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Research on test method of heat transfer coefficient for refrigerator gasket

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ABSTRACT

This paper presents an experimental research on test method of heat transfer coefficient for refrigerator gasket. An experimental apparatus based on Reverse Heat Loss Method (RHLM) was established. Tests were carried out on three distinct refrigerators to develop a high accuracy and repeatability test method. To ensure the accuracy of test, heat flux test zones of cabinet walls were divided strictly in conformity with the thermal resistance distribution. It was observed that the deviations of the heat transfer coefficient were less than ±5%, by using small volume size refrigerator as the test box and the characteristic heat flux method to simplify the test zones. Through testing gaskets with the different structures, the heat transfer coefficient of common gasket was about 0.045 W/m·°C. The results also indicate that the auxiliary air cell, multi-air cells, stiffener and auxiliary edge can reduce the heat leakage of gasket. The fan in the cabinet contributed to an increase of about 17% to heat leakage of gasket.

Recherche sur la méthode d'essai du coefficient de transfert de chaleur pour le joint du réfrigérateur

Mots-clés: Joint de réfrigérateur; Coefficient de transfert de chaleur; Méthode de perte de chaleur inversée; Méthode d'essai

1. Introduction

Refrigerators consume a large amount of energy every year around the world. The consumption of electricity in a refrigerator is related to its insulation property. Over the past two decades, significant efforts have been devoted to improving the thermal resistance of advanced insulation material for refrigerator cabinet (Hossieny et al., 2019; Trias et al., 2018). But not much information is available on the gasket, although the heat leakage attributed to gasket is a high percentage of the total thermal load. The accurate measurement method of heat transfer coefficient is the basis for evaluating a gasket insulation performance. And this parameter is also the data that the refrigerator designers hope to get (Bansal et al., 2011). However, it is very difficult to test the heat transfer coefficient of gasket with conventional methods due to complexity of structure. Thus, most refrigerator designers usually calculated the heat leakage of gasket as empirical value, such as 15% of the total heat load (Wu, 1998), or heat transfer coefficient of 0.065~0.080 W/m·°C (Griffith et al., 1995). Some of them even ignored the heat leakage of gasket (Engin et al., 2019).

Most scholars were keen to obtain heat transfer characteristics of gasket by numerical simulation, and experimental test method was rare. Boughton (1996) assumed that the boundaries between gasket and cabinet, and gasket and door, were adiabatic. The two-dimensional finite difference state equation of gasket was written by FORTAN (Ozisik, 1980), and the heat leakage in single direction was 2.5 W, accounting for only 2.7% of total heat leakage in refrigerator. Obviously, gasket heat transfer path is multi-directional, not only from the external environment to gasket, but also from the door and box to gasket. Guadalupe et al. (2011) combined a quasi-one-dimensional theoretical model with CFD numerical and experimental results to evaluate the heat leakage of gasket. The variation of results at different section indicated that the distances between the door and cabinet were not uniform. Kim et al. (2011) compared three cases of CFD simulation through changing the

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Nomenclature

RHLM	reverse heat loss method
q	heat flux, (W/m^2)
E	direct voltage of heat flux sensor, (μV)
S	sensitivity of heat flux sensor, (μV/W/m^2)
T	temperature, (°C)
A	area, (m^2)
Q	heat load, (W)
K	heat transfer coefficient, (W/m·°C)
l	length, (m)
h	convective heat transfer coefficient, (W/m^2·°C)
δ	thickness, (m)
λ	thermal conductivity, (W/m·°C)
q	characteristic heat flux, (W/m^2)
P	total power, (W)
W	actual running power, (W)
U	direct voltage of fan, (V)
I	direct current of fan, (A)
Greek symbols	
u	uncertainty
Subscripts	
o	original
flu	heat flux sensor
wal	wall of cabinet
gas	gasket
hea	heater made of electric heating wire
out	outside of cabinet
in	inside of cabinet
fan	fan inside cabinet
n	number of test zones in walls
m	number of test zones in single wall

boundary conditions. The calculated result of temperature using measured temperature showed a large difference from a uniform free steam temperature. Yan et al. (2016) established three-dimensional model to investigate the heat transfer characteristic and thermal load near the freezer gasket region. It was found that the heat leakage of gasket was 10.57 W and 6.68 W, with the compressor turned on and off. Gao et al. (2017) presented an approach combined Reverse Heat Loss Method (RHLM) experiment and CFD simulation to obtain the heat leakage through the refrigerator gasket area. The results showed that the average effective heat leakage on the gasket surface was 0.2 W/m·°C. Ghassemi (1993) added insulation around gasket areas through installing cotton batting and duct tape, which could reduce all possible heat leakage to a negligible level. Thus, by comparing the power consumption of refrigerator with and without additives, it was concluded that the gasket heat leakage accounts for about 5% of the total leakage in refrigerator. Hessami and Hilligweg (2003) used many heat flux meters to measure the heat transfer through the walls of a household refrigerator under inside heating steady state condition. The results indicated that 87% of the heat leakage load in the cabinet took place through the walls and remaining heat leakage load was considered through gasket. This was an early example of scholars using RHLM to test the heat leakage on the wall of refrigerator and indirectly obtaining the heat leakage of gasket. Then, other researchers like Melo et al. (2000), Tao and Sun (2001), Ma et al., 2012), and Thiessen et al. (2014) also attributed the difference between the power of the internal heater and the heat leakage load through the cabinet walls was heat leakage of gasket and edge. And the existence of edge leakage is inevitable around the gasket areas, which should also be counted as part of the gasket leakage (Sim, 2014).

Table 1
Instrumentation uncertainties in the experiment study.

Parameter	Instrument	Accuracy	Full scale
Temperature	T-type thermocouple	±0.5 °C	−50–150 °C
Heat flux voltage	Heat flux sensor	±3.0%	−150–150 kW/m^2
Heater power	Digital power meter	±0.5%	0–12 kW

From the previous research, it could be found that few investigations on the experimental test method of heat transfer coefficient for gasket. And most of the literatures just focused on the study of the heat transfer on the cabinet walls. Especially, the accuracy and repeatability of results were not discussed. The objective of this investigation is to develop a reliably test method of heat transfer coefficient for refrigerator gasket. Therefore, an experimental bench based on RHLM was established, and twelve groups of experiments were performed on three refrigerators with different conditions. Based on the proposed method, the heat transfer coefficients of gaskets with different structures were compared. Tests were also carried out with the fan turned on and off in the refrigerator to study the influence of fan on gasket heat leakage. We hope that the experimental test method can provide some guideline for gasket and refrigerator manufacturers.

