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ABSTRACT

A model capable of simulating evaporators with multiple modules under frosting conditions was developed and validated. Effects of various geometrical parameters on fan and compressor power consumption were quantified and some general guidelines for design were developed. These general guidelines were used to design new prototypes for a given external load keeping in mind the packaging restrictions and constraints on tube and fin availabilities.

1. INTRODUCTION

This document presents the use of a validated numerical design tool for a medium temperature open display case evaporator. The most significant aspect of current medium temperature open display case evaporators is the frost accumulation and the resultant change in air-side heat transfer and pressure drop characteristics. The change in air-side pressure drop due to frosting leads to higher fan power requirements and also results in lower airflow rate through the air curtain wherever constant-speed fans are used. Any drop in air-curtain airflow leads to higher infiltration and hence higher latent and sensible loads on the evaporator. Thus there is a necessity to model the evaporator in frosting.

Most of the early research on frosting was experimental in nature, and focused on local phenomena. A comprehensive review of the effect of frosting on heat exchangers is provided by Kondepudi and O'Neal (1987). Efforts to empirically arrive at improvements in heat exchanger performance in frosting conditions were studied by Ogawa, Tanaka, and Takeshita (1993). Some of the methods suggested were staging, cutting, and/or extending the fins. Most of the work was in refrigerator and heat pump evaporators designed for much lower temperatures.

Initial modeling of mass transfer was restricted to condensation in a/c coils or modeling of frost formation on flat structures. Verma (2002) modeled secondary refrigerant based systems and DX coils in frosting. This paper builds on Verma's work.

The challenge in display case evaporator modeling is the fact that the geometries of current heat exchangers lie outside the range of available correlations for heat transfer and pressure drop. This necessitates intelligent modification to the available correlations.

This document first briefly addresses the key aspects of the model and its validation. Then the design objectives and the design parameter space are specified. Within this parameter space, a series of guidelines are then arrived at using quantitative prediction for energy consumption. Finally, these guidelines are used to arrive at designs for newer prototype coils.

2. THE NUMERICAL MODEL AND VALIDATION

The numerical model uses a quasi-steady finite-volume approach to simulate cross-counterflow DX evaporators in frosting conditions. Given the finite-volume method, the model can handle variable fin and tube properties through a modular approach. The numerical model is implemented using Engineering Equation Solver (EES) which is a Newton-Raphson based nonlinear simultaneous equation solve (Klein, 1995).

A few simplifying assumptions are used in the modeling.

1. Frost growth modeling is simplified by considering only mature frost growth.
2. Fin and tube frost thickness does not vary within any given finite volume.
3. As temperature increases, frost density grows to that of ice at 0 ^oC, then, density remains constant while thickness increases. Frost thickness thus increases monotonically.

4. Only the leading edge finite volume contains a superheated segment.

The heat and mass transfer modeling method and governing equations are given by Verma (2002) and not reproduced here. The essence is the use of the coupled heat and mass transfer equations along with a two-lump model of the frost on the fins and tubes. The correlations used are also discussed in Verma (2002). The main correlations are also listed in the References (Hayashi (1977), Gnielinski (1976), Souza (1995), Wattelet (1994)).

The model was validated against data taken at 2-minute intervals from a well-instrumented medium-temperature vertical display case, for two evaporators having very different configurations (referred to as the baseline coil and Prototype-1). The data from these experiments (Faramarzi et al, 2003) provided both the input data for the model and additional data for comparison with the modeling results. Redundant data was available, and was used to further analyze the data.

Without getting into the details, the gist of the validation process is presented here. The variables chosen as input to the model were the geometry of the coil, air inlet temperature, initial air flow rate (determined from an energy balance), the fan curve, duct pressure loss, refrigerant inlet enthalpy, refrigerant exit pressure and the infiltration.

Both the baseline coil and Prototype-1 coil led to the same conclusions. The following is a list of the model's capabilities and failings

1. To achieve a particular discharge air temperature, the model predicts a lower evaporating and surface temperature than suggested by the experimental values.
2. The mass of frost matches within 4%.
3. The model airflow rate decays slower than indicated by the energy balance.
4. The qualitative trends of the various variables are predicted very well
5. The difference in simulated and measured outlet air temperatures is ~1-2 C.

