2002

Real Time Upright Freezer Evaporator Performance Under Frosted Conditions

T. N. Aynur
Arcelik A.S.

C. Inan
Arcelik A.S.

H. Karatas
Arcelik A.S.

N. Egrican
Istanbul Technical University

C. Lale
Arcelik A.S.

R16-4

REAL TIME UPRIGHT FREEZER EVAPORATOR PERFORMANCE UNDER FROSTED CONDITIONS

*Tolga Nurettin Aynur, Graduate Student in ARCELIK A.S., R&TD Center,
Istanbul Technical University, Mech. Eng. Faculty / Istanbul / TURKEY, Tel: 90 216 359 31 44
E-Mail: tolga.aynur@arcelik.com , tetolga@hotmail.com *Author for Correspondence

Dr. Cemil Inan, ARCELIK A.S., R&TD Center, 81719, Tuzla, Istanbul, TURKEY
E-Mail: cemil.inan@arcelik.com

Hakan Karatas, ARCELIK A.S., R&TD Center, 81719, Tuzla, Istanbul, TURKEY
E-Mail: hakan.karatas@arcelik.com

Dr. Nilufer Egrican, Istanbul Technical University, Mech. Eng. Faculty / Istanbul / TURKEY
E-Mail: egrican@mkn.itu.edu.tr

Cetin Lale, ARCELIK A.S., R&TD Center, 81719, Tuzla, Istanbul, TURKEY
E-Mail: cetin.lale@arcelik.com

ABSTRACT

Frost formation on an upright freezer evaporator is investigated experimentally on real refrigerator geometry. It's effects on overall heat transfer value (UA), airside pressure drop and airflow rate is observed with respect to time. That is, the real time evaporator performance under frosted conditions is evaluated. Airflow rate is obtained with respect to time by using fan characteristic curve.

INTRODUCTION

When moist air comes across a surface, which has a surface temperature below dew point of air, the moisture in the air condenses on the surface. If the surface's temperature is below the freezing point of the water, the condensed moisture on the surface changes its phase from liquid to solid, namely from water to frost. This phenomenon so-called frost formation is mostly encountered in the field of refrigeration and air-conditioning and has a significant adverse effect upon the heat transfer and airside pressure drop. During the cooling cycle, frost accumulates on the evaporator of the system. The frost does not only insulate the heat transfer surface of the evaporator, but also becomes thick enough to restrict and blocks the airflow. Moreover, the compressor decreases the evaporation temperature to increase the heat transfer during frosting. However, this precaution increases the energy consumption of the system. Furthermore, frost layer on the evaporator has to be melted away periodically to prevent the above effects. However, during defrosting, energy has to be spent for heating the evaporator.

Due to these adverse effects, frost formation phenomenon has been investigated by several researchers. Aluminum heat exchanger fins under frosting conditions were investigated experimentally [1] by using a laser scanning system. It was found that frost thickness close to the leading edge and fin base was higher than the frost that accumulated on the other parts of the fin. In another study, frost growth around a cylinder was investigated numerically [2]. It was stated that frost thickness near the leading edge was thicker than the trailing edge. In addition to these studies, the whole no-frost evaporator geometry was investigated [3,4,5]. It was determined that the overall heat transfer value (UA) and evaporator airside pressure drop increased during frosting [3,4]. These conclusions matched with mathematical model result [4], which was based on constant airflow rate and homogenous frost accumulation on the evaporator. The other model [5] that was based on non-homogenous frost distribution, real time airflow rate and pass-by-pass evaporator frosting evaluation gave different trends. It was found that, UA value increased in the first hours and a decreasing trend followed this augment. Airside pressure drop trend gave reasonable results with the other studies.

Almost all of the experimental studies were performed in wind tunnels, however there are some differences between wind tunnels and real refrigerator geometries. For instance in refrigerators, airflow through the evaporator is not same in wind tunnels due to the refrigerator geometry and airflow rate follows a decreasing trend due to the frosting, however in wind tunnels constant airflow rate can be maintained through the evaporator. In addition to airside, refrigerant side in refrigerators cannot be adjusted easily as it can be in wind tunnels. Because of these facts, a real refrigerator is utilized in this study to obtain more reliable and realistic results.

