Investigation of Airflow and Temperature Distribution in the Freezer Cabinet of a Domestic No-Frost Refrigerator

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INVESTIGATION OF AIRFLOW AND TEMPERATURE DISTRIBUTION IN THE FREEZER CABINET OF A DOMESTIC NO-FROST REFRIGERATOR

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ABSTRACT

Uniformity of temperature distribution in a loaded freezer cabinet is one of the most important factors affecting energy consumption of a refrigerator. Present study focuses on the airflow behavior and the temperature distribution inside the freezer compartment of a domestic no-frost refrigerator. Energy consumption increases in a freezer cabinet if the temperature difference between the warmest load package and average of all packages is high. The objective is to reduce the energy consumption by providing a uniform temperature distribution and also to keep the food fresh for a longer time. In this study, the air flow and heat transfer during on-time and off-time periods inside the freezer compartment is modeled by considering turbulent and laminar flow conditions in 3D transient CFD analyses. The initial and boundary conditions are provided from temperature controlled room and PIV measurements. The CFD analyses obtained are verified by experimental measurements.

INTRODUCTION

The utilization of no-frost refrigerators have increased nearly in all industrialized countries for the last 25 years. The design of a no-frost refrigerator includes criteria such as reducing the overall energy consumption, decreasing on-time period of the compressor, and increasing shelf life of foods. Homogeneous air circulation in the cabinet is one of the parameters which affect the overall energy consumption dominantly. In a no-frost refrigerator, air is collected from the freezer compartment and sent over to the evaporator in order to be cooled. Heat is transferred from air to the refrigerant passing through the channels of the evaporator. The cooled air is blown back to freezer and fresh-food compartments through air supply channels by a fan.

Some of the previous work in literature focused on design and analysis of air management inside the cabinet by computational fluid dynamics (CFD). Generally, in CFD analyses, the steady state solutions were investigated as in the work of Fukuyo et al. [1], Laguerre et al. [2], Ding et al. [3], Lacerda et al. [4], and Lee et al. [5]. In the work of Fukuyo et al. [1], a fan and a jet channel had been added to the conventional air supply system to investigate temperature uniformity and obtain a high cooling rate by improving the air supply system in the fresh food cabinet.

Laguerre et al. [2] studied the natural convection for the refrigerators that do not have ventilation systems. Several experimental and CFD analyses have been performed for three configurations such as empty cabinet, cabinet with glass shelves and cabinet with loaded products. The laminar airflow and a constant evaporator temperature were assumed in the CFD analysis. When radiation was taken into consideration in CFD analysis, the predicted air temperatures were in good agreement with the experimental values.

Ding et al. [3] found that temperature variation will decline and more uniform temperature distributions will be achieved if the distance between the shelves and the evaporator and the distance between shelves and refrigerators door are reduced. Mea-

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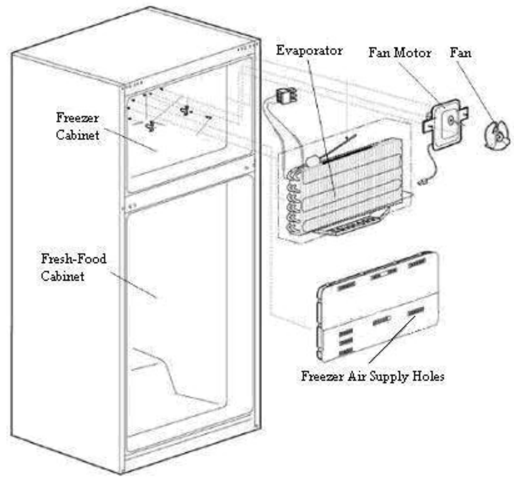

Figure 1. COMPONENTS OF A NO-FROST REFRIGERATOR

surement of air flow in a freezer compartment under real operating conditions was carried out by Lacerda et al. [4] and Lee et al. [5] using PIV (particle image velocimetry). Due to the fact that physical properties of air change with temperature variations, it was observed that the flow field was strongly influenced during the "on" and "off" operation cycles of compressor.