2. Experiment work

2.1. System description

All tests were carried out in the constant temperature and humidity chamber. Therefore, the ambient temperature could be set. And RHLM was adopted in order to maintain the steady state (Sim and Ha, 2011; Thiessen et al., 2018). Sim and Ha (2011) found that the variation of temperature differences between cabinets and ambience have linearly increasing characteristics with the heat input. That meant the heat leakage from the refrigerator to ambience is directly related by the temperature differences between cabinets and ambience. The ratio of heat input and the temperature difference is the heat transfer coefficient of refrigerator. When the heat input was raised, the temperature inside the cabinet would be increased too. Thus, it also can be concluded that the heat transfer coefficient is constant and insensitive of temperature through using RHLM. Therefore, due to the insensitive of temperature, RHLM was adopted to test the heat transfer coefficient of gasket. The schematic diagram of experiment setup consists of three parts, as shown in Fig. 1. Uncertainties of the sensors in the experiment study are listed in Table 1.

There are three different refrigerators as the test box for gasket heat transfer coefficient testing in this experiment study. The key dimensions of experimental refrigerators are listed in Table 2.

Seven kinds of gaskets with different structures were tested, as shown in Figs. 2 and 3. Gaskets with the shapes of structure 1, structure 2, structure 3 and structure 4 are self-contained gaskets in freezer cabinet of 535 L refrigerator, refrigerating cabinet of 535 L refrigerator, 133 L refrigerating refrigerator and 108 L freezer/refrigerating refrigerator respectively. Moreover, in order to compare the insulation performance of gaskets with different structures, the gaskets manufacturer offered other four suitable size gaskets with structure 5, structure 6 and structure 7 for 108 L freezer/refrigerating refrigerator.

The heat flux sensor outputs a voltage signal and the ratio of voltage (E) to sensitivity (S) is the heat flux (q). The calculation method is shown in Eqs. (1) and (2). Different sensors have different initial sensitivities (S_o) which are provided by the manufacturer.

$$q = E/S \tag{1}$$

$$S = S_o \times (0.0034 \times T_{flu} + 0.917) \tag{2}$$

Table 2
Key dimensions of experimental refrigerator as the test box.

Test box	Compartment	Volume	Insulation thickness	Refrigeration type	Gasket size
535 L Side-by-side	Freezer	175 L	75 mm	Frost-free	360 mm × 1660 mm
Refrigerator-freezer	Refrigerating	360 L	55 mm	Frost-free	450 mm × 1660 mm
133 L single door refrigerator	Refrigerating	133 L	40 mm	Direct-refrigerating	500 mm × 740 mm
108 L single door refrigerator	Freezer or refrigerating	108L	80mm	Direct-refrigerating	465 mm × 925 mm

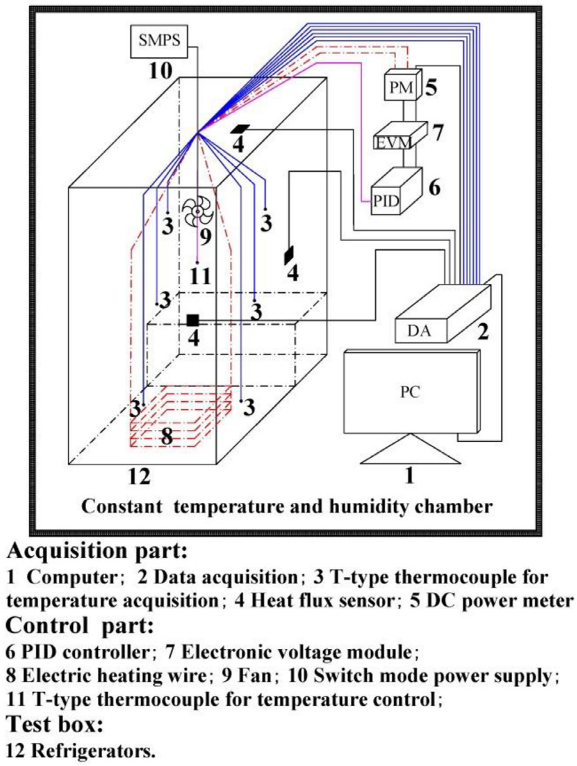

Fig. 1. Schematic diagram of the experimental setup.

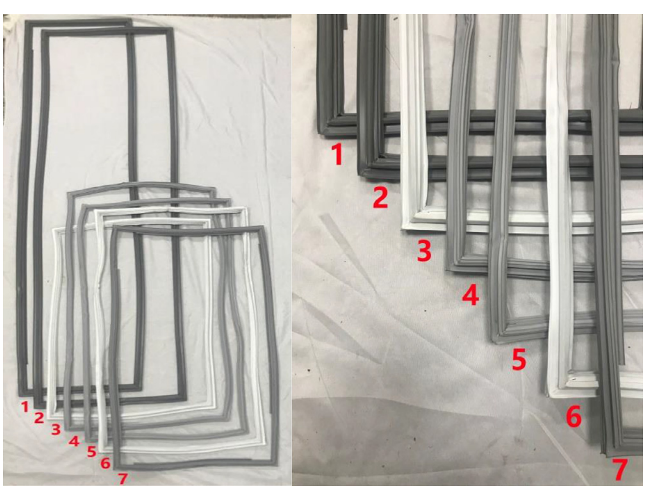

Fig. 2. Pictures of experimental gaskets.

2.2. Experiment principle

Generally, the heat leakage areas of a refrigerator mainly contains the wall, the door gasket, the edge, the copper pipe connected to outside compressor and the possible unseal gap of gasket. Apart from the former three leakage paths, the last two leakage paths need to be reduced as zero as possible. The “hot bridge” is formed between freezer air and ambient air through ABS and steel shell. This heat leakage often calls as edge leakage. The heat leakage of edge accounts for a small proportion of total heat leakage of gasket areas, and the existence of edge leakage is inevitable around the gasket areas. Furthermore, when using the same box to test the heat leakages of gaskets with different structures, each experimental result all contains the same edge leakage and is comparable. So, the edge leakage is counted as part of the gasket leakage. Then, to deal with the heat open loop problems of the compressor chamber, the cover plate of compressor chamber, the compressor, the copper pipes and the refrigerant were all removed. The heat fluxes of rear wall and top wall of compressor chamber were measured. The experimental results showed that the heat fluxes of side walls and bottom wall were zero. In addition, to reduce heat leakage of the copper pipes, all the pipes holes were filled with foam. Last but not least, the sealing propriety of the gaskets was also should be considered. Due to the structure deformation of box and door, the gaps between the gasket and box possible exist. In this paper, the infiltration leakage was not the object of study and should be removed, too. To ensure the good sealing effect, the test boxes and gaskets were all new. There was no obvious gap at during each testing process. And some adiabatic silicone was added at the contact face between gasket and cabinet to seal the possible tiny gaps.

