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EFFECT OF REFRIGERANT MAL-DISTRIBUTION IN FIN-AND-TUBE EVAPORATORS ON SYSTEM PERFORMANCE

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ABSTRACT

Refrigerant mal-distribution in fin-and-tube evaporators for residential air-conditioning (RAC) is investigated numerically in this paper. A model of the system is developed in the object-oriented modeling language Modelica. The models of the compressor and expansion valve are static, whereas the condenser is a dynamic moving boundary model. The evaporator model is a dynamic distributed one-dimensional homogeneous equilibrium model, in order to capture the distribution phenomena. Fin-and-tube heat exchangers usually have a complex circuitry, however the evaporator will be simplified to be two straight tubes. The refrigerant mal-distribution is then induced to the evaporator by varying the vapor quality at the inlet to each feeder tube, the pressure drop through each feeder tube and the air-flow across each tube. Finally it is shown that air-flow mal-distribution can be compensated by an intelligent distributor, that ensures equal superheat in both tubes. The refrigerant is R410a.

Keywords: Mal-distribution, air-conditioning, evaporator, modeling, Modelica.

NOMENCLATURE

A	Area [m ²]
D	Outer tube diameter [m]
d	Inner tube diameter [m]
F	Distribution factor [-]
G	Mass flux [kg/m ² s]
h	Specific enthalpy [J/kg]
L	Tube length [m]
M	Mass [kg]
\dot{m}	Mass flow rate [kg/s]
n	Number of control volumes [-]
p	Pressure [Pa]
\dot{Q}	Heat flow rate [J/s]
T	Temperature [°C]
t	Time [s]
U	Internal energy [J]

U	Overall heat transfer coefficient [W/m ² K]
V	Velocity [m/s]
\dot{V}	Volumetric flow rate [m ³ /s]
x	Vapor quality [-]
Δp	Pressure drop [Pa]
η	Efficiency [-]

Subscripts

<i>air</i>	Air
<i>cond</i>	Condenser
<i>corr</i>	Correlation
<i>evap</i>	Evaporator
<i>f</i>	Friction
<i>fr</i>	Frontal
<i>ft</i>	Feeder tube
<i>in</i>	Inlet
<i>is</i>	Isentropic
<i>m</i>	Mean
<i>out</i>	Outlet

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<i>sh</i>	Superheat
<i>tot</i>	Total
1	Tube 1
2	Tube 2

INTRODUCTION

Energy consumption and refrigerant charge in refrigeration systems are becoming increasingly important for environmental, legislative and economic reasons. Hence compact dry-expansion multi-channel heat exchangers are of interest for future refrigeration technology. The use of more channels in evaporators gives rise to mal-distribution phenomena, which have shown to reduce evaporator capacity and thus system energy efficiency. Refrigerant mal-distribution can be caused by different reasons, such as air-flow mal-distribution, non-uniform air-temperature, fouling, improper heat-exchanger or distributor design and installation, or interactions of these.

Several studies on mal-distribution in evaporators and its impact on cooling capacity and coefficient of performance (COP) have been made. Payne and Domanski [1], Lee et al. [2] and Kim et al. [3] studied air-flow mal-distribution, and Nakayama et al. [4] and Li et al. [5] studied refrigerant flow distribution in distributors. Particularly Payne and Domanski [1] and Kim et al. [3] demonstrated that individual superheat control of each circuit could recover most of the cooling capacity and COP.

Typically A-coils are employed in RAC systems as the indoor coil, i.e. as the evaporator. The coil forms an A-shape in order to minimize the size of the air-duct. A drawback is that the air-flow becomes non-perpendicular to the face coil, resulting in air-flow mal-distribution. Steady state models capable of handling user-defined air-flow mal-distribution exist in the literature such as Domanski [6], and Jiang et al. [7]. Recently Domanski and Yashar [8] used computational fluid dynamics (CFD) and particle image velocimetry (PIV) to obtain the air-velocity profile through an A-coil. Likewise AbdelAziz et al. [9] applied CFD to analyze the air-flow in an A-coil. Both showed that air-flow mal-distribution occur.

Domanski and Yashar [8] applies a novel optimization system called ISHED (intelligent system for heat exchanger design) to optimize refrigerant circuitry (tube connections) in order to compensate air-flow mal-distribution. They show that cooling capa-

city is increased by 4.2% compared to an interlaced type of circuitry.