However, steady state solutions do not represent the real temperature difference that occurs in the local regions inside the freezer cabinet. In this present work, transient CFD analyses have been performed for the freezer compartment of a no-frost refrigerator and results are compared with the experimental measurements performed in temperature-controlled room. The compatibility of experimental measurements and CFD results yields a design strategy which can be used to predict the homogenization of air circulation inside the cabinet.

EXPERIMENTAL MEASUREMENTS

Temperature Controlled Room measurements

A domestic no-frost refrigerator has freezer and fresh food compartments which are cooled by a refrigerant flowing inside an evaporator as shown in Fig. 1. A fan placed above the evaporator manages air circulation through the air supply channels. CFD analyses are performed to simulate the conditions that experimental measurements are achieved in a temperature-controlled room. In the freezer, instead of real food, simulation packages are used to determine the temperature distribution occurred during the air circulation. The simulation packages are placed in the freezer cabinet according to TS-EN-ISO-15502 standard [6]. The temperatures are monitored through the thermocouples placed in the center of packages. The loading plan of packages and the packages with thermocouples which are numbered are shown in Fig. 2. The packages are attributed with properties sim-

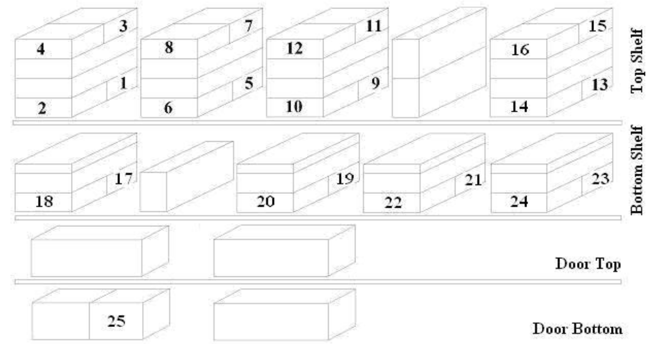

Figure 2. PACKAGE LOADING PLAN AND THERMOCOUPLES

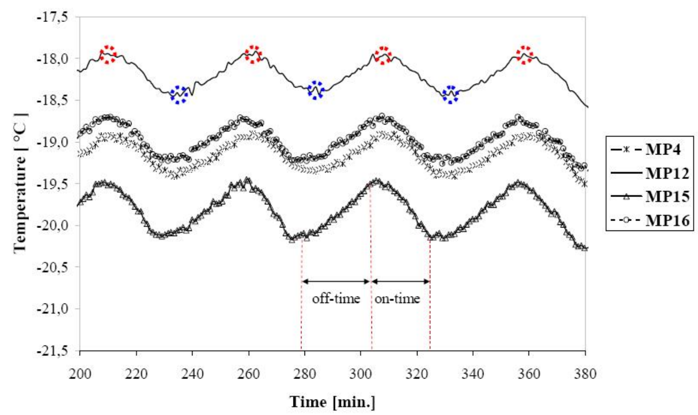

Figure 3. TEMPERATURES OF SELECTED M-PACKAGES AT STEADY CONDITIONS DURING ON-TIME AND OFF-TIME OPERATING CYCLES

ilar to meat products. Figure 3 illustrates temperatures measured in simulation packages at steady state conditions during on-time and off-time period of compressor. For each simulation package temperature is averaged over a couple of operating cycles at 25^{o}C ambient temperature as shown in Fig. 4. The temperature gradient between the warmest and coldest packages is found to be 2.5K and 3K at the end of on-time and off-time period of the compressor, respectively. As a result of the experiments, the warmest packages are found to be the 8th and 12th packages which are located at the upper shelf.