EXPERIMENTAL SETUP AND PROCEDURE

Experimental Setup Descriptions

Refrigerator cabinet

In this study, an upright freezer charged with R134a is utilized in the experiments, however a few modifications shown in figure 1 have been made. For instance, the separation wall so-called mullion between the freezer and the fresh food cabinet is cut out, so that instead of the two cabinets, a relatively big one is obtained. Inner volume of this modified refrigerator is about 464 L.

Evaporator

An aluminum finned-tube evaporator is used in this refrigerator. The outside diameter of the refrigerant tube is 8 mm and the heat transfer surface area is 0.81 m^2. Despite modification of the refrigerator, the location of the evaporator in the refrigerator is not changed.

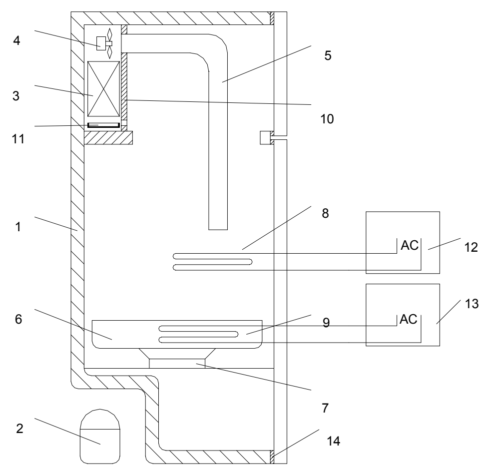

Figure 1. Experimental set up. (1-insulation wall, 2-compressor, 3-evaporator, 4-fan, 5-circular duct, 6-water tray, 7-scale, 8-air heater, 9-water heater, 10-evaporator front cover, 11-defrost water tray, 12-voltage modulator of air heater, 13-voltage modulator of water heater, 14-door gasket)

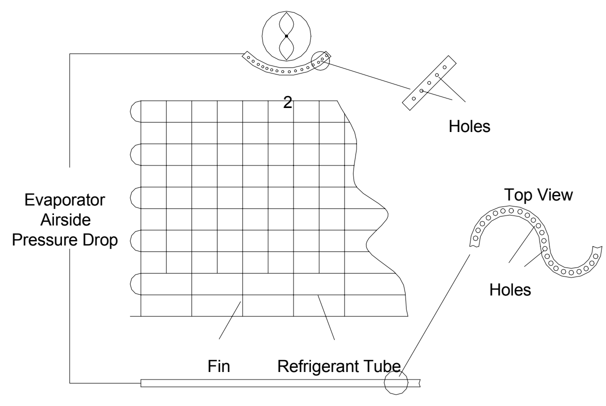

Figure 2. Locations of the evaporator airside pressure drop tabs

Evaporator Fan

An axial flow type fan is used above the evaporator as seen in figure 1. In figure 3, characteristic curve of the fan with its housing is given. This curve is obtained in a wind tunnel, which is based on AMCA 210-85 standards.