This study focuses on understanding the effect of refrigerant mal-distribution in the evaporator and methods of compensation. The objective is to quantify the influence of the distributor and the air-flow on refrigerant mal-distribution, UA-value and COP. A system model is made capable of simulating refrigerant mal-distribution in the evaporator. The evaporator model is ideal, i.e. two straight tubes, in order to perform simple and basic investigation of refrigerant mal-distribution in the evaporator and its effect on system performance. The refrigerant mal-distribution is then induced to the model by varying the vapor quality distribution in the distributor, the bending of the distributor feeder tubes and the air-flow distribution across the tubes. Finally the hypotheses implied by Payne and Domanski [1] and Kim et al. [3] as mentioned above is investigated, that is "controlling individual superheats results in recovered cooling capacity and COP at air-flow mal-distribution".

MODEL DESCRIPTION

The main interest in this study is the refrigerant mal-distribution and its effect on system performance. Thus the model of the compressor and the expansion valve are static and not particularly detailed, i.e. the volumetric flow rate and the isentropic efficiency are constant. The model of the condenser is a bit more detailed and relies on the requirement to model refrigerant mass distribution between the condenser and evaporator in this project. The moving boundary model formulation as proposed by Zhang and Zhang [10] was chosen for this.

The evaporator model needs to capture the mass flow distribution through the tubes, thus pressure drop needs to be modeled. The simplest form of the distributed models is chosen for this, i.e. the homogeneous equilibrium model. In the following the evaporator model is explained in more detail. Information about the moving boundary model used as condenser is available in Zhang and Zhang [10], where the homogeneous void fraction is employed together with similar heat transfer correlations, to be comparable to the evaporator model (see Table 1).

The evaporator model

The main assumptions of the distributed evaporator model is:

- The refrigerant flow is one-dimensional.
- The refrigerant vapor and liquid are in thermodynamic equilibrium.
- The refrigerant flow is homogeneous.
- The refrigerant kinetic and potential energies are negligible.
- The heat transfer coefficient on the air-side is uniform on each segment.
- The air is dry, incompressible and does not accumulate mass or energy.
- The axial heat conduction of the wall is negligible.

Further assumptions with regards to the two-phase flow formulation are explained in the following. The inertia term in the momentum equation (the derivative term w.r.t. time) is important for modeling propagation of pressure fluctuations and other effects of very small time scales (see Richter [11]), which is not important in the current study and will be neglected. Also accelerational pressure drop is typically small compared to frictional pressure drop and will also be neglected (see Jiang [12]). The one-dimensional homogeneous two-phase flow formulation for horizontal tubes is then

$$\frac{dM}{dt} = \dot{m}_{in} - \dot{m}_{out} \quad (1)$$

$$\frac{dU}{dt} = (\dot{m}h)_{in} - (\dot{m}h)_{out} + \dot{Q} \quad (2)$$

$$(p_{in} - p_{out}) = \Delta p_f \quad (3)$$

where equation 1, 2 and 3 are the mass conservation, energy conservation and momentum equation, respectively. The model has been implemented as described by Jensen [13] and has been verified in steady state by Coil-Designer (Jiang et al. [7]). Additional information needs to be given such as heat transfer coefficient and friction coefficient. The model also includes the thermal mass of the tube wall and employs the effectiveness-NTU relations for each segment similarly to Jiang [12] for cross-flow heat exchangers. Table 1 gives an overview of the correlations used.

The evaporator also consists of a distributor and a manifold, that serve to split and join the refrigerant flow, respectively. The pressure drop across the

Table 1: Overview of correlations

Air-side	
Heat transfer coefficient	Wang et al. [14]
Fin efficiency	Schmidt [15] (Schmidt approximation)
Two-phase	
Heat transfer coefficient	Shah [16]
Friction coefficient	Müller-Steinhagen and Heck [17]
Single phase	
Heat transfer coefficient	Gnielinski [18]
Friction coefficient	Blasius [19]

distributor feeder tubes is modeled by the Müller-Steinhagen and Heck [17] correlation, where the inner diameter of the feeder tube is 3 mm and the length is 250 mm. Other pressure drops from a sudden enlargement or a tee branch etc. are not considered. Both energy and mass conservation is applied to the distributor and the manifold.

