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ABSTRACT

Finned tube evaporators are used in a wide range of application such as commercial and industrial cold/freezed storage rooms with high traffic loading under frosting conditions. During the literature survey it has been realized that there is not enough experimental and visual data regarding the unit cooler tested with casing and fan integrated as in real operating conditions. As it is known, the usage of fan and evaporator external case and non-optimum distributor design may affect air and refrigerant side distribution, capacity and frost formation along the fins with respect to degree of maldistribution. In this paper, an evaporator with an integrated fan was manufactured and tested at under frosting conditions by only changing air flow rate at ambient balanced type test laboratory in contrast with testing at wind tunnel and more uniform flow distribution. During the test, operation performed according to three different air flow rate separately. The parameters concerning test operation such as the changes of air temperature, air relative humidity, surface temperature, air-side pressure drop and refrigerant side capacity etc. were followed in detail for each air flow rate. At the same time, the digital and thermal images were captured in front of heat exchanger; thus, the frost temperature distributions, blockage ratios and frost thicknesses occurring throughout all surfaces were investigated. In this study the effect of air flow rate on frosting has been investigated. The test and visual results showed that the trendline of air side pressure drop has increased slowly at the first stage of test operations, then this phenomena has increased linearly up to a top point and then the linearity has disrupted instantly and this point has admitted beginning of defrost operation for each case. On the other hand, the refrigerant capacities were declined continuously during the test operations. When examining visual and thermal images captured at certain time intervals, it was shown an uneven frost layers and detected the temperature distributions in front of evaporator as different degree for each case via thermal and digital camera. Similarly, the frost thicknesses distribution with time occurring at the specific area of each circuit was identified via Matlab Image Process Program. Moreover, the initial period of frost growth, which have significant impact on further frost growth, was investigated via a microscope camera.

1. INTRODUCTION

Frost formation and its growth are undesirable but unavoidable phenomena in many applications such as cold storage unit coolers and heat pumps. This phenomenon leads to a decrease of heat transfer rate caused by the combined effect of thermal resistance of frost and increment of air pressure drop. In order to protect the foods against temperature and humidity fluctuations in the room, periodic defrost process will be carried out at suitable time intervals for the sake of a balanced energy efficiency and food protection. However, the parameters having significant impact on frost growth should be clarified exactly to detect optimum time of defrost period.

As could be followed in the literature, it has been realized that the effects of air flow rate on frost formation and growth is still unclear and there are some conflicts among the researches; besides, there are lack of experimental studies considering real fan effect. The previous studies generally dealt with investigations under constant air flow conditions.

Some author such as Chen *et al*. (2003), Aljuwayhel *et al*. (2008), Schmidt and Kristensen (2014), Da Silva (2012), Huang *et al*. (2008) and Groll *et al*. (2011) investigated the frost growth on tube-finned evaporators under variable air flow conditions. As these studies have highlighted the change of heat transfer rate and pressure drop depending on frost formation, some of them have examined the effect of fan types on frost growth.

According to the study considering the effect of air flow rate on frosting, Yan *et al*. (2003), Cui *et al*. (2011), Padhmanabhan *et al*. (2011) claimed that the amount of frost formation has increased as air flow rate decreased. This is because the surface of the heat exchanger becomes colder for a lower flow rate due to a lower heat transfer rate. Opposite to this situation, Haijie *et al*. (2014) has stated that the rising of the mass transfer coefficient and the wall temperature at the same time, the frost accumulation rate increases. This situation is consistent with Da Silva (2012) and Tashiro *et al*. (2014).

The different views on frost formation rate depending on air flow rate are caused by two phenomena, which are mass transfer coefficient and absolute humidity difference (driving force). According to Ye *et al*. (2014), situation of frost rate depending on air flow rate could be determined by dominant factor between these two phenomena.

Within the scope of this study, the frost formation phenomenon on unit cooler considering the effect of air flow rate as similar to real refrigeration systems had been investigated. During the test process carried out for different fan frequency, thereby for different air velocity, all the parameters belonging to tests have followed and compared within themselves. The frost thicknesses, blockage ratios and thermal images have been detected as well. Consequently, the effect of air flow rate on the unit cooler under real operation condition has been experimentally determined.

