Numerical and Experimental Investigation of Air Distribution System of Larder Type Refrigerator

Funda Erdem Şahnali, Ş. Özgür Atayılmaz, Tolga N. Aynur

***Abstract*—**Almost all of the domestic refrigerators operate on the principle of the vapor compression refrigeration cycle and removal of heat from the refrigerator cabinets is done via one of the two methods: natural convection or forced convection. In this study, airflow and temperature distributions inside a 375L no-frost type larder cabinet, in which cooling is provided by forced convection, are evaluated both experimentally and numerically. Airflow rate, compressor capacity and temperature distribution in the cooling chamber are known to be some of the most important factors that affect the cooling performance and energy consumption of a refrigerator. The objective of this study is to evaluate the original temperature distribution in the larder cabinet, and investigate for better temperature distribution solutions throughout the refrigerator domain via system optimizations that could provide uniform temperature distribution. The flow visualization and airflow velocity measurements inside the original refrigerator are performed via Stereoscopic Particle Image Velocimetry (SPIV). In addition, airflow and temperature distributions are investigated numerically with Ansys Fluent. In order to study the heat transfer inside the aforementioned refrigerator, forced convection theories covering the following cases are applied: closed rectangular cavity representing heat transfer inside the refrigerating compartment. The cavity volume has been represented with finite volume elements and is solved computationally with appropriate momentum and energy equations (Navier-Stokes equations). The 3D model is analyzed as transient, with k-ε turbulence model and SIMPLE pressure-velocity coupling for turbulent flow situation. The results obtained with the 3D numerical simulations are in quite good agreement with the experimental airflow measurements using the SPIV technique. After Computational Fluid Dynamics (CFD) analysis of the baseline case, the effects of three parameters: compressor capacity, fan rotational speed and type of shelf (glass or wire) are studied on the energy consumption; pull down time, temperature distributions in the cabinet. For each case, energy consumption based on experimental results is calculated. After the analysis, the main effective parameters for temperature distribution inside a cabin and energy consumption based on CFD simulation are determined and simulation results are supplied for Design of Experiments (DOE) as input data for optimization. The best configuration with minimum energy consumption that provides minimum temperature difference between the shelves inside the cabinet is determined.

***Keywords*—**Air distribution, CFD, DOE, energy consumption, larder cabinet, refrigeration, uniform temperature.

F. E. Şahnali is with the School of Mechanical Engineering, University of Yıldız Technical, İstanbul, Turkey (phone: +90537-788-66-85; e-mail: funda.erdem@arcelik.com)

Ş.Ö. Atayilmaz is Professor, Department of Mechanical Engineering, University of Yıldız Technical, İstanbul, CO 34349 Turkey (e-mail: atayil@yildiz.edu.tr).

T. N. Aynur is a manager in the Thermodynamics & System Based Simulation Department, Arcelik Company, Tuzla, CO 34950 Turkey (e-mail: tolga.aynur@arcelik.com).

I. Introduction

It is believed that increasing the cooling performance and saving in energy consumption of domestic refrigerators are getting more important. One of the most important factors that affect the cooling performance of a refrigerator is air velocity and temperature distribution in the cooling chamber. Food keeping quality increases, as temperature distribution inside the refrigerator compartments gets uniform.

The aim of research is that energy efficiency was investigated with an A+ energy class larder type refrigerator. System components and refrigeration cycle of the refrigerator were explained, information about standard [1] on energy consumption testing was given, and the parameters affecting the energy consumption and the pull-down time of the cabinet are investigated both experimentally and numerically.

Three dimensional models for the air path inside the cabinet were studied. These models were separated into finite volumes and solved computationally with appropriate air flow and energy equations. Effects of air temperature, fan speed and shelf type on the energy consumption and pull-down time were investigated and validated with the experimental results. After the analysis, the main effective parameters for the temperature distribution inside the cabin and energy consumption based on CFD simulation were determined and simulation results are supplied for DOE as input data for optimization.

II. Literature Review

Within the scope of literature review, experimental and numerical studies related to homogeneous velocity and temperature distribution in the refrigerator were investigated.

Few authors, including [2], [3], studied on a static refrigerator in which heat transfer is carried out by natural convection. The refrigerating cabinet of the refrigerator was examined both numerically and experimentally for cases where the cabin was shelfless, shelved and packaged. The analysis results were based on convection between the wall and the air, radiation heat transfer between the inside walls of the cabin. The temperature and speed distribution of the air inside the cabin were examined. Experimental and numerical results were compared only for temperature distribution. It has been observed that the air flow between shelves and products in the cabinet with glass shelf is very low.

Fukuyoa et al. [4] conducted studies on the development of the air supply system to ensure homogeneous and rapid cooling in a domestic refrigerator. In this scope, blower and jet nests were added to the system. The jet pockets optimized the air velocity and distribution by circulating the air in the cab at high speeds. In addition, these nests also improved the heat transfer on the food surface. In this context, the air distribution in the cabin was analyzed by CFD with the alternative design.

