Journal of Thermal Engineering
Yildiz Technical University Press, Istanbul, Turkey
Vol. 2, No. 5, pp. 946-953, October, 2016.

http://eds.yildiz.edu.tr/journal-of-thermal-engineering/Articles
Manuscript Received June 20, 2015; Accepted July 07, 2015

This paper was recommended for publication in revised form by Regional Editor Derya Burcu Özkan

NUMERICAL INVESTIGATION OF THE BOTTOM CABINET OF A HOUSEHOLD REFRIGERATOR

Haluk Yılmaz
Anadolu University
Eskişehir, Turkey

L. Berrin Erbay
Eskişehir Osmangazi University
Eskişehir, Turkey

*** Bahadır Doğan**
Eskişehir Osmangazi University
Eskişehir, Turkey

Keywords: Household refrigerator, Condenser, Bottom Cabinet, Cooling
** Corresponding author: Bahadır Doğan, Phone: +90 222 2393750/3523, Fax: +90 222 2293613*
E-mail address: bdogan@ogu.edu.tr

ABSTRACT

In this study, the bottom cabinet of a commercial household refrigerator is investigated numerically. The numerical model is formed in two stages. The first stage includes the fan motion, and the second stage includes the heat rejection process at the condenser. The compressor is included in the numerical model to get the accurate simulation of the real conditions. A protective lid covering the condenser and the fan is taken into consideration for both stages of the numerical solution. In the numerical analysis, the total amount of heat transfer, temperature, pressure and velocity gradients and streamlines are studied in detail. Due to the location and the design of the fan, a static fluid region occurs at the mid-section of the condenser. Therefore, the number of wires on the mid-section of the condenser tubes and the cost can be decreased. It is observed that the compressor is also cooled by the discharged air from the condenser.

INTRODUCTION

Especially over the last ten years, energy regulations have been inured to reduce the residential energy consumptions. Household refrigerators which have been used by hundreds of millions are the most important appliances in residential life in terms of energy consuming. Due to the energy efficiency requirements of the EU regulations and the entry into force of the modification of consumption standards in 2011, sales of the B-class products having the energy index value of 55 and in 2014, sales of the A-class products having the energy index value of 42 were prohibited. These regulations have been also changing the preferences of the consumers. From 2008 to 2009, the market share of the A++ class products increased 2.2% to 5.9% and market share of the A+ products increased to 30.8% to 37.7%. In parallel, the market share of the A and the lower class products decreased from 67% to 56.4%. The A+++ products which are not remain before has strengthened its position in the market during the last five years. Between 2010 and 2014, the production of the A+++ products increased from 0% to 2%. In this process, the main aim of the manufacturers are the increasing of energy efficiency of each component in the household refrigerators. Increasing the heat transfer capacity of the condensers and the evaporators and decreasing the irreversibility during the compression process of the refrigerants in the compressors and during the throttling in the capillary are the major methods of the improvement of energy efficiency. On the other hand, the physical properties of the household refrigerators such as the insulation of the whole body, the location of the condensers, the compressors and the evaporators also effect the energy efficiency of the refrigerator.

In the open literature, many studies focus on the improvement of energy usage and the saving costs in the field of household refrigerators, condensers and evaporators. Erbay et al. [1-4] performed several studies on the thermal and hydraulic performance of a mini-channel flat tube condenser with multi louvered fins. Yılmaz [5] investigated the thermal performance of a condenser of the commercial household refrigerator. One of the main objectives of the study is to analyze the performance of the condenser via numerical solution under the operating conditions. 190 W heat transfer into the air from the condenser was seen in the condenser area via numerical model under tropical climate conditions. As a result of the theoretical calculation, it was found that the heat gain of refrigeration cycle is 198 W. Radermacher and Kim [6] reviewed the recent developments in the field of domestic household refrigerators based on a survey of publications and patents. Bansal et al. [7] provided an overview of the current status of the technology of five major household appliances. They stated that the overall energy use in refrigerator-freezers, dishwashers, clothes washers, clothes dryers and electric ovens can be reduced respectively by more than 50%, 17%, 43%, 50% and 45%. Gonçalves et al. [8] developed a numerical model for simulating the steady state conditions of household cooling systems. They present the effects of some key parameters on the system performance by using EES software.

Hermes and Melo [9] investigated the transient behavior of household refrigerators by numerically. The refrigerator energy consumption was found to be within ±10% agreement with the experimental data. Hermes et al. [10] presented a simplified model to assess the energy performance of vapor compression refrigerators. The numerical results were found to be within ±5% agreement with the experimental data. It was shown that the energy consumption can be decreased by as much as 7.5% by using a lower capacity compressor. Laguerre et al. [11] studied the heat transfer by natural convection in domestic refrigerators without ventilation by numerically and experimentally. They found that when radiation was taken into consideration in simulation, the predicted air temperatures were in good agreement with the experimental data. Yang et al. [12] performed a numerical simulation of the performance of the household refrigerator. The simulations showed that the design of air duct and its locations may impose a detrimental role on the temperature uniformity within the refrigerator.