2. TEST METHODOLOGY

2.1 Description of Heat Exchanger

Table 1 shows the dimensional characteristics of unit cooler used in this study and pictures has been illustrated in Figure 1. As stated in Table 1, fin type was selected as plain for the sake of better visualization and easy measurement of frost formation on fin surface. The evaporator is equipped with a 500 mm fan with a maximum flow rate of 9500 m3/h.

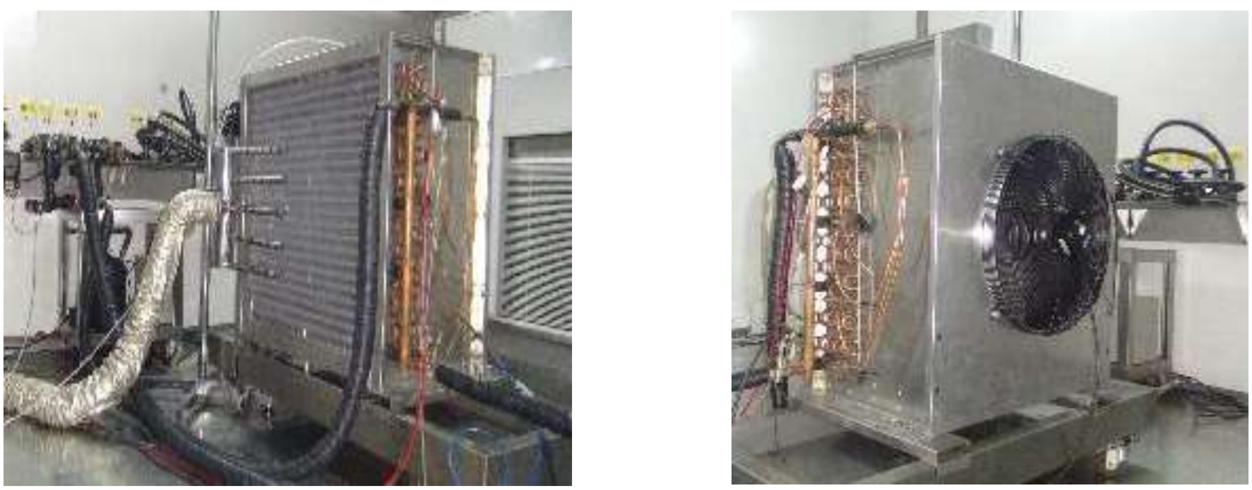


Figure 1: The general view of unit cooler inside the calorimetric room

Table 1: Geometric parameters of unit cooler

Geometric parameters	Values
Number of rows	4
Number of tubes per row	22
Transversal tube pitch	35 mm
Longitudinal tube pitch	35 mm
Tube length	745 mm
Tube diameter (inner/outer)	11.86 mm/ 12.5 mm
Fin thickness	0.15 mm
Fin spacing	7 mm
Fin type	Flat
Fin height	770 mm
Tube arrangement	inline

2.2 Test Procedures

The test set-up is illustrated in Figure 2. The right side of this figure demonstrates the calorimetric room (ambient balanced room), where the test operations were performed. The calorimetric room setup consists of a test section, where fin and tube heat exchanger was installed; an air handling unit in order to maintain a constant air temperature and humidity; a refrigeration section to regulate the temperature and flow rate of the refrigerant fed to the test unit during the experiments.

