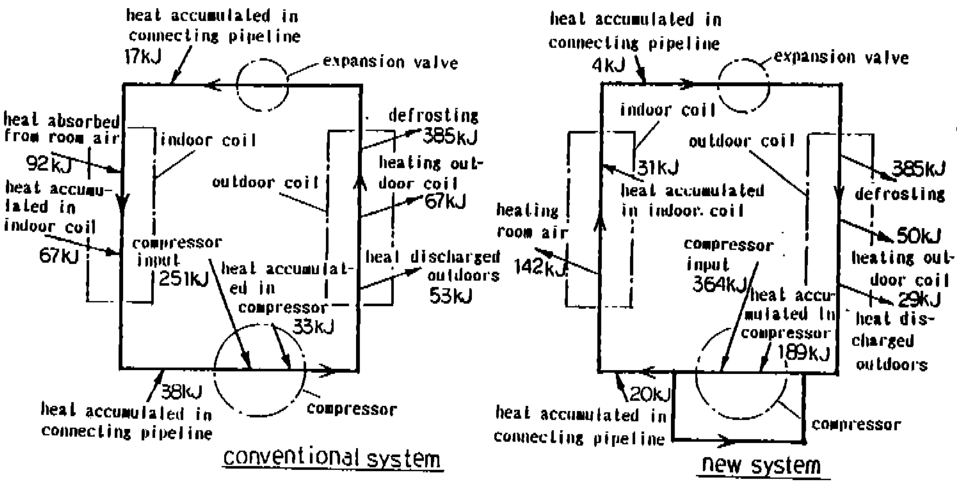

Fig. 6 - Heat balance during defrosting

the condensation temperature of the refrigerant is approximately equal to the thawing temperature of frost which is 0°C, Pdis is as low as 0.53 MPa/1/. Accordingly, Psuc is very low, 0.1 MPa, and the circulating amount of refrigerant is small, so that $\bar{W}$ is also small.

Since this means a small caloric value of the heat applied to the frost, the defrosting time is long.

In the new defrosting system, a different situation prevails. Thus, as the condensation temperature of the refrigerant in the outdoor coil is approximately equal to the thawing temperature of frost, Psuc is as high as 0.41 MPa and Pdis is also maintained high, 1.2 MPa. Therefore, the circulating amount of refrigerant is large and the value of $\bar{W}$ is also large. Consequently, the following and other advantages are materialized. 1) As the amount of heat applied to frost is large, the defrosting time becomes shorter; 2) as the indoor coil remains to be the high pressure side with the reversing valve kept in the heating position even during the defrosting operation, room heating continues while defrosting is performed; and 3) the time interval from resetting to heating operation to stabilization of the refrigeration cycle is short.

Now, the two systems are compared in regard to the heat balance during defrosting operation. Fig. 6 shows the heat balance during defrosting operation in the conventional defrosting system and that in the new defrosting system. In the new defrosting system, the compressor input accounts for about 60% of the total charged energy to the circulating refrigerant. Moreover, during defrosting operation, the compressor body is cooled by the biphasic refrigerant (gas-liquid mixture) drawn into the compressor, with the result that the compressor functions as a heat accumulator. This accumulated compressor heat accounts for about 30% of the total charged energy. On the other hand, the caloric value of defrosting heat and that of heat used for room heating account for about 65% and about 25%, respectively, of the total discharged energy from the circulating refrigerant.

6. DEFROSTING CHARACTERISTICS OF THE NEW SYSTEM

We performed various experiments using a heat pump embodying the new defrosting system and found that one of the most important factors influencing the heating capacity and defrosting time during defrosting operation is the relative refrigerant flows in the main circuit and the bypass circuit. Therefore, we varied the ratio of refrigerant flow in the main circuit to that in the bypass circuit by adjusting the throttling amount of the electromotive expansion valve and that of the capillary disposed in the bypass circuit and measured the heating capacity and defrosting time under the varying conditions.