3D Particle Image Velocimetry (PIV) measurements

Air flow rates blown to the freezer from the air supply holes shown in Fig. 5 are determined by 3D PIV measurements. Two cameras (Dantec Dynamics Hisense MKIi) are used to obtain 3D velocity vectors at each cross section of air supply holes. Nd-YAG laser with 8Hz energy pulse is used. Oil particles which

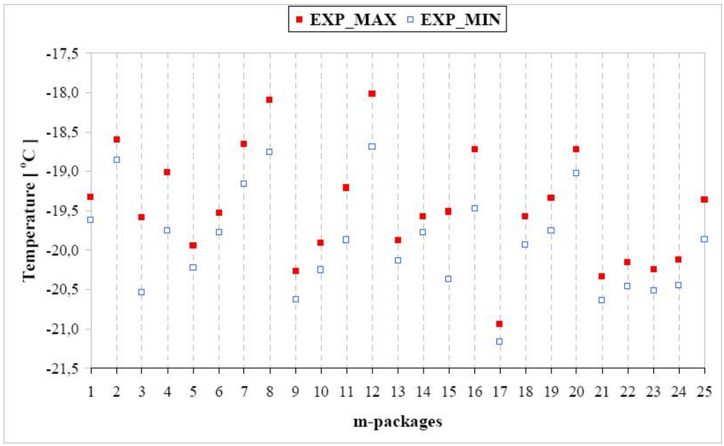

Figure 4. EXPERIMENTAL RESULTS FOR ON-TIME AND OFF-TIME OF THE COMPRESSOR

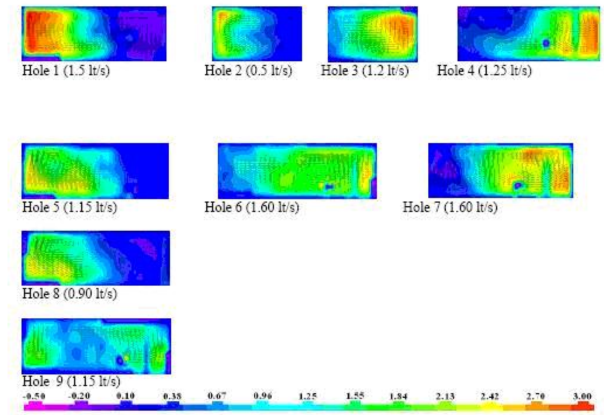

Figure 6. PIV VELOCITY MEASUREMENTS AT AIR SUPPLY HOLES

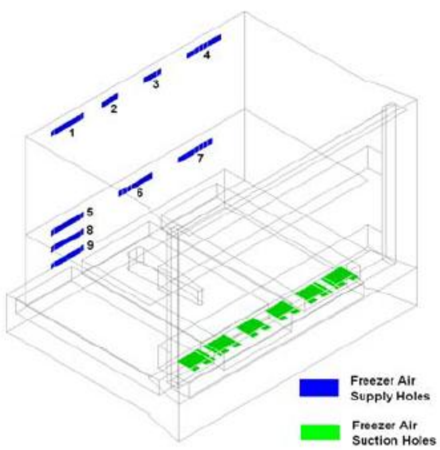

Figure 5. AIR SUPPLY HOLES AND SUCTION HOLES INSIDE THE FREEZER

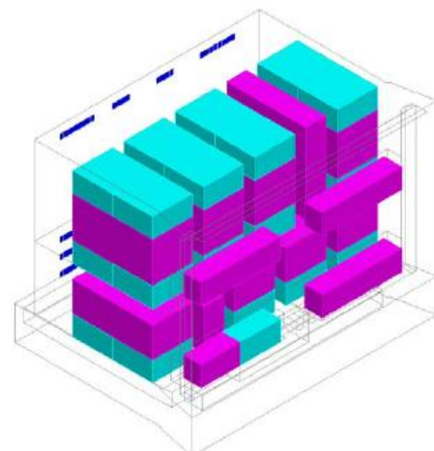

Figure 7. THE LOAD PACKAGES IN THE FREEZER

have diameter of 25 micrometer were used for seeding. The repeatability and reproducibility (R&R) of these measurements are guaranteed by Gauge R&R study according to 6 sigma criteria which is GR&R < 30. The volumetric flow rates are calculated from velocity profiles at each air supply hole and are demonstrated in Fig. 6. The total air flow rate that is sent into the freezer compartment is approximately 11 lt/s. The measured flow rates for each air supply hole are used as boundary conditions in CFD analyses.