Gupta et al. [5] studied both parts of the refrigerator's freezer and fresh food compartment, using the final volumes and non-structural mesh methods. The simulation results with test results were found to be negligible in size compared to the temperature differences. In simulation, average freezer temperature value was observed to be slightly higher than experimental test results.

Jaramillo et al. [6] used three-dimensional model and time dependent approach to improve air temperature distribution inside the cabin in order to reduce energy consumption. In addition, the position of the inlet and outlet air vents has been improved. To homogenize the air and temperature distribution in the cabin, two new blow holes were made and analyzed in this way. After adding two new holes, temperature of the vegetable compartment decreased.

Hermes et al. [7] examined the air flow and temperature distribution in the intended cooling cabinet and in the freezer cabinet. The stream was considered incompressible and the Boussinesq approach and "finite volume" method were used. Simulation results apply to all single-door cabinets. The comparison of numerical results with experimental data was included in the study. When the numerical and experimental data were compared, it was seen that the results are close to each other. The difference was found to be a maximum deviation of 1.7 °C.

III. Mathematical Equations

In order to study the heat transfer inside the aforementioned refrigerator, forced convection theories were applied. Closed rectangular cavity representing heat transfer inside the refrigerating compartment and cavity volume were solved computationally with appropriate momentum and energy equations (Navier-Stokes equations).

It was considered that the problem was solved in time. In the numerical analysis, forced convection was accepted since the fan was used in the refrigerant chamber and the k-ε Turbulence Model was used because of turbulence effects. Radiation was neglected in numerical analysis in order to simplify the solution and shorten the duration of the solution. The heat transfer that circulating in refrigerant compartment was included in the model according to the results obtained from the experimental data.

Equations (1)-(5) represent Navier-Stokes equations that are used in numerical analysis.

A. Continuity Equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

B. Momentum Equation

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho_o}\frac{\partial P}{\partial x}+v\nabla^2 u \tag{2}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{\rho_o}\frac{\partial P}{\partial y}+v\nabla^2 v+g\beta(T-T_o) \tag{3}$$

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{1}{\rho_o}\frac{\partial P}{\partial z}+v\nabla^2 w \tag{4}$$

C. Energy Equation

$$\frac{\partial}{\partial \mathrm{t}}\left[\rho\left(e+\frac{1}{2}V^2\right)\right]+\nabla[\rho(e+\frac{1}{2}V^2)\vec{V}=-\nabla(\rho\overrightarrow{V})+\rho\vec{f}\vec{V}+\dot{w}_{visc}+\rho\dot{q}+\dot{q}_{visc} \tag{5}$$

The k-ε turbulence model has been the most widely used turbulence model in practice, due to its comprehensive and economical nature and its high degree of accuracy. The assumptions underlying the k-ε model are as follows; the flow is turbulent and the molecular viscosity is negligible. Equations (6), (7) give transport equations for k and ε in the k-ε model.

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_j}(\rho k u_j)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\epsilon-Y_M+S_k \tag{6}$$

$$\frac{\partial}{\partial t}(\rho\epsilon)+\frac{\partial}{\partial x_j}(\rho\epsilon u_j)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\epsilon}\right)\frac{\partial \epsilon}{\partial x_j}\right]+\rho C_1 S_\epsilon-\rho C_2\frac{\epsilon^2}{k+\sqrt{v\epsilon}}+C_{1\epsilon}\frac{\epsilon}{k}C_{3\epsilon}G_b+S_\epsilon \tag{7}$$

Equation (8) represents that the only difference according to the other models is that Cμ is not a constant but is calculated.

$$C_\mu=\frac{1}{A_0+A_s\frac{kU^{\infty}}{\epsilon}} \tag{8}$$

IV. Theoretical Studies

When theoretical calculations were made, the heat gains obtained by the HeatGain program results and the compressor calorimeter test results were used. However, it has been calculated that the fresh food compartment fans in the current product will also cause the energy consumption of the compressor to operate. For the present case (25 °C ambient temperature), the cabinet heat gain was found to be 34.5 W, when the available polyurethane insulation thicknesses were entered and the temperature difference between the outdoor and the indoor environment was 21 K (see Table I).

The heat transfer coefficient for air was 8 W/m²K and the heat transfer coefficient for the cabin was 9,37 W/m²K, using the polyurethane thermal conductivity coefficient of 24 mW/mK in the direction of theoretical calculations.

TABLE I
UA and Heat Gain Table

UA (W/mK)	Heat Gain (W)	Temperature of cabinet (℃)	Ambient temperature (℃)
1,6	34,5	4	25
1,8	50,2	4	32

Considering the evaporation and condensation temperatures of base and alternative cases, the cooling capacity and the power taken for each case by the help of the compressor map were theoretically calculated and used as input during CFD analysis. Initially, the enthalpy difference and the cooling capacity were calculated using the lnP-h diagram of the R600a refrigerant for the operating conditions of -12 °C evaporation, 37 °C condensation, 0.2 °C subcool and 5 °C superheat temperatures. Since cooling capacity of the evaporator tubes is different from that of the refrigerant with the lowest degree of dryness when the cooling capacity is calculated, each evaporator tube was calculated based on the weighted average of the individual time and pipe area divided by specific time intervals.

