Energy Conversion and Management 51 (2010) 1422–1427

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

An investigation of a top-mounted domestic refrigerator

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ARTICLE INFO

Article history:
Received 25 June 2009
Accepted 29 January 2010
Available online 26 February 2010

Keywords:
Top-mount
Temperature uniformity
Refrigerator

ABSTRACT

A detailed numerical simulation of the performance of household refrigerator having a top-mount configuration is made in this study, and an experiment is carried out with a real refrigerator for verification. The results indicate the temperature distributions from the numerical simulation are qualitatively in line with the experimental measurements. Cyclic temperature variations occur amid freezer, refrigerating, and vegetable compartments due to on/off control strategy. However, it is found that the temperature variation for freezer and refrigerating compartment are in phase with each other while the temperature variation in vegetable compartment is roughly out-of-phase with the other two compartments. The simulations show that the design of air duct and its locations may impose a detrimental role on the temperature uniformity within the refrigerator, yet the airflow is strongly influenced by gravity. It is also found that the refrigerating compartment possesses the worst temperature non-uniformity. For improving the temperature non-uniformity, a modified design incorporating the air duct design with appropriate locations of the inlet openings in the freezer and refrigerating compartment is proposed. Though these modifications the maximum temperature difference and the root mean-square of temperature variation in refrigerating compartment, had been reduced from 7.17 °C to 3.57 °C and from 3.17 °C to 1.55 °C, respectively.

1. Introduction

For the past several decades, the amount of household refrigerator had increased dramatically for the ubiquitous needs of clean and fresh foods. However, the rise of great demand of household refrigerator inevitably consumes a lot of electricity when pursuing a higher living standard. As far as economics and environment is concerned, it is therefore critically important to make the household refrigerator more energy efficient. Consequently many countries had entailed standards [1–3] to encourage consumers to buy more energy efficient products to lower greenhouse gas emissions and more effectively use of energy. It has been identified that the performance of household refrigerator depends strongly on temperature and air distribution inside the storage chamber [4]. Therefore, many subsequent investigations had been made to improve and to seek for the optimal design of such household refrigeration systems. The most common ways for detailed investigations include Particle Image Velocimetry (PIV), and Computational Fluid Dynamics (CFD). However, the major problem associated with Particle Image Velocimetry [5–7] is probably with its complex measurement techniques and difficult to de-couple the effect between air velocity and temperature.

In the meanwhile, there had been some numerical approaches or experimental studies concerns on the optimal design of refrigeration system [8–11]. In practice, detailed information such as temperature or velocity distribution within household refrigerator is unavailable in typical experimental studies. Hence, it would be easier to resort to numerical modeling due to its simplicity and comparatively cost-effective feature. Through the simulation, one can obtain related detailed temperature/velocity distribution subject to design concerns. As a consequence, iterative process for prototype development can be shortened, and the Computational Fluid Dynamics method is therefore regarded as a powerful design tool, and some recently successful designs had been reported [12,13] through these tools. However, for making CFD even more powerful and reliable, it is often desirable for a constant verification and validation with the experimental data. Hence, the aim of the present work is to combine the efforts from CFD tool and experimental data, and to come up a better design modification of the household refrigerator.

2. Materials and methods

2.1. Experimental set-up

In this study, a very common top-mounted domestic refrigerator with three doors design is used as the base model for

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doi:10.1016/j.enconman.2010.01.016

Nomenclature

g	gravitational acceleration, m s^{-2}
h	convection heat transfer coefficient, W m^{-2} K^{-1}
k	thermal conductivity, W m^{-1} K^{-1}
L	characteristic length, m
Nu	Nusselt number
Pr	Prandtl number
Ra	Rayleigh number
RH	relative humidity, %
T_{db}	dry bulb temperature, K
T_s	surface temperature, K
T_∞	fluid temperature, K
$\Delta T_{r,max}$	maximum temperature variation within refrigerating compartment, K
$\Delta T_{r,mean}$	root mean square temperature variation within refrigerating compartment, K

