



The mass and heat transfer process through the door seal of refrigeration☆



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ABSTRACT

As one of the main reasons causing leakage heat load in a refrigerator, mass and heat transfer through refrigerator door seal is of great importance to be studied. In this paper, a model is presented for numerical simulation of mass and heat transfer process through refrigerator door seal, and an experiment apparatus is designed and set up as well for comparison. A two-dimensional model and tracer gas method are used in simulation and experiment, respectively. It can be found that the relative deviations of air infiltration rate between the simulated results and experimental results were less than 1%, and the temperature difference errors at two special points of the door seal were less than 2.03 °C. In conclusion, the simulated results are in good agreement with the experimental results. This paper initially sets up a model that can accurately simulate the heat and mass transfer through the refrigerator door seal, and the model can be used in refrigerator door seal optimization research in the follow-up study.

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1. Introduction

Refrigerator heat load consists of latent heat load, electric heat load and leakage heat load. Among these, leakage heat load is the biggest one, accounting for about 75%. Although refrigerator door seal can decrease leakage heat load much, there still exist mass and heat transfer through the gap between seal and chamber, which results in a lot of energy loss. In order to decrease leakage heat load as much as possible, it is of great importance to accurately predict the mass and heat transfer through door seal.

In recent years, a few related studies have been carried out. Afonso *et al.* [1] introduced tracer gas method to quantify the rate of air infiltration through refrigerator door seal. Kim *et al.* [2] simulated the temperature field of refrigerator door seal by Fluent. Han *et al.* [3] studied four kinds of common refrigerator door seal by using two-dimensional CAE models. Li *et al.* [4] calculated the heat leakage amount through seal on the basis of Fourier law. However, the gap between seal and chamber was not seriously considered in their models, so that mass transfer process had not been taken into consideration.

The purpose of this paper is to present a model by which the mass and heat transfer process through refrigerator door seal can be simulated with high accuracy. To verify the reliability of the proposed model, an experiment apparatus was also set up for comparison.

2. Method of Simulation

For simplification, a two-dimensional model of refrigerator door seal was set up for simulation in Fluent. The computational domain, boundary conditions and mathematical foundations of the model are as follows.

2.1. Computational domain

The simulation model is composed of seal, door body and chamber body (Fig. 1). The seal is assembled between the chamber body and door body, while a gap, the passageway of air infiltration, may occur between the seal and chamber body because of mechanical assembling (Fig. 2).

As shown in Fig. 1(b), anti-dew pipes are set up near the seal in the freezer chamber to prevent the door from being frozen. Table 1 tabulates the physical parameters of various materials for different parts in the model shown in Fig. 1.

2.2. Boundary conditions

Pressure boundaries are used for the inlet and outlet of flow domain, while the specific parameters (temperature and pressure) are listed in Table 2. The wall of anti-dew pipes in the freezer chamber is isothermal at 38 °C. Besides, Wall 1, Wall 2 and Wall 6 are assumed in the third boundary condition, and the temperatures and convection heat transfer coefficient values are presented in Table 3.

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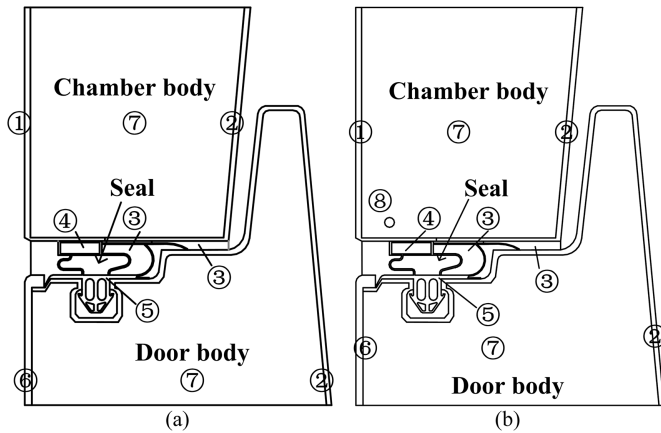


Fig. 1. Calculation models of each refrigeration chamber (a) refrigeration chamber (b) freezer chamber. ① Steel; ② ABS; ③ Air; ④ Magnet; ⑤ PVC; ⑥ Aluminum; ⑦ PU-foam; ⑧ Copper.