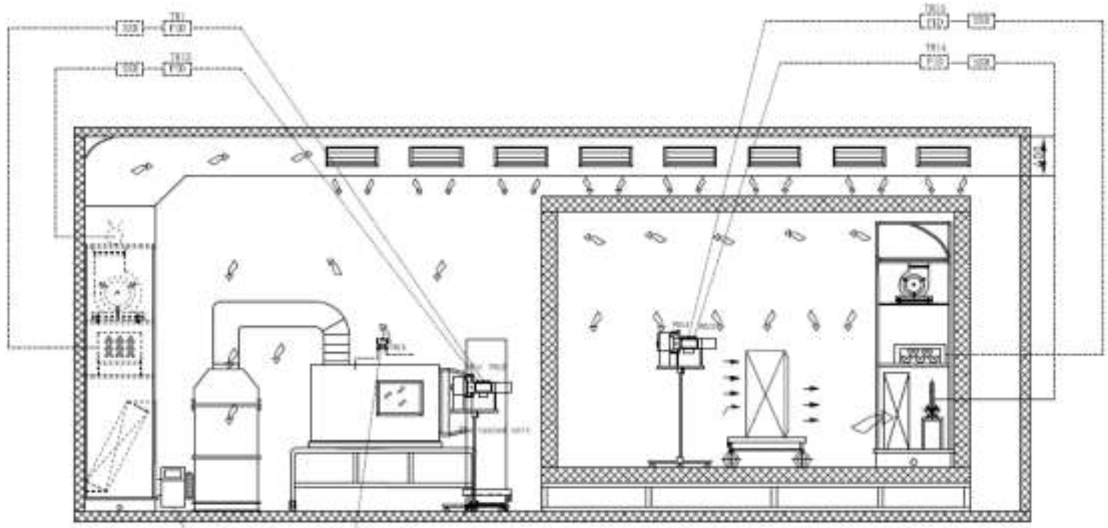


Figure 2: Schematic representation of the experimental facility

The refrigerant used in these experiments was R404A and refrigerant system has an auxiliary line, which is operated with water for the purpose of controlling the refrigerant temperature at some part of installation such as condenser sub cooling temperature. Besides, this refrigerant line has shell and tube condenser used for system's condenser and auxiliary evaporator in order to adjust evaporation pressure.

The type of expansion valve is very important for frosting test because this apparatus affects the characteristic of frosting test significantly. An electronic expansion device (EXV) had been used in this study. With an EXV expansion device, a better control of the superheat with regard to a TXV is possible and therefore a better usage of the evaporator surface area is possible for steady-state and frost-up tests. (Groll *et al.*, 2011)

During the test operation, as the air inlet temperature has been controlled automatically, the relative humidity has been controlled as manual because the automatic control of relative humidity results in undesirable fluctuations in wide range. Eventually, it has been ensured very stable room condition. On the other hand, the frosting images has been taken a digital camera mounted on a manual traverse mechanism, which is able to move from bottom to up manually.

2.3 Experimental Conditions

Table 2 shows the experimental condition applied in this study. The only change among conditions is operating frequency which enables to perform the test process at different air velocities. The different fan air velocities have been adjusted via a frequency inverter mounted to axial fan. Besides, the superheat temperatures followed during the test operations have been about 0.5 °C because it has been desired uniform tube surface temperature in order to evaluate the frost thickness and blockage ratio at windward of unit cooler more accurately.

Table 2: Experimental conditions

Test	**Air Velocities [m/s]**	**T_a [°C]**	**RH [%]**	**T_s [°C]**
1	2.52	3	77,5	-9
2	3.10	3	77,5	-9
3	3.37	3	77,5	-9

2.4 Experimental Uncertainty

The uncertainty of related parameters was calculated according to Stephanie Bell's study 'A Beginner Guide to Uncertainty of Measurement' (Bell, 1999). The results are tabulated at Table 3. In order to obtain a good measurement of the frost thickness, it is important that the camera positioned normal to the plane of the coil. Therefore, the camera having 4224×2376 image resolution is mounted on a 2-axis traverse system. However, the uncertainty of frost thickness measurements are caused by two reasons. One of them is the error from reading the pixel values, the other is the error of reference meter measurement. As the maximum error of reading is about 1 pixel, thereby, corresponding frost thickness is about 0.06 mm, maximum error of reference meter is 0.2 mm. Thus, the overall uncertainty for fin and tube frost thicknesses is about 0.208 mm.