IMPLEMENTATION

The models are implemented in Dymola 7.1 [20]. Dymola solvers are able to integrate large-scale differential and algebraic equations (DAEs) efficiently. Dymola is based on the Modelica language and supports object-oriented programming, that is important for model reuse and extension. Equations can be written in a casual manner and supports event driven procedures. Dymola has been well tested within the field of air-conditioning and refrigeration (Eborn et al. [21], Richter [11]). Thermophysical properties are provided by the RefEqn package (Skovrup [22]). A given heat exchanger geometry within the applicable range of the correlations is chosen for both evaporator and condenser, where the considered fin type is louvered. The only difference is that the total length of the condenser is twice as long as the total length of the evaporator. Typically the geometric difference is around one and a half, however the air velocity is higher for the condenser, thus a geometric difference of two is reasonable when the air velocity is set equal. The main input parameters to the models are depicted in Table 2.

In the following three important parameters will be introduced to analyze the mal-distribution in the evaporator. These will be referred to as the distribution parameters. The first and the second are distribution parameters used for the distributor. The first is used

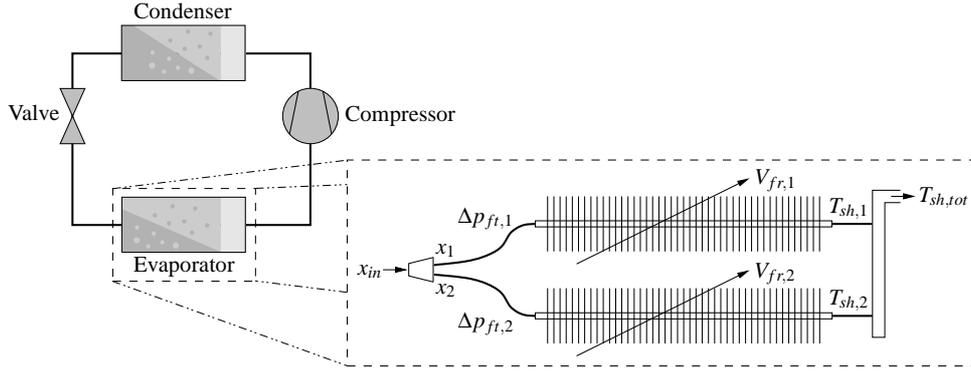


Figure 1: Sketch of model setup

Table 2: Main input parameters

Compressor	
Volumetric flow rate, \dot{V}	3.25 m ³ /h
Isentropic efficiency, η_{is}	0.8
Condenser	
Inlet air temperature, $T_{air,cond}$	310 K (36.85 °C)
Total tube length, L_{cond}	28 m
Evaporator	
Inlet air temperature, $T_{air,evap}$	300 K (26.85 °C)
Tube length, L_{evap}	7 m (per tube)
Number of control volumes, n	30 (per tube)
Evaporator and condenser	
Mean air frontal velocity, V_m	1.5 m/s
Tube inner diameter, d	7.6 mm
Tube outer diameter, D	9.6 mm

for distribution of liquid and vapor phases and is defined as $F_x = x_2/x_{in}$. Figure 1 shows a sketch of the model setup including these symbols. When F_x is equal to one, the vapor quality into the feeder tubes is equal, when F_x is equal to zero, only liquid comes into feeder tube 2.

The second parameter is used to induce different pressure drops in the feeder tubes, that could be caused by different bending. This distribution factor is defined, so that $\Delta p_{ft,1} = F_{ft}\Delta p_{ft,1,corr}$ and $\Delta p_{ft,2} = \Delta p_{ft,2,corr}$, which means that the factor is multiplied to the pressure drop correlation for tube 1 only. When F_{ft} is equal to one, the pressure drop across the feeder tubes are equal. When F_{ft} is above 1, the pressure drop becomes higher across feeder tube 1.

The third distribution parameter is a similar parameter used for the air-flow distribution and is defined as $F_{air} = V_{fr,1}/V_m$. When F_{air} is equal to one, the air-flow is distributed equally across the 2 tubes, when

F_{air} is equal to zero, air only travels across tube 2.

This leads to 3 cases that are simulated in this study, where each distribution parameter is varied individually. F_x and F_{air} are varied from 1 to 0.1 and F_{ft} is varied from 1 to 5.5. The total superheat $T_{sh,tot}$ is controlled to 5 K throughout the simulations by the mass flow rate through the expansion device. A fourth case is also investigated, that is, control of individual superheats to 5 K by controlling individual mass flow rates at air-flow mal-distribution. This requires that the pressure drop across each tube is no longer the same, in order to control individual mass flow rate. This is so to say done by an intelligent distributor.