Greek symbols

α	thermal diffusivity, m^2 s^{-1}
β	volumetric thermal expansion, K^{-1}
ν	kinematic viscosity, m^2 s^{-1}
ρ	density, kg m^{-3}

simulation/experimentation. The cross-sectional view of the refrigerator including three separate compartments, airflow path and temperature designation within the refrigerator is schematically shown in Fig. 1. The refrigerator has three compartments encompassing a freezer, a refrigerating and a vegetable compartment. As shown in figure, the cold air is first circulated from the freezer and then into the refrigerating compartment, and finally moved into vegetable compartment. The advantage of the top-mounted domestic refrigerator is its simple structure can effectively distribute lower temperature airflow into refrigerating and vegetable compartments by gravitation. The experiment apparatus is based on ISO 7371 [1] setup to measure the temperature characteristics of the household refrigerator. The refrigerator is operated at an ambient temperature of $T_{db} = 30 \pm 1$ °C and RH = 75 ± 5%. Where T_{db} is the dry bulb temperature and RH denotes the relative humidity, and the mean freezer compartment temperature is maintained at -18 ± 0.5 °C. When the operation reaches steady state, the designated temperatures in the refrigerator (shown in Fig. 1, T_1–T_5) are measured and recorded. During the isothermal test, the variation of these thermocouples was within 0.2 °C. In addition, all the thermocouples were pre-calibrated by a quartz thermometer having 0.01 °C precision. The accuracies of the calibrated thermocouples are of 0.1 °C. All the data signals are collected and converted by a data acquisition system (a hybrid recorder). The data acquisition system then transmitted the converted signals through GPIB interface to the host computer for further operation.

2.2. Mathematical modeling

The three doors of the top-mounted domestic refrigerator is of square configuration with a length/width/height of 746 mm/805 mm/1835 mm, and its rated storage volume is 620 l. In this study, simulation is carried out by a commercially available software (EFD.Lab). The detailed structure layout of the refrigerator and the corresponding mesh structure are schematically shown in Figs. 2 and 3, respectively. The basic principles in flow module are originated from the conservation of mass and momentum (Navier–Stokes equations). In the meantime, EFD.Lab employs trans-

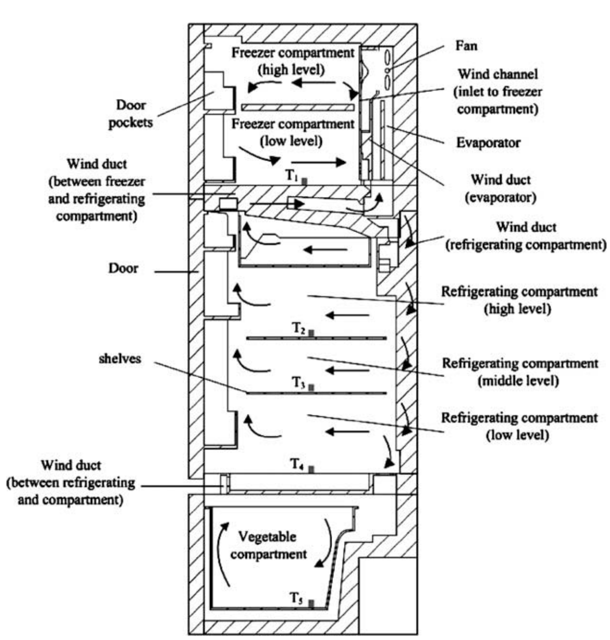

Fig. 1. Compartment and airflow in top-mounted domestic three doors refrigerator.

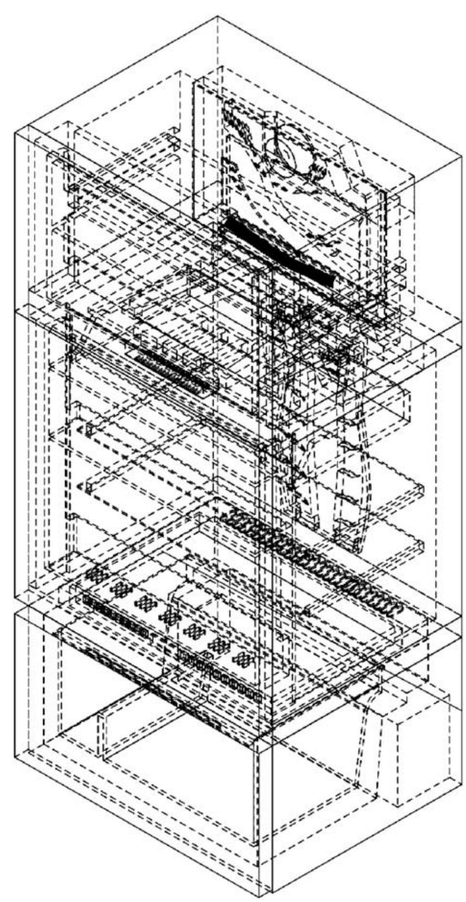

Fig. 2. Detailed structure of the simulated refrigerator.