As a result of the calculations, the current state cooling capacity was found to be about 148 W. When the experimental working conditions were examined, it is stated in the upper part of the report that the current cabinet had 26% runtime and the heat gain at 25 °C was 34.5 W. Also from here, the cooling capacity was calculated experimentally as 135 W by the formula "capacity = heat gain/RT" and 9% difference with the theoretical calculation. On the other hand, in order to validate the experimental cooling period with CFD for each case, the theoretical working times were calculated using the above information. The energy consumption of the 24-hour cycle in the specified operating conditions and operating rate can be found by the equation "power x working rate x 24" (see Table II).

TABLE II
COOLING CAPACITY AND POWER RESULTS FOR EACH CASE

Heat Gain (W)	Ambient Temperature (℃)	Fan speed (rpm)	Evap/Cond Temp (℃)	Shelf type	Q (W)	Fan power (W)	Total power (W)
34,5	25	1320	-12,2/37,4	Glass	148	1,7	81
	25	1500	-11,5/37,4		152	2	83
50,2	32	1320	-9,3/45,8	Glass	168	1,7	86
	32	1500	-9/46		177	2	87
34,5	25	1320	-12,4/37,5	Wire	148	1,7	81,5
	25	1500	-11,7/37,5		152	2	83,4
50,2	32	1320	-9,3/46	Wire	168	1,7	86,5
	32	1500	-9/46,2		177	2	87,3

V. EXPERIMENTAL METHOD

Almost all of the domestic refrigerators operate on the principle of the vapor compression refrigeration cycle and removal of heat from the refrigerator cabinets is done via one of the two methods: natural convection or forced convection. In this study, airflow and temperature distributions inside a 375L no-frost type larder cabinet, in which cooling was provided by forced convection, were evaluated both experimentally and numerically.

As in Fig. 1, larder type refrigerator was used for experimental and numerical studies. The experimental setup comprises the components of evaporator, cooling air fan, wire on tube condenser, compressor, defrost heater and sensor. There was a natural convection wire condenser in the cabinet used in the operation and the inside of the cabinet was cooled by a vertical WOT (wire on tube) evaporator. The radial fan with DCDC motor used in the cabin and the air sucked from the upper part was passed through the evaporator for cooling.

Fig. 1 Refrigerator

For the single-door refrigerators referred to in the thesis, tests were carried out according to the latest edition of the European Council "EN ISO 15502" energy consumption standards [8]. Fig. 2 represents that the cooling device shall be placed on a wooden platform painted with a matte black color. Provided that the platform is not less than 0.05 m, the test chamber shall be on the ground and at least 0.3 m away from the vertical divider.

Air temperature should be measured using copper or brass cylinders 20-30 mm below the platform and 35 mm away from the sides. The relative humidity of the environment should not exceed 75%. The environment should be at 25 °C and the temperature oscillation should not be higher than ± 0.5 K.

Four different parameters and two levels for each case were considered in the scope of the study (see Table III). At the end of the analysis, the best configuration was determined to provide minimum energy consumption.

TABLE III
VARIABLE PARAMETERS AND LEVELS

Cases	Ambient Temperature (°C)	Fan speed (rpm)	Shelf Type
1	25	1320	Glass
2	25	1500	Glass
3	32	1320	Glass
4	32	1500	Glass
5 (base case)	25	1320	Wire
6	25	1500	Wire
7	32	1320	Wire
8	32	1500	Wire

Energy consumption tests for base case were carried out in accordance with ISO 15502 (wire shelf, 25 °C ambient temperature and 1320 rpm fan cycle). Temperature measurements were taken with the thermocouple from the places required (condenser pipes, compressor surface, evaporator passes, dryer inlet, suction, blowing, cabin shelves) during the tests.

The system's evaporation temperature was -12.5 ºC and the condenser temperature was 37.5 ºC according to the work done at 25 ºC ambient temperature and 4 ºC thermostat state. The product operated at 26% RT in this thermostat position. According to the test results, the coldest shelf in the current situation was found to be ~1.8 °C (lower shelf) and the warmest shelf was ~4 °C (upper shelf). In the same way, for the other six cases, energy tests were carried out (see Table IV).

