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Prediction of Condensation and Frosting Limits in Rotary Wheels for Heat Recovery in Buildings

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ABSTRACT

Rotary heat exchangers in air handling systems, particularly those with hygroscopic rotors, can be operated at essential lower outdoor temperature than static heat exchangers without any need for defrosting. The reason for rotary heat exchangers starting to frost at lower outdoor temperatures than static heat exchangers is proved to be the moisture transfer in the rotor. The supply air temperature limit presented here for frosting as well as excess water is calculated by a numerical method. The frosting-defrosting process is also discussed.

INTRODUCTION

When using heat exchangers for heat recovery in air handling systems, more or less frosting will be obtained on exchanger surfaces in the winter. In static heat recovery units, such as plate heat exchangers, frosting will occur as soon as water starts condensing out of the air on surfaces at temperatures below the freezing point. Condensation depends on the exhaust air being cooled down to its dew point. Condensate will form as soon as the surface temperature of the heat exchanger has dropped below the dew point. The frost layer grows in thickness as more condensate precipitates out onto the cold surfaces, and this phenomenon is known as frosting. The frost must be removed by changing the condition so that the ice will melt.

Rotary heat exchangers, on the other hand, can operate with frost on the rotor surfaces without permanent frosting occurring, since the frost is sublimated on the supply air side. The moisture transfer in these heat exchangers thus results in the limiting temperature for incipient frosting being lower than in static heat exchangers.

The moisture transfer efficiencies and the limiting temperature for excess water and frosting presented here have been calculated in accordance with the method for calculating heat and moisture transfer presented at an earlier date by the author (Holmberg 1977, 1979). Calculations have been carried out for rotary heat exchangers with non-hygroscopic as well as hygroscopic rotors (Figure 1). The non-hygroscopic rotor is made of untreated aluminum, whereas the hygroscopic rotor is made of chemically treated aluminum forming an oxide layer about 10% of the rotor weight. The rotor consists of a structure of triangular passages with a channel length of 200 mm, a foil spacing of 1.6 mm, and a material thickness of 0.06 mm.

The temperature and moisture efficiencies are defined on the supply air side in accordance with Eurovent document 10/1:

$$\eta_z = \frac{z_2 - z_1}{z_3 - z_1}$$

where subscripts 1 and 2 designate the inlet and outlet, respectively, on the supply air side, whereas subscripts 3 and 4 designate the inlet and outlet, respectively, on the exhaust air side. The supply air velocity, v_2, in the heat exchanger is defined at the outlet (subscript 2), and the exhaust air velocity, v_3, at the inlet (subscript 3). The airflows are thus related to the application side of the heat exchanger, i.e., the design flows in the event of leakage.

The input data of the calculation method defined in Holmberg (1979) consist here of the mass flow rate ratio of the two airstreams, $W_c/W_h = v_2/v_3$; the overall number of transfer units, $N_{tu,o} = 11.0/v_{min}$; the diffusion resistance ratio, $(\beta_x A)_c/(\beta_x A)_h = 1$; the conduction area ratio, $A_{kc}/A_{kh} = 1$; and the atmospheric pressure, $p_t = 101.3$ kPa, together with the heat capacity rate ratio of the rotor, $C_r/C_n = 1.15$ n/v_{min}, the conduction parameter, $\Lambda = 0.15/v_{min}$, and the mixing parameter, $\theta = 0.866$, for the nonhygroscopic rotor and the mass capacity rate ratio of aluminum oxide, $W_a/W_n = 0.1$ n/v_{min}, the heat capacity rate ratio of the rotor, $C_r/C_n = 1.0$ n/v_{min}, the conduction parameter, $\Lambda = 0.12/v_{min}$, and the mixing parameter, $\theta = 1.0$, for the hygroscopic rotor. n denotes the rotational speed, and v_{min} the minimum velocity of v_2 and v_3.