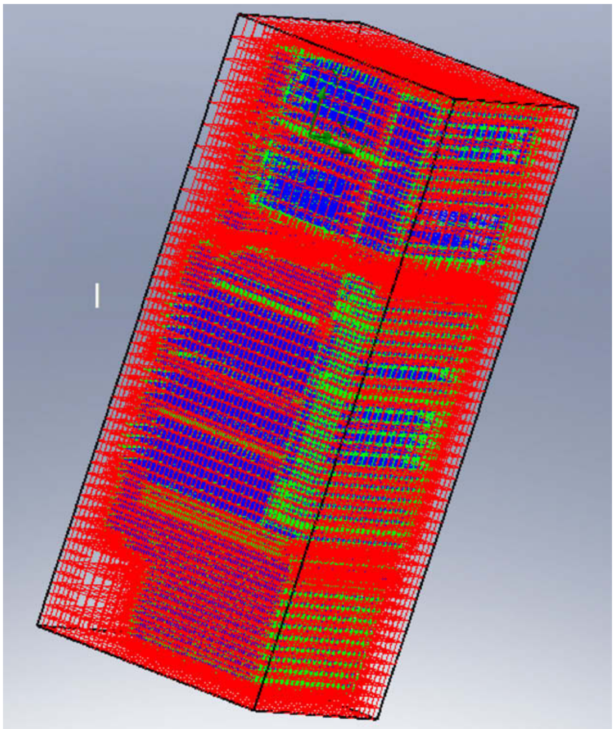

Fig. 3. Grid system for the refrigerator.

port equations for the turbulent kinetic energy and its dissipation rate (k–ε model) for turbulent flow situation [14].

The thermal boundary conditions are based on measured outside surface temperature on refrigerator. Generally, it took less than two hours for the measured temperature to reach the steady state as shown in Fig. 4. In operating refrigeration system, the condenser tube is arranged around the wall of refrigerator, yielding a higher outside surface temperature than its ambient temperature. Hence, natural convection is regarded as the responsible heat transfer mechanism outside the refrigerator surface. For a vertical plate, the natural convective correlation recommended by Churchill and Chu [15] is:

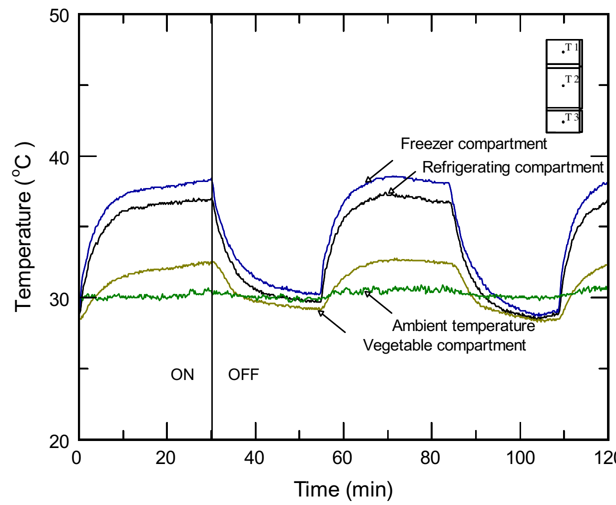

Fig. 4. Variation of the outside surface temperatures of the tested refrigerator vs. time.

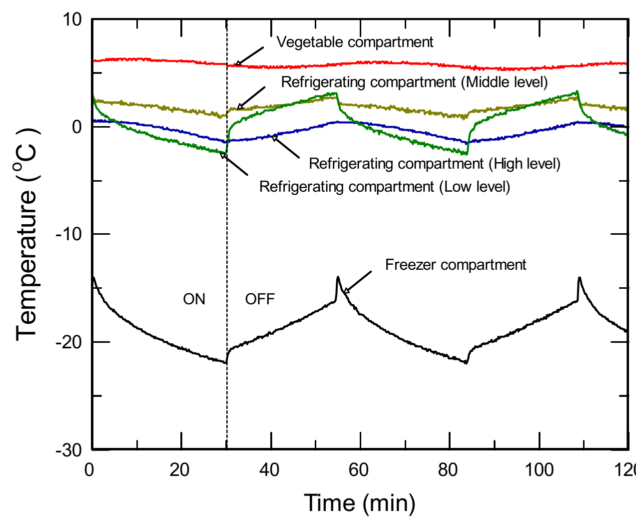

Fig. 5. Variation of inside air temperatures inside the refrigerator vs. time.