CFD ANALYSIS

As it was shown in Fig. 5, there is only one shelf in the freezer cabinet and two shelves on the freezer door. The suction region is composed of three channels as shown in Fig. 5. Air in the freezer cabinet is collected through the channels which are on both sides of the suction region. Nevertheless the channel which is located in the middle is used to collect the warmer air coming from the fresh-food cabinet. There is insulation between these channels. In Fig. 7 the simulation packages which are loaded in the freezer cabinet are shown. The packages in which temperature change is monitored are colored in light blue. In CFD analyses only the flow domain in the cabinet is modeled and circulation of the air through the evaporator and air supply channels are neglected. The geometric model used in CFD analyses is simplified to provide a better quality of flow domain mesh. Mesh is created in Gambit 2.3.16 and flow analyses are performed in Fluent 6.3.26. However, the features which may have an effect on

Table 1. MATERIAL PROPERTIES

Properties	Polyurethane	Plastic	M-Packages
Density [kg/m^3]	40	1050	3106
Specific heat [J/kg-K]	1100	1500	1047
Thermal conductivity[W/m-K]	0.022	0.136	1.58

flow and temperature distribution are saved with attention. Computational grid is composed of 4 million hybrid cells which are 60% hexahedral and the rest tetrahedral elements. In order to investigate mesh dependency, the CFD analysis is performed with a computational grid consisting of 5 million elements as well. Temperature values in the center of the simulation packages are used to assess the convenient mesh. Analyses showed that the mesh with 4 million cells provides the satisfying results with a reasonable difference for the temperature values.

In this study, the on-time period of the freezer compartment is the first 21 minutes when the air circulation is driven by the fan. In that case the flow is considered as turbulent and k-epsilon standard turbulence model is employed. During off-time period which is the following 29 minutes, fan is not running, therefore the air is driven due to the density variation and flow is considered as laminar. Boussinesq equation is employed to simulate natural convection.

Mass, momentum and energy equations are solved for this heat transfer problem. The flow properties of the air which is blown from air supply holes to the freezer cabinet are taken from 3D PIV measurements and calorimetric room measurements. In on-time period, cold air volumetric flow rate for the freezer cabinet is 11 lt/s while 2 lt/s air is entering to the fresh-food cabinet. The thicknesses and thermal conductivities of insulation materials, plastic liners, shelves, packages are known parameters and listed in Tab. 1 .

The insulation on the walls and the thickness of the shelves are not modeled but "Shell conduction" option of Fluent is utilized. Therefore the conductance of the insulation and shelves are taken into account while the mesh size for 3D analyses is kept affordable. In temperature controlled room experiments, it is seen that shelves bend due to the package loading, therefore, the packages which are adjacent to the side walls contact to the walls with nearly half of their side area. This bending is not modeled in CFD geometry and a full contact is assumed between packages and walls, however, the thermal conductivity of the contact surfaces are taken as half of the real value. Nevertheless, because of the plain structure of shelves, the packages and shelves include contact conductance without any regulations on the thermal conductivity. In this study, the ambient temperature is kept constant at 25^{o}C and the heat transfer coefficient for the outer wall of the refrigerator is chosen as 8 W/(m^{2}K). Since simulation of infiltration and thermal bridging at gasket region is very complicated, they are not included in the CFD model but an adjustment is made to increase the heat transfer coefficient at the gasket region of the door with reference to former CFD studies and experiments.