MOISTURE TRANSFER

In a rotary heat exchanger with a rotor made of *non-hygroscopic* material, moisture transfer will occur only when the exhaust air has been cooled down to its dew point. The condensate will then form as a thin film of water

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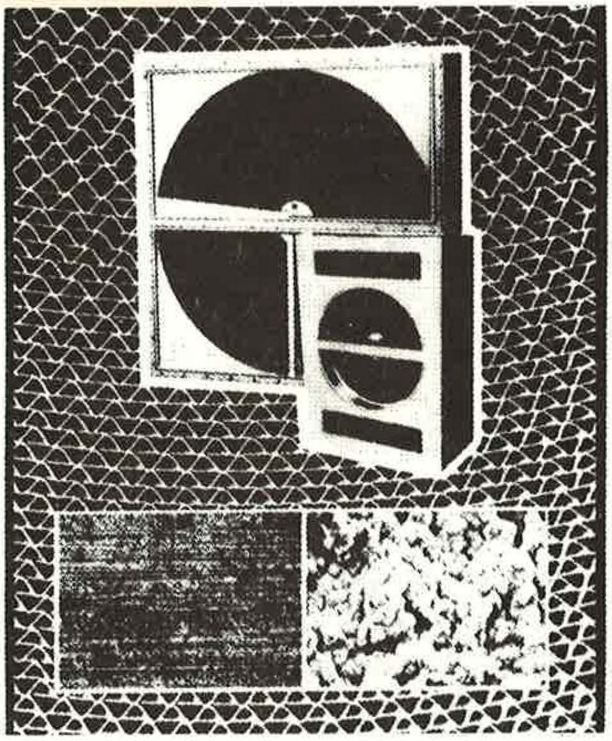

Figure 1 *Rotary heat exchanger with integrated bypass dampers. The background shows the triangular passages of the rotor matrix. The inset micrographs are taken through an electron microscope and show the surface structure of the aluminum before (left) and after (right) the surface treatment that produces the hygroscopic oxide layer.*

on the rotor surface. As the rotor rotates, the condensate will be transferred to the supply air side, where it will be evaporated. In comfort ventilation systems, moisture transfer will only take place at low outdoor temperatures, as illustrated by the calculated humidity efficiencies in Figure 2. The temperature efficiency in these examples is approximately 75%. It will be noted that the humidity efficiency increases rapidly with decreasing outdoor temperature. Moisture transfer takes place equally efficiently at rotor temperatures below or above 0°C, since the condensate formed as frost on the rotor surface will be sublimated on the supply air side. However, the moisture efficiency reaches an upper limit (which is lower than the temperature efficiency) and remains basically constant in the event of a further drop in the outdoor temperature, since the supply air becomes saturated and is incapable of picking up additional moisture. Under these conditions, what is known in the next section as "excess water" will be formed. Calculated temperature and moisture efficiencies have proved to agree well with measured values for the rotor described here.

Figure 3a shows the changes in the conditions of the supply and exhaust airflows in a Mollier chart for winter conditions at an outdoor temperature that is so low that moisture transfer occurs in the cold part of the rotor, i.e., the part facing the supply air inlet side. In the case of "cooling recovery" during the summer, no moisture transfer will take place.

If the rotor is made of *hygroscopic* material, moisture is transferred by the hygroscopic layer adsorbing water vapor from the exhaust air, storing it, and then emitting it to the supply air by desorption. Moisture will thus be transferred as soon as the supply air has a lower water vapor ratio than the exhaust air, which is usually the case. Rotors impregnated with lithium chloride absorb moisture predominantly by absorption. The humidity efficiency of a hygroscopic rotor may be as high as the temperature efficiency, provided that the moisture capacity of the rotor (i.e., the product of the quantity of hygroscopic material and the speed) is sufficiently high and the diffusion resistance is low. This results in the outlet conditions of the airflows ending up on an intermediate line between the two inlet conditions in a Mollier chart, which is illustrated in Figure 3b. The figure also illustrates that moisture transfer can take place equally efficiently under extreme summer conditions. However, the moisture efficiency is not equal to the temperature efficiency in all hygroscopic rotors, particularly if the entire heat transfer area is not hygroscopic, if the hygroscopic layer is too thin or too inert, or if the hygroscopic solution with which the rotor might be impregnated is drained away (see the next section). Rotors that have practically equal moisture and temperature efficiencies will be referred to as fully hygroscopic rotors below.