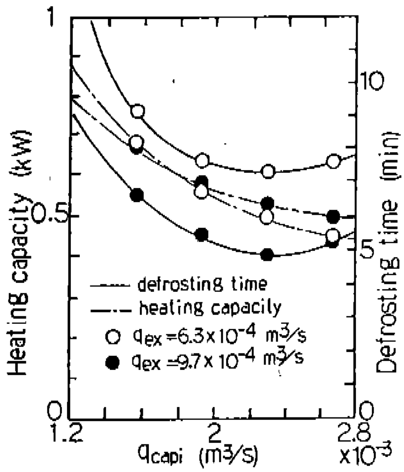

Fig. 7 - Heating capacity and defrosting time vs. q_{capi}

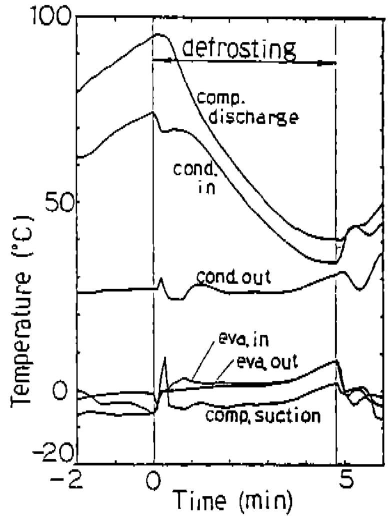

Fig. 8 - Temperature changes during defrosting

Since the refrigeration cycle is in transient state during defrosting operation, the change in refrigeration cycle during defrosting was also analyzed on the basis of experimental data.

Relative flows of refrigerant in the main circuit and bypass circuit

Fig. 7 shows the relationship between the throttling amount of the capillary in the bypass circuit and the defrosting time and heating capacity at two different settings of the electromotive expansion valve. The horizontal axis of Fig. 8 represents the air flow rates through the capillary when air at a temperature of 21°C is passed with a pressure differential of 0.098 MPa between the entrance and exit of the capillary. Air flow rates are also shown for the electromotive expansion valves.

The defrosting time is minimal when the capillary air flow q_{capi} is approximately 2.4×10^{-3} m³/s.

Moreover, the defrosting time decreases with an increase of the air flow rate q_{ex} of the electromotive expansion valve.

On the other hand, the heating capacity is also decreased with an increase of q_{capi}.

Change in refrigeration cycle during defrosting operation

Fig. 8 shows the changes in temperature at various parts of the refrigeration cycle. Thus, a transient state prevails during defrosting operation, with the refrigeration cycle changing continually. Since the refrigerant drawn into the compressor is a biphasic entity, it is difficult to experimentally determine its quality. Therefore, refrigerant quality was estimated by the following procedure. A given time during defrosting operation is taken as τ. Then, we have;

$$Qre(\tau) = Qcomp(\tau) + Qhc(\tau) \quad \ldots (1)$$

where

$$Qcomp(\tau) = C \cdot W(\tau) \quad \ldots (2)$$

$$Qhc(\tau) = \overline{t_v}(\tau) \cdot Hcomp \quad \ldots (3)$$

Further,

$$Qre(\tau) = Gcomp(\tau) \cdot (i_{dis}(\tau) = i_{suc}(\tau)) \quad \ldots (4)$$

Using experimental data, $Qre(\tau)$ can be calculated from the equations (1), (2) and (3). As the refrigerant is a superheated

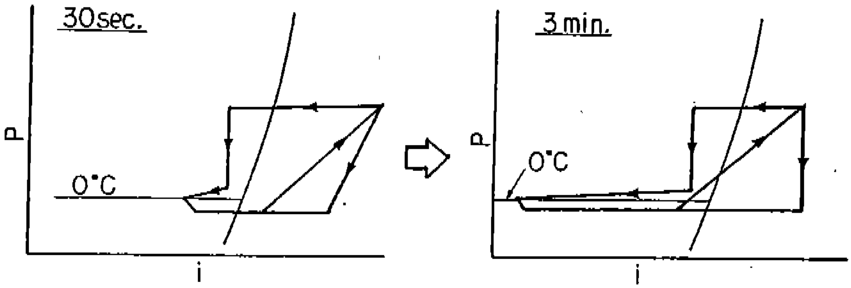

Fig. 9 - Pressure-enthalpy charts of the refrigeration cycles at 30 sec. and 3 min. after start of defrosting

vapor, $i_{dis}(\tau)$ can also be calculated. Gcomp(τ) is measured with a turbine type flowmeter. From these values, $i_{suc}(\tau)$ can be calculated.

The heating capacity during defrosting operation can be expressed by the following expression.

$$Qheat(\tau) = q \cdot \gamma \cdot Cp \cdot (t_{out}(\tau) - t_{in}(\tau)) \quad ... \; (5)$$

An alternative expression is:

$$Qheat(\tau) = Gmain(\tau) \cdot (i_{cin}(\tau) - i_{cout}(\tau)) \quad ... \; (6)$$

From the equation (5), Qheat(τ) is calculated. Referring to the equation (6), Gmain(τ) is measured with a turbine type flowmeter. As the refrigerant is a superheated vapor, $i_{cin}(\tau)$ can also be calculated. From the above values, $i_{cout}(\tau)$ can be calculated.