Melo et al. [13] tried to obtain a non-dimensional correlation to calculate the natural convection coefficient between the outer surface of static wire-on-tube type condenser and environment air. The correlation was used to present the effects of tube diameter and number of tube rows and wires on the heat rejection capacity of the condenser. They found that the heat transfer coefficient decreases with the increase of the number of wires. They stated that the effect of tube diameter was only significant for a large number of tube rows. Borges et al. [14] developed a model for calculating the energy consumption of the household refrigerators and a sensitivity analysis was carried out to identify opportunities for energy savings. The model predictions were found to be within a maximum deviation of ±2%.

Lin et al. [15] presented a dynamic model for the multi-compartment indirect cooling household refrigerator. They calculated the differences between the predictions and the experimental data are within 2°C for compartment air temperature. Negrao and Hermes [16] presented an innovative design methodology for household refrigeration systems focused on both energy savings and cost reduction. The results of the simulation were verified with the experimental data obtained for a single-door vertical freezer. It was found that the freezer becomes less costly where highly efficient compressors are used. Waltrich et al. [17] performed an optimization methodology for sizing the components of refrigeration cassettes for light commercial applications. The optimization led to two improved cassette designs, which were assembled and tested. Mitishita et al. [18] advanced an optimization methodology focused on both energy and cost savings in frost-free refrigerators. The system performance for four different compressors was analyzed. It was found that the variable speed compressor should not cost 26% more than a single-speed one with the same piston displacement. Mansour et al. [19] performed a study of the conjugate effects of ejector performance characteristics, the activation pressure-temperature conditions at the generator and the interaction with the compressor on refrigeration systems. A performance analysis and a parametric optimization were performed to select promising cycles.

Due to the experimental analysis of household refrigerators are usually carried out through costly, time-consuming standardized test procedures, a faster and less costly alternative is the use of numerical analysis. In this study, the bottom cabinet of a commercial household refrigerator is investigated numerically. The bottom cabinet of the household refrigerator is modeled with a commercial CFD package. The heat transfer and the fluid flow characteristics of the bottom cabinet was investigated. In addition, the theoretical calculation is compared with the results obtained from the numerical solutions.

METHOD

In this study, the air flow is investigated in Cartesian coordinates and assumed as 3-D, steady state, and single phase. In the solution, computational fluid dynamics (CFD) package program is used with realizable *k-ε* (RKE) turbulent model. This model has been found to be substantially better than that of the standard *k-ε* model for the rotational air flow [20]. Enhanced Wall Treatment Option is used. For the numerical solution of conservation equations, finite volume method is used. As the SIMPLE algorithm and second order discretization and multiple step precision are preferred, convergences are limited to around 10^{-5} for the solutions. Also, solver options are chosen, Segregated Solver, Least Squares Cell-Based and Hybrid initialization. The no-slip boundary condition is applied. A finite element package program of ANSYS14 [20] is used for the numerical analysis. The fundamental fluid and energy equations are solved at the steady state, 3-D and turbulence conditions.

The modeled transport equations for k and ε in the realizable *k-ε* model are;

$$\frac{\partial}{\partial x_j}\left(\rho k u_j\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon - Y_M + S_k \quad (1)$$

$$\frac{\partial}{\partial x_j}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \rho C_1 S \varepsilon$$
$$-\rho C_2 \frac{\varepsilon^2}{k+\sqrt{\nu \varepsilon}} + C_{1\varepsilon}\frac{\varepsilon}{k} C_{3\varepsilon} G_b + S_\varepsilon \quad (2)$$

$$C_1 = max\left[0.43, \frac{\eta}{\eta+5}\right] \quad (3)$$

$$\eta = Sk/\varepsilon \quad (4)$$

$$S = \sqrt{2S_{ij}S_{ij}} \quad (5)$$

$$C_{3\varepsilon} = \tanh\left|\frac{v}{u}\right| \quad (6)$$

In these equations G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients, G_b is the generation of turbulence kinetic energy due to buoyancy, Y_M represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate. $C_{1\varepsilon}$=1.44 and C_2=1.9 are constants. σ_k=1.0 and σ_ε=1.2 are the turbulent Prandtl numbers for k and ε. μ_t is the turbulent viscosity.

Numerical Model

Considered geometry of the numerical model is shown in Figure 1. This model is considered into two stage during the numerical simulations.