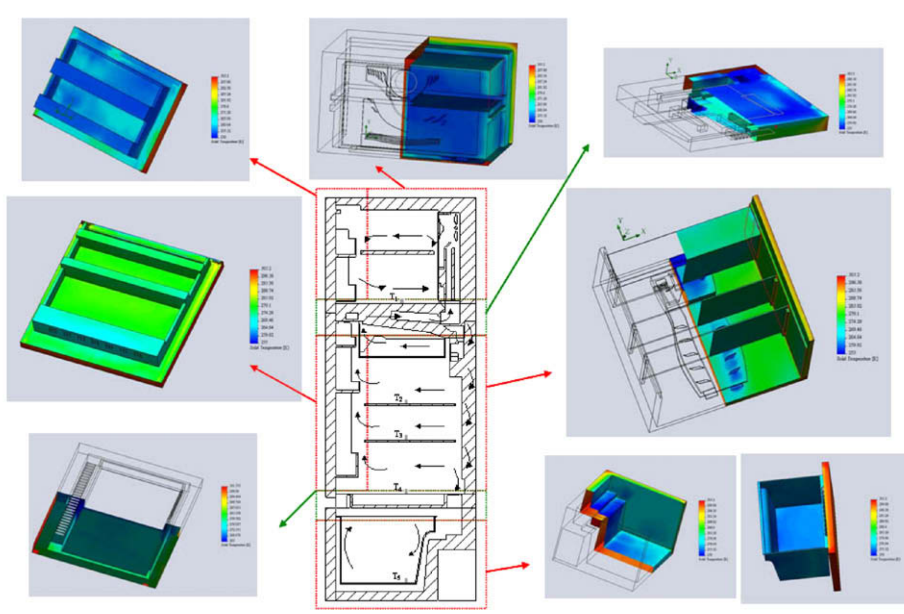

Fig. 6. The temperature distribution on solid surface inside the refrigerator.

