Heat-Pump Evaporators Under Frosting Air Pressure Drops and Heat-Exchange Correlations

T. Argaud
Electricite De France

P. Panheleux
Electricite De France

D. Clodic
Ecole des Mines de Paris

HEAT-PUMP EVAPORATORS UNDER FROSTING AIR PRESSURE DROPS AND HEAT-EXCHANGE CORRELATIONS

AUTHORS

Thierry ARGAUD
Electricité De France, les Renardières, ADEB, E15, BP 1 77818 Moret sur loing FRANCE
thierry.argaud @ edf.fr

Patrick PANHELEUX
Electricité De France, 6 Quai Watier, MFTT, I84, BP 49 78401 Chatou FRANCE
patrick.panheleux @ edf.fr

Denis CLODIC
Ecole des Mines de Paris, Centre d'Energétique, 60 boulevard Saint-Michel – 75272 Paris Cedex 06, FRANCE
denis.clodic @ cenerg.ensmp.fr

ABSTRACT

Heat pumps often work in frosty conditions with decreasing performances. This frosting is characterised by thin frost layers mainly on the leading edge of fins. The narrow spacing of fins causes the clogging-up of the coil. It is the main effect, the frost layer thermal resistance being negligible. Yet, most studies focus on the frost growth, density and conductivity for large deposits on flat plates.

The goal of this study is to determine the pressure drops and the heat transfer parameters for different coils under frosting conditions. The main geometric parameters are pitch, type and surface treatment of fins. Three finned-tube heat exchangers have been studied under frosty conditions in an experimental set-up.

NOMENCLATURE

Le	Lewis number		V	air speed	$m.s^{-1}$
Re	Reynolds number		V_o	air speed between fins	$m.s^{-1}$
			s	fin spacing	m
A	area	m^2	S_{fr}	front area	m^2
G	mass speed	$kg.m^{-2}.s^{-1}$	T	temperature	K or °C
h	air exchange coefficient	$W.m^2.K^{-1}$			
h_m	mass transfer coefficient	$W.m^2.K^{-1}$	ΔP	pressure drops	Pa
H	latent energy	$J.kg^{-1}$	μ	viscosity	$kg.m^{-1}.s^{-1}$
Q_m	mass flow rate	$kg.s^{-1}$	ρ	air density	$kg.m^{-3}$
p	fin pitch	m			

INTRODUCTION

Frost on heat pumps decreases the heating performances. Yet, since the beginning of refrigeration, frost has been observed on heat exchangers. But, the use of refrigeration systems in heating mode is more recent, so, the heat pump frosting is known with less accuracy. This study focuses on the frosting of typical heat pump evaporators.

First, the distinctive feature of heat pump frosting is shown. Then, the experimental set-up reproducing the heat pump coils working conditions is described. Finally, the first results are given for three different coils. It is the beginning of a larger study on finned-tube heat exchangers.

DISTINCTIVE FEATURE OF HEAT PUMP FROSTING

Before using refrigeration for air-cooling or heating, refrigeration systems have been used in industry, typically in the food industry. An important frosting occurs to maintain food at low temperature. So, a lot of researchers (e.g. :Mao [1]) have studied the frosting in food industry conditions. In table 1, some differences between opened food cabinets and heat pumps are described :

FOOD CABINET	HEAT PUMP
Ambient air temperature about +15°C	Outside air temperature below +7°C
High humidity	Low humidity
Evaporation temperature about -20°C	Evaporation temperature ~ 10K below the air one
Frosting everyday	Frosting in winter if the relative humidity is higher than 60 % The more the air temperature decreases, the lesser frosting is.
Frosting studied on flat plates (large fin pitch)	Frosting studied on coils (fin pitch < 2 mm) smooth, waved, louvered or slit fins
large frost layers, high density, thermal conductivity through the layer	thin frost layers, clogging-up at the leading edge

Table 1 : Comparison between two different frosting

In the food industry, frosting is important and manufacturers make large fin spacing coils. So, most studies are about large frost layers on flat plates. For large deposits, if the air temperature is high, it is possible to observe the melting of frost at the layer surface, named « full growth period ». So, the growing of the layer is typical and its density and thermal conductivity are an important part of studies.