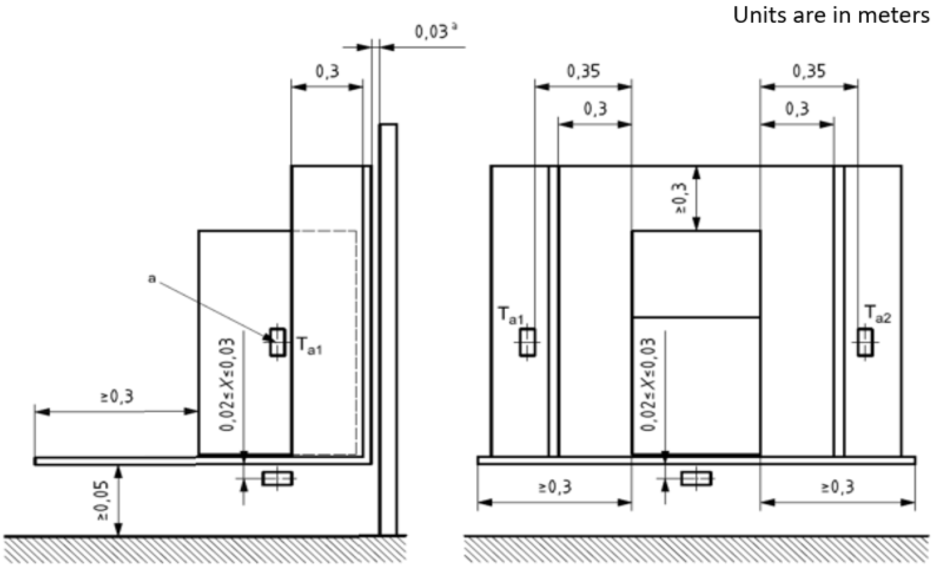


Fig. 2 Cooling device positioning in energy consumption experiments

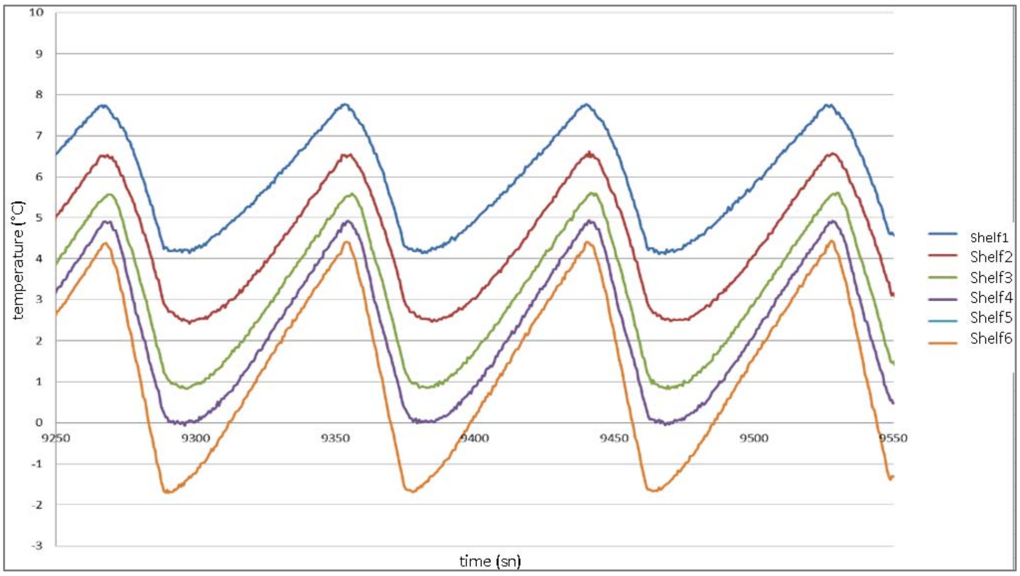
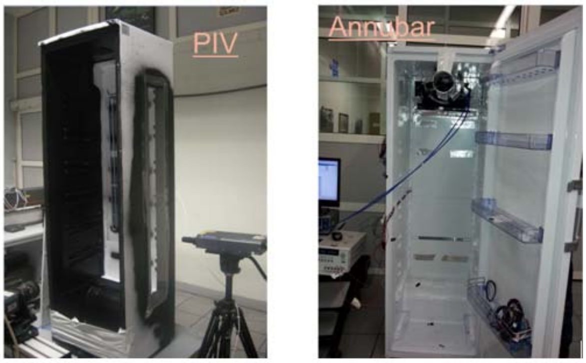


Fig. 3 Current situation shelf temperature (Tambient: 25 °C)

TABLE IV
EXPERIMENTAL RESULTS FOR CURRENT AND ALTERNATIVE SITUATIONS

Cases	RT (%)	Off time (min)	On time (min)
1	0,24	62	20
2	0,24	61	19
3	0,34	47	24
4	0,33	48	23
5 (base case)	0,26	64	22
6	0,25	64	21
7	0,35	46	25
8	0,33	47	23

The flow visualization and airflow velocity measurements inside the original refrigerator were performed via SPIV and Annubar measurement system. Annubar flow measurement method is a metric for measuring flow with differential pressure principle. The Annubar flow measurement sensor has pressure relief holes. The pressure coming from the pressure receiving holes in the sensor inlet is transmitted to the (+) part of the pressure transmitter, while the pressure from the sensor output holes is transmitted to the part of the differential pressure transmitter. The differential pressure transducer dissolves the Bernoulli equations by taking the difference of these (+) and (-) pressures and measures the flow rate of the process.