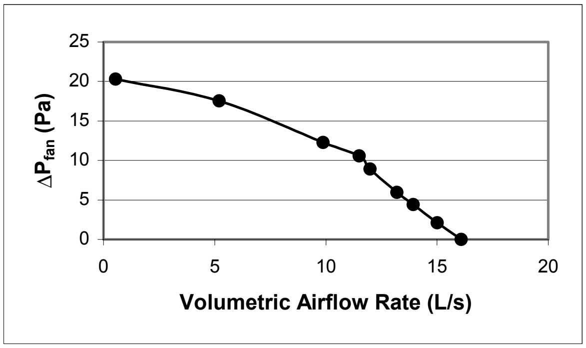

Figure 3. Characteristic curve of the fan with its housing

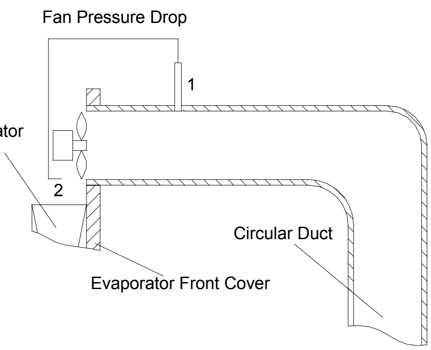

Figure 4. Locations of the fan pressure drop probes

Circular Duct

The duct system of the refrigerator itself no longer exists due to the modification. Because of this fact, a circular duct is mounted in front of the evaporator fan. With the help of the duct, the air dragged by the fan can be directed through the bottom of the refrigerator, so that more homogenous air temperature and air distribution can be obtained inside the refrigerator. The diameter of the duct is selected to be suitable for the fan housing.

Air Heater

A heater with a power of 160 W is used to adjust air temperature. It is supplied by a voltage modulator, which enables to regulate the heater power, and controlled by an on-off type controller.

Water Tray, Water Heater and Scale

During the experiments, cabinets' doors are kept closed. A water tray is placed in the refrigerator and located on a scale with an accuracy of ±0.05gr. A heater with a power of 120 W, supplied by a voltage modulator, is submerged in the water tray. With the help of this heater, which is controlled by an on-off type controller, the amount of evaporated water can be adjusted, so that different frosting conditions can be obtained and parametric studies can be performed. The insulation section of the refrigerator door that corresponds to the scale's display is replaced by a transparent stretch film, so that during the experiment, the amount of the evaporated water can be measured from outside. A line is sketched in the tray to check the water level, thus the level is always kept constant.

Measurement Equipment

Thermocouples and Relative Humidity Sensors

Refrigerator air temperatures from various locations, evaporator air inlet and outlet temperatures and temperature inside the duct is measured with the help of type T thermocouples. In addition, refrigerant temperatures are measured from evaporator surfaces by thermocouples. Besides, two relative humidity (R.H.) sensors are utilized to determine the inlet and outlet R.H. of the evaporator. A data logger with an accuracy of ±0.3°C is utilized for data logging.

Pressure Taps

Fan outlet pressure is measured by pneumatic tube at the circular duct as indicated (1) in figure 4. For fan inlet pressure, a hand made U shaped cupper tube is utilized as indicated (2) in figure 2. It is located under the fan. The probe's diameter is so small that it does not affect the airflow. A lot of petty holes were made on its surface by drilling, thus it measures the mean inlet pressure. The tube is also used to measure the evaporator outlet pressure as seen in figure 2. At the evaporator inlet section, a continuous S shaped cupper tube is located under the evaporator as indicated (3) in figure 2. The evaporator and fan pressure drops are measured by two micromanometers. Their accuracies are ±0.7 Pa and ±0.3 Pa.

Data Acquisition

During the tests, recording interval for the micromanometers are chosen as 2 minutes, and the temperature and R.H. recording intervals are chosen as 1 minute. On the other hand, evaporated water is measured with 15 minutes intervals from the scale display.

Experimental Procedure

Before the experiments, doors are kept open for a while to stabilize the temperature and R.H. between the ambient air and air in the refrigerator. After the stabilization, the evaporator front cover is placed in front of the evaporator as shown in figure 1 and sealed to the refrigerator walls in order to prevent air leakage. Then, the duct is inserted into the fan housing and sealed to the evaporator front cover. The thermocouples that are used to measure the evaporator air inlet temperature are placed in the suction channel, which is on the evaporator front cover. The water level in the tray is checked and then air and water heater controllers are set to the desired temperatures. After these preparations, the refrigerator doors are closed and then compressor, fan, data logger and micro manometers are operated. In general, compressor in a refrigerator operates thermostatically, however in this study it operates 100% runtime. During the experiments, water is heated to a temperature that was set to the controller. During the heating, water evaporates from the tray and this evaporated water forms frost on the evaporator. After the test, the compressor and fan are stopped as well as micromanometers and data logger. Then, the defrost heater is operated. This operation is automatically ceased by a control sensor, which was set to 15°C. During defrosting, melted frost drips on the defrost tray as indicated (11) in figure 1. After the defrosting period, the water in the tray is taken for weighting.