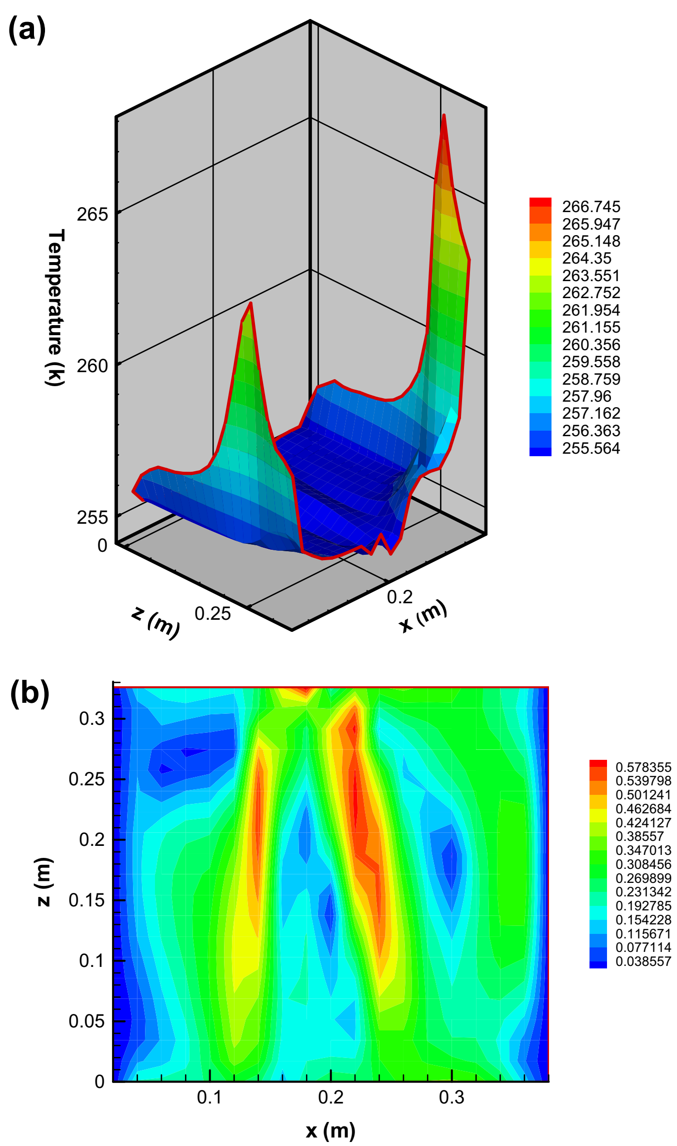

Fig. 7. (a) Temperature and (b) velocity distribution in high level of freezer compartment.

$$\overline{\mathrm{Nu}_L} = \left\{0.825 + \frac{0.387\mathrm{Ra}_L^{1/6}}{[1 + (0.492/\mathrm{Pr})^{9/16}]^{8/27}}\right\} \tag{1}$$

$$\bar{h} = \frac{\overline{\mathrm{Nu}_L} \cdot k}{L} \tag{2}$$

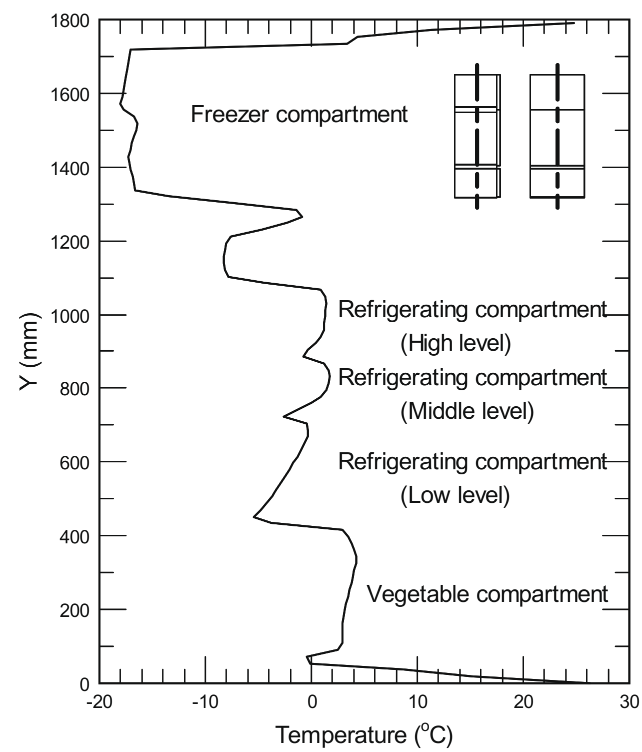

Fig. 8. Temperature distribution on the center line of the refrigerator.

where $\bar{h}$ is the convection heat transfer coefficient, k is the thermal conductivity, Nu is the Nusselt number, Pr is the Prandtl number and Ra is the Rayleigh number. The Rayleigh number can be obtained from the following equation:

$$\mathrm{Ra}_L = \frac{g\beta(T_s - T_\infty)L^3}{\nu\alpha} \tag{3}$$

where g is the gravitational acceleration, L is the characteristic length, T_s is the surface temperature, T_∞ is the ambient fluid temperature, α is the thermal diffusivity, β is the volumetric thermal expansion and ν is the kinematic viscosity.

Based on foregoing equations and experimental results, the convection heat transfer coefficient is estimated to be 2.5 W m^{-2} K^{-1}, and P–Q curve of the circulation fan (AC fan of 8412NGH) inside the refrigerator was included in the software database, and a heat generation rate of −135 W from the evaporator based on experimental results is implemented into simulation. The insulation material is polyurethane (PU) with a specific heat of 1200 J kg^{-1} K^{-1}, density of 30 kg m^{-3} and thermal conductivity of 0.019 W m^{-1} K^{-1}. The working fluid is air.