(a) (b)

Fig. 4 (a) Current status PIV; (b) Annubar air flow measurement

In the present case, the evaporator fan rotates at 1320 rpm, and according to PIV, there was a total of 7.3 l/s air flow. For the same situation, Annubar measurement result was found to be 7.2 l/s (see Table V).

TABLE V
ANNUBAR MEASUREMENT RESULTS

Fan speed (rpm)	Voltage (V)	Air flow rate (liter/sec)
646	4	3,4
873	5	4,6
1201	7	6,8
1467	8	7,8
1645	9	8,9
1823	10	10
1293	7,3	7
1320	7,4	7,2
1500	8,3	8,2

VI. NUMERICAL MODELING

In order to study the heat transfer inside the aforementioned refrigerator, forced convection theories covering the following cases were applied: closed rectangular cavity representing heat transfer inside the refrigerating compartment, the cavity volume has been represented with finite volume elements and was solved computationally with appropriate momentum and energy equations (Navier-Stokes equations). The 3D model was analyzed as transient, with k-ε turbulence model and SIMPLE pressure-velocity coupling for turbulent flow situation.

In the CFD analysis, fluid domain was created in ANSYS geometry. Cut-cell method with ~12 million elements was used in ANSYS meshing for fluid domain of refrigerator. Fig. 5 shows the shape and grid of the fluid domain. While generating structured grid, inflation was applied to boundaries of air outlet and inlet in order to visualize air flow.

It is crucial that, in the CFD analysis, density of air was considered temperature dependent. During the flow analysis, the radial fan in the cabin was defined by the moving reference frame (MRF) method; the rotation speed was 1320 rpm and the rotation axis was z = + 1. Properties of polyurethane used as insulation material were defined as 31 kg/m^3 density, 1045 J / kgK specific heat and 0.024 W/mK thermal conductivity. The evaporator inside the air distribution duct was modeled as an isothermal wall. As a result of the theoretical calculations, the heat fluxes obtained were entered as inputs.

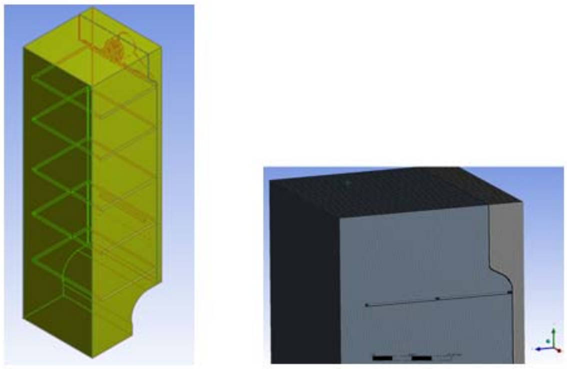


Fig. 5 The shape and mesh domain of the refrigerator

The heat transfer between the cabin's rear and side surfaces and the external environment was modeled. The outdoor temperature was assumed to be 32 °C for the rear wall, 35 °C for the cabin groove area and 25 °C for the side surfaces. Lastly, the heat transfer coefficient between the outdoor environment and the outer surface of the air refrigerator was calculated. Accordingly, since the cabinet heat gain, side surface area and temperature difference are known, the coefficient of heat transfer is as follows:

$$q'' = h \; x \; \mathrm{A} \; \mathrm{x} \; \Delta(T_{ortam} - T_{kabin})$$

$$35 \text{ W} = \text{h}*0{,}75 \text{ m}^2*(25\text{-}4)\ °\text{C}$$

$$\text{h} = 2.2 \text{ W/m}^2\text{K}$$

When calculating, only polyurethane was taken into account and the total heat transfer coefficient was found. During the analysis, the polyurethane thicknesses were not entered and the total heat transfer coefficient and surface temperatures were entered.

Heat transfer coefficient of convection was calculated by using experimental data. Moreover, turbulence effect in the channel flow was modelled by k-ε turbulence models. The current state model was analyzed for 3 different mesh values to determine the change of the values obtained by the flow analysis depending on the mesh size. From the results it can be seen that the blowing speed does not depend on the mesh size and the maximum difference is within 1% band.

Fig. 7 represents the change in air flow of refrigerator along the horizontal and the vertical midplane in the flow domain. Looking at airflow distribution, it is clear that the velocity of the air on the left side is higher and the middle parts are lower.

In numerical analysis, the total flow rate in the cabin was found to be 7.5 l/s @1300 rpm and the air velocity in the outlet was found to be 1.04 m/s. The air from the blowing duct was distributed over the door to the upper and lower parts.

Especially at the bottom, it is seen that the flow is weak and dead zones are formed. Moreover, experimental results have a good agreement with CFD analysis substantially. Maximum 4% deviation was obtained under steady the state air flow rate of in the fluid domain. After validating CFD results, effects of different parameters on air flow rate and temperature distribution in the channel were investigated. These parameters were fan speed, shelf type and ambient temperature.