EXCESS WATER

Excess water will occur in hygroscopic as well as non-hygroscopic rotors as soon as the supply air is unable to pick up all of the condensate precipitated out of the exhaust air. The excess water can thus be said to be surplus condensate in the rotor. The limitation in the capacity of picking up water is due to the supply air being saturated at some point in its path through the rotor. Excess water will thus be formed in a cross-sectional zone of the rotor. Since the variations in the radial direction of the rotor are negligible, this zone can basically be illustrated along the periphery of a random cylindrical element of the rotor, as shown in Figure 4.

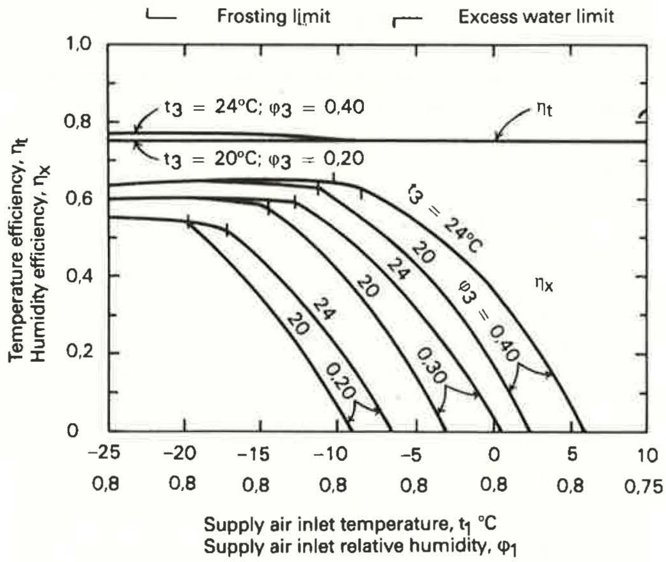

Figure 2 *Temperature and humidity efficiencies for non-hygroscopic rotor at rotor speed 15 rpm and air velocities* $v_2 = v_3 = 3.0$ *m/s (at* $\rho = 1.2$ *kg/m³)*

The excess water is obtained as liquid or solid depending on the rotor surface temperature. At temperatures above 0°C, the amount of surplus condensate increases with each revolution and the condensate water droplets float together, forming bigger droplets. As soon as these droplets become big enough, they will be drained out of the rotor in some way, depending on the design of the rotor and casing. To enable a hygroscopic rotor to operate under conditions giving rise to excess water, the hygroscopic properties of the rotor must remain unaffected by saturating the rotor with water. Such a rotor material is the oxidized aluminum sheet. In rotors impregnated with some hygroscopic solution, such as lithium chloride, no excess water is permissible, since the impregnating agent would be drained out of the rotor together with the excess water.

If and when excess water occurs in a fully hygroscopic rotor can be determined from a Mollier chart. As illustrated in Figure 5b, the intermediate line between the two inlet conditions must intersect the saturation curve. On the other hand, it is appreciably more difficult to determine the limit at which excess water will occur in hygroscopic rotors with limited moisture efficiency or in non-hygroscopic rotors. A careful determination must be carried out experimentally or by means of some calculation model (Holmberg 1977, 1979).

However, the limit for excess water in a non-hygroscopic rotor can be determined approximately in a Mollier chart as follows: Since the change in condition of the exhaust air first takes place along a constant vapor content x_3, excess water will already occur when the intermediate line between the supply air inlet condition 1 and the exhaust air condition 3′ (Figure 5a) intersects the saturation curve. At typical exhaust air conditions occurring in comfort ventilation and a temperature efficiency of 75%, condition 3′ is set to (t_{d3} + 4 °C; x_3) where t_{d3} is dew point of the exhaust air.

The amount of excess water, Δx_v (kg of water/kg of dry supply air), obtained under different operating conditions has been calculated for non-hygroscopic as well as fully hygroscopic rotors (Figure 6). The temperature efficiency in these examples is approximately 75%. Since the non-hygroscopic rotor does not transfer moisture with all of its heat transfer area, the amount of excess water will be greater in this case. This is illustrated in Figure 5a by the saturation curve being intersected by the relevant intermediate line by a greater amount than in the case of a fully hygroscopic rotor (Figure 5b). In Figure 6 also the supply air limiting temperature for excess water can be read at $\Delta x_v = 0$.