Table 3: Estimated uncertainty values belonging to related parameters

Parameters	**Maximum Uncertainty**
Heat transfer rate [kW]	2.35%
Air side static pressure difference [Pa]	$\mp$ 5 Pa
Frost thickness [mm]	$\mp$0,208 mm

3. RESULTS AND DISCUSSION

The test process has been repeated twice for each test, thus the experimental outcomes has been obtained by taking the average except the air flow rates and the most important parameters during test operation has been illustrated at Figure 3. As the first 15 minutes was admitted as stabilization period in order to reach desired conditions after this period, desired condition for each test has been ensured almost constant throughout test operations. Moreover, the superheat temperatures during test has been ensured as quite stable because of the excellent control of EXV and superior design of refrigerant distributor.

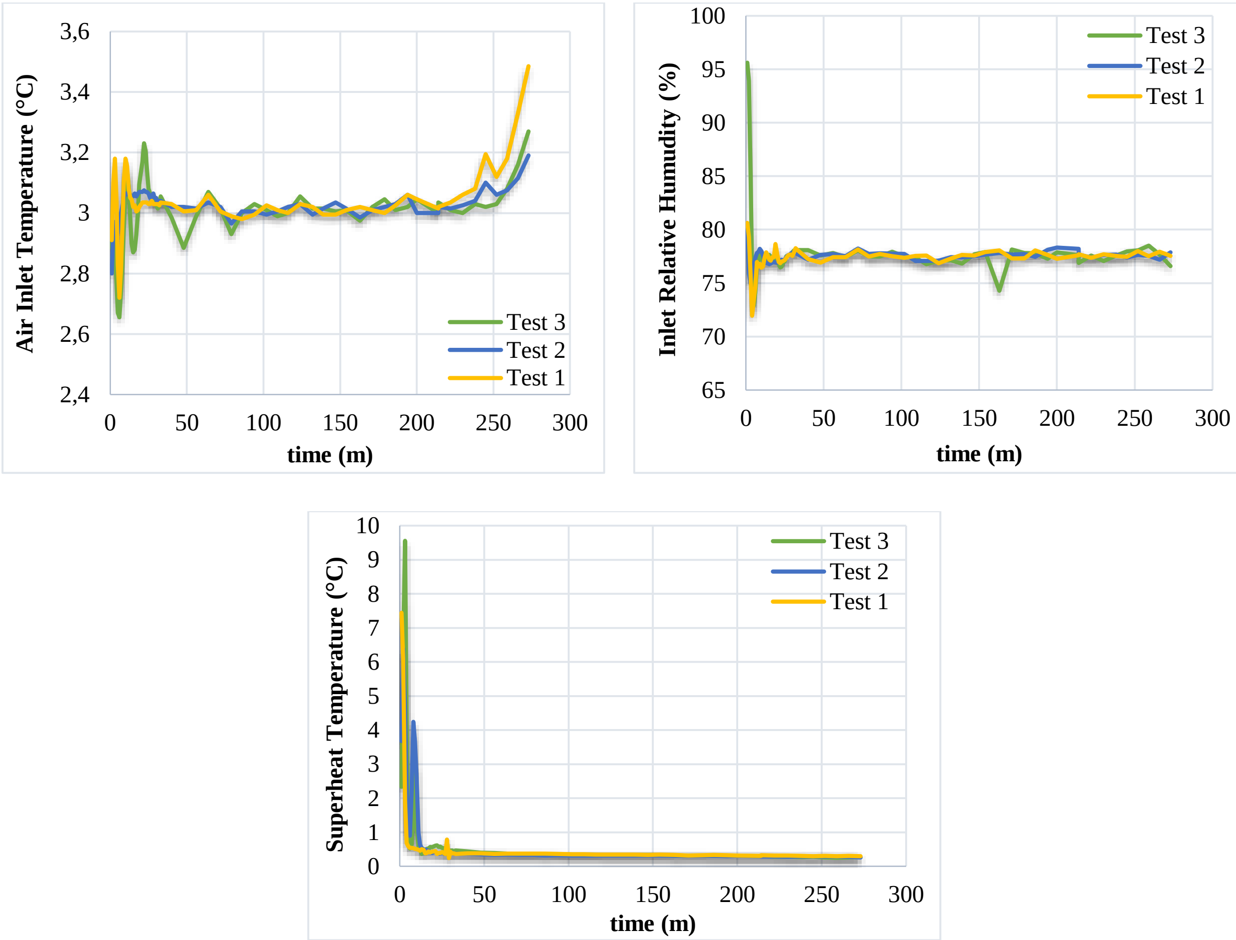


Figure 3: The changes of air inlet temperature, inlet relative humidity and superheat temperature with time

However, although the control of relative humidity at windward of unit cooler is rather difficult, the average value is almost same as shown in Table 4. The average values of air inlet temperature, superheat temperature and evaporation pressure could also be followed in the same table.