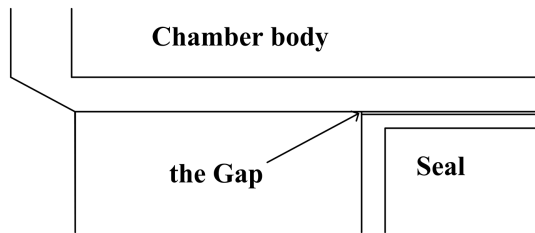


Fig. 2. The gap between seal and chamber body.

Table 1
Physical parameters of the materials in the models

No.	Material	$\rho/\text{kg}\cdot\text{m}^{-3}$		$\lambda/\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$		$c_p/\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$
		Refrigeration chamber	Freezer chamber	Refrigeration chamber	Freezer chamber	
①	Steel	8030		16.27		502.48
②	ABS	1050		0.191		1600
③	Air	1.2176	1.2979	0.02566	0.02432	1005
④	Magnet	3700		0.5		1000
⑤	PVC	1300		0.14		1839
⑥	Aluminum	2719		202.4		871
⑦	PU-foam	30		0.026		1800
⑧	Copper	8978		387.6		381

Table 2
The inlet and outlet conditions of the models

Case	Inlet		Outlet	
	$T/^\circ\text{C}$	p/Pa	$T/^\circ\text{C}$	p/Pa
Refrigeration chamber	20	0	5	-10
Freezer chamber	20	0	-18	-100

Note: Relative pressure values are shown in the table.

Table 3
The boundary conditions of Wall 1, Wall 2 and Wall 6

Wall	Boundary conditions	
	Refrigeration chamber	Freezer chamber
Wall 1 and Wall 6	$T_\infty = 293.15\text{K}, h = 7\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$	$T_\infty = 293.15\text{K}, h = 7\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$
Wall 2	$T_\infty = 278.15\text{K}, h = 6.5\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$	$T_\infty = 255.15\text{K}, h = 6.5\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$

2.3. Mathematical foundation

The simulation model of seal is developed under the following assumptions:

- The flow is laminar and incompressible.
- The two-dimensional flow is steady.
- Fluid properties are constant.
- Radiative transfer is neglected.

On the basis of the above assumptions, the flow in the gap is governed by the following equations, which are the continuity, the Navier-Stokes as well as energy equations.

(1) Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

(2) N-S equations:

$$\rho \frac{\partial u}{\partial t} + \rho \operatorname{div}(u\mathbf{U}) = \operatorname{div}(\mu \operatorname{grad} u) - \frac{\partial p}{\partial x} + S_u \quad (2)$$

$$\rho \frac{\partial v}{\partial t} + \rho \operatorname{div}(v\mathbf{U}) = \operatorname{div}(\mu \operatorname{grad} v) - \frac{\partial p}{\partial y} + S_v \quad (3)$$

(3) Energy equation:

$$\rho \frac{\partial T}{\partial t} + \rho \operatorname{div}(\vec{u}T) = \operatorname{div}\left(\frac{\lambda}{c_p} \operatorname{grad} T\right) + S_T \quad (4)$$

3. Method of Experiment

3.1. Tracer gas technique

Tracer gas technique has the advantages of simple operation, uncomplicated control and relatively high accuracy. Mainly being used in buildings for energy auditing before [5–9], this method can be illustrated as follows: after one kind of gas is introduced in the space where the air infiltration is to be measured, the CO_2 concentration of the air in the chamber over time can be measured by an appropriate gas analyzer in this work.