Some heat pumps are used in summer in the air-cooling mode where frost does not exist. Their fin spacing is small. Moreover, to reduce the size of coils and heat pumps, fin spacing are smaller and smaller. The full growth period does not exist. Instead of a uniform frost layer, there is an accumulation of frost at the leading edges. The main frost effect is the clogging-up of the heat exchanger, bringing a lower convective heat exchange. Senshu [2] has modelised this effect but with a uniform layer. To increase the heat exchange, manufacturers create louvered and slit fins. The clogging-up can be more important.

THE EXPERIMENTAL SET-UP

An experimental set-up has been built. The goal is to test several finned-tube heat exchangers under realistic frosting conditions. The experimental frosting conditions are described in table 2. The following parameters are kept constant during experiments :

TEST CONDITIONS	LOWER	UPPER
air speed	0.4 $m.s^{-1}$	1.2 $m.s^{-1}$
dry air temperature	-1°C	5°C
dew air temperature	dry air temperature - 5K	dry air temperature
R22 evaporation temperature	dry air temperature - 12K	dry air temperature - 5K
R22 mass flow rate	G > 200 $kg.m.s^{-1}$	decrease in R22 temperature due to pressure drop < 2K

Table 2 : Air/R22 parameters for heat pump frosting conditions

The heat exchange is both measured on the air side and on the refrigerant one. The mass of frost is measured by both two dew temperatures in the coil inlet and outlet, and by the water resulting from the frost melting. A micro video camera films the leading edge during frosting. The coil front area is 600 mm x 400 mm.

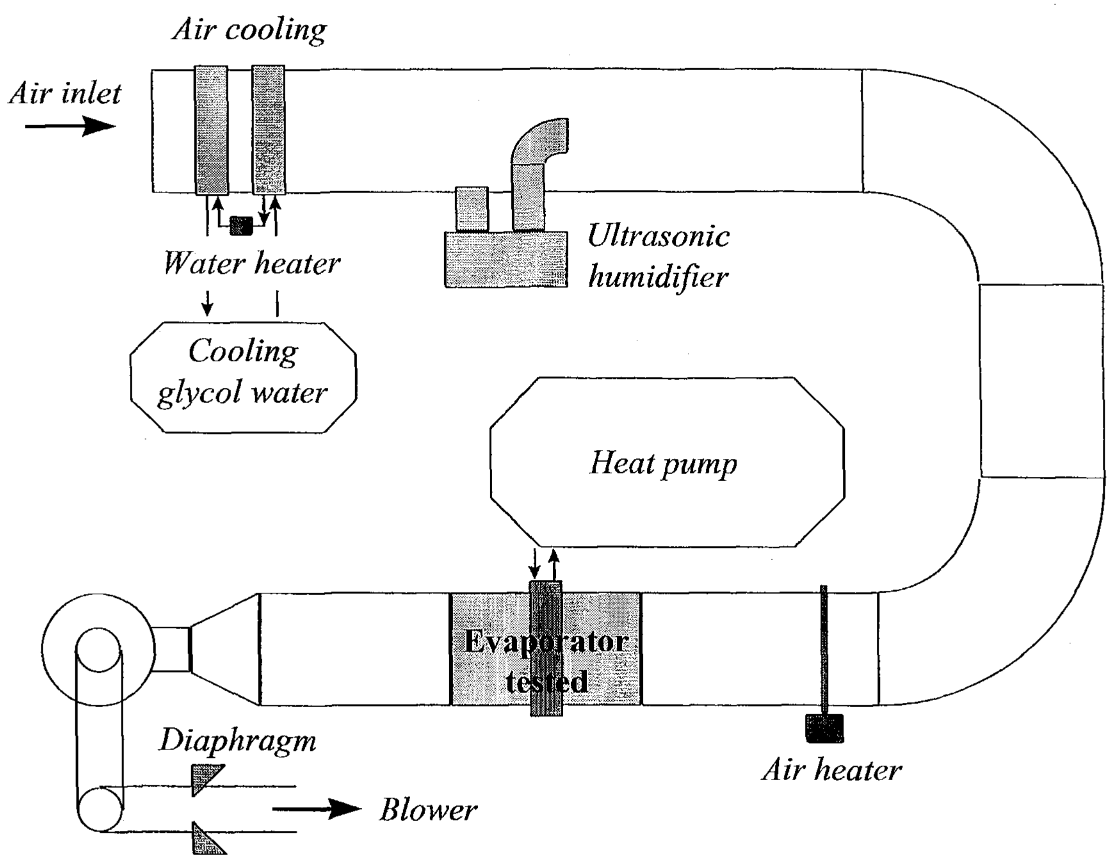

Figure 1 : The experimental set-up

Both pressure drops and heat exchange are studied. To make easier the study :

- the frosting has to be uniform on the front area,
- the heat transfer coefficient inside the tubes has to be 100 times greater than the outside one.