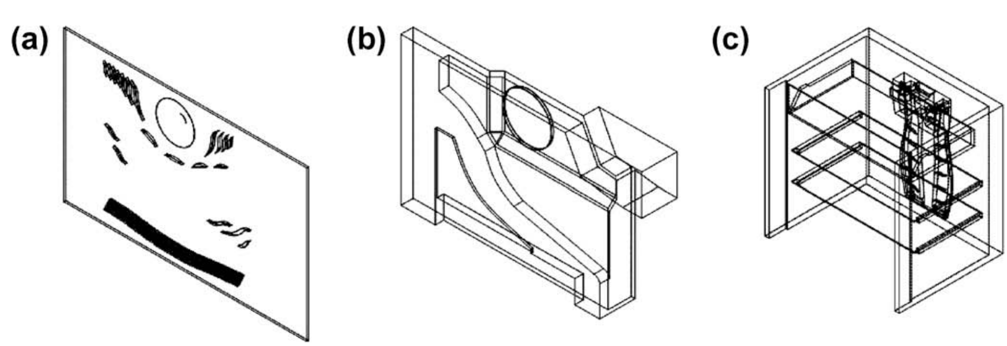

Fig. 9. The original air duct design of: (a) inlet of freezer compartment (b) evaporator and (c) refrigerating compartment.

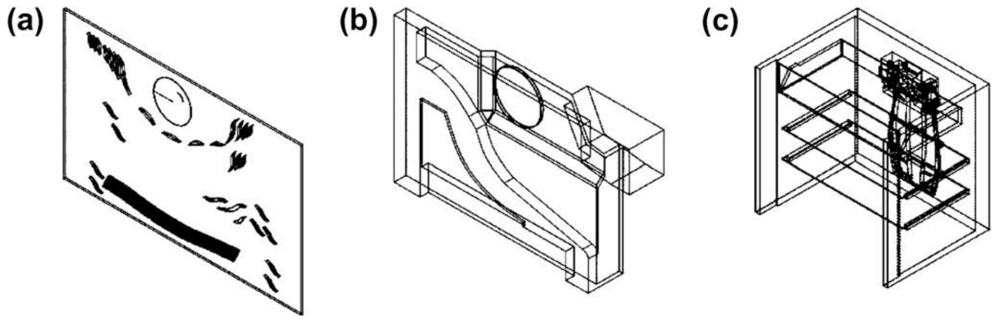

Fig. 10. The modified air duct design of: (a) inlet of freezer compartment (b) evaporator and (c) refrigerating compartment.

3. Results and discussion

A typical test result for temperature variation vs. time inside the three compartments is shown in Fig. 5. Notice that the ambient temperature is maintained at $T_{db} = 30 \pm 1$ °C. As expected, the temperature variation in each compartments undergoes a periodic change due to the on/off control strategy of the compressor. In the meantime, the temperature variation of freezer compartment is the highest amid the three compartments, followed by the refrigerating compartment and the vegetable compartment shows the smallest variation of about 1–2 °C. It is interested to know that despite the temperature of the three compartments all shows periodic change, the temperature variations are not in phase with each other. As seen in Fig. 5, the temperature variation in freezer compartment and refrigerating compartment are actually in phase while the temperature variation in the vegetable compartment, though quite small, shows out-of-phase characteristic when compared to the other two compartments. The difference arises from the airflow circulating methods being used. For the freezer and refrigerating compartment, cold air flow from the evaporator is delivered by axial fan while natural circulation is used in the vegetable compartment. It is well known that the heat transfer coefficient for forced air flow is roughly an order of magnitude higher than that of natural convection. In this regard, the temperature response in the vegetable compartment falls considerably behind the other two compartments and exhibits an out-of-phase characteristic.

Temperature variation within the refrigerator is very crucial to the preserved food quality. Normally better temperature/velocity uniformity engenders better preserved quality. In thus sense, it is therefore quite essential to examine the relevant distribution of the present top-mount refrigerator. Initially it is suspected that considerable temperature differences might appears at the corners, and the simulation results shown in Fig. 6 confirms that expectations. In fact, most of the wall temperatures from the corners of all compartments suffer from this non-uniformity as depicted in Fig. 6. It appears that the temperature non-uniformity is related to the velocity non-uniformity. For further understanding about the significant variation of temperature uniformity caused by the worse non-uniformity region, the temperature and airflow velocity at upper level of freezer level are plotted in Fig. 7 for illustration. The region is in front of the supplied fan where a large amount of cool air is forced into freezer compartment to lower the temperature. Subsequently the airflow flows to the lower level of the freezer compartment and a significant difference in temperature over 10 °C appeared at the corner region just in front of freezer compartment inlet. Some further detailed variation from the freezer compartment down to vegetable compartment is shown in Fig. 8. It is clearly seen in the figure that the worse temperature uniformity occurred in refrigerating compartment. The maximum temperature difference is more than 6 °C between upper and lower level of refrigerating compartment. Note that the lowest tempera-