When the experiment refrigerator is running stably, the mass conservation equation of CO_2 inside can be expressed by,

$$\left(\frac{dc}{dt}\right) \cdot dt \cdot V = q \cdot dt + c_{\text{atm}} \cdot Q_v \cdot dt - c \cdot Q_v \cdot dt \quad (5)$$

To change the concentration of CO_2 inside, N_2 with certain quantity is introduced in the tested space at the beginning. Because there is no extra production of CO_2 in chamber, the net production rate of CO_2 (q) in the chamber is null, and we can obtain the following equation,

$$\left(\frac{dc}{dt}\right) \cdot dt \cdot V = c_{\text{atm}} \cdot Q_v \cdot dt - c \cdot Q_v \cdot dt \quad (6)$$

After integrating the terms, we can rewrite the equation as follows,

$$V(c(t_1) - c_0) = Q_v c_{\text{atm}} t_1 - Q_v \int_0^{t_1} c dt \quad (7)$$

where t_1 is test time and c_0 is the initial CO_2 concentration in the tested chamber.

Considering that the air infiltration rate Q_v and the CO_2 concentration in air c_{atm} are constant, thus we have

$$Q_v = \frac{V(c(t_1) - c_0)}{c_{\text{atm}}t_1 - \int_0^{t_1} c dt} \quad (8)$$

3.2. Test facilities

Fig. 3 illustrates the experimental apparatus used in this study. On the basis of ASTM standard about tracer gas method [10–13], the apparatus fundamentally are consisted of a refrigerator section, a gas supply section and a sampling section.

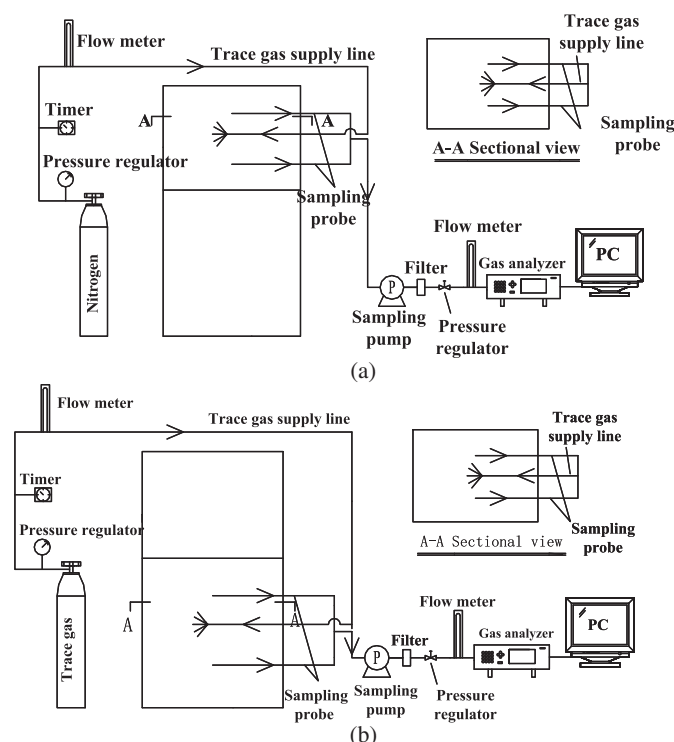


Fig. 3. Schematic view of experimental system of air infiltration rate measurement on (a) refrigeration chamber and (b) freezer chamber.

In experiment, Frestec BCD217CHS2B direct cooling refrigerator was selected with one refrigeration chamber (internal volume of 131 L) and one freezer chamber (internal volume of 86 L). In order to ensure that there is no leakage through other refrigerator structures, we sealed the drainage hole that leads the condensate inside to a drying pan sitting on the top of the compressor. The air samples were collected simultaneously on four sampling points inside the chamber. Moreover, average value of the four is taken as the experimental value, for an accurate measurement of CO_2 concentration.

The CO_2 concentration was measured by a IRME-S infrared gas analyzer. The measurement range of the gas analyzer is $0\text{--}5 \times 10^{-3} \text{ m}^3 \cdot \text{m}^{-3}$. In addition, the repeatability and zero/span shift errors of the gas analyzer are 0.5% and 1% of the full scale value, respectively. The resolution of the instrument is $1 \times 10^{-6} \text{ m}^3 \cdot \text{m}^{-3}$.