In our experiments, the R22 is used, because the tubes temperature has to be uniform. The refrigerant flow rate is controlled by a variable speed compressor and a distributor to get low refrigerant pressure drops. Moreover, another water-refrigerant exchanger (superheater) takes place after the tested coil to end the evaporation and add a superheat.

The electronic expansion device is used as a fixed valve to keep the same evaporation temperature during the frosting cycle. When the air exchange decreases, the superheater prevents small superheat. Usually, for a real heat pump, the more the frost grows, the lower is the air flow rate. In our case, the air flow can be kept constant to help the experiment interpretations.

FIRST EXPERIMENTS

Since the frost deposit is important at the leading edge, three-one-row coils have been first studied :

COILS	n°1	n°2	n°3
Fin pitch	1.8 mm	2.5 mm	2.5 mm
Fin geometry	smooth	waved	louvered
Surface treatment	hydrophilic	hydrophilic	hydrophilic

The fin thickness is 0.1 mm

Table 3 : Characteristics of the first coils tested

It is easier to study frosting when coils are dry at the beginning, and it is possible to do experiments with a good repeatability. Yet, the heat pumps frosting occurs in cycles of frosting - defrosting. Thus, it is more realistic to study frosting after a defrosting. Indeed, it is very difficult to dry the fins on heat pumps or to characterise the surface state after a defrosting. Even with an hydrophilic treatment, some micro-droplets or a micro-film can still exist. The time and energy cost seem too high to try to completely dry the coils. Several studies are about the incipient frost formation (e.g. : Sèki [3], Tao [4]). During this period, micro subcooled droplets appeared. This stage creates a delay in the frost growth. When there are cycles, the delay is not observed. The water still present on fins after defrosting never seems to be subcooled, but frozen as soon as its temperature goes down, under 0°C, giving its solidification energy to the coil. The sizes of these water droplets or films are greater than of those appearing on the first period on a dry fin.

Pressure drops

Pressure drops are commonly written : $\Delta P = f \dfrac{\rho}{2} v^2$

« f » depends on air speed. After several dry air experiments, it was found : $f = \alpha \dfrac{2}{\rho} v^{-n}$

« n » is between 0.3 and 0.5 for the three exchangers. The mean value of 0.4 gives good results for every coils, not only when the air is dry, but also with frost. The figure 2 shows the differences for the pressure drops between a frosting with a dry coil at the beginning and a wet coil after a defrosting.