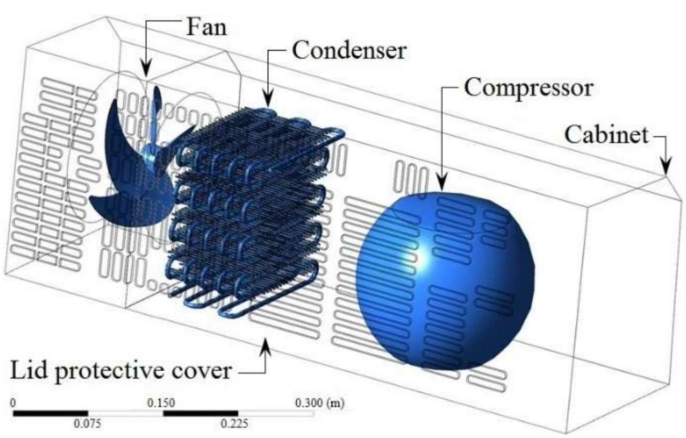

Figure 1. Considered geometry of the bottom cabinet of a household refrigerator

The first stage includes the rotational air flow by the fan motion, and the second stage includes the heat rejection process from the condenser. The output data obtained from the first stage is used as the input data of the second stage. In second stage of numerical simulations, the hermetically sealed compressor is also considered in the model to get the accurate simulation of the real conditions. A protective lid covering the fan and the condenser are taken into consideration for both stages of the numerical solution.

Boundary Conditions

In the first stage of the numerical simulation, there is not defined a boundary condition of velocity inlet at the inlet holes of the protection lid. Air motion is only provided due to the pressure difference crated by the fan. A reference frame is defined to simulate the fan motion of 1400 rpm. The properties of air are assumed to be constant at 320 K as shown in Table 1.

Table 1 Properties of the air at 320 K

ρ (kg/m^3)	1.0948
c_p (J/kgK)	1006.2
k_{air} (W/mK)	0.02778
μ (kg/ms)	1.9404×10^{-5}

The pressure field obtained from the outlet of the first stage is defined as an inlet boundary condition for the second stage. Air is assumed at a temperature of 316 K at the inlet of the condenser section. The surface temperature of the condenser is defined based on the experimental measurements. The surface of the condenser is equally divided into 22 parts from the inlet port to the outlet port. Figure 2 shows the first five parts of the condenser surface from the inlet port. The experimentally measured temperature data as shown in Table 2 are defined as a boundary condition to these surfaces.

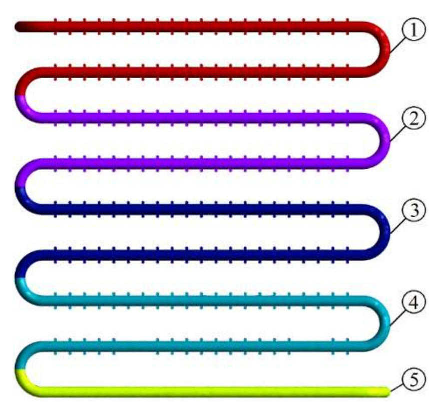

Figure 2 Surface temperature of the condenser

For the compressor, the constant surface temperature of 360 K is defined based on the experimental measurements and the inlet and outlet holes on the protection lid are defined as the outlet to the atmosphere.

Table 2 Surface temperature of the condenser under actual operating conditions

	T (K)		T (K)
1	343.0	12	331.2
2	340.0	13	330.8
3	338.0	14	330.3
4	336.3	15	330.2
5	335.1	16	329.8
6	335.0	17	329.5
7	334.2	18	329.2
8	333.5	19	328.9
9	332.8	20	328.6
10	332.2	21	328.3
11	331.7	22	328.0

Verification of Numerical Results

The tetragonal mesh structure is preferred due to the curved surfaces of the condenser and the compressor. A finer mesh structure is provided at the inlet and the outlet parts of the cabinet, and at the surface of the fan. Table 3 shows the deviation of volumetric flow rate of the air at different mesh sizes. The number of nodes having %1.16 deviation by the previous examination is used for the numerical simulation.

It is found that the heat transfer rate from the condenser to the air is 190.94 W by using the finest mesh structure having node number of 658945. This result is in good agreement with the experimental calculation as given below. The thermo-physical properties of R-600 which is the working fluid in the real cycle are given in Table 4.