Table of Design Points

	A	B	C	D	E	F	G
1	Name	Update Order	P1 - Body Sizing Element Size	P2 - out-h?z	P3 - out-velocity	Exported	Note
2	Units		mm	m s^-1	m s^-1		
3	Current	1	0,5	1,04	1,04		
4	DP 1	2	0,4	1,03	1,03		
5	DP 2	3	0,6	1,04	1,04		
*							

Fig. 6 Mesh dependency

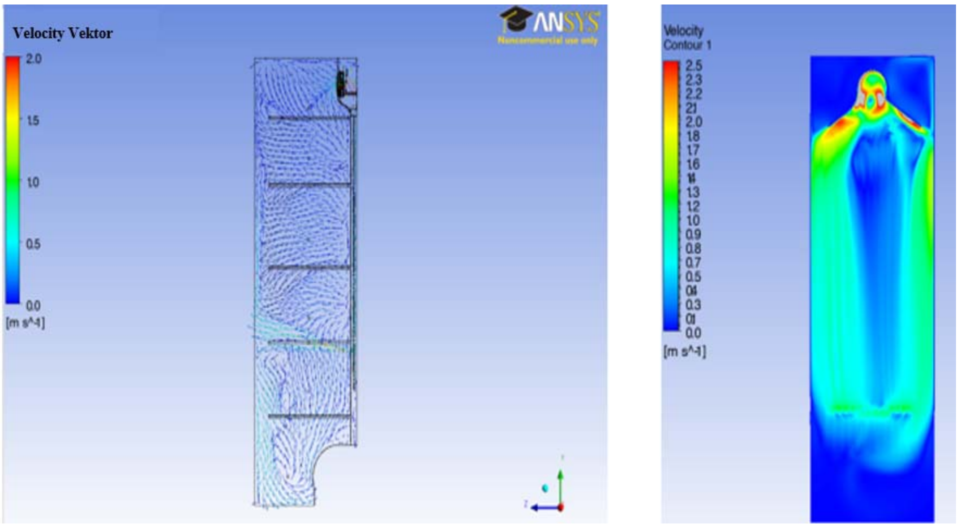


Fig. 7 Velocity vectors and contours for base case

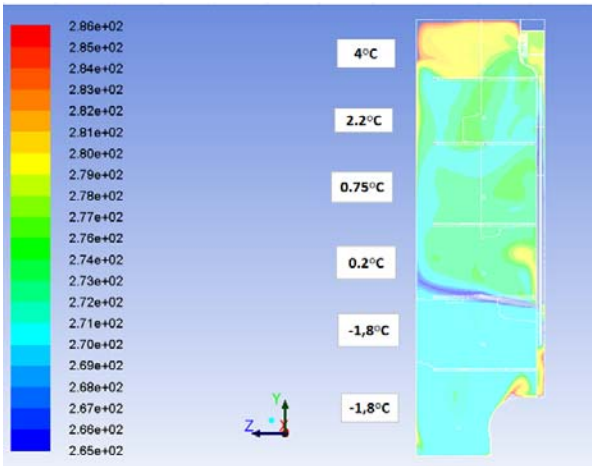


Fig. 8 Temperature distribution for base case (wire shelf, 1320 rpm, 25°C)

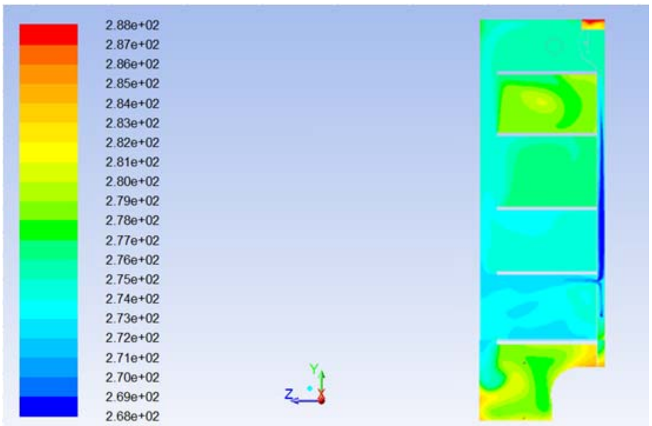


Fig. 9 Temperature distribution for alternative case (glass shelf, 1320 rpm, 25 °C)

Time-dependent temperature distribution analyzes have been performed since flow analyzes have been completed for base and alternative situations. As a theoretical calculation, a heat flux of -660 W/m^2 was entered for the evaporator and the refrigerator was analyzed during the pulldown period.

During the analysis, the temperature was monitored in the cabin for the current situation and the analysis was stopped. The analysis was completed at about 13 minutes for the current wire rack 1320 rpm fan cycle and 25 °C ambient temperature. Maximum 16% difference was obtained between the CFD and the experimental result when compared to the cooling time.