Complete validation of the model was not possible due to the absence of direct airflow measurements and the uncertainties in the fan curves. Since the calculated refrigerant-side resistance is small (ΔT~0.75 C from the refrigerant to outer surface of the frost) and fairly certain, the air-side heat transfer coefficient would have to be underestimated by 40% or more to explain the difference between the measured and predicted air exit temperatures. Since spot measurements revealed large variations in air velocity, it is likely that the uniform transverse air flow assumption is mainly responsible for the difference between measured and predicted evaporator performance.

3. PARAMETERS FOR DESIGNING AND COMPARING COILS

The efforts to improve performance are directed at

1 Reducing the total heat transfer resistance. This is done by reducing the air-side resistance by increasing $(hA)_{air}$, reducing the refrigerant-side resistance by focusing on ΔT_{ref} and ΔT_{sat} and by reducing the resistance through the frost. Reducing the resistances lead to higher T_{evap} which leads to lower frost accumulation and hence lower compressor and fan power requirements.

2 Distributing the frost to maintain uniform free flow area. Nonuniform frosting results in huge penalties in terms of ΔW_{fan}. Frost distribution is visualized by plotting maximum velocities along the airflow direction.

The parameters that can be varied to achieve the above objectives are presented in Table 3.1.

No.	**Parameter**	**Range**	**Notes**
		COIL PARAMETERS	
1	Tube Diameter	10mm (3/8") microfinned tubes	Few 8mm tubes also considered
2	Transverse tube spacing	1.25" (31.75mm), 1.5" (38 mm)	
3	Longitudinal tube spacing	1.08" (26mm), 1.3" (33mm)	
4	Transverse tube row no.	4,5,6	
5	Longitudinal tube row no	10,12,15,18	
6	Duct height	5", 6", 7.6"	
7	Fin thickness	0.1397 mm (0.0055") to 0.2413 mm (0.0095")	Few 0.0105" fins checked
8	Fin density	2 to 7 fpi	
9	Number of modules	2, 3, 4	3 to 9 rows deep
10	Coil width	8 feet	Usual value for display

			cases
11	Coil depth	0.3m (12”) to 0.48m (19”)	
12	Number of circuits	3, 4, 5, 6, 8	
13	Inter-module spacing	12mm (0.5”)	All coils
14	Tube and fin material	Cu tubes, Al fins	Cu fins: costly
15	Position of coil	Bottom mounted, back mounted	
OPERATING PARAMETERS			
1	Steady state load	4 kW	
2	Discharge air temperature	-1.5 C, -2.5 C	
3	Store conditions	24 C, 55% R.H.	ASHRAE
4	Airflow rate	750 cfm, 700 cfm	Variable speed fans
5	Infiltration	180 cfm	Faramarzi et al (2003)

Table 3.1 Parameters and their range

The quantitative results generated by the simulations include fan and compressor power. These are used along with the frost distribution figures to compare coils. It is sufficient to use the compressor and fan power instead of the total energy consumption (which includes the pulldown loads due to defrosting of the coils) because the inclusion of the pulldown penalty does not change the comparison between coils.

4. EXPLORING THE PARAMETER SPACE

This section is presented as the study of a series of changes within the parameter space presented in Section 3 aiming to reduce the heat transfer resistance and/or make the frost distribution more uniform.

4.1 Effect of changing the tube spacings

Here we explore changing the transverse and longitudinal spacing without changing the overall core dimensions. This is done to understand the effect of spacing alone on performance. Table 4.1.1 and 4.1.2 show the effect of increased transverse and longitudinal tube spacings

Duct=7.6", Longitudinal spacing = 1.08", DAT= -1.5 C, 3/8" tubes, 0.0095" Al fins, 6 circuits							
Fin Staging	Transverse Spacing	Tevap	Frost mass	Fin frost	Tube frost	Wcomp	Ideal Wfan
	inches	[C]	[kg]	[m]	[m]	[W]	[W]
3.5 hour runtime							
10 x 6 4@2fpi, 6@4fpi	1.25”	-3.87	6.5	0.00104	0.00146	1016	7.4
10 x 5 4@2fpi, 6@4fpi	1.5”	-4.05	6.43	0.00099	0.00151	1021	6.23

Table 4.1.1 Effect of increasing transverse tube spacing (same core volume)

An increase in the transverse tube spacing results in a drop in the fin efficiency causing an increased air-side resistance and a consequent drop in evaporating temperature. The advantage of increasing the transverse tube spacing is the reduction in frost formation on the fin and lower maximum velocities resulting in lower fan power requirements. Thus on its own, increasing the transverse tube spacing is beneficial only if the fan efficiencies are low (in this example, the fan has to be less than 26% efficient). But usually the increase in transverse tube spacing is done in conjunction with some other change.