Table 4: The average values belonging to test operations

Test	T_{ai} [°C]	RH_{ai} [%]	T_{sh} [°C]	P_{evap} [MPa]
1	3.04	77.53	0.37	0.446
2	3.03	77.50	0.37	0.446
3	3.00	77.44	0.40	0.446

3.1 Frost Thickness and Blockage Ratios

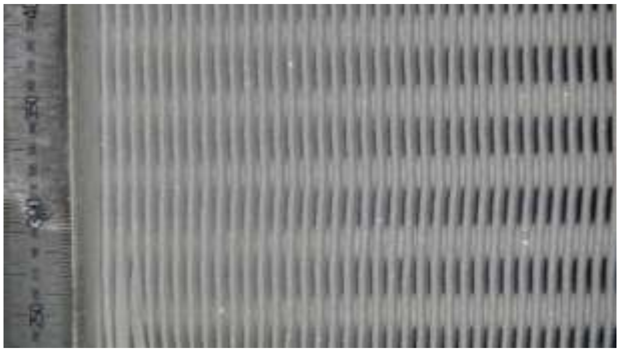
Frost thickness development during the test operation is rather important phenomenon, because it has vital importance on system performance due to the fact that the growing frost results in both the decreasing of air flow rate and increasing thermal insulation effect. However, Janssen (2013) has stated that another reason for the significance of frost thickness is its direct involvement within the differential equations governing frosting and defrosting, so that researchers need accurate measurement of frost characteristics.

In the literature, various methods regarding the frost measurements had been described; visual methods executed with a camera and image process toolbox or physical measurement, for instance micrometers. In this study, the frost thickness measurements have been performed via visual method because the physical measurements for finned tube heat exchanger are quite difficult and have less accuracy when compared with visual methods.

Accordingly, first, the images belonging to specific area of each circuits have been captured by a digital camera, of 20.4 Megapixel image quality, Then, the frost thicknesses on both fin and tube have been measured for related area of each circuits via Matlab Image Processing Toolbox. Eventually, the average frost thickness formed at windward of unit cooler could be obtained at a certain instance when the fan shows instable effects, which is going to be mentioned at next section in detail. As it can be shown in Image 1, in order to find the pixel count at per unit length, the imaging equipment is calibrated by means of a reference meter, placed near side of unit cooler. For the fin, the non-frost thickness between two fins is obtained by taking this reference length, thereby, the frost thickness for fin is calculated as following equation. On the other hand, the tube thickness is directly obtained according to reference length.

$$\delta_f = \frac{(FP - L_g) - FT}{2} \tag{1}$$

$$\delta_t = \frac{\delta_t - D_o}{2} \tag{2}$$



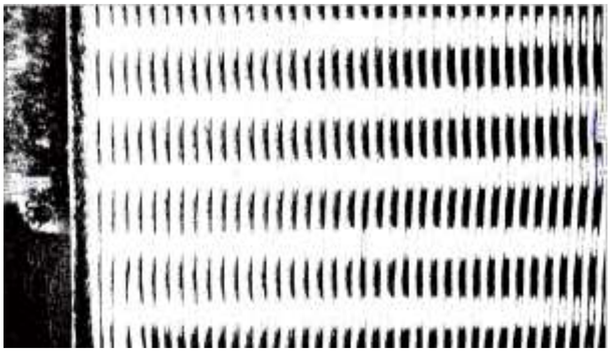


Image 1. The illustration of the digital and post processed images

At the end of the image analysis, average fin and tube thickness for each test operation had been obtained separately. As shown in Figure 4, the fin frost thicknesses for the test operations corresponding to number of 1, 2 and 3 are 1.4477 mm, 1.5062 mm and 1.4506 mm and the tube frost thicknesses are 1.4581 mm, 1.4495 mm and 1.4729 mm, respectively.