According to CFD result, the top shelf maximum temperature was 4 °C and the bottom shelf temperature was -2 °C. According to the experimental results, the top shelf temperature is 4.1 °C and the bottom shelf temperature is -1.7 °C at off mode.

Fig. 10 presents the CFD analysis results with the experimental results of each case. The results show that the minimum difference between the analysis and the experimental results was 6% and the maximum difference was 16% when the cooling time was compared on some occasions. This value is acceptable (± 20%) [6].

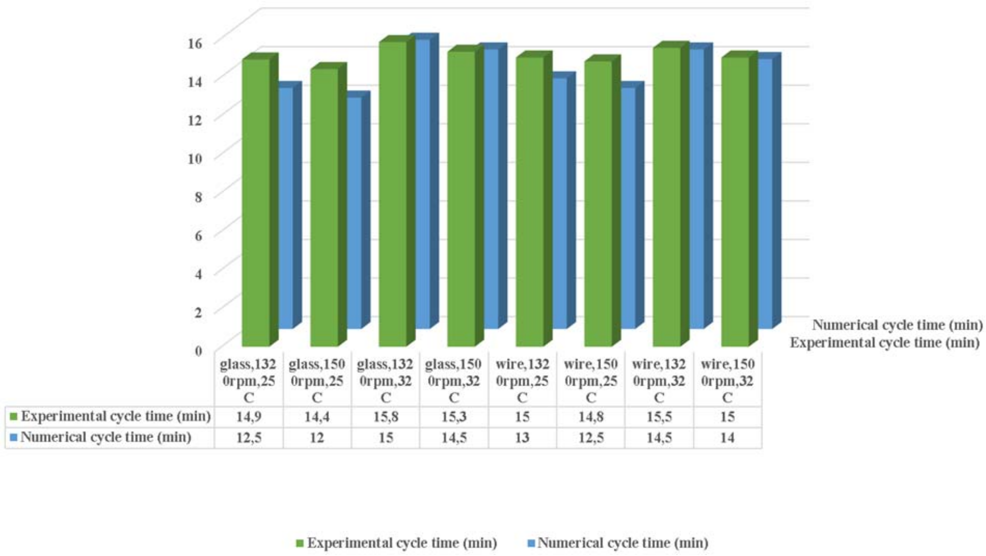


Fig. 9 Comparison of experimental and CFD results

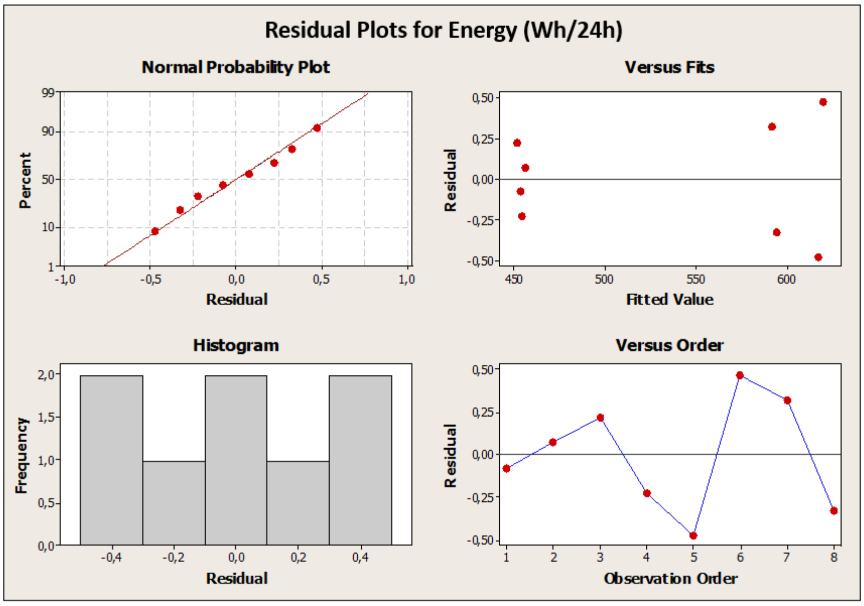


Fig. 10 Reduced model graph

Increasing the fan speed for the same shelf type and ambient temperature reduced cooling time by 4%, resulting in a decrease in energy consumption.

Increasing the ambient temperature in the same shelf type and fan speed increases cooling time by 12% and energy consumption by 35% accordingly.

VII. DOE Analyzes

Sensitivity analysis was performed in the Minitab program to investigate the effect of parameters determined during the thesis study on cooling time and energy consumption. Full Factorial Design method has been applied in order to determine parameters that affect energy consumption.

Main effects and binary interactions on energy consumption in the model are:

- Ambient temperature
- Fan Speed
- Shelf Type
- Fan Speed x Shelf Type

A four-in-one graph was drawn after the reduced model and the fit and residuals were obtained. It has been determined that there is no increase in RES (residual) as the FITS values increase, according to the graph between residuals and normal FITS-RES.