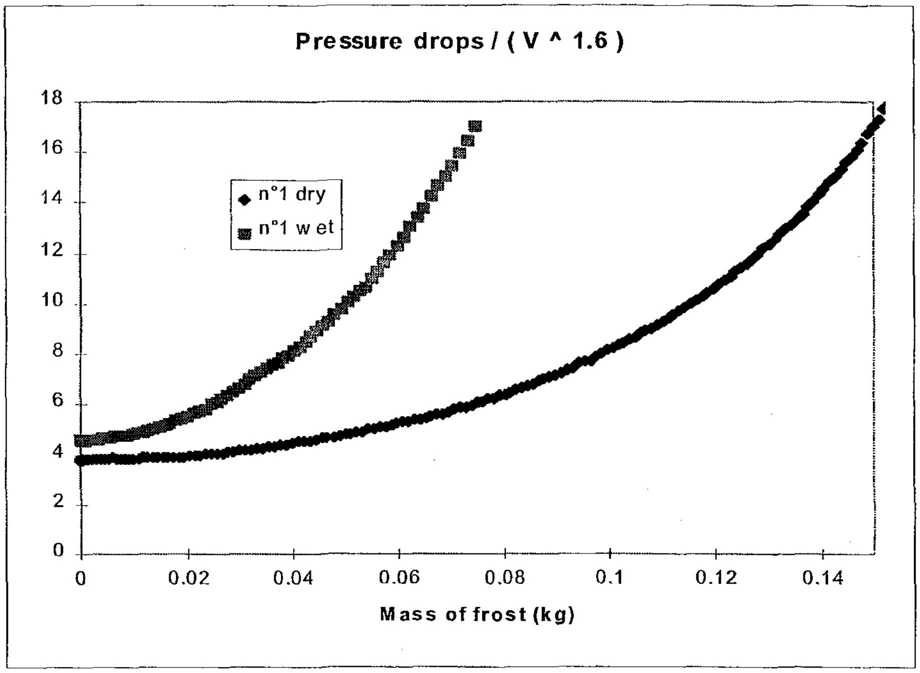

Figure 2 : First coil results, with a dry surface and after a defrosting (wet surface)

The more the fin spacing raises, the lesser is the difference between a dry or wet coil at the incipient frosting. Louvered fins give an effect of lower spacing than waved fins. So, the surface treatment is more important when the fin pitch decreases.

For the 2.4 mm fin spacing, a wet surface is similar with an incipient frost layer. For the 1.7 mm fin spacing, at the beginning of frosting the pressure drop is not only higher than with a dry coil, but the raising is faster. A possible explanation can be the clogging-up of the coil, the frost being concentrated at the leading edge. For large spacing, the phenomenon seems to be the increasing of the fin thickness with a uniform frost layer, the air flow can still go through the coil. For lower spacing, the phenomenon focuses on the smaller and smaller air channel between the fins leading edge.

Usually, in a real use of heat pump, frosting occurs after defrosting. So, in the following, frosting has been studied after defrosting with several frosting-defrosting cycles.

The figure 3 shows the parameter « α » linked with the mass of frost, after a defrosting for the three coils :

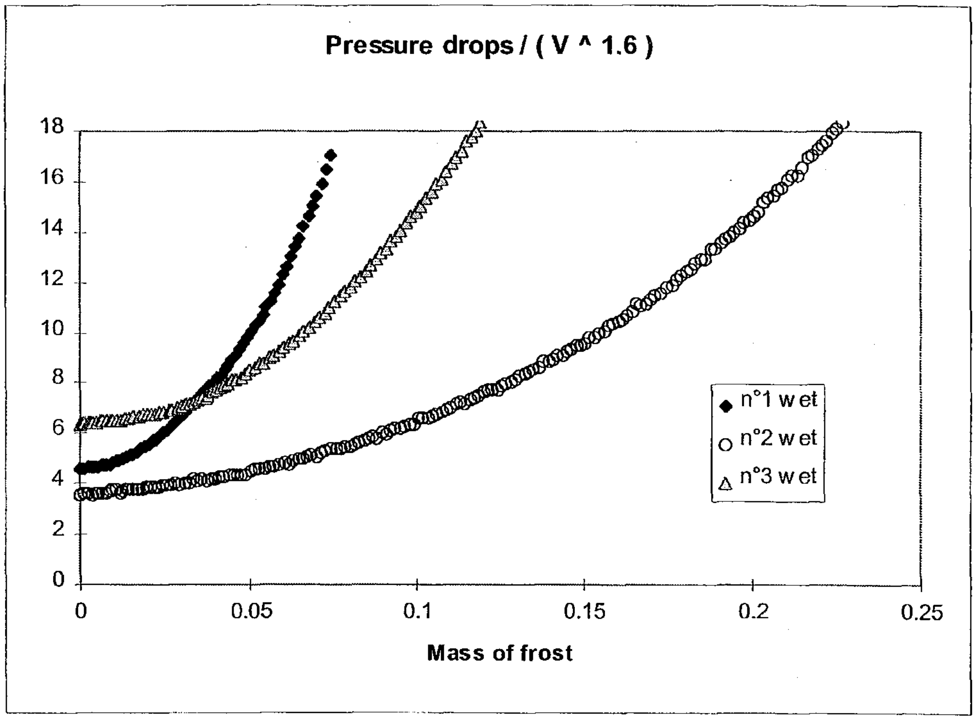

Figure 3 : Frosting after defrosting for the three coils

The result is :

- for the same fin spacing, the pressure drops raise faster for louvered coil than for the waved one,
- the first coil has lower pressure drops than the third one at the beginning of frosting, but because of its small fin pitch, the clogging-up happens earlier.