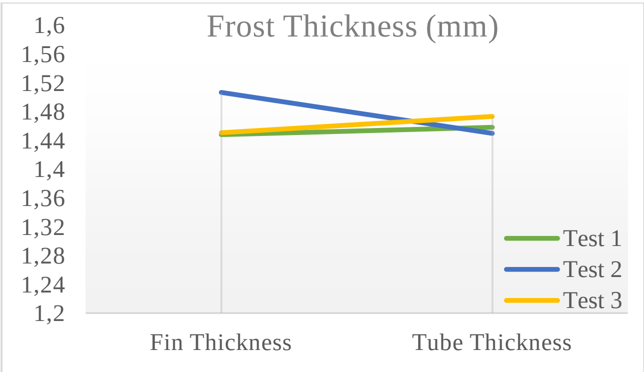


Figure 4: The thicknesses forming at fin and tube for each test operation

Another phenomeno directly related with fin and tube thickness is blockage ratio, which occures on the windvard of unit cooler, which is evaluated in a lot of studies. According to Equations (3), (4) and (5), the blockage ratios of frost for each test operation have been calculated as about 50%, which was measured at the beginning of the fan stall, as illustrated in Figure 5 and this result is consistent with study performed by Huang (2008). Related author has also stated that the defrost period should start when the half of flow area is blocked by frost.

$$\text{space area}_{\text{non-frost}} = (X_t - D_o) \text{ x } (\text{FP} - \text{FT}) \quad (3)$$

$$\text{space area}_{\text{after frost}} = (X_t - D_o - 2\delta_t) \text{ x } (\text{FP} - \text{FT} - 2\delta_f) \quad (4)$$

$$\text{BR} = 1 - \frac{\text{space area}_{\text{after frost}}}{\text{space area}_{\text{non-frost}}} \quad (5)$$

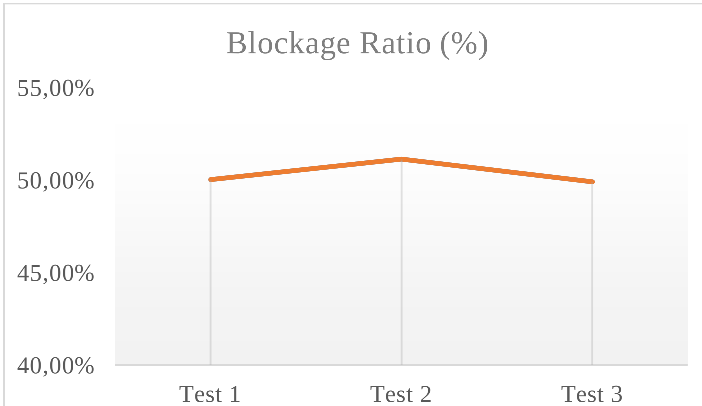


Figure 5: The blockage ratios for each test operation

Another data was taken visually via a microscope in order to observe the shape of the first formation of frost at starting time of test operation because some researchers have stated that initial frost formation has significant impact on further growth of frost (Janssen, 2013). Unfortunately, because of the fact that the frost formation has occurred in the stabilization period, which is about 15 minutes, these data couldn't be obtained legibly.

Also, when looked at the thermal images taken during test operation for entire tests, it has been observed quite uniform distribution throughout the unit cooler surface for the test operation carried out at the highest air velocity. Image 2 shows that the improvement of specific area as the test time progress. This situation results from improvement of operation of refrigerant distributor as the mass flow rate decreases. Furthermore, Bayrak and Konukman (2016) and Jianying *et al.* (2008) have pointed out that the air side maldistribution by about 21% doesn't have significant affect the refrigerant internal heat transfer characteristic according to these images.