FROSTING

At rotor temperatures below 0°C, the moisture condensing out of the air on the rotor surfaces will be in the form of frost. If the supply air is saturated and is unable to pick up all frost on sublimation, excess water will thus occur in the form of a frost layer. If this layer of frost is to grow in thickness at any cross section of the rotor, i.e., if frosting is to occur, the mean temperature of the rotor during one revolution must be sufficiently low at this cross section to make it impossible for the frost layer to melt during the warmer part of the revolution. The zone in the rotor in which frosting occurs can schematically be illustrated as shown

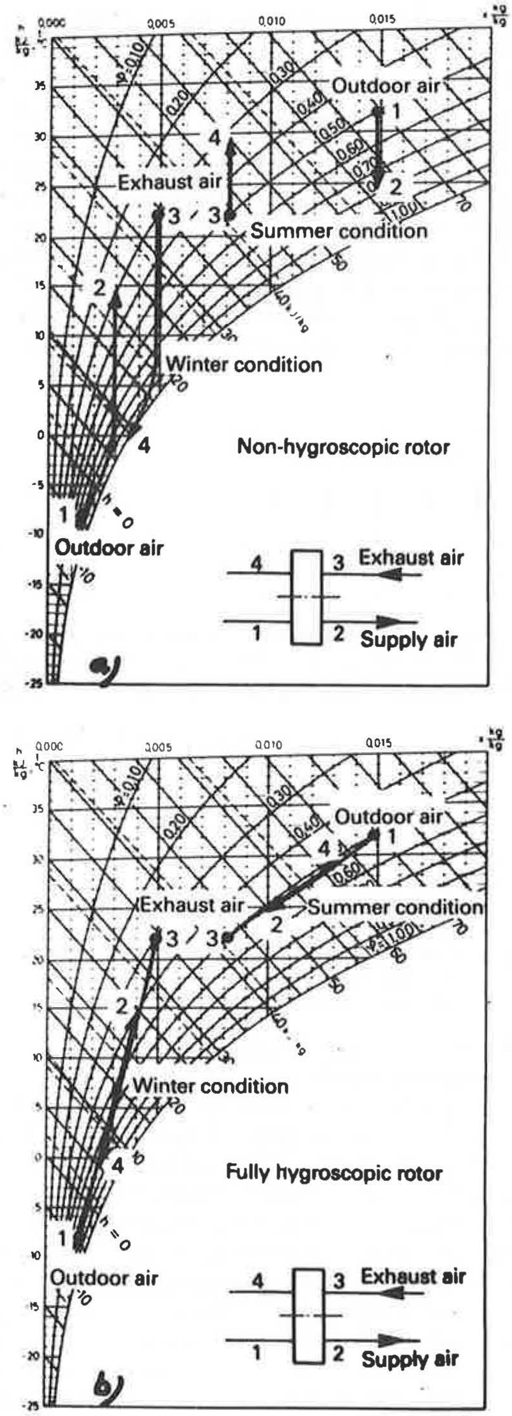

Figure 3 *Example of changes in air condition during winter and summer operation with a) non-hygroscopic and b) fully hygroscopic rotor, respectively.*

in Figure 4. It is reasonable to assume that the mean temperature of the rotor during one revolution must be lower than 0°C in all cross sections of the rotor in this zone. Provided that the excess water in liquid form outside this zone is not displaced into the zone and freezes in it, the *criterion for frosting* can be formulated as follows:

Frosting will occur if excess water is formed in the rotor if at least part of this water is formed in a cross-sectional zone in which the mean temperature of the rotor during one revolution is lower than 0°C.

This criterion for frosting has provided very good agreement between calculated values and those measured in the laboratory for the limiting temperatures of non-hygroscopic as well as hygroscopic types of rotors. The validity of this frosting criterion is further confirmed by measurements carried out on a hygroscopic rotor with duct passages (Ruth et al. 1975) as well as a non-hygroscopic rotor consisting of an aluminum network (von Kruse and Vauth 1976). Calculations presented in Figure 7 show that the frosting limit of the hygroscopic rotor is up to 10°C lower than that of the non-hygroscopic rotor, due to the fact that there is less risk of excess water in the hygroscopic rotor.