Further, the following equation is given according to heat balance.

$$Gmain(\tau)\,(i_{suc}(\tau) - i_{eout}(\tau)) = (Gcomp(\tau) - Gmain(\tau)) \cdot (i_{bout}(\tau) - i_{suc}(\tau)) \quad ... \; (7)$$

From the equation (7), i_{eout} can be calculated.

Using the above procedure, the refrigeration cycle during defrosting operation was determined. The pressure-enthalpy charts of the refrigeration cycles at 30 seconds and 3 minutes after start of defrosting operation are presented in Fig. 9.

Influence of the position of the bypass

In the new defrosting system, the bypass circuit interconnects the discharge and suction lines of the compressor. This bypass arrangement is compared with a bypass arrangement interconnecting the compressor discharge line and the outdoor coil entrance.

Fig 10 is a pressure-enthalpy chart of the refrigeration cycle in the arrangement of a bypass circuit interconnecting the compressor discharge line and the entrance of the outdoor coil (hereinafter referred to as the entrance bypass system).

Fig. 11 shows the changes in temperature at various parts of the refrigerant circuit during defrosting operation in the entrance bypass system with the same throttling amount of the bypass circuit capillary as used in the new defrosting system.

Fig. 12 shows the change in quality of the refrigerant drawn into the compressor during defrosting operation as estimated by the procedure described earlier.

In the entrance bypass system, the quality of the refrigerant drawn into the compressor is very low compared with the new defrosting system. Moreover, in the new defrosting system, the quality of the refigerant drawn into the compressor can be controlled by changing the ratio of Gmain to (Gcomp-Gmain) through adjusting the throttling amounts of the electromotive expansion valve and the bypass circuit capillary. In the entrance bypass system, the quality of the refrigerant can hardly be adjusted, for all the refrigerant flows through the outdoor coil and is utilized for defrosting. Therefore, the return of a liquid-rich refrigerant to the compressor is not desirable from the standpoint of compressor dependability.

7. DECREASE OF INDOOR TEMPERATURE DURING DEFROSTING OPERATION

Fig. 13 shows the changes in mean indoor temperature obtained by simulation. The house used as a simulation model was a standard Japanese-style prefabricated house and the outdoor atmospheric temperature was 0°C. The indoor space was 14 m^2. First, the heating operation was carried out to establish an indoor temperature of 21°C and, then, the defrosting operation was carried out. The heating capacity during defrosting operation by the new defrosting system was assumed to be about 550 W. In the conventional system, a heat absorption of about 150 W from the indoor atmosphere was assumed.

In the new defrosting system, heating takes place even during defrosting operation and the defrosting time is short. Consequently, the decrease of indoor temperature is held to about 2.5°C as compared with 6°C in the conventional system.

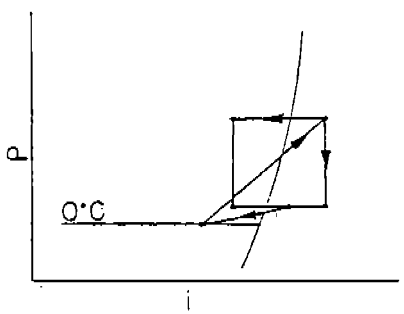

Fig. 10 - Pressure-enthalpy chart of the refrigeration cycle (entrance bypass)

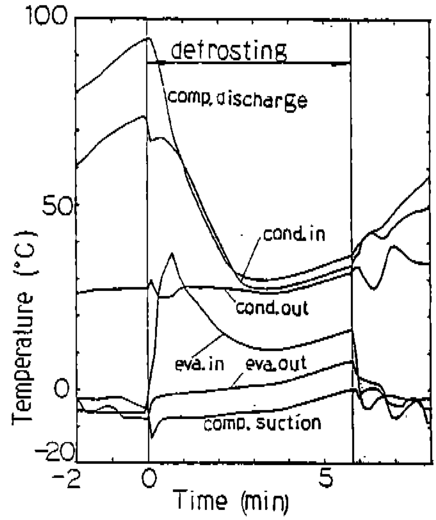

Fig. 11 - Temperature changes during defrosting (entrance bypass)