For initialization a linear enthalpy and a linear pressure drop along the tubes are used for the evaporator. These correspond to an inlet quality of 0.2 into the distributor, a pressure of 10 bar into the distributor and individual tube superheats of 5 K, i.e. no mal-distribution. The initial condenser pressure was set to 25 bar, together with a superheated length of 15%, two-phase length of 75% and a subcooled length of 10%. Fast transients happen at the beginning of each simulation, however at no mal-distribution (i.e. $F_x = F_{ft} = F_{air} = 1$), from which the simulations starts, the steady state becomes the same for all cases, and is thus comparable. After initialization the steady state at no mal-distribution gives a pressure in the distributor at 10.95 bar (10.4 °C), inlet quality at 0.26, manifold pressure at 10.43 bar (8.8 °C), condenser pressure at 29.08 bar (48.2 °C) and a subcooling at 5.7 K. It is important to note that this steady state is determined by the initialization, which again determines the mass of refrigerant in the system. Another initialization would give another steady state and hence mass of refrigerant.

RESULTS

Mal-distribution from the distributor

In this section the results of the aforementioned case 1 and 2 will be presented, i.e. mal-distribution of liquid and vapor phases in the distributor (F_x , case 1) and mal-distribution caused by different feeder tube bending (F_{ft} , case 2). Figure 2 depicts the mass flux distribution through each tube as F_x goes to zero (case 1) and F_{ft} goes to six (case 2).

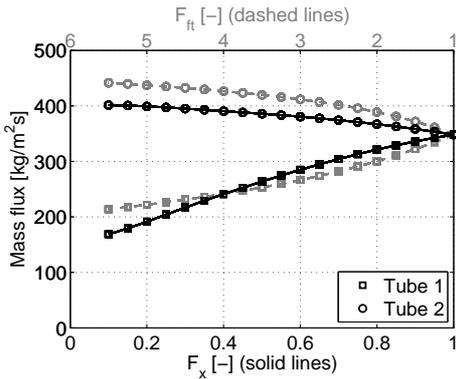


Figure 2: Mass fluxes vs. F_x and F_{ft} (case 1-2)

It shows that mass flux distribution is dependent on the quality factor F_x , so that more mass comes through the tube with lower inlet quality (tube 2) and less mass comes through the tube with higher inlet quality (tube 1). When F_{ft} goes to six a similar mass distribution trend is seen. This is determined by the pressure drop across the tubes. If the inlet vapor quality becomes lower the pressure drop will become lower, because of the difference in liquid and vapor pressure drop, which is higher for the vapor phase. In order to ensure equal pressure drop through the tubes, the mass flux becomes higher for the tube with lower inlet vapor quality. Similarly when F_{ft} goes to six the pressure drop across tube 1 becomes higher, and more refrigerant will travel through tube 2 to maintain equal pressure drop.

The consequence of different mass flux distribution is seen on figure 3, which shows the individual and total superheats. At $F_x = 0.8$ liquid is coming out of tube 1 (case 1) and at $F_{ft} = 2.75$ liquid is coming out of tube 1 (case 2). These points are important because the two-phase zone decreases, when full evaporation is not reached. A higher superheated zone in tube 1 is required in order to evaporate this surplus liquid, thus the UA-value is decreased. The UA-value also decreases when the total mass flux

decreases. The total mass flux decreases in both cases in order to maintain the total superheat of 5 K. The UA-value degradation is showed on figure 4, together with the COP of the system. It shows that mal-distribution of inlet vapor quality is more significant than different feeder tube pressure drop. Note that a compact fin-and-tube heat-exchanger usually consists of more passes and thus U-bends. This would reduce the influence of the distributor pressure drop even more compared to the inlet vapor quality.

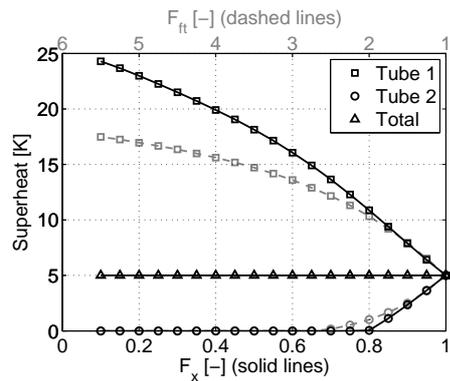


Figure 3: Superheats vs. F_x and F_{ft} (case 1-2)

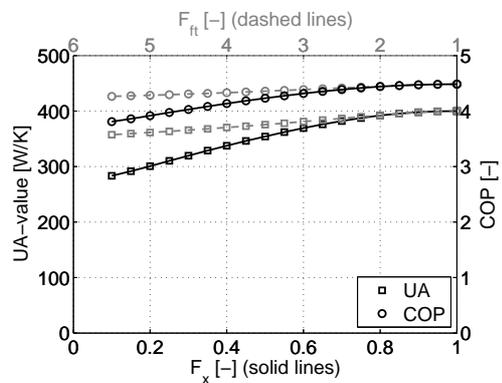


Figure 4: Evaporator UA-value and COP vs. F_x and F_{ft} (case 1-2)

The UA-value decreases 29% and 11% as F_x goes to 0.1 and F_{ft} goes to 6, where the COP decreases 15% and 5% as F_x goes to 0.1 and F_{ft} goes to 6. Because the UA-value decreases, the evaporating temperature decreases. This is seen on figure 5 and 6.