The highest energy consumption was achieved with 1320 rpm fan cycle, 32 °C ambient temperature and wire rack (619.8 Wh/24 h). The lowest energy consumption was achieved with 1500 rpm fan cycle, 25 °C ambient temperature and glass shelf (451.8 Wh/24 h).

Fig. 12 shows that the "shelf type" factor was found to have the highest effect over the variability of 49.3%.

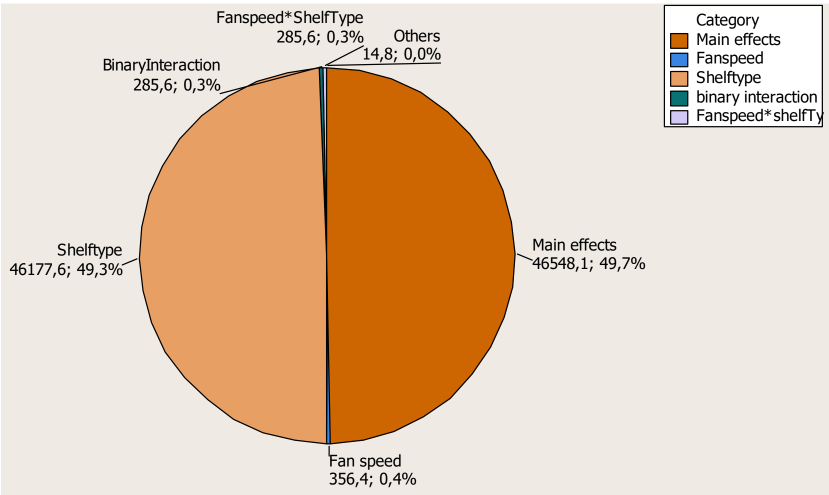


Fig. 11 Main effects and binary interaction on energy consumption

VIII. Conclusions

In this study, the effects of parameters in a single door refrigerator cabinet on cooling time and energy consumption were studied experimentally and numerically. The parameters were examined on air flow distribution and the temperature difference between shelves in transient conditions.

Energy consumption tests for base case were carried out in accordance with ISO 15502 (wire shelf, 25 °C ambient temperature and 1320 rpm fan cycle). The system's evaporation temperature was -12,5 ºC and the condenser temperature was 37.5 ºC for 25 ºC ambient temperature and 4 ºC thermostat state. The product operated at 26% RT in this thermostat position. According to the test results, the coldest shelf in the current situation was found to be ~ 1.8 °C (lower shelf) and the warmest shelf was ~ 4 °C (upper shelf).

The flow visualization and airflow velocity measurements inside the original refrigerator were performed via SPIV and Annubar measurement system. Moreover, experimental results have a good agreement with CFD analysis substantially. Maximum 4% deviation was obtained under steady the state air flow rate of in the fluid domain. After validating CFD results, effects of different parameters on air flow rate and temperature distribution in the channel were investigated. These parameters were fan speed, shelf type and ambient temperature.

As consequences of parametric works in CFD, increasing the fan speed for the same shelf type and ambient temperature reduced cooling time by 4%, resulting in a decrease in energy consumption. Increasing the ambient temperature in the same shelf type and fan speed increases cooling time by 12% and energy consumption by 35% accordingly.

In this study, it has been seen that experimental temperature values and CFD temperature values are in harmony with each other. According to the experimental results, the temperature difference in the wire shelf case is $\Delta T = 5.6$ K, which is obtained as $\Delta T = 5.8$ K in CFD results.

During modeling, it is assumed that there is no distance between the sides of the glass shelves and the rear surfaces. This situation prevents air leakage between the shelves. It is recommended to compare effect of this distance on the temperature distribution by repeating the analysis and experimental tests.

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Funda Erdem Şahnali acquired her Bachelor and MS degree on Mechanical Engineering in 2012 and 2014 from Yıldız Technical University. She works as a Senior R&D specialist in Arcelik Company R&D Center since 2012. Now, she is a PhD student at the Yıldız Technical University. She works on heat and mass transfer enhancement, modelling and optimization of fluid dynamics and heat transfer mechanism.

Şevket Özgür Atayılmaz is a Professor at Mechanical Engineering Department, University of Yıldız Technical, Turkey. He received his Ph.D. from the Yıldız Technical University at Turkey. He has co-authored more than 50 journal and conference publications. His research interests include experimental and numerical two phase and single phase convective heat transfer, enhancement of heat and mass transfer by different techniques.

Tolga N. Aynur earned his B.S. on Mechanical Engineering in 2000 and Ph.D. in 2008 at the Mechanical Engineering Faculty, Istanbul Technical University. He has worked as a faculty researcher in CEEE, University of Maryland from 2005 to 2009. His research interest is energy efficiency improvement of vapor compression cooling and heating systems. He is currently working as an R&D manager of Thermodynamics and System Based Modelling team at the R&D Center of Arçelik.