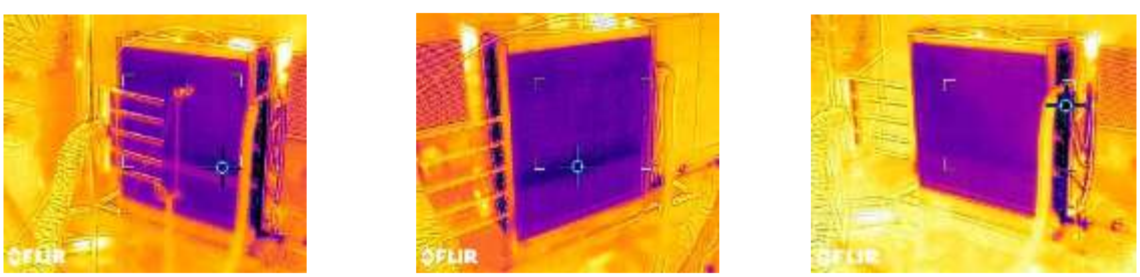


Image 2: The improvement of 3th circuit with time

3.2 The Change of Air-Side Pressure Drop and Heat Transfer Rate

Figure 6 demonstrates the change of air pressure drop and heat transfer rate during the test operation. According to this figure, as the pressure drop for the first 40 minutes increases slowly, the same phenomenon increases steeply after that point, which means that the frost grow rate has increased this time. This situation could be interpreted as that the amount of frost formation increases as the air flow rate decreases because the surface of unit cooler becomes colder and decrease of surface temperature is more dominant when compared with the effect of decrease of mass transfer coefficient.

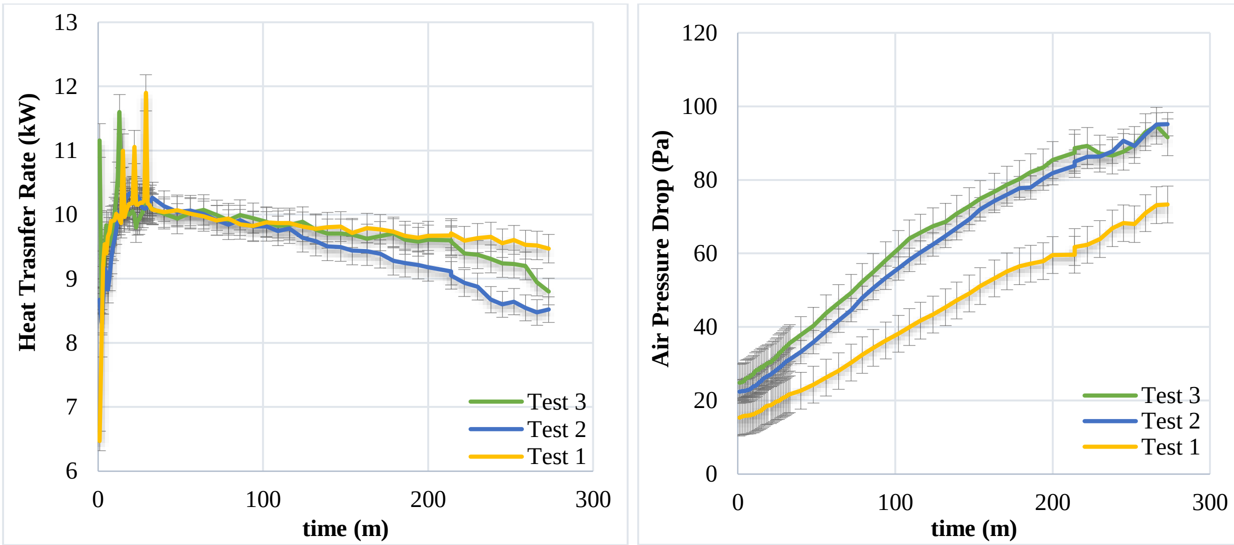
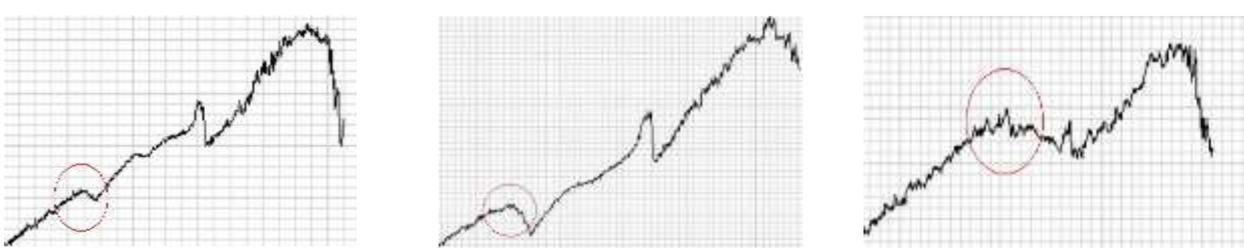