Duct=7.6", Transverse spacing = 1.5", DAT= -1.5 C, 3/8" tubes, 0.0095" Al fins, 6 circuits								
Fin Staging	Longitudinal Spacing	Tevap inlet	Tevap outlet	Frost mass	Fin frost	Tube frost	Wcomp	Ideal W fan
	inches	[C]	[C]	[kg]	[m]	[m]	[W]	[W]
3.5 hour runtime								
12 x 5 4@2fpi, 8@4fpi	1.08"	-3	-3.5	6.39	0.00085	0.00133	999	6.39
10 x 5 4@2fpi, 6@4fpi	1.3"	-3.52	-3.9	6.45	0.00082	0.00136	1011	5.99

Table 4.1.2 Effect of increasing longitudinal tube spacing (same core volume)

Similarly, when the longitudinal tube spacing is increased the lower fin efficiency increases the resistance on the air side while the refrigerant side area is reduced. The small reduction in refrigerant and air-side pressure drops do not make up for the loss in evaporating temperature.

4.2 Increasing core volume.

The core volume can be increased by using a deeper coil. Given that a coil deeper than 14" has to be mounted in the back wall of the case, such a deeper coil has to be thinner (shorter height). Tables 4.2.1, 4.2.2 compare the effect of such an increase in depth.

Duct=7.6", Transverse spacing = 1.5", Longitudinal spacing=1.3" DAT= -2.5 C, 3/8" tubes, 0.0095" Al fins, 6 circuits Flow rate=750 cfm					
Fin Staging	Tevap inlet	Tevap outlet	Frost mass	Wcomp	Ideal W fan
	[C]	[C]	[kg]	[W]	[W]
6 hour runtime					
12x5 3@3 3@4 6@6	-3.31	-3.8	12.7	1028	13.03
12x5 3@2 5@4 4@5	-3.6	-4.1	12.6	1034	12.42

Table 4.2.1 7.6" high 12 row coils

Tables 4.2.1 and 4.2.2 show that making a coil deeper at the cost of height leads to a massive increase in the fan power requirement due to the increase in velocities. Even though the other resistances are comparable, the higher fan power requirement makes this infeasible.

Duct=5", Transverse spacing = 1.2", Longitudinal spacing=1.3" DAT= -2.5 C, 3/8" tubes, 0.0095" Al fins, 4 circuits Flow rate=700 cfm					
Fin Staging	Tevap inlet	Tevap outlet	Frost mass	Wcomp	Ideal Wfan
	[C]	[C]	[kg]	[W]	[W]
6 hour runtime					
15x4 5@3 5@4 5@5	-3.4	-4.5	12.8	1039	30.6
15x4 5@3 7@4 3@5	-3.4	-4.6	12.9	1041	31

Table 4.2.2 5" high 15 row coils

It is possible to reduce the fan power requirement by increasing the transverse tube spacing. This can be achieved either by making the coil deeper while maintaining a coil of 5" height (i.e. a 18x3 coil) or by increasing the duct height to 6". Increasing the depth to 18 rows led to a large refrigerant pressure drop and hence increased the compressor power requirement. Hence this option did not improve on the performance shown in Table 4.2.2. However increasing the coil height by 1" did reduce the fan power without increasing the compressor power requirement as shown in Table 4.2.3.

Duct=6", Transverse spacing = 1.5", Longitudinal spacing=1.3" DAT= -2.5 C, 3/8" tubes, 0.0095" Al fins, 4 circuits Flow rate=700 cfm					
Fin Staging	Tevap inlet	Tevap outlet	Frost mass	Wcomp	Ideal Wfan
	[C]	[C]	[kg]	[W]	[W]
6 hour runtime					
15x4 5@3 5@4 5@5	-3.25	-4.4	12.9	1034	11.6
15x4 5@3 7@4 3@5	-3.31	-4.5	12.9	1035	11.5

Table 4.2.3 Comparison of 15 row coils of 6" height

The improvement in performance is due to lower air-side pressure drop and the increase in air-side area over the coils in Table 4.2.2. Compared to the coils in Table 4.2.1, the refrigerant side resistances are higher. This issue is addressed in Section 4.4.