At the same time, the temperatures at seal edge and the corner of chamber body (Fig. 4) were measured by calibrated thermocouples (200 μm diameter, precision $\pm 0.2^\circ\text{C}$).

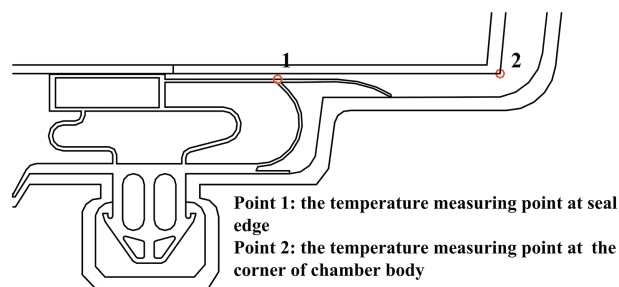


Fig. 4. The location of thermocouples installed.

All thermocouples were connected to a data acquisition system that read all temperatures in interval of 20 s.

4. Results and Discussion

4.1. Air infiltration analysis

For the reason that a two-dimensional model was applied in this study, the simulated results were expressed in per unit length, and the experimental results also are set in per unit length for unity. Table 4 listed the simulated and measured air infiltration rates through seal, together with the deviations between them.

Table 4

The simulated results and experimental results of air infiltration rate through seal

	Refrigeration chamber $q_m/\text{kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$	Freezer chamber $q_m/\text{kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$
Simulated results	3.33×10^{-6}	1.89×10^{-6}
Experimental results	3.38×10^{-6}	1.86×10^{-6}
Relative deviations/%	0.75	0.8

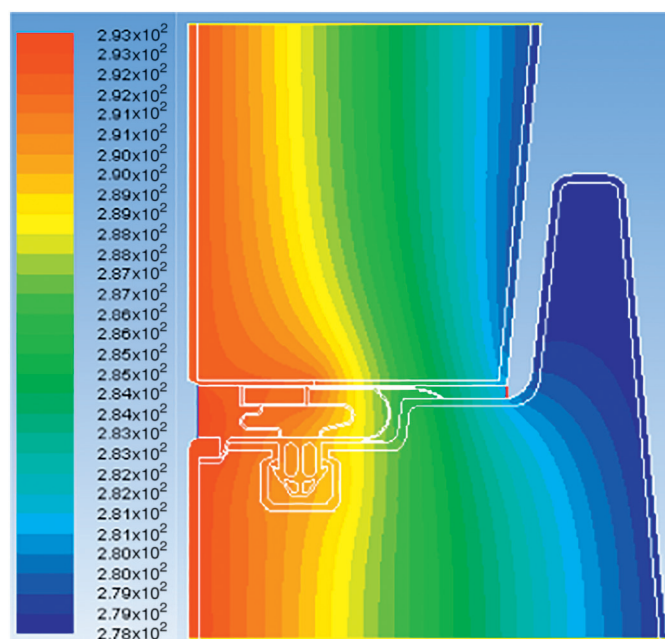
As to the simulated results shown in Table 4, the rate of the air infiltration on the refrigeration chamber and that on the freezer chamber were $3.33 \times 10^{-6} \text{ kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$ and $1.89 \times 10^{-6} \text{ kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$, respectively. When the refrigerator was running stably, the corresponding experimental values were $3.38 \times 10^{-6} \text{ kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$ and $1.82 \times 10^{-6} \text{ kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$, respectively. Both of values were extremely closed to the experimental results, and the relative deviations on the refrigeration chamber and on the freezer chamber were 0.75% and 0.8%, respectively. These results indicated that the simulated results were in good agreement with the experimental results and the model proposed in this paper could calculate the air infiltration through seal accurately.