The results show that the different feeder tube bending has a minor impact on UA-value and COP, whereas the distribution of liquid and vapor phases has a higher impact. The two are not considered by

the authors to interact significantly, i.e. the distribution of liquid and vapor phases is a separation phenomena in the distributor, and thus not affected by different feeder tube pressure drop.

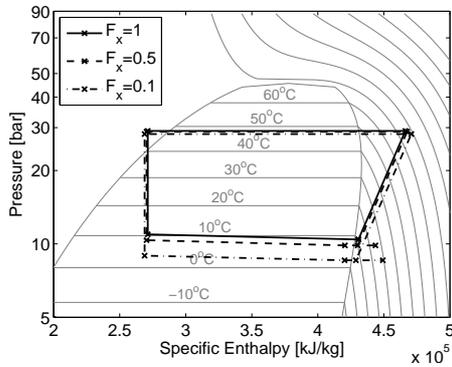


Figure 5: log p - h diagram at different F_x (case 1)

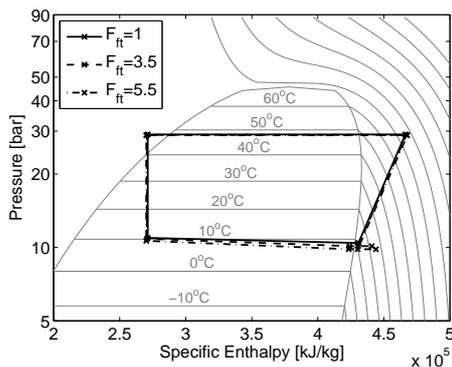


Figure 6: log p - h diagram at different F_{ft} (case 2)

Mal-distribution from the air-flow and compensation

In this section the results of the aforementioned case 3 and 4 will be presented, i.e. air-flow mal-distribution (F_{air} , case 3) and compensation of air-flow mal-distribution by controlling individual superheats (F_{air} , case 4). Figure 7 shows the mass flux distribution.

Interestingly the mass fluxes are almost equal for case 3 with no individual control of the superheat, however reduced significantly. The reduction of mass flux together with different superheated zones have a degrading effect on the UA-value, as depicted on figure 8, however it can be compensated considerably if individual superheats are controlled as in

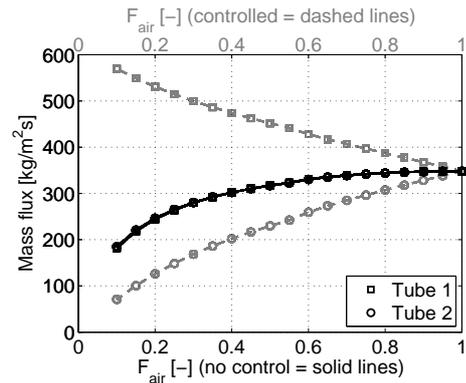


Figure 7: Mass fluxes vs. F_{air} (case 3 and 4)

case 4. For this case the refrigerant mass flux distribution is optimized to compensate the air-flow mal-distribution.

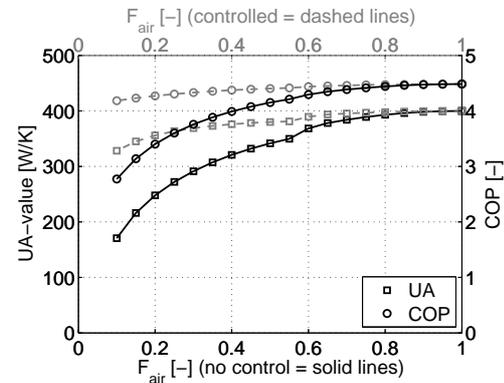


Figure 8: Evaporator UA-value and COP vs. F_{air} (case 3 and 4)

The UA-value and the COP decrease 57% and 38% as F_{air} goes to 0.1 for case 3, where the UA-value and the COP decrease 18% and 7% as F_{air} goes to 0.1 for case 4. Again the UA-value reduction decreases the evaporating temperature. This reduction causes a degradation of the COP. The log p - h diagrams of case 3 and 4 are depicted on figure 9 and 10.