Heat exchange

The mass of frost is directly linked with the mass transfer when the coil temperature, the air temperature and the dew one are given. Thanks to the Lewis law, the total energy ($\varphi + \varphi_m$) is proportional with the mass of frost.

The dry heat transfer is : $\varphi = h\,A\,\Delta T$

The mass heat transfer is : $\varphi_m = h_m\ A\ \Delta w\ H$

where : $h_m = \dfrac{h}{Cp\ Le^{2/3}}$

For the first coil, which has smooth fins, the experiments show the increase of the heat exchange by a ratio of 50%. That is to say that, the dry exchange is multiplied by 1.5 and the latent energy has to be added. The Lewis law with Le = 0.9 is still working and the total exchange is very improved. Moreover, with a constant air flow rate, the heat exchange is constant during the frosting. So, the frosting problem is really the clogging-up of the coil and not the thermal resistance in the frost layer. Payne [5] use this property, by raising the fan power as the frost layer grows.

It is possible to show that the Reynolds number is almost constant during the frost grows between the fins at the leading edge. Indeed, at the leading edge the hydraulic diameter is :

$$\phi = 2s$$

So, the Reynolds is :

$$\mathrm{Re} = \frac{\rho V_o (2s)}{\mu}$$

We have too :

$$V_o s = V p = \frac{Q_m}{\rho S_{fr}} p$$

$$\mathrm{Re} = \frac{2 p Q_m}{\mu S_{fr}}$$

Frost brings a roughness surface. That probably increases the heat exchange by an extended surface or by turbulence. No differences are observed in heat exchange between a frosting on a dry surface or after a defrosting.

For the waved fins, the exchange increase is lower but it is also constant during the frosting cycle. For the third coil with louvered fins the same phenomenon is observed. The louvers blocking does not seem to decrease the heat exchange.

CONCLUSION

This study deals with frosting for heat exchangers designed for heat pumps systems. The main phenomenon is the clogging-up of the coil.

An experimental set-up has been built to study the heat pump frosting. Three finned-tube heat exchangers have been tested. Smooth, louvered and waved fins for two fin pitches have been compared.

The major results are :

1. the important increase of pressure drops when the coil surface is wet (after a defrosting in spite of hydrophilic fins), for a mass of frost given, overall for small fin pitches. A surface treatment may reduce the incipient frost formation,
2. the clogging-up of an average smooth fin pitch (1.8 mm) is faster than a louvered coil with a large fin pitch (2.5 mm) sized for the same pressure drops without frost,
3. the frost increases the total heat transfer, not only by the mass transfer, but also by the increase of the dry heat transfer on a roughness surface,
4. during frosting, the growth of the frost layer does not really reduce the heat transfer when the air flow rate is kept at the same value.

Now, small fin pitches (1.2 mm), slit fins and another hydrophilic treatment have to be tested. To further investigate the heat pump frosting, we have to study both pressure drops and heat exchange together. The advantages in a controlled blower to avoid the air flow rate decrease have to be studied too. The results will help to know how to size a good coil including frosting conditions.

REFERENCES

[1] MAO Y., BESANT R. et REZKALLAH K., "Measurement and correlations of frost properties with airflow over a flat plate", *ASHRAE Transaction*, 98, p2 (1992), p. 65-78

[2] SENSHU T., YASUDA H., OGUNI K. et ISHIBANE K., "Heat pump performance under frosting I & II", *ASHRAE Transactions*, 96, p1 (1990), p. 324-336

[3] SEKI N., FUKUSAKO S. et MATSUO K., "Incipient phenomena of frost formation", *Bulletin of JSME*, 27, 233 (1984), p. 2476-2482

[4] TAO Y., BESANT R. et MAO Y., "Characteristics of frost growth on a flat plate during the early growth period", *ASHRAE Transactions*, 99, p1 (1993), p. 746-753

[5] PAYNE V. et O'NEAL D., "The effect of outdoor fan airflow on the frost/defrost performance of an air-source heat pump", *Recent research in heat pump design ASME AES*, 29 (1993), p. 107-115