Because of the irregular shape of gasket, it is very difficult to directly test the heat transfer of gasket, so the indirect test method is adopted. The difference between heater power (Q_{hea}) and heat leakage of walls (Q_{wal}) is equal to the heat leakage of gasket (Q_{gas}), given by Eqs. (3) and (4).

$$Q_{wal} = \sum_{i=1}^{n} q_i \times A_i \tag{3}$$

$$Q_{gas} = Q_{hea} - Q_{wal} \tag{4}$$

The regulation deviations are ±0.5 °C for ambient temperature and ±0.2 °C for interior temperature, so the results of each single test may be slightly different. Additionally, gaskets in different refrigerators have different length. In order to indicate the heat transfer characteristics the insulation property of gaskets more appropriately, as shown in Eq. (5), heat transfer coefficient (K_{gas}) is proposed.

$$K_{gas} = Q_{gas} / [l \times (T_{in} - T_{out})] \tag{5}$$

Heat transfer types include conduction, convection and radiation. The surface of experimental gaskets and refrigerators is light color to minimize radiative effects. Therefore, main heat transfer types are conduction and convection. From Eq. (6), heat flux is affected by internal surfaces thermal resistance ($1/h_{i,in}$), external surfaces thermal resistance ($1/h_{i,out}$), wall thermal resistance (δ_i /λ_i), and temperature difference ($T_{i,in}$- $T_{i,out}$) between internal and external surfaces.

$$q_i = \frac{1}{\left(\frac{1}{h_{i,in}} + \frac{\delta_i}{\lambda_i} + \frac{1}{h_{i,out}}\right)} \times (T_{i,in} - T_{i,out}) \tag{6}$$

Thermal resistance at different zones is different. There will be a lot of measurement zones to ensure the accuracy of walls

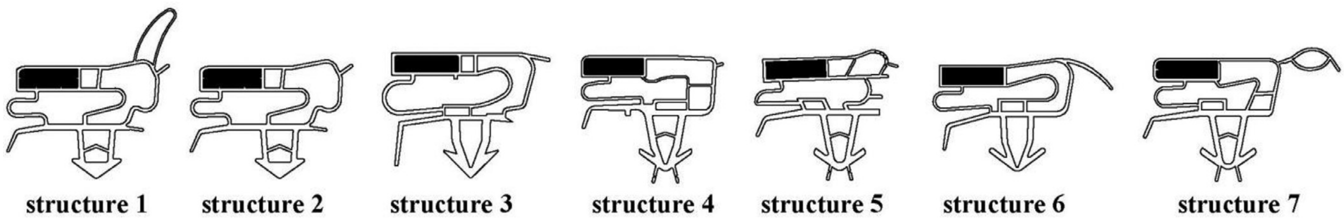

Fig. 3. Section shapes of gasket.

Table 3
Experimental tests steps.

Test box	Order	Test zones	Test times	Temperature set value(°C)	Fan	Experimental purposes
535 Lbig volume refrigerator	Exp.1	43	2	68/45	Off	Preliminary attempt (strcture1,strcture2)
	Exp.2	102	2	68/45	Off	Obtaining characteristic heat flux (strcture1,strcture2)
	Exp.3	22	5	68/45	Off	Repeated testing (strcture1,strcture2)
133 Lsmall volume refrigerator	Exp.4	48	2	45	Off	Obtaining characteristic heat flux (strcture3)
	Exp.5	8	5	45	Off	Repeated testing (strcture3)
	Exp.6	8	5	45	On	Influence of fan on heat leakage (strcture3)
108 Lsmall volume refrigerator	Exp.7	65	2	45	Off	Obtaining characteristic heat flux (strcture4)
	Exp.8	8	5	45	Off	Repeated testing (strcture4)
	Exp.9	8	5	45	On	Influence of fan on heat transfer (strcture4)
	Exp.10	8	5	45	Off	Repeated testing (strcture5)
	Exp.11	8	5	45	Off	Repeated testing (strcture6)
	Exp.12	8	5	45	Off	Repeated testing (strcture7)

heat leakage. However, the accumulated measurement errors will be very large, resulting in low repeatability of the test. To ensure accuracy and repeatability simultaneously, characteristic heat flux method was proposed. Calculating method of characteristic heat flux is given in Eq. (7).

$$\overline{q}_j = \sum_{i=1}^{n} q_i \times A_i / \sum_{i=1}^{n} A_i \tag{7}$$

2.3. Experiment arrangement

When refrigerator is in actual operation, room temperature is mostly 25 °C and freezer/refrigerating temperature is −18 °C/5 °C. In order to keep the same temperature difference between freezer/refrigerating compartment and ambience at 43 °C/20 °C, setting temperature of ambience is 25 °C and the freezer temperature is heated to 68 °C while refrigerating temperature is heated to 45 °C. Experimental tests steps are listed in Table 3.

2.4. Temperature uniformity regulation

The experimental platform can test the heat transfer coefficient of gasket for frost-free refrigerators and direct-refrigerating refrigerators, with the fan turned on and off. According to Eq. (6), in order to reduce number of heat flux measuring zones, the first step is to homogenize temperature in cabinet. The spatial temperature can be homogenized by installing fan, but the heat transfer type between internal surface and hot air changes from natural convection to forced convection (Sim and Ha, 2011). This is feasible for frost-free refrigerators, but changes the heat transfer type of direct-refrigerating refrigerators. Thus, three distribution modes of electric heating wire have been tried, as shown in Fig. 4. The results showed that under the first two modes, the maximum temperature differences at different locations inside freezer cabinet and refrigerating cabinet were all over **10 °C**. When electric heating wires were only distributed in the lower part of compartment evenly, temperature differences at different positions could be achieved within 2 °C.

3. Results and discussion

3.1. 535 L refrigerator as the test box

3.1.1. Preliminary attempt

Exp.1 was a preliminary attempt, which did not strictly divide the test zones according to the thermal resistance distribution of cabinet walls. As shown in Fig. 5, refrigerator walls were divided into 43 test zones.

Two aspects needed to be noticed: (1) Zone B4, B5, E3, F4 and F5 were located at the junction of freezer cabinet and refrigerating cabinet. Considering temperature difference between freezer cabinet and chamber, refrigerating cabinet and chamber were 43 °C and 20 °C, respectively. 43/63 of the heat leakage load in these zones belonged to freezer cabinet and 20/63 to refrigerating cabinet; (2) The heat leakages of zone D6, D7 and D8 were the heat flow out of freezer cabinet, which belonged to “leakage” heat. While for the refrigerating cabinet they were “inflow” heat, and belonged to heating power. Experiment results are listed in Table 4.