THEORETICAL ANALYSIS

The volumetric airflow rate is obtained from characteristic curve of fan with its housing, with respect to fan pressure drop (ΔP_{fan} [Pa]). The function of the characteristic curve is given in the following.

$$\dot{V} = 16.15 + (-0.66 \cdot \Delta P_{fan}) + (0.04 \cdot (\Delta P_{fan}{}^2)) + (-2.43 \cdot (10^{-3}) \cdot (\Delta P_{fan}{}^3)) \ \text{[L/s]}$$

In addition to volumetric airflow rate, air mass flow rate can be found by

$$\dot{m} = \dot{V} \cdot \rho$$

Air density in the above equation is calculated from the following equation [4], with respect to average temperature (T_{ave}) of fan air inlet (T_I [°C]) and outlet (T_o [°C]).

$$\rho = \frac{348.357}{273.15 + T_{ave}} \ \text{[kg/m}^3\text{]}$$

Overall heat transfer value of the evaporator can be calculated from the following equation,

$$UA = \frac{\dot{Q}}{\Delta T_{LM}} \ \text{[W/°C]}$$

In the above equation, heat transfer rate, from ambient air to the refrigerant, can be obtained from

$$\dot{Q} = \dot{m} \cdot (h_I - h_o) \ \text{[W]}$$

where h_I and h_o are the air inlet and outlet enthalpies of the evaporator, respectively. They can be obtained from the equation below, which is valid and used for humid air.

$$h = c_P \cdot T + w \cdot (2501 + 1.805 \cdot T) \ \text{[kJ/kg]}$$

Specific heat of air (c_p) can be substitute into the above equation from [6], and the temperatures are measured from evaporator air inlet and outlet sections, and the mean temperatures of these sections are used.

Water vapor saturation pressure is necessary for humidity ratio. It can be obtained from,

$P_{WS}=1000.e^{\alpha}$

where $\alpha=A.T^2+B.T+C+D^{-1}$. Here A, B, C and D are constant coefficients and can be found from [7] with respect to temperature ranges. Water vapor pressure can be calculated by

$P_W=\varphi.P_{WS}$

where φ is relative humidity.

With the help of these calculated values, humidity ratio [kg/kg] can be obtained from

$$w = 0.62198 \cdot \left(\frac{P_W}{P - P_W} \right)$$

As for logarithmic mean temperature, the following equation is used.

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}$$

Here, $\Delta T_1=T_o-T_{evai}$ and $\Delta T_2=T_I-T_{evao}$, where T_{evai} and T_{evao} are refrigerant inlet and outlet temperatures, respectively. They are measured by thermocouples from inlet and outlet tube surfaces.

RESULTS AND DISCUSSION

During the tests, +5°C evaporator air inlet temperature and different amounts of evaporated water conditions are investigated. The test conditions are given in table 1. Values that are given as evaporated water in table 1 are the values that are measured at the end of 8-hour test period.

0 gr evaporated water can be obtained by removing of the water tray from the refrigerator. Since 0 gr-evaporated water does not form any frosting on the evaporator, it is chosen to be the baseline condition, thus, UA value, evaporator airside pressure drop and airflow rate that correspond to this condition are baseline values of the evaporator.

Table 1. Test Conditions

Evaporator Air Inlet Temperature (°C)	Evaporated Water (gr)				
+5	0	200	270	370	480

Referring to figure 5, it can be stated that evaporation of water from the tray is approximately linear with time. On account of this fact instead of amounts of evaporated water, the evaporation rate, that is ratio of the amount of the evaporated water to time, can be used, so that real time evaporated water can be obtained. Thus, 0gr/hr, 25gr/hr, 33.75gr/hr, 46.25gr/hr and 60gr/hr can be utilized instead of 0, 200, 270, 370 and 480gr, respectively.