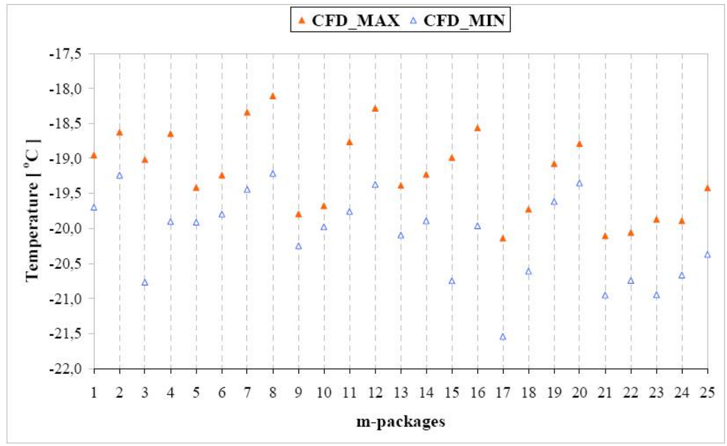

Figure 8. TEMPERATURE VALUES FROM THE CFD RESULTS FOR THE ON-TIME AND OFF-TIME PERIODS

CFD RESULTS

The CFD analyses are run in parallel on a cluster of 10 itanium nodes. "Metis" algorithm is utilized for mesh partitioning. The turbulent flow analysis which lasts 21 minutes is performed in 138.000 CPU seconds. Moreover, 195.000 CPU seconds are needed to solve the 29 minutes laminar flow analysis. The time duration that will be used for "on-time" and "off-time" of the compressor is obtained from the experiments performed in temperature controlled room. In analyses, first, the on-time period is simulated as turbulent flow while compressor and fan are operating. The initial condition for on-time period analysis is given as same as the temperature values obtained from the experimental measurements taken at the end of off-time period. At the end of on-time period which lasts for 21 minutes, the temperature values are read at the center of the simulation packages.

The transient laminar flow conditions which natural convective heat transfer in the freezer cabinet prevailed, is simulated for the following 29 minutes off-time period. The initial conditions are taken from the experimental measurements when the compressor stopped at the end of 21 minutes. The temperatures for the simulation packages obtained from transient CFD calculations for both on-time and off-time periods are illustrated in Fig. 8. In Fig. 9, the temperature distribution over the packages are shown at the end of "on-time" period. As it is seen in both experiments and computations, the temperature values of the packages which are adjacent to the air supply holes, are cal-