Two problems have arisen: (1) Ratio of gasket heat leakage to total leakage was smaller than the values of other scholars. And even the heat leakage of refrigerating gasket was negative at test 2; (2) There was a deviation of heat transfer coefficient between two tests, particularly refrigerating gasket.

The division of heat flux test zones in the Exp.1 was unscientific. From Fig. 6, it is observed that the structure of the door is irregular and can be divided into gasket area, bulge, walls intersection, etc. Heat fluxes in these zones are obviously smaller than that of drawer zones. So the results tested according to Fig. 5 would be smaller than real value or negative. A more detailed and reasonable division method is shown in Fig. 6 to deal with the problem of inaccuracy.

For the problem of poor repeatability, Eq. (8) can be used to calculate the uncertainty of heat transfer coefficient (Kline, 1953), where y represents the calculated function with the independent variables x_i, and dx_i is the uncertainty of the variable x_i.

$$\frac{\delta_y}{y} = \sqrt{\sum_{1}^{n} \left(\frac{dy}{dx_i} \frac{\delta x_i}{y} \right)^2} \tag{8}$$

Fig. 4. Distribution modes of electric heating wire.

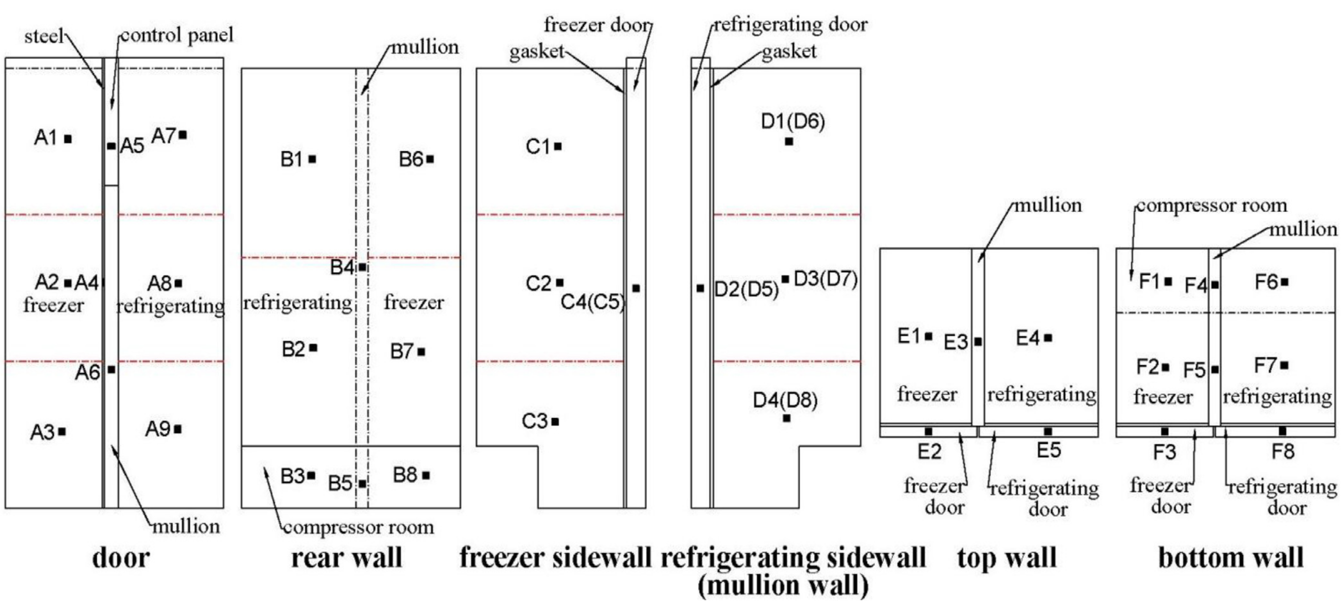

Fig. 5. Exp.1 test zones division.

Table 4
Heat transfer parameters of gasket at Exp1.

Heat transfer parameters		Test 1	Test 2
Heater power/W	Freezer cabinet	41.933	41.098
	Refrigerating cabinet	22.741	22.788
Heat leakage/W	Freezer cabinet wall	38.668	38.417
	Refrigerating cabinet wall	23.182	21.102
	Freezer gasket	3.265	2.681
	Refrigerating gasket	−0.237	1.686
Ratio of gasket heat leakage to total leakage	Freezer gasket	7.79%	6.52%
	Refrigerating gasket	−1.03%	7.40%
Temperature difference between cabinet and ambience/°C	Freezer cabinet	43.15	42.32
	Refrigerating cabinet	19.67	20.39
Heat transfer coefficient/(W/m·°C)	Freezer gasket	0.01873	0.01568
	Refrigerating gasket	−0.00286	0.01959

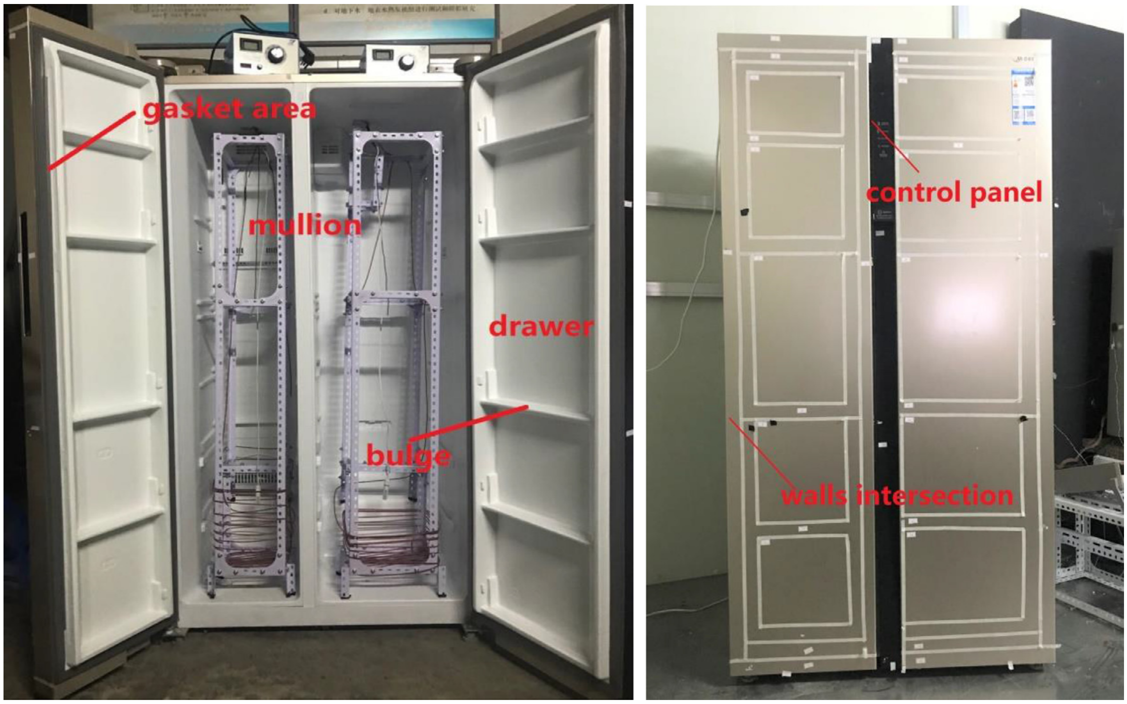

Fig. 6. Example of test zones division for refrigerator doors.