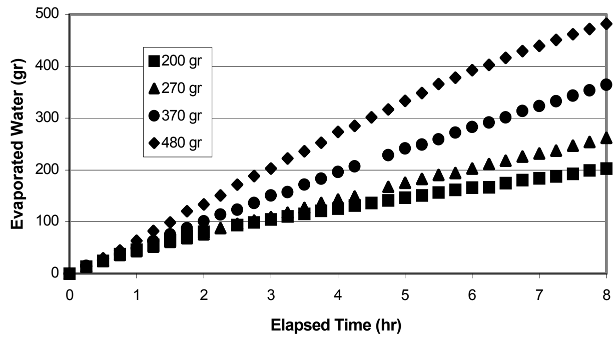

Figure 5. Various amounts of evaporated water with respect to time

During the 0 gr-evaporated water test, it is found that there is not any frost accumulation on the evaporator, thus it can be assumed that there is not any humidity infiltration from the door gaskets to the cabinet. For each test, the differences between defrosted water measurements and evaporated water measurements are about 10% and this situation supports this assumption. Due to these facts, frosting rates of the evaporator can be obtained from the evaporated water measurements.

In figure 6, the inlet and outlet R.H. values for different evaporation rates are given. After the steady state condition (about 1 hour), the values follow almost constant trends. As seen, the outlet R.H. values are always higher than the inlet ones for the same evaporation rates. It can be stated that higher evaporation rates result higher inlet and outlet R.H. values, consequently this higher R.H. values cause more frosting on the evaporator.

In figure 7, evaporator airside pressure drops for different evaporation rates are given. As seen, pressure drop is almost constant for 0gr/hr, because the R.H. for this condition is not adequate to accumulate frost on the evaporator. However, evaporator pressure drops are 4 times and 7 times larger than the baseline value for 46.25gr/hr and for 60gr/hr evaporation rates, respectively at the end of 8-hour test period. For this reason, it must be stated that, higher evaporation rates, namely R.H., result more frosting, likewise more pressure drop.

Referring to figure 8, it must be stated that frosting affects the airflow rate. As seen for the baseline condition, the airflow rate is almost constant and about 12 L/s. However, airflow rate continuously decreases for the other evaporation rates, and it decreases sharply for higher evaporation rates. It decreases from 12 to about 9 L/s for 33.75gr/hr and to about 4 L/sec for 46.25gr/hr after 8-hour test period. As for 60gr/hr, airflow rate gets 0 L/s value at about 5 hours. It is thought that the air passage of the evaporator is blocked by frost. Since air straighter is not used in the set up, fan pressure drop and consequently airflow rate values fluctuate during the tests.

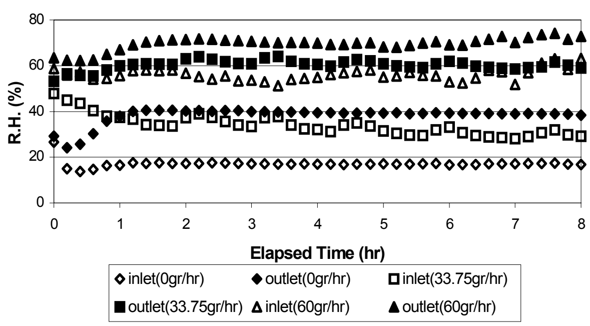

Figure 6. Evaporator inlet and outlet R.H. values for different evaporation rates