4.3 Varying fin density along length of the coil

Fin density can be used to increase the air side area. It can also be used to achieve better distribution of frost. Both these possibilities are explored in this section

First, consider the possibility of improving the frost distribution of the coils discussed in Table 4.2.3 using a different fin density. The frost distribution can be improved by either lowering the fin density or by making the fins thinner in areas of high frosting. Reducing the fin thickness is a non optimal use of metal and also leads to an increase in compressor power requirement. Figure 4.3.1 compares the frost distributions achieved by lowering the fin density and by reducing the fin thickness. It is seen that the frost distributions are equally good. The energy and material usage were lower when the better distribution of frost was achieved (see Figure 4.3.1) using lower fin density in the portions of high frosting (energy numbers not shown.).

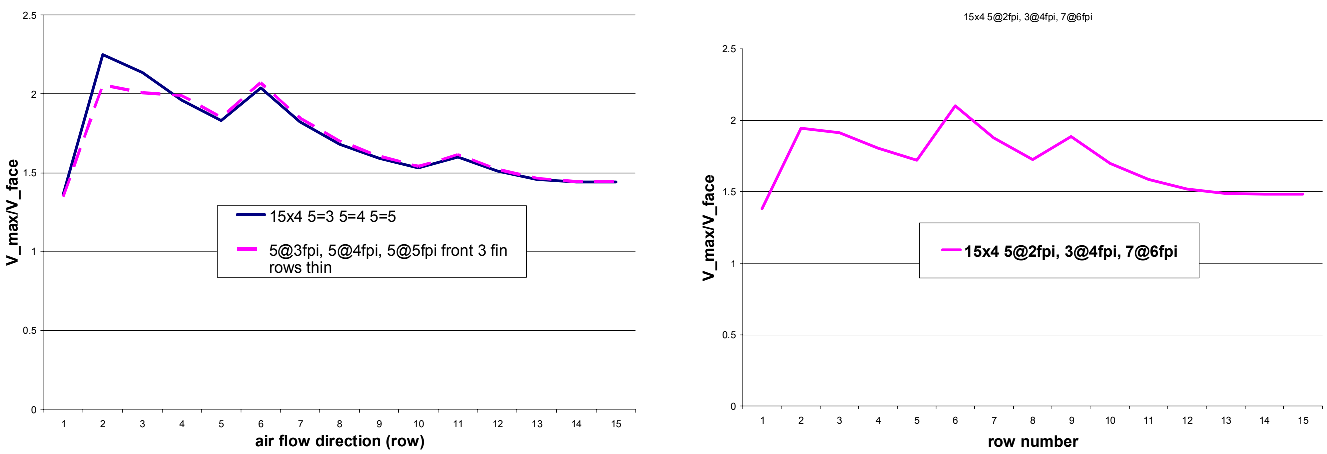

Figure 4.3.1 Velocity distribution in 6" high 15 row coils (left: original and thinner fin stock, right: using lower fin density at the front)

This method of improving frost distribution using varying fin densities can be thought of as "shifting" fin area from the front to the back. However there is a limit beyond which increasing fin density at the back will not help redistribute frost. This limit is dictated by "pinching" conditions at the trailing edge. Increasing fin density at the back beyond this limit does not help as there is no additional mass transfer. This limit was 7fpi at the back of the coil for the given load conditions and airflow rates.

4.4 Increasing the number of circuits

It is possible to lower the refrigerant side resistance by lowering the pressure drop or by increasing the refrigerant side heat transfer coefficient. Reducing tube diameters was found to increase the pressure drop to an extent that the gain in heat transfer coefficient was negated. Increasing the number of circuits lowers the pressure drop but it also reduces the refrigerant heat transfer coefficient. Figure 4.4.1 captures these two effects in detail. Though the inlet evaporating temperatures are lower, the reduction in pressure drop results in a higher exit temperature for the refrigerant which leads to warmer surfaces at the leading edge on the air-side. This reduces the frosting at the leading edge, thus improving the frost distribution. Hence there is a reduction in both compressor and fan power consumption with the added benefit of more uniformity in frosting.

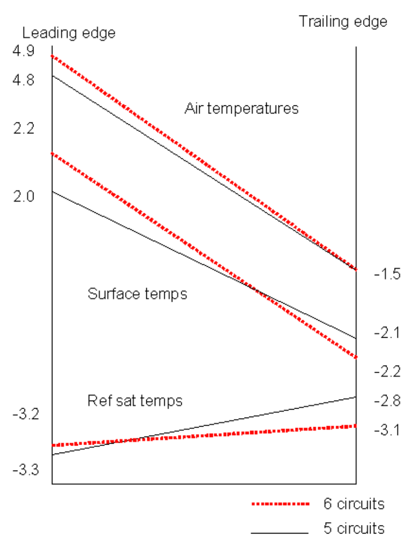

Figure 4.4.1 Comparison of two 12x5 coils with different number of circuits

Thus Section 4 outlined various ways to achieve the objectives listed in Section 3. Increasing fin density at the back to a limit of ~6fpi and using a larger number of circuits were the steps that helped reduce the air and refrigerant side resistances. Frost distribution improvement can be attained by lowering the refrigerant pressure drop and opening up the coil in the transverse and longitudinal directions.