Figure 6: The change of air side pressure and heat transfer rate with time for each test operation

Furthermore, as the air pressure drop curve is lower for test 1 as expected, the curves of test 2 and test 3 are very close to each other. However, the augmentation of pressure drop at test 2 is higher than other values. If it is examined in detail considering the heat transfer curves, it can be seen that the heat transfer rate curve belonging to test 2 degrades faster. It may be based on the more effective frost growth at this velocity value than others and this situation could be attributed the value of critical air velocity for evaporators as highlighted by Ye *et al.* (2014). In other words, the tendency of the mass transfer rate of frost to increase or decrease depending on air velocity was identified by the dominant factor regarding the change in the mass transfer coefficient and absolute humidity difference (driving force).

According to same figure, the heat transfer rate decreases continuously. Within the first 110 minutes no significant effect of air flow rate on heat transfer rate has been observed, after that point, the curve belonging to test 2 decreases more depending on aforementioned reasons, test 1 and test 3 are about similar throughout the test operation. Therefore, the running at 2.52 m/s is the most suitable selection in scope of this study.

On the other hand, axial fans have a limit of pressure drop, if it is exceeded, it could be some detrimental impacts such as increasing of noise and vibration, structural fatigue and damaging of ductwork and other components. Figure 8 shows the stall condition for each test operation. The starting point of stall specified with red circle can be speculated the time of starting of defrost process at frosting tests even if the critical test parameters are stable. Therefore, the blockage ratios have been evaluated by taking reference to this point. The interesting point is that this time is about 50% blockage ratio as mentioned in the above section.



a) b) c)

Figure 7: Fan stall situations a) Test 1 b) Test 2 c) Test 3

4. CONCLUSIONS

In this experimental study, the effect of air flow rate on unit cooler under frosting condition has been observed by using visual methods and image processing. As a conclusion;

- According to experiments, the air velocity has no significant effect on heat transfer rate for a certain period of time, but it has been observed that it gains importance at the further times because of the effect of air velocity on frost growth. Also, it could be an air velocity need to be avoided for unit coolers running under frost condition. For this study, as the fan air velocity specified as 3.10 m/s should be avoided, the air velocity specified as 2.52 m/s could be superior point because the product running at this value has minimum loss of capacity.

- Considering the fan stall issue, when the limit of pressure drop permitted is exceeded it could have detrimental impacts on fan. Therefore, the starting point of stall could be optimum point for running of defrost process because half channel area is blocked by frost as mentioned some studies.

- The effect air velocity on initial frost growth hasn't been detected due to the fact the frost formation has occurred at stabilization period of the test operation.

- Further studies have to investigate the effect of refrost condition, which has detrimental impacts on change of heat transfer rate of unit cooler.

NOMENCLATURE

BR	blockage ratio	(%)	**Subscript**	
D	diameter	(mm)	a	air
EXV	electonic expansion valve	(-)	ai	air inlet
FP	fin pitch	(mm)	evap	evaporation
FT	frost thickness	(mm)	f	fin
L	length	(mm)	g	gap
RH	relative humudity	(%)	m	minute
T	temperature	(°C)	o	outlet
TXV	thermostatic expansion valve	(-)	s	surface
X	transversal pitch	(mm)	sh	superheat
δ	frost thickness	(mm)	t	tube

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