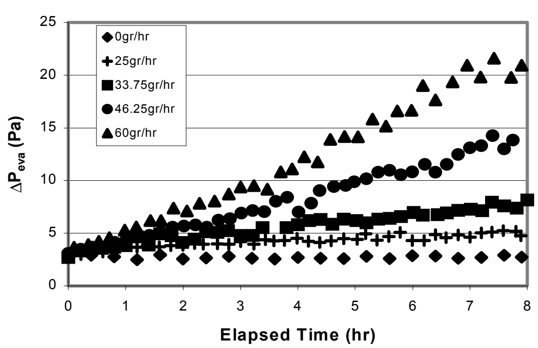

Figure 7. Evaporator pressure drops for different evaporation rates

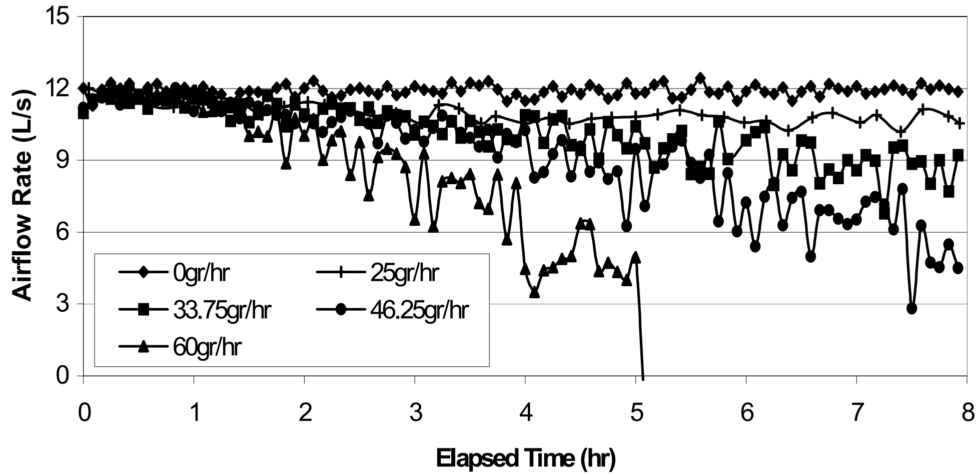

Figure 8. Airflow rate for different evaporation rates

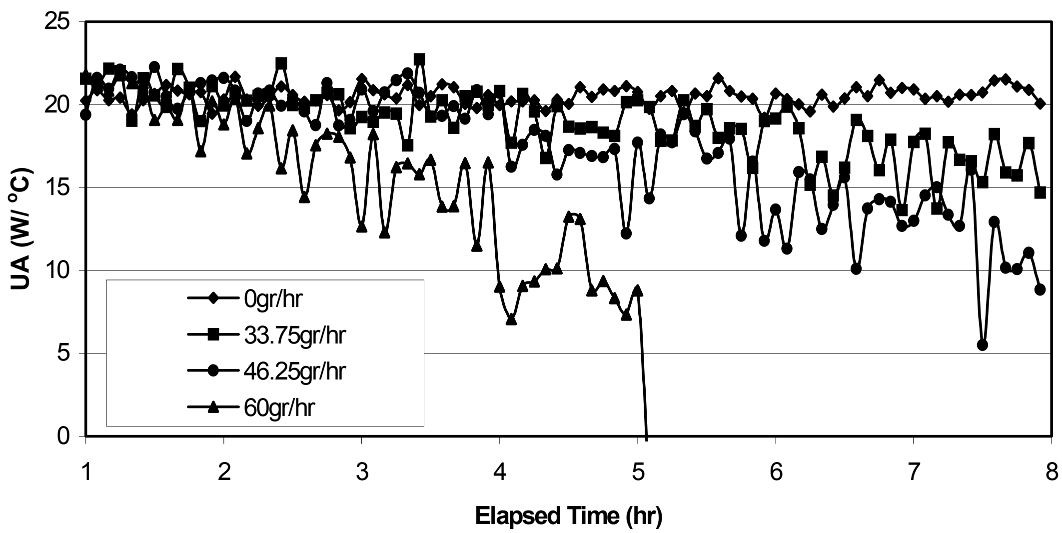

Figure 9. UA values for different evaporation rates

In figure 9, UA values for different evaporation rates are given. The first hour of the tests are omitted from the figure, because this part includes the starting effects and system is assumed to be steady state after the first hour. Referring to figure 9, it can be said that, frosting affects the evaporator performance negatively, because airflow rate and evaporator performance are directly related to each other. As seen, UA values follow decreasing trends for all evaporation rates except the baseline condition. The UA value decreases from about 21W/°C to about 15W/°C for 33.75gr/hr and about 9W/°C for 46.25gr/hr in 8 hours. However as for 60gr/hr, there is not any reasonable UA value after 5 hours on account of the airflow rate.

CONCLUSIONS

- A method to measure real time airflow rate in a refrigerator is described. This method is based on characteristic curve of fan with its housing. With the help of real time airflow rate, real time evaporator performance can be obtained.