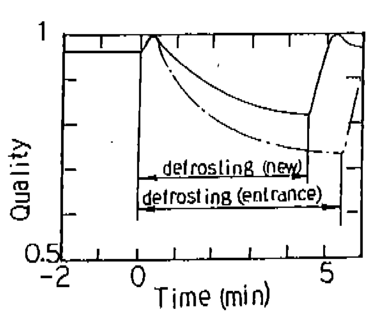

Fig. 12 - Quality change during defrosting

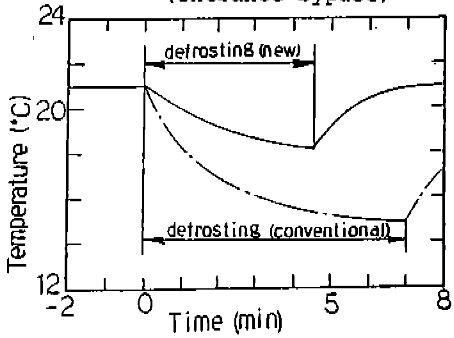

Fig. 13 - Changes in mean indoor temperature during defrosting

8. CONCLUSIONS

1. The heat balance of the conventional system and that of the new system were analyzed. In the new defrosting system, the compressor input accounts for about 60% of total charged energy, while the defrosting heat accounts for about 65% of total discharged energy, and the heat used in heating accounts for about 25%.
2. The heating capacity during defrosting operation decreases with an increasing refrigerant flow in the bypass circuit capillary while the defrosting time assumes a minimal value.
3. In the new defrosting system, the quality of the refrigerant drawn into the compressor is controllable.
4. In the new defrosting system, a heating capacity of about 550 W is maintained during defrosting operation and the defrosting time is shorter. As a result, the decrease of indoor temperature can be held to 2.5°C and, therefore, a remarkable improvement in indoor comfort is realized.

NOMENCLATURE

C	:	Constant
Cp	:	Isobaric specific heat capacity of air [J/kg•K]
Gmain, Gcomp	:	Circulating amount of refrigerant in main circuit, in compressor [kg/s]
Hcomp	:	Heat capacity of compressor [J/K]
Psuc, Pdis	:	Pressure of refrigerant drawn into and discharged from compressor [Pa]
q, q_{capi}, q_{ex}	:	Air flow of indoor fan, capillary, expansion valve [m^3/s]
γ	:	Specific gravity of air [kg/m^3]

* * *

(τ)	:	An instant value at a given time τ during defrosting
$i_{suc}(\tau)$, $i_{dis}(\tau)$, $i_{cin}(\tau)$, $i_{cout}(\tau)$, $i_{eout}(\tau)$, $i_{bout}(\tau)$	:	Specific enthalpy of refrigerant drawn into and discharged from compressor, at entrance and exit of indoor coil, at exit of outdoor coil, at exit of bypass circuit [J/kg]
$Qcomp(\tau)$, $Qhc(\tau)$	:	Heat input from compressor input, accumulated compressor heat [W]
$Qre(\tau)$	:	Total heat input within compressor [W]
$t_{out}(\tau)$, $t_{in}(\tau)$	:	Temperature of air after and before passage through indoor coil [K]
$\overline{t_v}\ (\tau)$	:	Rate of decrease in mean temperature of compressor body [K/s]

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NOUVEAU PROCÉDÉ DE DÉGIVRAGE POUR LES POMPES À CHALEUR RÉSIDENTIELLES

RESUME : Le présent article traite un nouveau système de dégivrage pour les pompes à chaleur résidentielles actionnées par inverteur avec la source de chaleur d'air.

Le réseau refrigérant de la pompe à chaleur garnie du nouveau systeme de dégivrage se compose d'un circuit réfrigérant principal comprenant un compresseur actionne par inverteur, une vanne à quatre voies, un échangeur de chaleur d'intérieur, un détendeur électromobile entraîné par moteur pas a pas, un échangeur à l'extérieur, ainsi que d'un circuit by-pass reliant l'amont et l'aval du compresseur par l'intermédiaire d'une vanne à duex voies.

L'analyse de differents résultats des essais en a constaté que la pompe à nouveau systéme de dégivrage a pu conserver la capacité de chauffage a l'ordre d'un sixième du régime normal, tout en assurant la fiabilité du compresseur, même au régime de dégivrage.

NEW DEFROSTING SYSTEM FOR HOUSE HEAT PUMPS

SUMMARY

The present paper describes a new defrosting system for house heat pumps put on by a reversor with hot air.

Heat pump refrigerant system working with this new defrosting system is composed of a main refrigerant circuit with compressor switch by invertor, a four-track gate, an inner heat exchange, an electromotive pressure-reducer moved by a step by step engine, an outdoor exchange system, and at final a by-pass pathway connecting upstream downstream compressor excremities by using a two-track gate.

Analyis of various experiments show that after incorporation of this new defrosting system the heating capacity is always one-sixth of standard level, with good compressor reliability even in defrosting conditions.