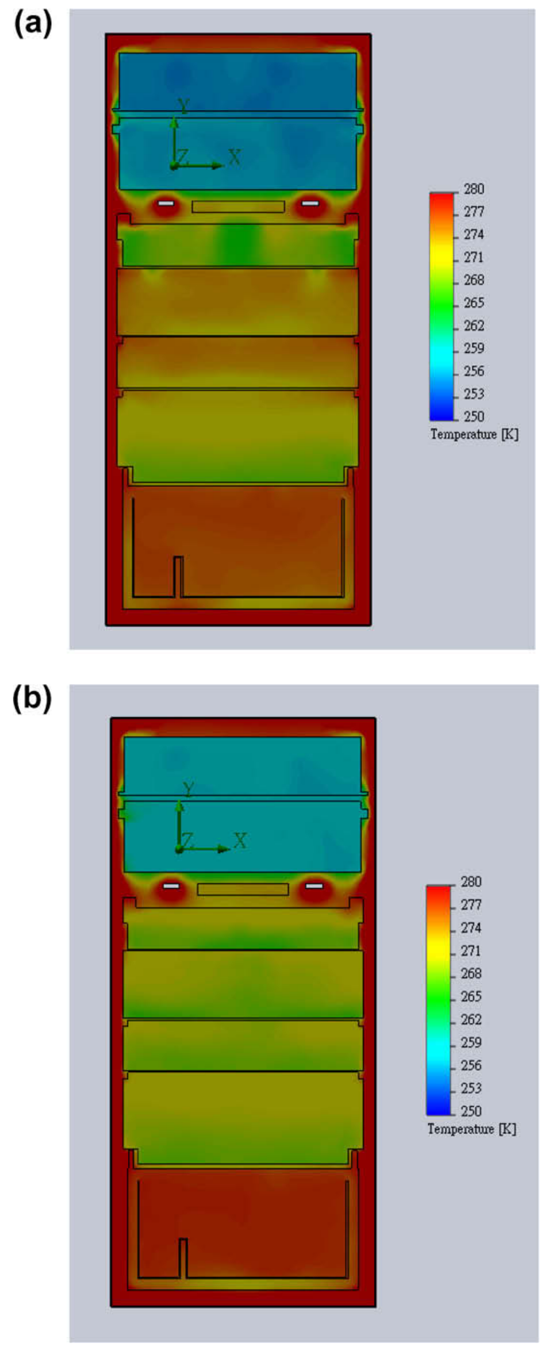

Fig. 11. Temperature distribution at the symmetry plane of: (a) original and (b) modified air duct design in refrigerator.

ture is at the lower level of the refrigerating compartment. This is probably due to stratification of the air flow within refrigerating compartment. In the meantime, additional sleeve in the refrigerating compartment may impair the air flow distribution and cause more pronounced temperature non-uniformity. The experimental results also reveal that the lower level of refrigerating compartment encompasses the lowest temperature region. From the forgoing discussion, the temperature uniformity is significantly affected by gravity and air flow path in refrigerator. To improve the temperature distribution in the refrigerating compartment, we had proposed a modified the design of air flow channel to convey air flow. The original air flow channel including inlet to freezer compartment, air flow channel in refrigerating compartment, and the flow channel of evaporator as show in Fig. 9, and a modified structure is shown in Fig. 10. The major difference between the modified structure and the original one is the air duct design around evaporator compartment and the locations of the inlet openings of the air discharge for the freezer and refrigerating compartment. The present modified structures alter the opening orientation in refrigerating compartment. Through this design, part of the cooled air flow is forced to flow form central direction to side direction. Fig. 11 shows the simulated temperature distribution amid the original and modified design. The simulation results indicate a significant improvement of thermal uniformity from modified design. Actually, the temperature non-uniformity, in terms of maximum temperature difference ($\Delta T_{r,max}$) and the root mean-square of temperature variation $\left(\sqrt{\frac{\sum(T-T_{average})^2}{N}}, \Delta T_{r,mean}\right)$, had been reduced from 7.17 °C to 3.57 °C ($\Delta T_{r,max}$) and from 3.17 °C to 1.55 °C ($\Delta T_{r,mean}$) in refrigerating compartment, respectively. With this simple alternation, one can realize the importance of management of air flow path.