- Frosting affects the airflow rate negatively in that it blocks the air passage of the evaporator and increases the airside pressure drop and consequently, decreases the airflow rate. After 8 hours, airflow rate is found to decrease about 25% for 33.75gr/hr and about 60% for 46.25gr/hr. It is observed that higher evaporation rates cause higher relative humidity values. This higher R.H. values result more frosting on the evaporator and decrease the airflow rate more rapidly.
- It is concluded that evaporator pressure drop increases exponentially during the frosting [3,4]. Constant airflow rate is thought to be the reason of this situation. Frosting reduces fin pitches and the maintained constant airflow increases the air velocity between the fins, thus pressure drop increases exponentially. However, in this study, it is found that pressure drop increases almost linearly instead of exponentially. The reason of this situation is thought to be the decreasing airflow rate. The decreasing of the fin pitches is accompanied with decreasing of the airflow rate.
- UA trends were found to increase steadily during the frosting [3,4]. Besides, a different UA curve based on variable airflow rate was observed [5]. This curve shows an initially increasing trend until a peak value and then a decreasing trend. However in this study, all of the UA values except the baseline value follow decreasing trends. After 8 hours, the UA value decreases about 25% for 33.75gr/hr and about 60% for 46.25gr/hr.
- After each test, defrosted water is weighted and the difference between the evaporated water from the tray and defrosted water is calculated. The differences are about 10% for all tests. In addition, it is seen from the baseline condition that there is not any humidity infiltration into the cabinet. Thus, frosting rate of the evaporator can be estimated from the evaporated water trend, which is approximately linear with time.
- One evaporator air inlet temperature with five different evaporation rates is investigated parametrically. To get a wide range of frosting effects, different air inlet temperatures with different evaporation rates should be studied parametrically. In addition, the compressor operates 100% runtime in this study, however in a real process, compressor operates cyclically. For this reason, a study based on cyclical run is planned and will be performed.
- It is found that higher relative humidity causes more frosting than lower relative humidity, and results quick airflow blockage. Blockage, even decreasing of the airflow rate deteriorates the cooling efficiency. For this reason, defrosting algorithms that take relative humidity into account should be improved. Defrosting periods based on relative humidity instead of constant defrosting periods should be used, thus energy consumption during the defrosting process can be reduced. Besides, frosting sensors that inspect the frost thickness on the evaporator gain importance. They can activate the defrosting process, when the frost thickness on the evaporator tubes or fins reaches to a critical level.

ACKNOWLEDGEMENTS

This study was performed in R&TD Center (ATGM) of ARCELIK A.S. We would like to thank to Mr. Sefik SENYUREK (Director of the center) and Mr. Yalcin TANES (Manager-Mechanical Technologies) for their continuous support and to all people in Fluid Dynamic Division for their help in preparing the system and for their participations during the discussions.

REFERENCES

[1] Thomas, L., Chen, H., Besant, R.W., “Measurement of frost characteristics on heat exchanger fins”, ASHRAE Transaction, Research 4301 (RP-824), 4302 (RP-824), 1999

[2] Ismail, K.A.R., Salinas, C., Gonçalves, M.M., “Frost growth around a cylinder in a wet air stream”, International journal of heat and mass transfer, Vol. 20, No: 2, pp 106-119, 1997

[3] Rite, R.W., and Crawford, R.R., “The effect of frost accumulation on the performance of domestic refrigerator-freezer finned-tube evaporator coils”, ASHRAE Transactions, Vol. 97, Pt 2, pp 428-437, 1991

[4] Seker, D., “The effects of frost formation on no-frost type household refrigerator evaporator”, Master Thesis, Istanbul Technical University, TURKEY, 1999

[5] Inan, C., “Evaporator performance under frosted conditions and evaluation of heat and mass transfer in a domestic refrigerator”, Ph-D Thesis, Istanbul Technical University, TURKEY, 2000

[6] Incropera, F.P, DeWitt, D.P., “Fundamentals of Heat and Mass Transfer, Fourth Edition”, 1996

[7] ASHRAE, “Psychrometrics - Theory and Practice”, USA, 1996