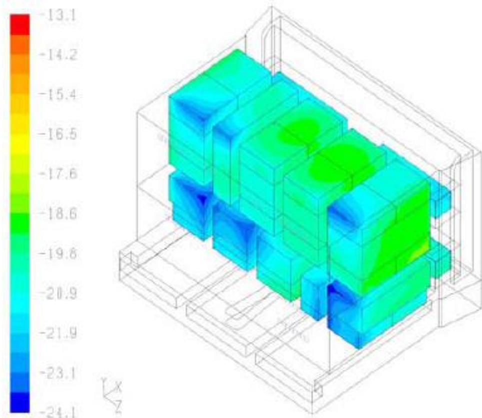

Figure 9. TEMPERATURE DISTRIBUTION OVER THE PACKAGES AT THE END OF ON-TIME

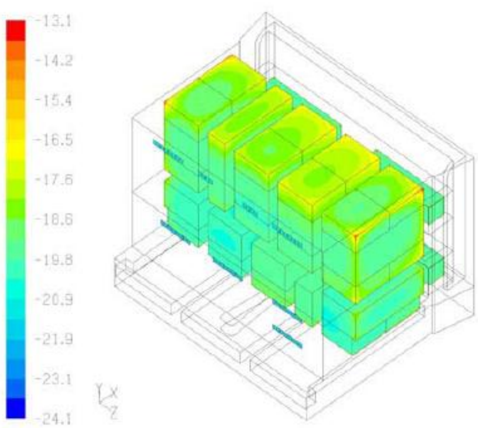

Figure 10. TEMPERATURE DISTRIBUTION OVER THE PACKAGES AT THE END OF OFF-TIME

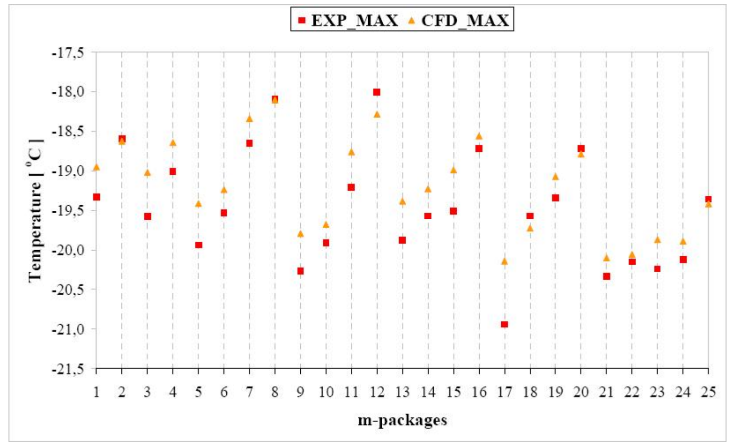

Figure 11. COMPARISON OF EXPERIMENTAL AND CFD RESULTS AT THE END OF OFF-TIME PERIOD

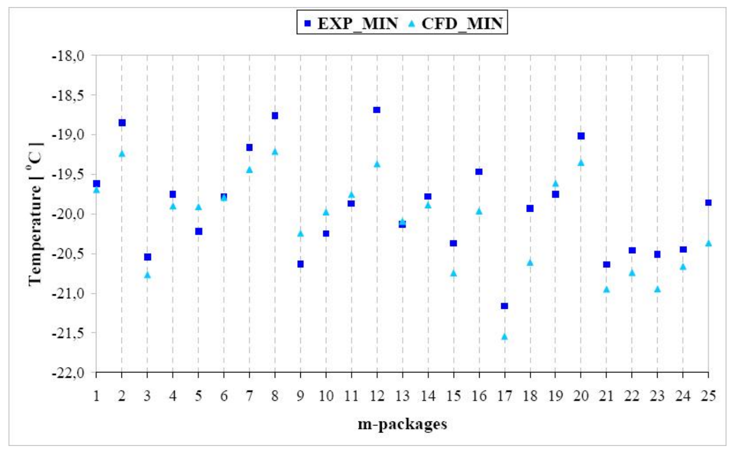

Figure 12. COMPARISON OF EXPERIMENTAL AND CFD RESULTS AT THE END OF ON-TIME PERIOD

culated to be colder than the other packages. The temperature of the package which is above the air supply channel which collects the warmer air from fresh food cabinet is higher than the other packages located on the bottom of the freezer. The package temperatures at the end of off-time period are shown in Fig. 10. It is observed that the packages locating in the middle of the upper shelf are obtained warmer compared to other packages. The comparison of temperature values between experiments and CFD analysis at the end of off-time period is compared in Fig. 11. Concerning the end of off-time period, the temperature difference between the warmest and coldest packages are 3K in the experiments, while it is 2K in the CFD analysis. When average of all package temperatures are considered, the difference between experiments and CFD analysis is 0.3K. As shown in Fig. 12, at the end of the on-time, the temperature difference between the warmest and coldest packages are 2.5K in the experiments, while it is 2.3K in the CFD calculations. When average of all package temperatures are considered, the difference between experiments and CFD analysis is calculated as 0.2K. These results proove that CFD computations and experimental measurements are in a very good agreement.

CONCLUSION

CFD analyses for 3D transient turbulent flow conditions in which forced convection heat transfer in the freezer cabinet is valid, are performed for the on-time period of a compressor. Also 3D transient laminar flow conditions with natural convection are analyzed for off-time period of the compressor as well. Both CFD analysis for turbulent and laminar flow conditions resulted in a good agreement with experimental measurements. Thus, the temperature distribution and air circulation is well simulated for the on-time and off-time periods of a freezer compartment in a no-frost refrigerator. In this present work, design parameters which are important for providing homogeneous air circulation in a freezer compartment are understood. The results of this study will yield a design optimization methodology to improve the on-time period of a no-frost refrigerator.

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