Table 3 Considered mesh sizes and the deviation of volumetric flow rate of the air at the exit of the cabinet

	Number of nodes	Skewness	Orthogonal quality	Volumetric flow rate (l/s)	% deviation in volumetric flow rate
1	377799	0.8478	0.1989	18.5	-
2	401201	0.8498	0.1951	17.4	%5.95
3	445394	0.9006	0.1788	18.8	%8.04
4	475234	0.8836	0.1804	19.6	%4.26
5	565931	0.8490	0.1780	22.4	%14.29
6	618860	0.8831	0.2060	21.55	%3.79
7	658945	0.8701	0.1994	21.8	%1.16

Table 4 Thermo-physical properties of R-600 at the inlet and the exit of the compressor

	Temperature, T (K)	Pressure, P (kPa)	Density, ρ (kg/m3)	Enthalpy, h (kJ/kg)
inlet	310	50	1.15	736.7
exit	400	800	14.80	887.4

Volumetric flow rate of the refrigerant is calculated as $\dot{V} = 550 \times 10^{-6}\ m^3/s$ through the compressor based on the Eq. 7,

$$\dot{V} = fV \tag{7}$$

where $f = 50\ Hz$ is the frequency and $V = 11.5 \times 10^{-6}\ m^3$ is the stroke volume of the compressor. Mass flow rate of the refrigerant is calculated as $\dot{m} = 632.5 \times 10^{-6}\ kg/s$ through the cycle using the Eq. 8,

$$\dot{m} = \rho\dot{V} \tag{8}$$

where ρ is the density of the refrigerant at the inlet of the compressor. Requested compressor power of $\dot{W}_{comp} = 95.32\ W$ is calculated to compress the refrigerant to the condenser pressure using the Eq. 9,

$$\dot{W}_{comp} = \dot{m}(h_2 - h_1) \tag{9}$$

where h_1 and h_2 are the enthalpies of the refrigerant at the inlet and the outlet of the compressor, respectively. Thus, the total heat transfer rate of $\dot{Q} = 198.32\ W$ is obtained to the refrigerant through the cycle based on the Eq. 10,

$$\dot{Q} = \dot{Q}_{freezer} + \dot{Q}_{freshfood} + \dot{W}_{comp} \tag{10}$$

where $\dot{Q}_{freezer} = 36\ W$ and $\dot{Q}_{freshfood} = 67\ W$ calculated experimentally are the rates of heat removal from the refrigerated spaces. It is seen that the difference of heat transfer rate to the refrigerant between the CFD result and the theoretical result is 3.8%. In addition, when the mass and the energy conservation are examined for the numerical simulation, 0.0244 W of loss in the total heat transfer and $3.47\text{x}10^{-7}$ kg/s of loss in the total fluid mass transfer occurred. These differences show that the mass and the energy conservation are provided.

RESULTS AND DISCUSSION

The air flow through the fan is analyzed in the first stage of the numerical simulation. The velocity vectors of the air flow due to the pressure difference created by the fan motion is shown in Figure 3. As it is seen that the fan sucks the air (Figure 3a) from the inlet holes of the protection lid and blows it to the condenser with a rotational flow (Figure 3b). The proportion of air motion in the center section of the fan blades is very little than that of the end point of the fan blades (Figure 3c) which is a reason of the pressure field occurs with the rotational flow.

Additionally, the air flow is effected by the negative pressure field occurs under the fan as seen in Figure 3d. This phenomenon is also seen in Figure 3b and 3c. It is seen that the air flow is accelerated passes through the region closed to the bottom wall of the cabinet at the exit of the fan. In the second stage of the numerical simulation, the effect of the rotational motion is considered for the air flow passes through the condenser as seen in Figure 4a and 4b. In this case, some disadvantages of the rotational flow are observed in terms of heat transfer.