Eq. (9) is derived from Eqs. (1) to (5), while Eqs. (10) to (15) are derived from Eqs. (8) and (9).

$$K_{gas} = \frac{Q_{hea} - \sum_{i=1}^{n} \frac{E_i}{S_{o,i} \times (0.00334 \times T_{flu} + 0.917)} \times A_i}{T_{in} - T_{out}} \tag{9}$$

$$u_K{}^2(Q_{hea}) = \left(\frac{\partial f}{\partial Q_{hea}}\right)^2 u^2(Q_{hea}) = \frac{u^2(Q_{hea})}{(T_{in} - T_{out})^2} \tag{10}$$

$$u_K{}^2(E_i) = \left(\frac{\partial f}{\partial E_i}\right)^2 u^2(E_i) = \sum_{i=1}^{n} \left(-\frac{A_i}{S_{o,i} \times (0.00334 \times T_{flu} + 0.917)(T_{in} - T_{out})}\right)^2 u^2(E_i) \tag{11}$$

$$u_K{}^2(T_{flu}) = \left(\frac{\partial f}{\partial T_{flu}}\right)^2 u^2(T_{flu}) = \sum_{i=1}^{n} \left(\frac{E_i \times A_i \times S_{o,i} \times 0.00334}{\left(S_{o,i} \times (0.00334 \times T_{flu} + 0.917)(T_{in} - T_{out})\right)^2}\right)^2 u^2(T_{flu}) \tag{12}$$

$$u_K{}^2(T_{in}) = \left(\frac{\partial f}{\partial T_{in}}\right)^2 u^2(T_{in}) = \left(\frac{-Q_{hea} + \sum_{i=1}^{n} \frac{E_i}{S_{o,i} \times (0.00334 \times T_{flu} + 0.917)} \times A_i}{(T_{in} - T_{out})^2}\right)^2 u^2(T_{in}) \tag{13}$$

$$u_K{}^2(T_{out}) = \left(\frac{\partial f}{\partial T_{out}}\right)^2 u^2(T_{out}) = \left(\frac{Q_{hea} - \sum_{i=1}^{n} \frac{E_i}{S_{o,i} \times (0.00334 \times T_{flu} + 0.917)} \times A_i}{(T_{in} - T_{out})^2}\right)^2 u^2(T_{out}) \tag{14}$$

$$u_K = \sqrt{u_K{}^2(Q_{hea}) + u_K{}^2(E_i) + u_K{}^2(T_{flu}) + u_K{}^2(T_{in}) + u_K{}^2(T_{out})} \tag{15}$$

The extended uncertainties of power meter, heat flux sensor and T-type thermocouple are ±0.5%, ±3%, and ±0.5 °C, respectively. In this study, when confidence level was 95% and coverage factor was 2, the uncertainties of freezer and refrigerating gasket were ±0.00473 W/m·°C (±25.27%) and ±0.00635 W/m·°C (±222.20%) at test 1, respectively. And the uncertainties of freezer and refrigerating gasket were ±0.00471 W/m·°C (±30.04%) and ±0.00549 W/m·°C (±28.03%) at test 2, respectively. Thus, instrumentation and propagated uncertainties in the Exp.1 had large influence on repeatability. In addition, the scales of data acquisition, the different installation state of each heat flux sensor, and the fluctuation of external temperature field also affect the repeatability (Mofat, 1988). To reduce the influence of interference factors, following aspects have been improved:

1) The key to ensure accuracy and repeatability simultaneously is the division method of test zones on cabinet walls. Thus, characteristic heat flux method is proposed, as shown at **Section 3.1.2**.
2) The scale of data collected must be more than 500 at each single test zone.
3) Ensure the same installation status of heat flux sensors at different test zones.
4) Put the experimental device away from the air outlet, light, heat source at outside, etc.