To be more detailed, the air infiltration rate on the refrigeration chamber was approximately twice as much as that on the freezer chamber. It was mainly resulted from a larger pressure difference between inside and outside air on the freezer chamber, which could cause the freezer seal to be much more compressed so that the gap between the chamber body and the seal was relatively narrower.

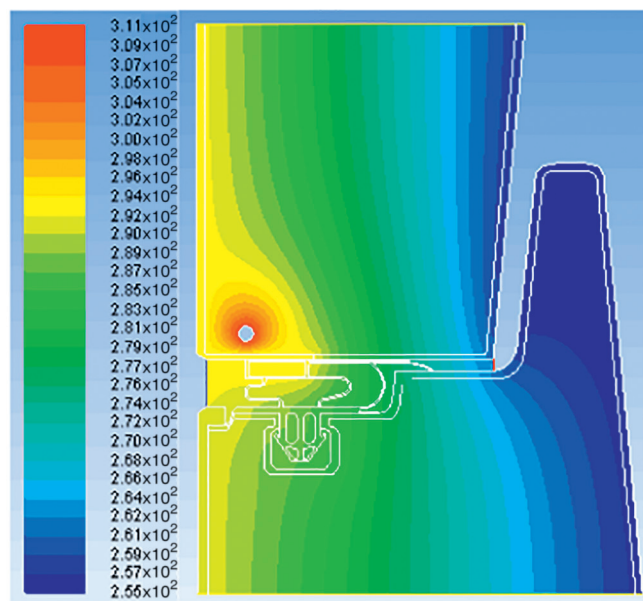
4.2. Temperature analysis

Fig. 5 displayed the simulated temperature contour of the refrigeration chamber and that of the freezer chamber individually, while the temperatures measured by thermocouples were presented in Fig. 6 for the refrigeration chamber and in Fig. 7 for the freezer chamber. The comparison of calculated and measured temperatures at Point 1 and Point 2 designated in Fig. 4 were tabulated in Table 5.

As shown in Fig. 5, the higher temperature region was the outside wall, while the lower temperature region was the inside wall both in the refrigeration chamber and in the freezer chamber. However, the



(a)



(b)

Fig. 5. Calculated temperature contours of (a) the refrigeration chamber and (b) the freezer chamber.

existence of the hot line (anti-dew pipes) resulted in the maximum temperature region surrounding the hot line in the freezer chamber. At Point 2, the maximum difference between measured and calculated temperatures that occurred in the freezer chamber was 2.03 °C, and that in the refrigerator chamber was 1.39 °C. At Point 1, both of the differences were even less than those at Point 2.

The difference between simulation results and experiment results might result from two main causes: one was because the inner wall of the chamber in simulation was supposed to transfer heat with the cooled air at a constant temperature of 5 °C in the refrigeration chamber (−18 °C in the freezer chamber) by convection regardless of the fact that the air temperature in the chamber was varying with time in experiment; the other was for the reason that the simulation model in this paper was two-dimensional, which would bring slight deviation as well.

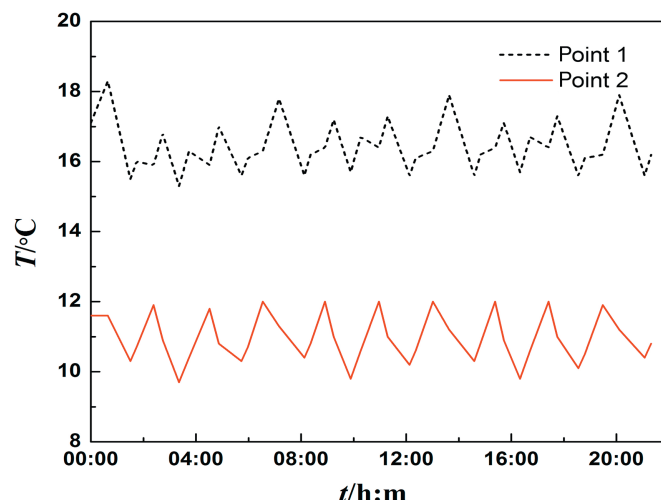


Fig. 6. The history of measured temperatures at seal edge and the corner of chamber body in the refrigeration chamber. h represents hour; m represents minute.