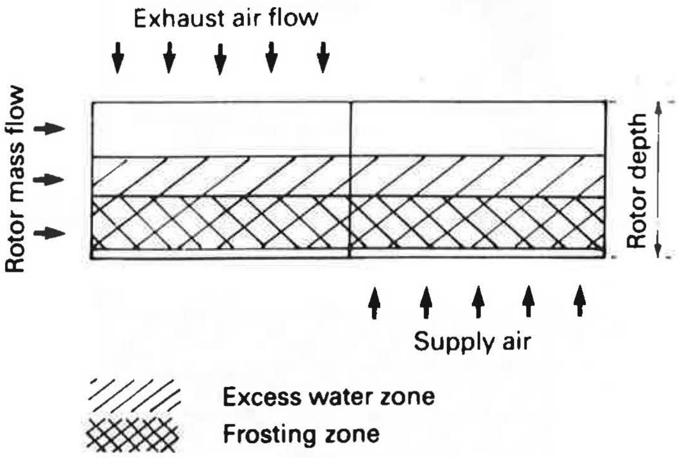

Figure 4 *Schematic view of the zones with excess water and frosting in a rotor. The figure shows an open envelope surface of an arbitrary cylinder in a rotor.*

If all excess water is formed in the zone in which the mean temperature of the rotor is equal to or lower than 0°C, the frosting limit can be determined in a Mollier chart in accordance with the same procedure as that described for the limit for excess water, since these limits coincide. On the other hand, the frosting limit cannot be determined as easily when only part of the excess water will occur in the form of frost within the zone at temperatures below 0°C and when the remainder of the excess water will occur in liquid form and will be drained out of the rotor. However, calculations and laboratory tests show that the supply air limiting temperature in this case can be assumed to be approximately -10°C in rotors designed for a temperature efficiency of 70% to 80% at equal supply and exhaust airflows and at exhaust air temperatures normally prevailing in comfort ventilation systems. Under these conditions, the supply air temperature limit for frosting will thus be approximately -10°C or equal to the temperature limit for excess water (below -10°C).

The requirement that excess water must form in the rotor for frosting to occur is fully analogous with the corresponding requirement regarding condensate in a plate heat exchanger. The moisture transfer of the rotor thus significantly lowers the frosting limit.

Figure 8 shows an example of air conditions that give frosting in non-hygroscopic as well as fully hygroscopic rotors. The time necessary to achieve a given increase in