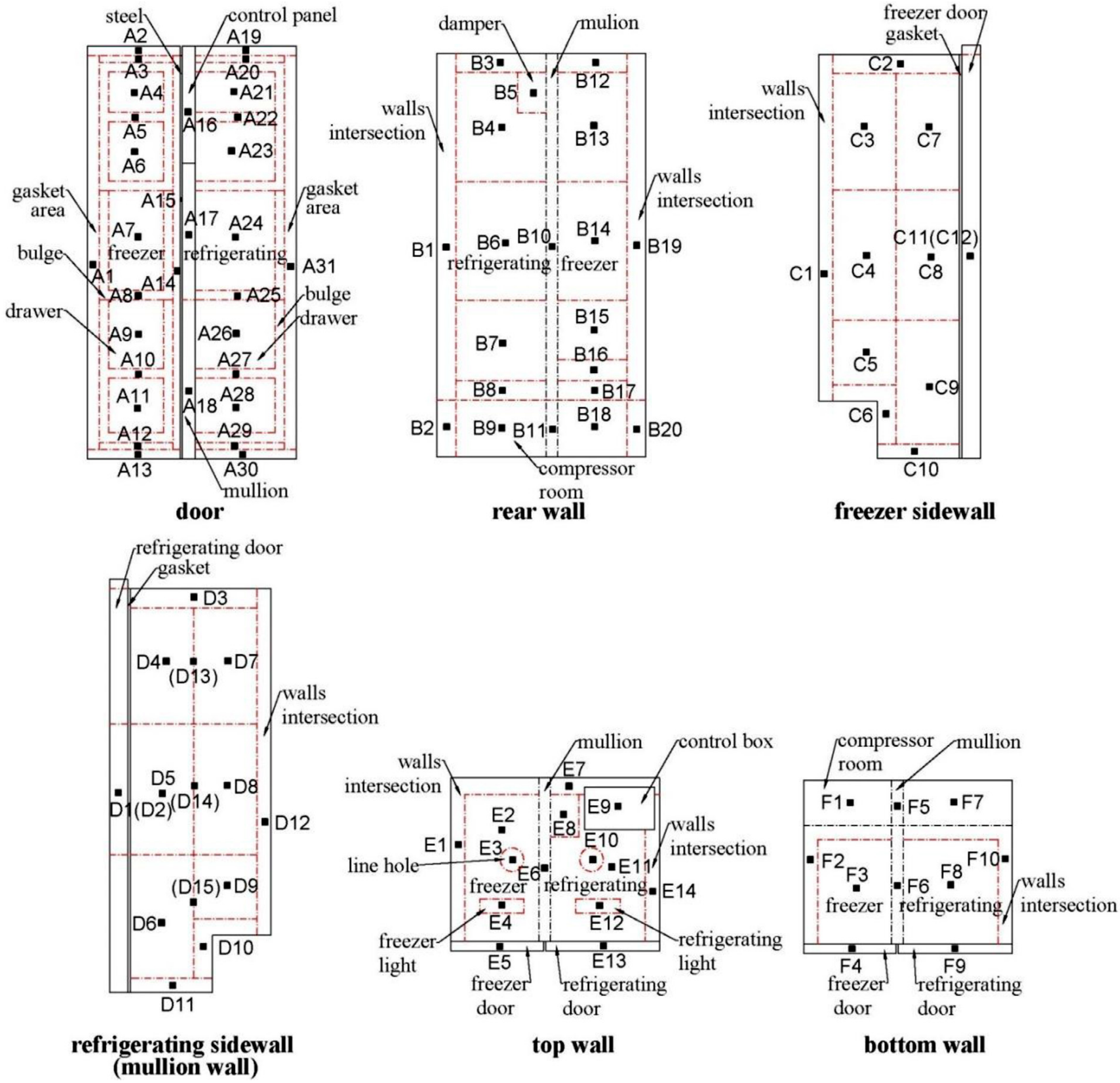

Fig. 7. Exp.2 test zones division.

In this paper, the accuracy of test results was influenced the errors produced by the ways of zones division and the built-up experimental errors. The main reason to divide heat flux test zones of the walls from 43 to 102 was to reduce the errors produced by the ways of zones division. However, the built-up experimental errors would be increased. Actually, the eventually experimental results showed that the accuracy of test results was improved when added the test zones due to the strong coincidence of heat flux test zones division and thermal resistance distribution. This meant that the errors produced by the ways of zones division were decreased rapidly though the built-up experimental errors had increased to a certain extent. As a consequence, the eventually accuracy of test results was improved. In addition, to reduce the complexity of testing and improve the repeatability of test results, the characteristic heat flux method is proposed.

3.1.2. Characteristic heat flux for 535 L refrigerator

102 test zones were determined strictly according to the distribution of cabinet thermal resistance, as shown in Fig. 7. Then, experiment results are listed in Table 5.

From Table 5, the ratios of gasket heat leakage to total leakage were similar to other scholars (Yan et al., 2016; Hessami and Hilligweg, 2003). The accuracy of measurement was improved, but the repeatability was still unsatisfactory. The instrumental uncertainty and the accumulated unknown errors in the testing process were increased with the increase of test zones. Thus, characteristic heat flux method was presented to simply test zones. The heat transfer coefficients of freezer gasket were closer than refrigerating gasket at twice test. But the structures of freezer and refrigerating gasket are similar, so it could be considered that the heat transfer coefficient of freezer gasket was more accurate. Thus, the heat fluxes that belonged to freezer cabinet were the average values from twice test, while refrigerating cabinet were the value from test 1. Simplified test zones were shown in Fig. 8, which were given from Table 6. Characteristic heat flux of each simplified zone was the value closest to the mean heat flux. But the characteristic heat flux in different big zones may be same. For instance, zone A'1 and zone A'2 in Fig. 8 had the same characteristic heat flux zone, A8 in Fig. 7. The characteristic heat flux of zone A'1 could locate near zone A8.

Table 5
Heat transfer parameters of gasket at Exp.2.

Heat transfer parameters		Test 1	Test 2
Heater power/W	Freezer cabinet	47.131	46.005
	Refrigerating cabinet	23.725	23.738
Heat leakage/W	Freezer cabinet wall	37.985	38.197
	Refrigerating cabinet wall	20.86	22.645
	Freezer gasket	9.146	7.808
	Refrigerating gasket	2.864	1.093
Ratio of gasket heat leakage to total leakage	Freezer gasket	19.41%	16.97%
	Refrigerating gasket	12.07%	4.61%
Temperature difference between cabinet and ambience/°C	Freezer cabinet	43.65	43.19
	Refrigerating cabinet	21.1	20.45
Heat transfer coefficient/(W/m·°C)	Freezer gasket	0.05186	0.04475
	Refrigerating gasket	0.03216	0.01267

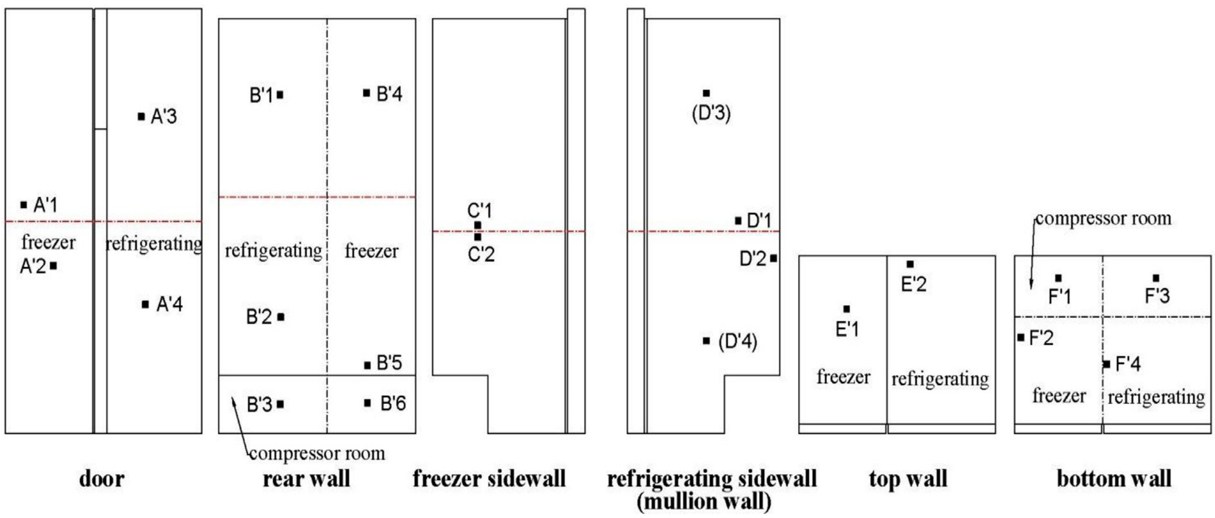

Fig. 8. Exp.3 test zones division.