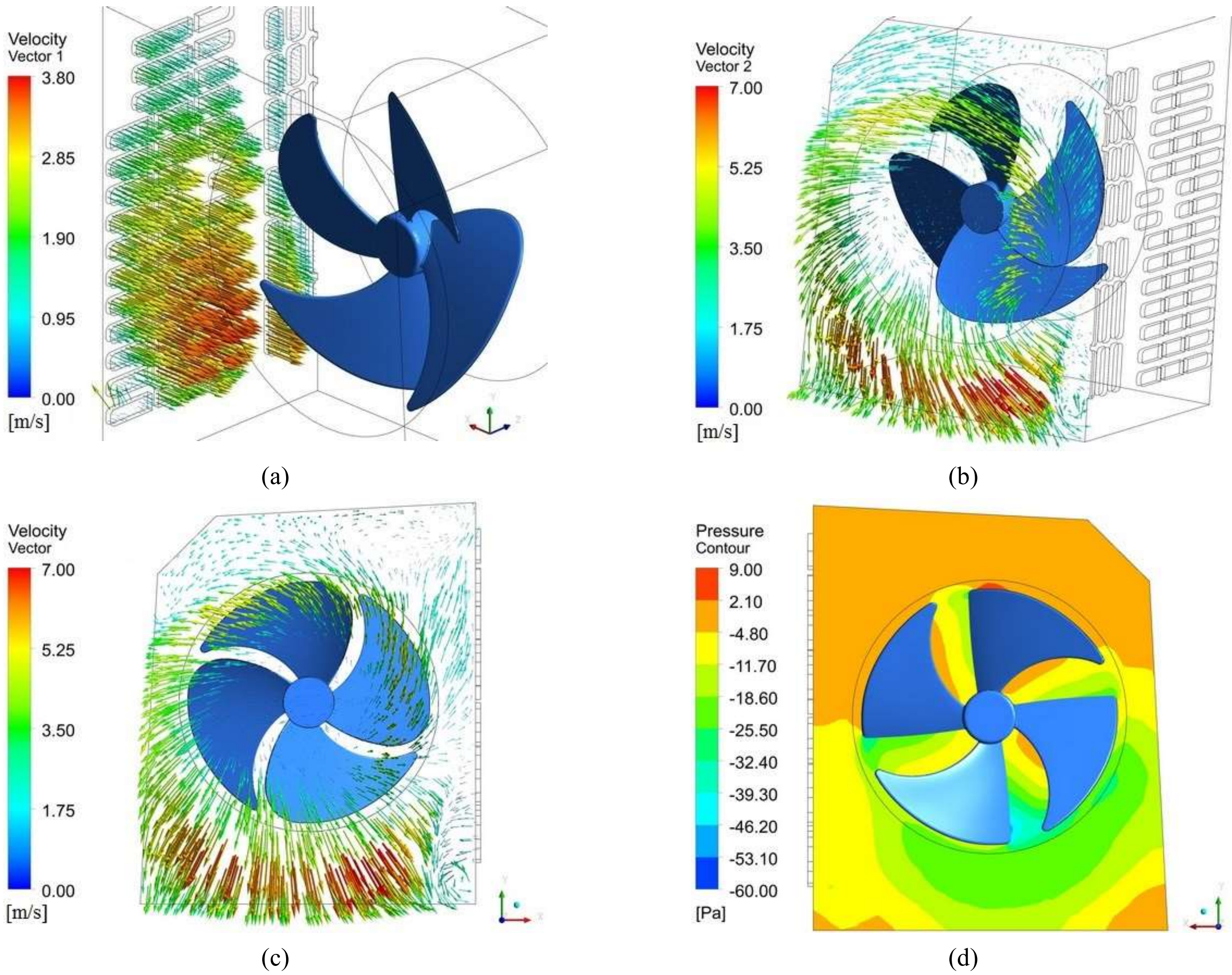

Figure 3 Velocity vectors and pressure contours of the air flow created by the fan