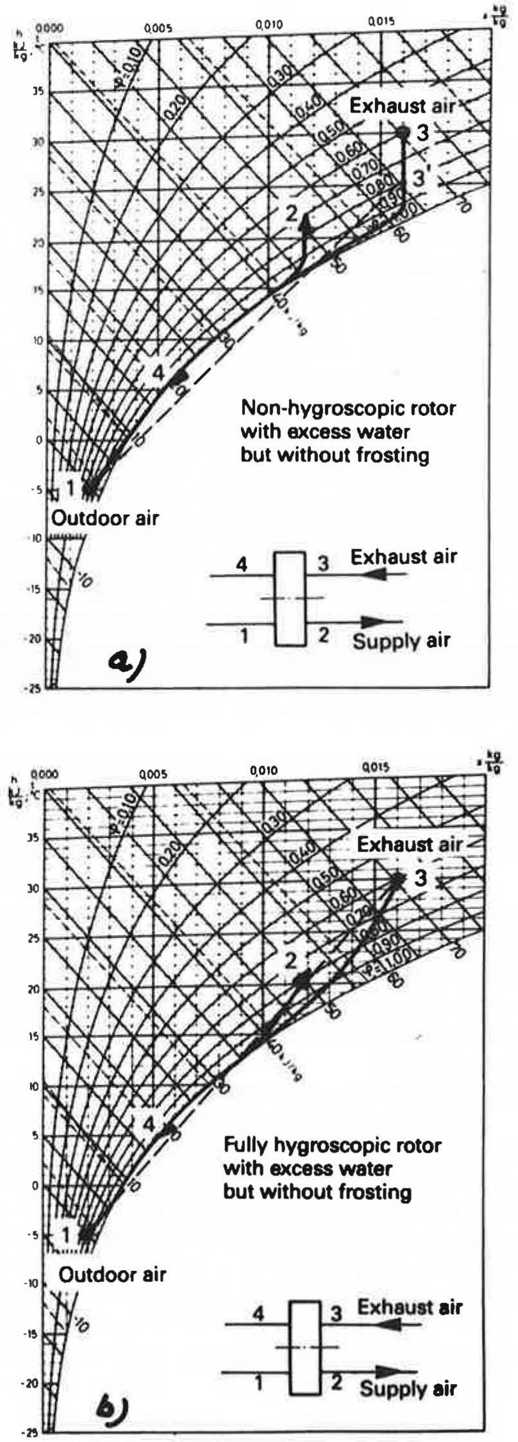

Figure 5 *Examples of charges in air condition at operations with a) non-hygroscopic and b) fully hygroscopic rotor in which excess water will be obtained but no frosting.*

pressure drop across a rotor due to frosting is inversely proportional to the amount of excess water occurring in the form of frost. As an example, laboratory tests show that it takes about four hours under the operating conditions shown in Figure 8a and about eight hours under the conditions shown in Figure 8b for the pressure drop to increase by 50% in the rotor. It should be noted that the specified frosting time is conditional on the temperature and moisture conditions being constant throughout the frosting

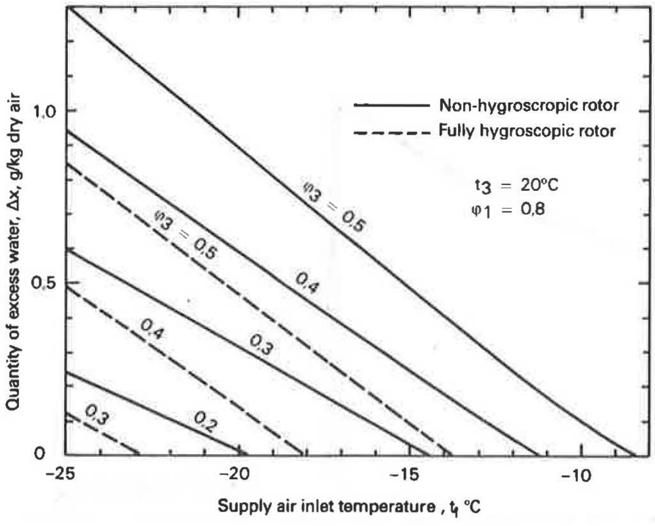

Figure 6 *Amount of excess water in non-hygroscopic rotor as well as hygroscopic rotor at rotor speed 15 rpm and air velocities* v2 = v3 = *3.0 m/s (at* p = *1.2 kg/m³).*

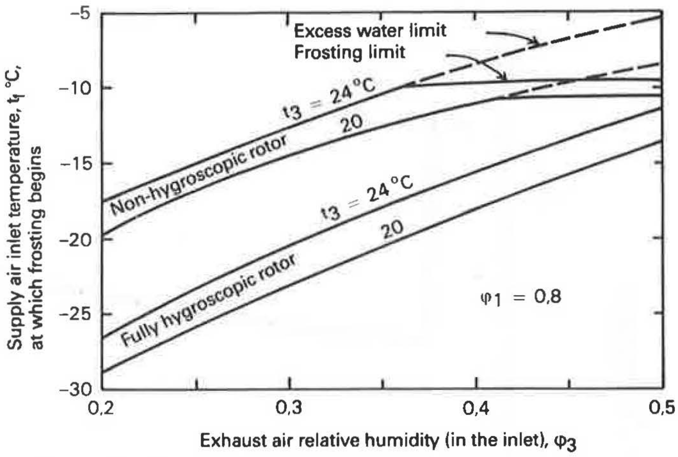

Figure 7 *Frosting limit for non-hygroscopic rotor as well as hygroscopic rotor at rotor speed 15 rpm and air velocities* v2 = v3 = *3.0 m/s (at* p = *1.2 kg/m³).*