5. RESULTS: EVAPORATOR DESIGN FOR GIVEN CORE VOLUME

This section presents the development of two prototype coils; one subject to manufacturing restrictions and the other a theoretical coil with these restrictions relaxed. Both the coils have the volume limitation outlined in Table 3.1. The following manufacturing constraints were considered while designing the first of these two coils (Prototype-2)

1. The maximum tube spacing available is 3.175 cm (1.25”) in the transverse and 2.7 cm (1.08”) in the longitudinal direction
2. The fins available are 0.1397 mm (0.0055”), 0.1905mm (0.0075”) and 0.2413mm (0.0095”) thick
3. The available core dimensions are 0.193*2.057*0.343 m (7.6 x 81 x 13.5”)
4. The number of rows in the airflow direction is restricted to even numbers.

These restrictions were combined with the guidelines from Section 4 to arrive at a design for Prototype-2. Table 5.1 compares the new design with an earlier Prototype (Neshan 2003) also tested by Faramarzi (2003).

Duct=6", Transverse spacing = 1.25", Longitudinal spacing=1.083" DAT= -2.5 C, 3/8" tubes, 0.0095" Al fins, 6 circuits Flow rate=700 cfm				
Fin Staging	Tevap inlet	Frost mass	Wcomp	Ideal Wfan
	[C]	[kg]	[W]	[W]
6 hour runtime				
Prototype-2 12x6 5@3fpi 7@4fpi	-4.1	12.7	1093	13.6
Prototype-1 10x6 4@2fpi 6@4fpi	-4.9	12.7	1104	19.3

Table 5.1 Comparison of Prototype-2 with Prototype-1

Experiments of Prototype-2 confirmed that the compressor energy usage per day dropped by 28% and that the system EER increased by 35% when compared to the original baseline coil. The main reasons for the better performance are the better frost distribution, higher air-side area due to the larger depth of the coil and higher fin density.

As a final exercise, the restrictions on the depth of the coil and inter-tube spacings observed while developing Prototype-2 are removed. The resulting coil was developed as a continuation of the parametric analyses presented in Section 4. Two coils with low energy consumption and a pretty uniform frost distribution are shown in Table 5.2.

As seen from Table 5.2, 5.1, and those in Section 4, the two coils presented in Table 5.2 achieve a lower refrigerant side pressure drop while not losing out on refrigerant side heat transfer coefficient. The coils developed in Table 5.2 are more ambitious than Prototype-2 in terms of cost (i.e. new fin geometries) and have not been built. But the fact that Prototype-2 did lead to benefits suggests that these 2 coils too will prove beneficial in terms of efficient operation. Cost factors are not within the scope of this analysis and will be a key factor in choosing coils. Nonetheless, the methodology presented here can be used to design coils with any restrictions as was shown in the development of Prototype-2.

Duct=6", Transverse spacing = 1.5", Longitudinal spacing=1.3" DAT= -2.5 C, 3/8" tubes, 0.0095" Al fins, 6 circuits Flow rate=700 cfm					
Fin Staging	Tevap inlet	Tevap outlet	Frost mass	Wcomp	Ideal Wfan
	[C]	[C]	[kg]	[W]	[W]
6 hour runtime					
15x4 3@2 4@3 8@6	-3.7	-4	12.5	1020	10.8
15x4 5@2 3@4 7@6	-3.6	-3.95	12.4	1015	11

Table 5.2 Coils with manufacturing restrictions removed

CONCLUSION

A validated numerical simulation models was used to design improved medium temperature display case evaporators. The simulations and subsequent testing suggests that increasing the air-side area with better frost distribution by varying fin density leads to much better performance. Deeper back-mounted coils with lower flow rates and a higher number of circuits prove superior to taller and less deep bottom-mounted coils.

NOMENCLATURE

Al	Aluminum	
DAT	Discharge Air Temperature	(C)
DX	Direct expansion evaporators	
Fpi	Fins per inch	(39.37 fins per meter)
T	Temperature	(C)
W	Work	(Watts)

Subscripts

ref	refrigerant
sat	saturation
evap	evaporator

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