Table 6 gives the characteristics heat flux of the 535 L side-by-side refrigerator. 102 zones were simplified to 22 zones. It was skillful to select the characteristics heat flux of each big zone. Firstly, the actual heat leakage of each big zone is the sum of the product of the small zones heat flux and area. And the mean heat flux is equal to the ratio of the actual heat leakage and the total area of each big zone. Then, the measured heat flux which is closest to the calculated mean heat flux is selected to the characteristic heat flux. The characteristic heat leakage is the product of the characteristic heat flux and the total area of each big zone. However, when all the characteristic heat leakages were calculated, there may be a certain deviation between the total actual heat leakage and the characteristic heat leakage total of the whole refrigerator. Thus, some selected characteristic heat fluxes should be adjusted slightly to reduce the total deviation. For instance, the mean heat flux in big zone A'3 is 4.456 W/m^2 but the characteristic heat flux of small zone A23 was selected. As a consequence, the eventual relative error between total actual heat leakage and total characteristic heat leakage was 0.15%, which had no effect on the accuracy of test results.

3.1.3. Repeated testing of gaskets with structure 1 and structure 2

After determing the characteristic heat fluxes, five repetitive tests were carried out according to Fig. 8. Results are listed at Table 7.

Same method was used to calculate the uncertainty of heat transfer coefficient of gasket as **Section 3.1.1**. The relative uncertainty of single test was about ±15% for freezer gasket and ±37% for refrigerating gasket. The heat transfer coefficient of each gasket was measured five times. Several data were relatively close in the five tests. These close data possibly were the eventually valid value. These potentially valid data were used as baseline in turn. When a potentially valid data was used as a baseline and the number of valid data with relative error of ±5% was the largest, the baseline was selected. If no data was close, the first data was used as a baseline, such as the relative error calculation of refrigerating gasket in Table 7. In addition, if any possible valid data as the baseline had no impact on the results, the first valid data in order was selected as the baseline, such as the relative error calculation of freezer gasket in Table 7.

From Table 7, the heat transfer coefficients of freezer gasket in test 1, test 4 and test 5 were very close (deviation within ±5%). Thus, it could be concluded that the valid heat transfer coefficients of freezer gasket was 0.04388 W/°C·m, namely the average of results from test 1, test 4 and test 5. However, the test results of refrigerating gasket were quite different. So, it was hard to judge which test results was valid. Instrumentation uncertainty was the main factor to affect the repeatability, but errors of the measured values included many other factors. The test results of freezer gasket had a certain probability of less than ±5%, but the probability of the refrigerating gasket was really small. The biggest difference between freezer cabinet and refrigerating cabinet was their volume, 175 L for the former and 360 L for the latter. We inferred that small volume size cabinet was less affected by convective heat transfer.

The poor repeatability of measurement was determined by many known or unknown factors, including instrumentation uncertainty and random errors which were produced by the

Table 6
Characteristic heat flux of 535 L refrigerator (△, denotes as a characteristic zone; *, denotes that the single test zone in Fig. 7 divide into several areas in Fig. 8).

Zones in Fig. 8	Zones at in Fig. 7	Area (m^2)	Heat flux (W/m^2)	Actual heat leakage(W)	Mean heat flux(W/m^2)	Characteristic heat leakage(W)
A'1 (Freezer door)	A1*	0.042	6.482	4.494	9.347	4.550
	A2	0.016	8.597			
	A3	0.009	6.049			
	A4	0.040	10.355			
	A5	0.071	8.031			
	A6	0.059	11.605			
	A7*	0.061	13.256			
	△A8*	0.021	9.464			
	A14*	0.025	6.699			
	A15*	0.007	36.142			
	C11*	0.057	7.089			
	C12*	0.057	8.277			
	E5	0.016	2.842			
A'2 (Freezer door)	A1*	0.042	6.482	4.957	9.557	4.908
	A7*	0.038	13.256			
	△A8*	0.025	9.464			
	A9	0.068	11.378			
	A10	0.067	9.155			
	A11	0.054	13.277			
	A12	0.009	10.592			
	A13	0.016	4.824			
	A14*	0.025	6.699			
	A15*	0.008	36.142			
	C11*	0.076	7.089			
	C12*	0.076	8.277			
	F4	0.016	3.511			
A'3 (Refrigerating door)	A16	0.028	4.742	2.478	4.456	2.747
	A17*	0.021	11.920			
	A19	0.017	1.865			
	A20	0.011	2.980			
	A21	0.058	3.970			
	A22	0.062	4.781			
	△A23	0.085	4.939			
	A24*	0.088	4.783			
	A25*	0.010	5.701			
	A31*	0.042	3.075			
	D1*	0.057	5.802			
	D2*	0.057	2.010			
	E13	0.019	1.409			
A'4 (Refrigerating door)	A17*	0.011	11.920	3.075	5.180	3.015
	A18	0.038	11.881			
	A24*	0.054	4.783			
	A25*	0.022	5.701			
	△A26	0.099	5.080			
	A27	0.051	5.775			
	A28	0.078	5.938			
	A29	0.011	3.409			
	A30	0.017	2.627			
	A31*	0.042	3.075			
	D1*	0.076	5.802			
	D2*	0.076	2.010			
	F9	0.019	2.510			
B'1 (Refrigerating rear wall)	B1*	0.068	4.502	2.998	7.323	3.130
	B3	0.031	5.681			
	△B4	0.157	7.645			
	B5	0.020	7.880			
	B6*	0.119	8.702			
	B10*	0.014	8.465			
B'2 (Refrigerating rear wall)	B1*	0.049	4.502	2.122	7.221	2.277
	B6*	0.073	8.702			
	△B7	0.131	7.750			
	B8	0.031	5.354			
	B10*	0.010	8.465			
B'3 (Refrigerating rear wall)	B2	0.019	3.760	0.896	7.748	0.970
	△B9	0.092	8.391			
	B11*	0.004	12.060			
B'4 (Freezer rear wall)	B10*	0.028	8.465	3.664	10.555	4.235
	B12	0.024	8.056			
	△B13	0.136	12.200			
	B14*	0.091	13.295			
	B19*	0.068	5.327			
B'5 (Freezer rear wall)	B10*	0.020	8.465	2.761	11.084	2.485
	B14*	0.056	13.295			
	B15	0.074	13.023			

(continued on next page)

Table 6 (*continued*)