DISCUSSION

The results of this paper should be used as guidelines to what occurs in evaporators, where mal-distribution is present, whenever it comes from the distributor or the air-flow. It is difficult to extend the results of this work to a given type of evaporator or system, in order to estimate the degradation of the UA-value and COP. The problem is that it is dif-

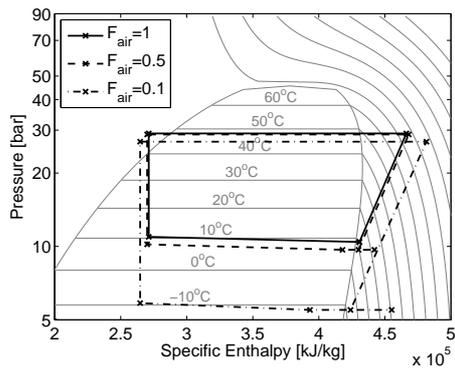


Figure 9: log p - h diagram at different F_{air} (case 3)

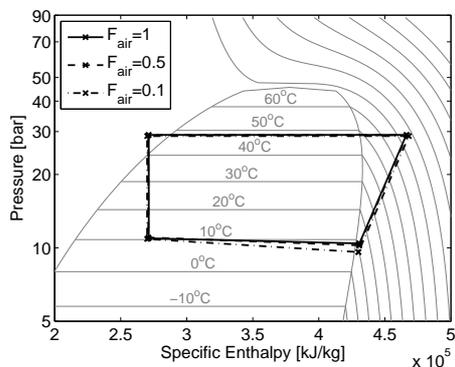


Figure 10: log p - h diagram at different F_{air} (case 4)

difficult to quantify and distinguish between the mal-distribution of the different sources, i.e. vapor quality distribution, feeder tube pressure drop difference (in bends) or air-flow mal-distribution. These can be different for different types of heat exchangers.

The trends of the results in this paper are similar to studies of Brix [23] and Kærn and Elmegaard [24], where the former showed these trends on a microchannel evaporator. However the inlet pressure was fixed for these analysis, which only focused on the evaporator as a separate component. Thus the system level influence was not addressed. The latter did not take into account the feeder tube pressure drop in fin-and-tube evaporators, hence addressed in the present study. The feeder tube pressure drop was in the present study about half of the total pressure drop through the evaporator. Thus the inclusion of the feeder tube pressure drop has significant effect when it comes to mal-distribution from the distributor. Because the mass flux is distributed evenly at mal-distribution from the air-flow, the feeder tube pressure drop has no particular effect here.

Kim et al. [3] also performed system level analysis of mal-distribution in evaporators, and showed that the cooling capacity and COP could be recovered to 99.9% at $F_{air} = 0.75$, by controlling individual superheat. The result of this study shows a UA-value recovery of 99.2% and a COP recovery of 99.6% at $F_{air} = 0.75$. The trends are also similar, thus the model appears verified with earlier work.

CONCLUSION

It can be concluded that mal-distribution in fin-and-tube evaporators reduces the UA-value and COP, whenever it comes from a mal-functioning distributor or a non-uniform air-flow. It turns out that the non-uniform air-flow significantly reduces the UA-value and COP, whereas the mal-functioning distributor has a smaller impact.

Four cases were studied, i.e. case 1 at mal-distribution of liquid and vapor phases, case 2 at mal-distribution by different feeder tube bending, case 3 at air-flow mal-distribution and case 4 at air-flow mal-distribution with compensation by individual superheat control. Case 1, 2 and 3 showed a degradation of up to 15%, 5% and 38% in COP, respectively, as F_x went to 0.1, F_{ft} went to 6 and F_{air} went to 0.1. Case 4 showed that the degradation of up to 38% could be compensated to only 7% as F_{air} went to 0.1.

With regards to the distributor, it is important to secure a good distribution of liquid and vapor phases, whereas the different feeder tube bending were found to have minor degradation effect.

The overall picture shows that the UA-value and COP reduction becomes significant when full evaporation is not reached in one of the tubes. Also air-flow mal-distribution can be compensated by controlling individual superheats.

This investigation gave rise to another important issue, that is, what is the benefits of optimizing refrigerant circuitry, if refrigerant distribution is controllable and how could the two interact?

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