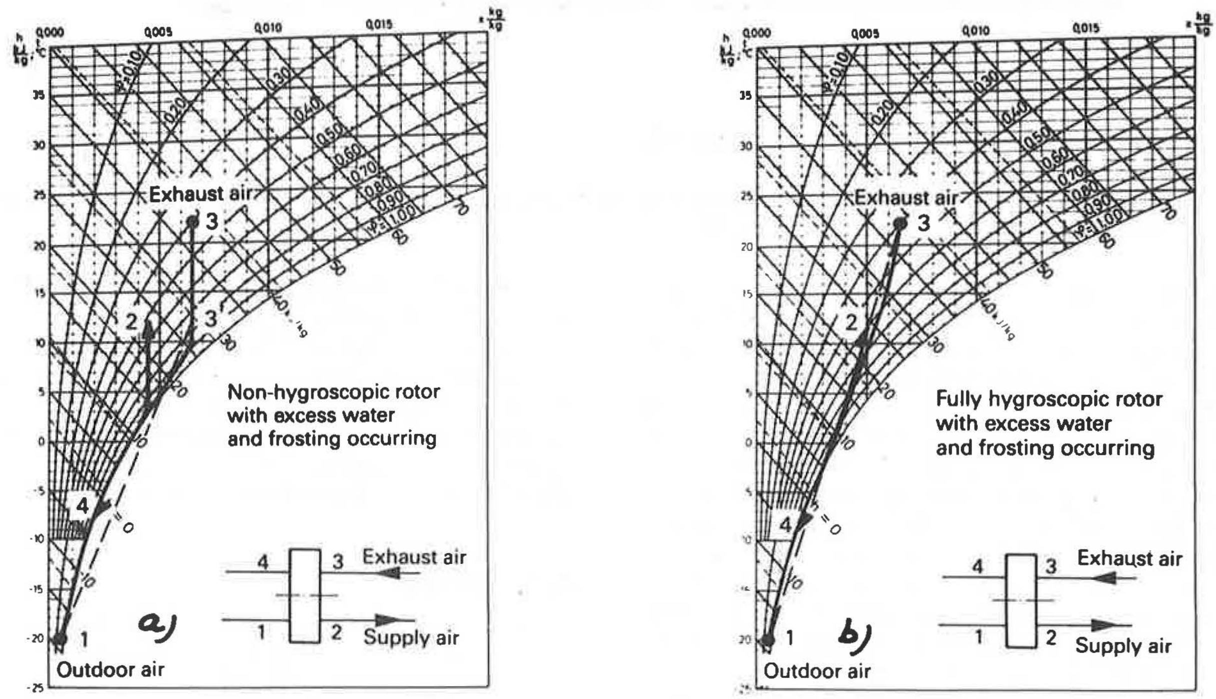

Figure 8 *Example of changes in air conditions at operations with a) non-hygroscopic and b) fully hygroscopic rotor, in which the excess water gives frosting.*

time. Since the temperature often varies during a 24-hour period, the actual frosting time may be much longer. Similarly, experience shows that the frosting limit is somewhat lower in practice, partially due to these variations, and a minor intersection of the saturation curve is therefore permissible without any frosting occurring.

DEFROSTING

The frosting problem can be solved either by entirely avoiding frosting or by defrosting the rotor after a certain amount of frost has been formed. Figure 9 shows an example of how the pressure drop may vary during a frosting/defrosting cycle. The defrosting time is very short as compared with the frosting time. Usual defrosting times vary between 5 and 15 minutes.

Frosting can be *avoided* by the supply air being preheated to the limiting temperature for frosting.

Defrosting of the rotor can be carried out in different ways. The usual method consists of the speed of the rotor being reduced to approximately 0.5 rpm as soon as the pressure drop across the rotor has exceeded a preset value.

Defrosting is started by a pressure switch, the setting of which should be approximately 50 Pa higher than the