4. Conclusion

This study presents a detailed numerical simulation of the performance of household refrigerator having a top-mount configuration, and the simulation is then substantiated with a real refrigerator by ISO 7371 which indicates the temperature distributions from the numerical simulation are qualitatively in line with the experimental measurements. The measured results indicated a cyclic variation of temperature amid freezer, refrigerating, and vegetable compartments due to on/off control strategy. However, it is found that the temperature variation for freezer and refrigerating compartment are in phase with each other while the temperature variation in vegetable compartment is roughly out-of-phase with the other two compartments. This is actually related to the different heat transfer modes between these compartments. The results of simulation also show that the design of air duct and its locations may impose a detrimental role on the temperature uniformity within the refrigerator, yet the airflow is strongly influenced by gravity. It is also found that the refrigerating compartment possesses the worst temperature non-uniformity. For improving the temperature non-uniformity, a modified design incorporating the air duct design around evaporator compartment and the locations of the inlet openings of the air discharge for the freezer and refrigerating compartment is proposed. Though these modifications the maximum temperature difference and the root mean-square of temperature variation in refrigerating compartment, had been reduced from 7.17 °C to 3.57 °C and from 3.17 °C to 1.55 °C, respectively.

Acknowledgements

The authors are indebted to the financial support from the Bureau of Energy and Department of Industrial Technology, the Ministry of Economic Affairs, Taiwan.

References

[1] ISO 7371—1985 (E). Performance of household refrigerating appliances—refrigerators with or without low temperature compartment. International standard ISO 7371: 1985/Amendment 1; 1987.
[2] US Code of Federal Regulations (CFR). Uniform test method for measuring the energy consumption of electric refrigerators and electric refrigerator-freezers, Appendix A1 to sub-part B of part 430—energy conservation program for consumer products. 10 CFR; 1990. p. 49–54 [chapter II].
[3] Chinese National Standards of Chinese Taipei. Electric refrigerators and freezers. National Test Standard CNS 2062; 1990.
[4] Gupta JK, Ram Gopal M, Chakraborty S. Modeling of a domestic frost-free refrigerator. Int J Refrig 2005;28:774–83.
[5] Lee IS, Baek SJ, Chung MK, Rhee D. A study of air flow characteristics in the refrigerator using PIV and computational simulation. J Flow Visual Image Process 1999;6:333–42.
[6] Lacerda VT, Melo C, Barbosa JR, Duarte POO. Measurements of the air flow field in the freezer compartment of a topmount no-frost refrigerator: the effect of temperature. Int J Refrig 2005;28:774–83.
[7] Laguerre O, Ben Amara S, Charrier-Mojtabi MC, Lartigue B, Flick D. Experimental study of air flow by natural convection in a closed cavity: application in a domestic refrigerator. J Food Eng 2008;85:547–60.
[8] Chen ZJ, Lin WH. Dynamic simulation and optimal matching of a small-scale refrigeration system. Int J Refrig 1991;14:329–35.
[9] Bansal PK, Wich T, Browne MW. Optimization of egg-crate type evaporators in domestic refrigerators. Appl Therm Eng 2001;21:751–70.
[10] Chen L, Wu C, Sun F. Heat transfer effect on the specific cooling load of refrigerators. Appl Therm Eng 1996;16:989–97.
[11] Graviss K, Zurada J. A neural network controller for optimal temperature control of household refrigerators. Intell Automat Soft Comput 1998;4:357–72.
[12] James SJ, Evans J, James C. A review of the performance of domestic refrigerators. J Food Eng 2008;87:2–10.
[13] Smale NJ, Moureh J, Cortella G. A review of numerical models of airflow in refrigerated food applications. Int J Refrig 2006;29:911–30.
[14] Mentor Graphics. EFD.Lab 8 fundamentals; 2007.
[15] Churchill SW, Chu HHS. Correlating equations for laminar and turbulent free convection from a vertical plate. Int J Heat & Mass Transfer 1975;18:1323–9.