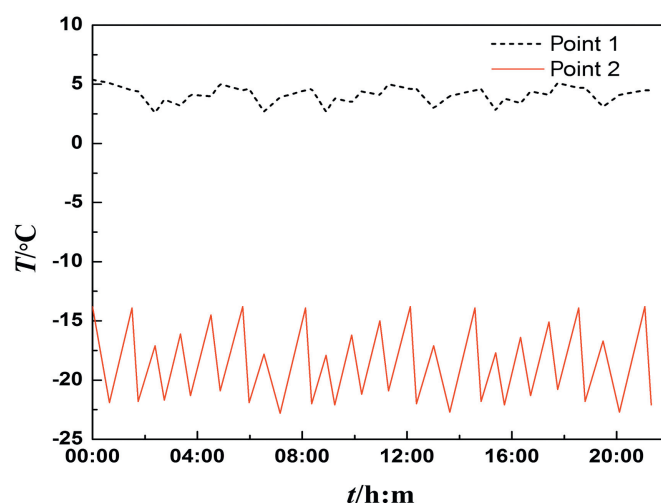


Fig. 7. The history of measured temperatures at seal edge and the corner of chamber body in the freezer chamber. h represents hour; m represents minute.

Despite the unavoidable tiny deviation stated above, the simulated results are of a higher precision compared with those in the previous related work. Also the values tabulated in Tables 4 and 5 give an evidence that the models proposed in this paper could accurately calculate the temperatures of the space both near and at seal. The optimization of the simulation model in this paper mainly comes from the improvement that the gap between the door seal and the chamber is firstly set up so that the heat and mass transfer in the gap, which has a huge influence on the simulated results, could be taken into full consideration. Additionally, the optimization on boundary conditions also works.

Table 5
The simulated results and experimental results of temperatures at special points

	Refrigeration chamber		Freezer chamber	
	Point 1	Point 2	Point 1	Point 2
Exp. temp./°C	16.12	10.79	4.75	−17.56
Cal. temp./°C	15.69	9.78	6.14	−15.53
Difference/°C	0.43	1.01	1.39	2.03

5. Conclusions

In the present paper, the mass and heat transfer through refrigerator door seal have been studied both simulatively and experimentally. In simulation, a two-dimensional optimized refrigerator door seal model is set up. In experiment, tracer gas method is employed, and then the air infiltration and temperature at some special points in both chambers are measured. The simulated and experimental results are as follows.

As to the field of mass transfer, the simulated values of the air infiltration rates on the refrigeration chamber and on the freezer chamber were $3.33 \times 10^{-6} \text{ kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$ and $1.89 \times 10^{-6} \text{ kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$, while the experimental values were $3.38 \times 10^{-6} \text{ kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$ and $1.86 \times 10^{-6} \text{ kg} \cdot \text{s}^{-1} \cdot \text{m}^{-1}$, respectively. The relative deviations between simulation and experiment on both chambers were less than 1%.

As to the field of heat transfer, the temperatures at the seal edge and at the corner of chamber were simulated and measured by thermocouples. The maximum difference between calculated and measured temperatures that occurred in the freezer chamber and in the refrigerator chamber were 2.03 °C and 1.39 °C, respectively, the values of which are less than those in previous work.

By comparisons, it appears that the simulated results are extremely close to the experimental results, which means that the proposed model has a good reliability. This paper initially sets up a modified model that can be used to precisely simulate mass and heat transfer through refrigerator door seal, and we may use this model to predict mass and heat transfer through door seal in other conditions as well. The innovation of this research not only lies in the modified model but also in the improvement that the air infiltration and heat loss can be analyzed simultaneously. In the future, the refrigerator door seal optimization on the basis of the model presented in this paper, will be under further study.