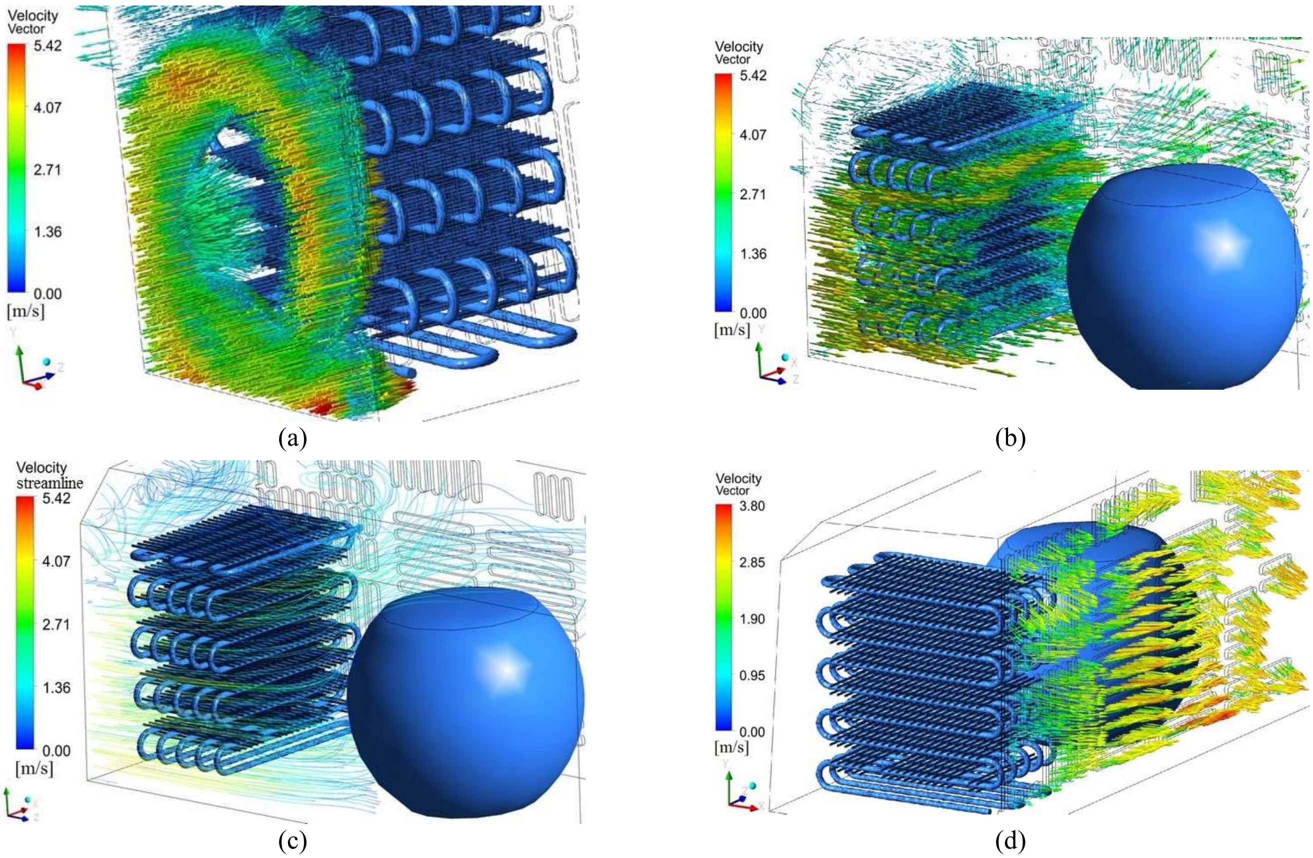

Figure 4 Velocity vectors and streamlines of the air flow at the condenser section

The air flow pumped by the fan cools the condenser expect the mid-section as seen in Figure 4b and 4c. Figure 4c shows that the air flow that passes near the outer frame of the condenser and then tends to go the compressor section and the outlet holes of the lid. Additionally, a small proportion of the air exits from the outlet holes of the lid before it is not reach the condenser as seen in Figure 4d.

Figure 5 shows the heat rejection process and the local temperature of the air at the inlet and the outlet of the condenser. Figure 5b and Figure 5c can cause a misunderstanding about the heat transfer performance of the condenser. The mid-section of the condenser can be seen as the most effective region in terms of heat transfer. As it is seen in Figure 4b, a small proportion of air passes through the mid-section of the condenser so that the temperature rise in the air is greater than the air passes through the outer frames of the condenser. Therefore, it can be wrong to say that the most effective region of the condenser is the mid-section.

Although the significant proportion of air passes through the upper side of the condenser (Figure 4b), there is a considerable temperature rise in the air at this region. The upper side of the condenser increasing the air temperature from 316 K to 330 K has the maximum contribution on the heat rejection process.

Figure 6 shows the mean velocity, pressure and temperature of the air through the bottom cabinet. It is clearly observed that the fan creates a pressure difference at the fan section so that the air is sucked from the inlet holes of the lid. The mean velocity of the air reaches to a velocity of 2.8-3.3 m/s with the rotational motion of the fan. After the rotational flow, the air tends to exit from the bottom cabinet (Figure 4d). Therefore, the momentum of the air decreases through the downstream and the mean velocity of the air decreases rapidly when it passes over the condenser and compressor. The mean temperature of the air increases at the condenser section due to the heat rejection in the condenser. The mean temperature of the air increases up to 321 K at the exit of the condenser. During the heat rejection process, there is no significant change in the mean pressure of the air. After the condenser section, first the temperature of the air decreases to 319 K, then it increases. It is seen that the discharge air flow from the condenser contributes to the cooling of the hermetic compressor.