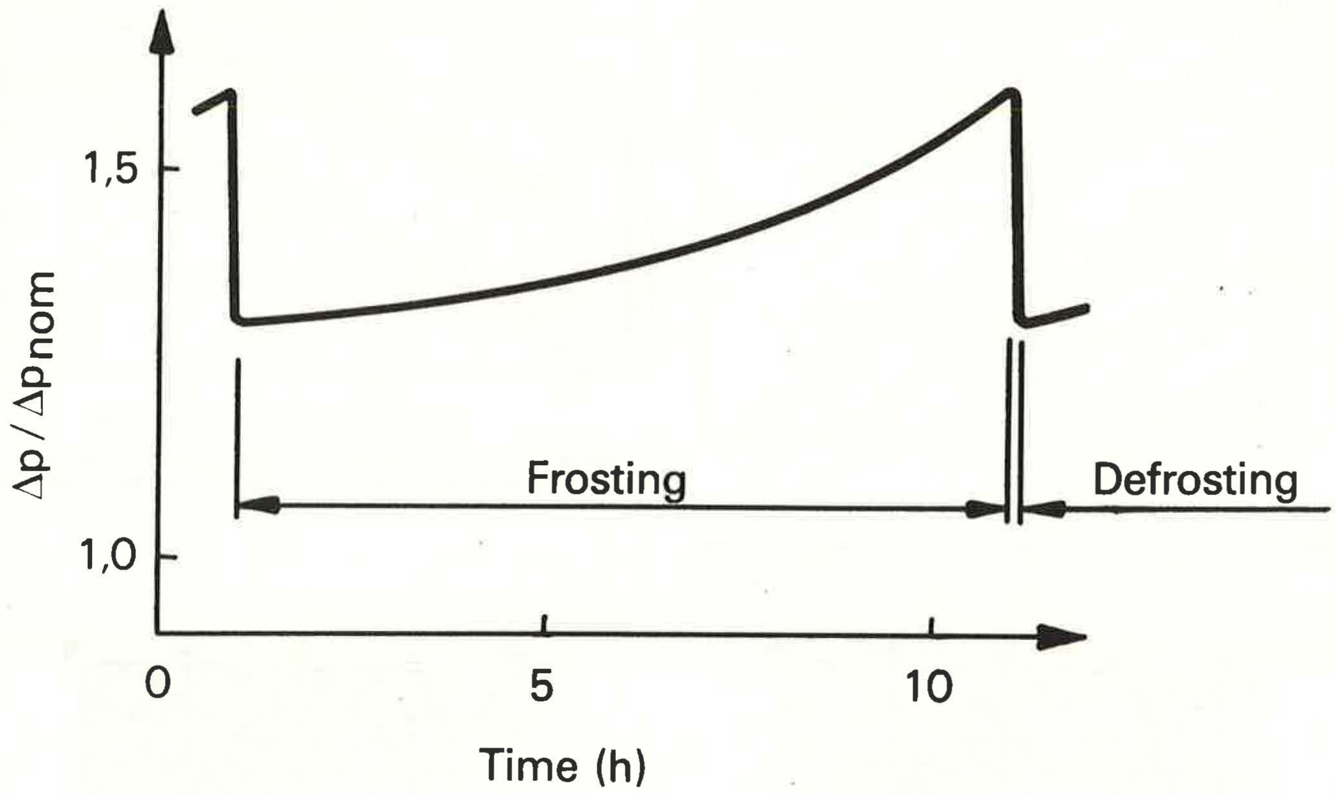

Figure 9 *Variation of the air pressure drop over the rotor during a frosting-defrosting cycle. *p/*pnom is the ratio between increased pressure drop (due to the frosting) and the nominal pressure drop.*

pressure drop prevailing after defrosting which, in turn, is approximately 30% higher than the normal pressure drop, due to the remaining excess water in the rotor. As an alternative, defrosting can be started by means of a timer and a thermostat a few times per 24 hours, whenever the outdoor temperature is below about -10°C. The duration of the defrosting should be controlled by means of a timer or a differential pressure switch. However, these methods can only be applied if the rotor can be run at variable speed.

Other methods must be employed for defrosting if the rotor is run at constant speed, e.g., a sufficient proportion of the supply air must bypass the heat exchanger to ensure that the mean temperature of the rotor during one revolution will be higher than 0°C in the frosting zone, which corresponds to an outlet temperature on the exhaust air side of about +5°C; and the outdoor air must be preheated to more than -10°C so the mean temperature of the rotor during one revolution will be higher than 0°C in the frosting zone.

During defrosting, the excess water which is formed as frost will be drained out of the rotor. As a result, not only excess water but also frosting must be avoided in rotors impregnated with a hygroscopic solution such as lithium chloride.

Which of the two alternatives, i.e., frosting-defrosting or avoidance of frosting, is chosen in a ventilation system is immaterial from the energy consumption aspect, since the reduction in energy use is negligible in both cases. From the energy power aspect, the frosting process is most advantageous, provided that no additional power need be supplied during defrosting, which is normally the case when the air is supplied to large premises.

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