Nomenclature

c	CO_2 concentration in the tested space, $\text{m}^3 \cdot \text{m}^{-3}$
c_{atm}	CO_2 concentration in air, $\text{m}^3 \cdot \text{m}^{-3}$
c_p	air heat capacity, $\text{J} \cdot \text{kg}^{-1} \cdot ^\circ\text{C}^{-1}$
c_0	initial CO_2 concentration in the tested space, $\text{m}^3 \cdot \text{m}^{-3}$
p	pressure, Pa
Q_m	mass air infiltration rate, $\text{kg} \cdot \text{s}^{-1}$
Q_v	volume air infiltration flowrate, $\text{m}^3 \cdot \text{h}^{-1}$
q	net production flowrate of CO_2 in test space, $\text{m}^3 \cdot \text{h}^{-1}$

q_m	the air infiltration rate through unit length seal, $\text{kg} \cdot \text{m}^{-1} \cdot \text{s}^{-1}$
S_T	viscous dissipation term
S_{ij}, S_v	the generalized source term of momentum conservation equation
T	temperature, K or °C
T_∞	temperature of the fluid far from surface, outside of the thermal boundary layer, K
t	time, s
$\mathbf{U}$	fluid velocity vector, $\text{m} \cdot \text{s}^{-1}$
u	fluid velocity component in x direction, $\text{m} \cdot \text{s}^{-1}$
V	volume, L
v	fluid velocity component in y direction, $\text{m} \cdot \text{s}^{-1}$
λ	thermal conductivity, $\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
μ	dynamic viscosity, $\text{Pa} \cdot \text{s}$
ρ	density, $\text{kg} \cdot \text{m}^{-3}$

References

- [1] C. Afonso, M. Castro, Air infiltration in domestic refrigerators: The influence of the magnetic seals conservation, *Int. J. Refrig.* 33 (4) (2010) 856–867.
- [2] H.S. Kim, J.S. Sim, J.S. Ha, A study on the heat transfer characteristics near the magnetic door gasket of a refrigerator, *Int. Commun. Heat Mass Transfer* 38 (9) (2011) 1226–1231.
- [3] L. Han, S.F. Chen, Z.G. Lv, et al., CAE analysis of the thermal conductivity and effecting factors of the door seal of refrigerators, *China Appliance S1* (2012) 170–173. (in Chinese)
- [4] J.S. Li, Y.Y. Yang, Energy-saving design of refrigerator door seal, *Household Appliances* 7 (2001) 19–19. (in Chinese)
- [5] C.F.A. Afonso, E.A.B. Maldonado, E. Skåret, A single tracer-gas method to characterize multi-room air exchanges, *Energy Build.* 9 (4) (1986) 273–280.
- [6] B. CKWRS, New approach for measuring airflows in buildings using a perfluoro-carbon tracer, *Fuel Energy Abstr.* 36 (5) (1995) 380–380.
- [7] K.W. Cheong, Airflow measurements for balancing of air distribution system – Tracer-gas technique as an alternative? *Build. Environ.* 36 (8) (2001) 955–964.
- [8] M.H. Sherman, On the estimation of multizone ventilation rates from tracer gas measurements, *Build. Environ.* 24 (4) (1989) 355–362.
- [9] N.O. Breum, Diagnosis of ventilation by single-tracer gas techniques, *Indoor Air* 3 (S1) (1993) 5–28.
- [10] ASTM, Standard Practices for Air Leakage Site Detection in Building Envelopes and Air Barrier Systems. ASTM E1186 - 03, ASTM, USA, 2009.
- [11] ASTM, Standard Test Method for Determining Air Leakage Rate by Fan Pressurization, ASTM, USA, 2010.
- [12] ASTM, Standard Test Method for Determining Air Change in a Single Zone by Means of a Tracer Gas Dilution, ASTM, USA, 2011.
- [13] ASTM, Standard Guide for Using Indoor Carbon Dioxide Concentrations to Evaluate Indoor Air Quality and Ventilation, ASTM, USA, 2012.