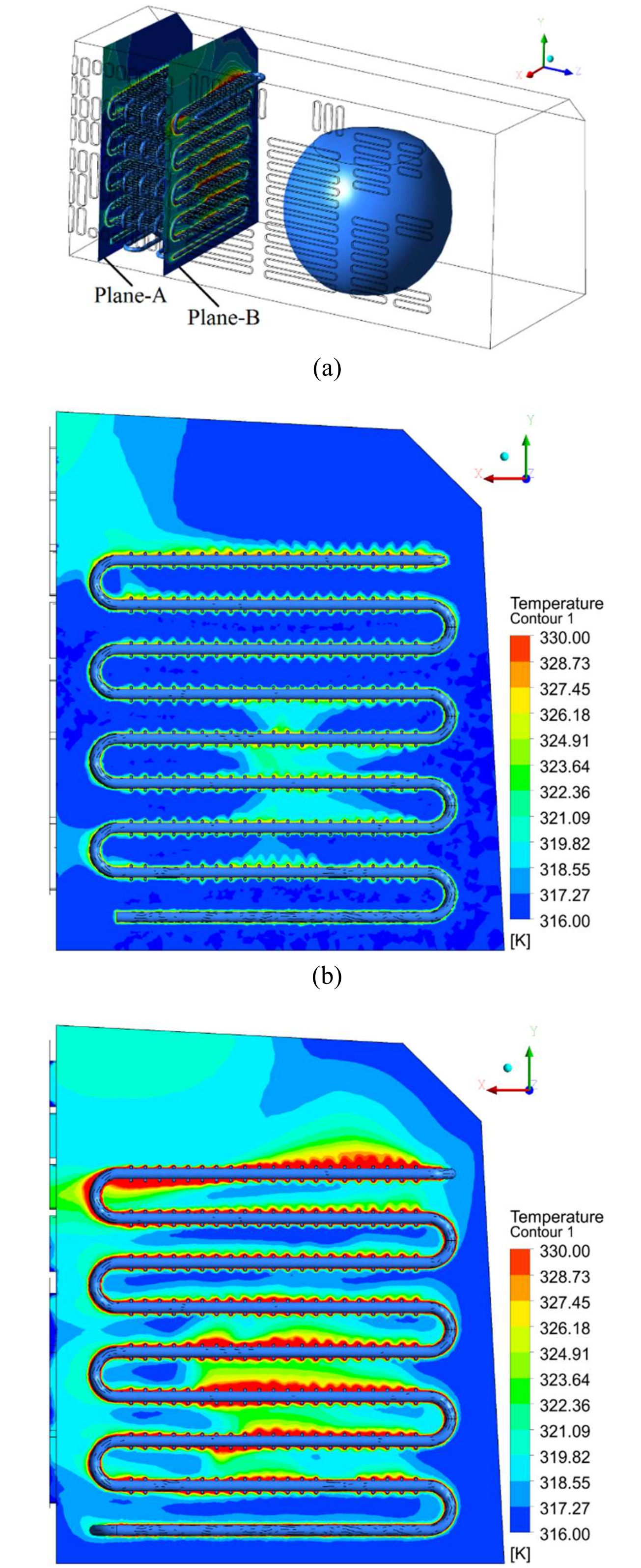

Figure 5 Heat rejection process in the condenser (a) 3-D view (b) detail of Plane-A (c) detail of Plane-B

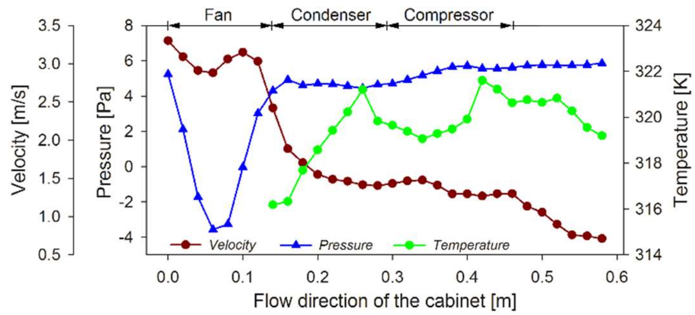

Figure 6 Mean velocity, pressure and temperature of the air in the cabinet

CONCLUSION

The study analyzed the bottom cabinet of a household refrigerator by numerically. The fluid motion sucked by the fan and the heat rejection process in the condenser are examined. The compressor and the protective lid are taken into consideration to simulate the actual cabinet. The total amount of heat transfer, temperature, pressure, velocity gradients and streamlines are studied in detail. The main conclusions and the contributions to the literature for these conditions can be summarized as follows:

- The effective region of the fan is the end points of the fan blades. The center of the fan cannot create a significant velocity field.
- Due to the rotational air flow, the mid-section of the condenser cannot be cooled effectively by the fan. Thus, the number of wires on the tubes in the mid-section of the condenser and the cost can be reduced.
- The presence of the holes on the lid between the fan and the condenser cause a pre-exit of the air before it is not reach the condenser. The cancellation of these holes can increase the rate of heat rejection in the condenser.
- For the further studies, more attention should be paid on the designing of the fan and the location of the holes on the lid for the effective cooling process of the condenser.

ACKNOWLEDGMENT

This research was performed under the Santez Project (00865-STZ.2011-1). The authors would like to thank the Ministry of Science, Industry and Technology and the Research and Development Department of the Arçelik A.Ş. Eskişehir Refrigerator Plant for their support.

NOMENCLATURE

c_p	constant pressure specific heat of air, J/kgK
f	frequency of the compressor, Hz
G	turbulence kinetic energy
h	enthalpy, kJ/kg
k	turbulence kinetic energy

k_{air}	thermal conductivity of air, W/mK
$\dot{m}$	mass flow rate, kg/s
P	pressure, Pa
$\dot{Q}$	heat transfer rate, W
S	strain rate, s^{-1}
S_k	strain rate for k, s^{-1}
S_ε	strain rate for ε, s^{-1}
T	temperature, K
u	mean velocity of the fluid, m/s
V	volume, m^3
$\dot{V}$	volume flow rate, m^3/s
$\dot{W}_{comp}$	power of the compressor, W
x,y,z	coordinates
Y_M	fluctuating dilatation

Greek letters

σ_k	turbulent Prandtl number for k
σ_ε	turbulent Prandtl number for ε
ρ	density, kg/m^3
ε	energy dissipation rate, m^2/s^3
μ	laminar viscosity, kg/ms
μ_t	turbulent viscosity, kg/ms
v	kinematic viscosity, m^2/s
η	dimensionless strain rate

REFERENCES

[1] Erbay, L B, Yılmaz, H. 2013, Numerical Investigation of the Performance of a Typical Condenser, Proceedings of the CLIMAMED VII., Mediterranean Congress of Climatization, pp.304-312, 3-4 October, Istanbul, Turkey.

[2] Uğurlubilek, N, Erbay L B Doğan B. 2013, Numerical investigation of the pressure drop characteristics in a heat exchanger, Proceedings of the 19. ULIBTK, pp.386-391, 9-12 September, Samsun, Turkey.

[3] Uğurlubilek, N, Erbay L B, Doğan B. 2013, Numerical investigation of the heat transfer characteristics in a heat exchanger, Proceedings of the 19. ULIBTK, pp.380-385, 9-12 September, Samsun, Turkey.

[4] Uğurlubilek, N, Erbay, L B, Altun, Ö, Doğan, B. 2013, Compact Heat Exchangers, Mühendis ve Makina 54(646), 37-48.

[5] Yılmaz, H. 2012, Numerical investigation of the performance of condenser in domestic refrigerator, Master Dissertation, Eskisehir Osmangazi University, Institute of Science, Turkey.

[6] Radermacher, R, Kim, K. 1996, Domestic refrigerators: Recent developments, International Journal of Refrigeration 19(1), 61-69.

[7] Bansal, P, Vineyard, E, Abdelaziz, O. 2011, Advances in household appliances - A review, Applied Thermal Engineering 31, 3748-3760.

[8] Gonçalves, J M, Melo, C, Hermes, C J L. 2009, A semi-empirical model for steady-state simulation of household refrigerators, Applied Thermal Engineering 29, 1622–1630.

[9] Hermes, C J L, Melo, C. 2008, A first-principles simulation model for the start-up and cycling transients of household refrigerators, International Journal of Refrigeration 31, 1341–1357.

[10] Hermes, C J L, Melo, C, Knabben, FT, Gonçalves, JM. 2009, Prediction of the energy consumption of household refrigerators and freezers via steady-state simulation, Applied Energy 86, 1311–1319.

[11] Laguerre, O, Amara, S B, Moureh, J, et al. 2007, Numerical simulation of air flow and heat transfer in domestic refrigerators, Journal of Food Engineering 81, 144–156.

[12] Yang, K -S, Chang, W -R, Chen, I -Y, et al. 2010, An investigation of a top-mounted domestic refrigerator, Energy Conversion and Management 51, 1422–1427.

[13] Melo, C, Hermes, C J L. 2009, A heat transfer correlation for natural draft wire-and-tube condensers, International Journal of Refrigeration 32, 546–555.

[14] Borges, B N, Hermes, C J L, Gonçalves, J M, et al. 2011, Transient simulation of household refrigerators: A semi-empirical quasi-steady approach, Applied Energy 88, 748–754.

[15] Lin, E, Ding, G, Zhao, D, et al. 2011, Dynamic model for multi-compartment indirect cooling household refrigerator using Z-transfer function based cabinet model, International Journal of Thermal Sciences 50, 1308–1325.

[16] Negrao, C O R, Hermes, C J L. 2011, Energy and cost savings in household refrigerating appliances: A simulation-based design approach, Applied Energy 88, 3051–3060.

[17] Waltrich, M, Hermes, C J L, Melo, C. 2011, Simulation-based design and optimization of refrigeration cassettes, Applied Energy 88, 4756–4765.

[18] Mitishita, R S, Barreira, E M, Negrao, C O R, et al. 2013, Thermoeconomic design and optimization of frost-free refrigerators, Applied Thermal Engineering 50, 1376–1385.

[19] Mansour, R B, Ouzzane, M, Aidoun, Z. 2014, Numerical evaluation of ejector-assisted mechanical compression systems for refrigeration applications, International Journal of Refrigeration. DOI: 10.1016/j.ijrefrig.2014.04.010.

[20] Ansys Fluent, User’s Guide, Release 14.0.1, Ansys Inc., Lebanon, New Hampshire.