Zones in Fig. 8	Zones at in Fig. 7	Area (m^2)	Heat flux (W/m^2)	Actual heat leakage(W)	Mean heat flux(W/m^2)	Characteristic heat leakage(W)
	B16	0.027	14.606			
	ΔB17	0.024	9.975			
	B19*	0.049	5.327			
B'6 (Freezer rear wall)	B11*	0.008	12.060	1.449	14.783	1.635
	Δ18	0.071	16.681			
	B20	0.019	8.919			
C'1 (Freezer sidewall)	C1*	0.044	4.610	3.816	8.713	3.706
	C2	0.043	6.581			
	C3	0.132	8.325			
	ΔC4*	0.043	8.460			
	C7	0.132	10.624			
	C8*	0.043	10.597			
C'2 (Freezer sidewall)	C1*	0.044	4.610	5.194	9.641	4.558
	ΔC4*	0.105	8.460			
	C5	0.073	8.169			
	C6	0.046	7.426			
	C8*	0.105	10.597			
	C9	0.140	13.764			
	C10	0.025	4.638			
D'1 (Refrigerating sidewall)	D3	0.043	6.066	2.652	6.054	2.602
	D4	0.132	6.493			
	D5*	0.043	6.757			
	D7	0.132	5.879			
	ΔD8*	0.043	5.941			
	D12*	0.044	4.665			
D'2 (Refrigerating sidewall)	D5*	0.105	6.757	2.528	4.691	2.514
	D6	0.140	4.270			
	D8*	0.105	5.941			
	D9	0.073	2.537			
	D10	0.046	3.678			
	D11	0.025	1.303			
	ΔD12*	0.044	4.665			
E'1 (Freezer top wall)	E1	0.063	6.005	2.997	10.639	3.314
	ΔE2	0.176	11.763			
	E3	0.008	19.842			
	E4	0.011	16.536			
	E6*	0.023	8.756			
E'2 (Refrigerating top wall)	E6*	0.012	8.756	2.794	7.488	2.128
	ΔE7	0.024	5.701			
	E8	0.022	9.381			
	E9	0.054	5.410			
	E10	0.008	4.837			
	E11	0.212	8.330			
	E12	0.011	8.394			
	E14	0.031	5.265			
F'1 (Freezer bottom wall)	ΔF1	0.071	13.962	1.054	13.581	1.083
	F5*	0.006	9.304			
F'2 (Freezer bottom wall)	ΔF2	0.049	6.380	1.374	6.728	1.303
	F3	0.139	7.224			
	F6*	0.017	3.623			
F'3 Refrigerating bottom wall)	F5*	0.003	9.304	0.637	6.960	0.629
	ΔF7	0.088	6.876			
F'4 (Refrigerating bottom wall)	ΔF6*	0.008	3.623	0.683	2.834	0.873
	F8	0.178	2.730			
	F10	0.054	3.054			
D'3 (mullion wall)	ΔD13	0.265	7.196	2.602	7.412	2.526
	D14*	0.086	8.075			
D'4 (mullion wall)	D14*	0.211	8.075	3.729	7.938	3.677
	ΔD15	0.259	7.827			
total			58.953		58.863	

Table 7
Heat transfer parameters of gasket at Exp.3.

Heat transfer parameters	Heat leakage /W		Temperature difference between cabinet and ambient/°C		Heat transfer coefficient/(W/m·°C)		Relative error (based on test 1)	
	Freezer	Refrigerating	Freezer	Refrigerating	Freezer	Refrigerating	Freezer	Refrigerating
Test 1	7.560	3.052	43.21	20.05	0.04330	0.03608		
Test 2	6.544	1.197	43.11	19.94	0.03758	0.01423	−13.21%	−60.56%
Test 3	9.914	3.284	42.24	19.58	0.05810	0.03974	34.18%	10.14%
Test 4	7.554	1.608	43.20	19.95	0.04328	0.01910	−0.05%	−47.06%
Test 5	7.698	2.046	42.29	19.48	0.04506	0.02489	4.06%	−31.01%
Valid value	7.604	2.235	/	/	0.04388	/		

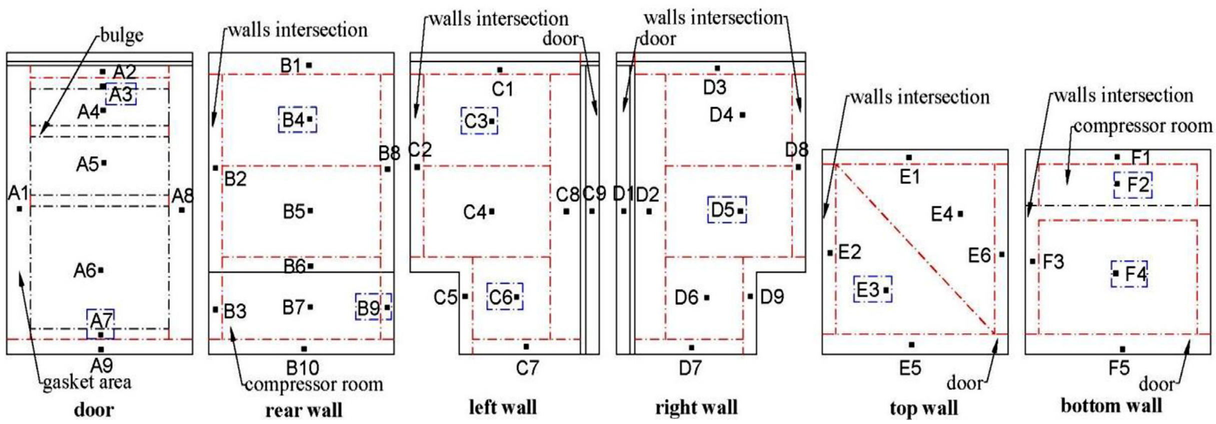

Fig. 9. Exp.4 test zones division.

difference of each test condition. The instrumentation uncertainty could be calculated by Eqs. (9) to (15). And the uncertainty of heat flux sensor was the main factor affecting the overall uncertainty of the system. Using the formulas above mentioned, it was found that the instrumentation uncertainty increased with the increase of the test zones. Thus, the accumulated errors which were produced by instrumentations were also increased with the increasing number of dividing zones. The random errors were caused by manual operation, installation status of instruments and thermal steady state consistency of each test. The manual operation errors would also be increased with the increase of dividing zones number because testing was more complex. Thus, it was an effective way to reduce the instrumentation uncertainty by simplified the test zones. The main instruments were T-type thermocouples and heat flux sensors. The positions of T-type thermocouples which were inside the cabinet were kept stationary. The temperatures of wall surfaces were measured by heat flux sensors which cloud measure the heat flux and temperature simultaneously. Thermal conductive silicone grease was used to attach the heat flux sensors to the wall, which may increase the a little wall thermal resistance, resulting in a very slightly smaller measured wall heat flux. The thermal steady states of all tests should be as consistent as possible. The slight variety of thermal steady state would affect the distribution of temperature field and flow field inside the cabinet, resulting in the difference of convective heat transfer. Therefore, it usually took over 20 h for waiting the cabinet interior temperature field and flow field to thermal steady state. However, although more than 20 h of continuously heating were performed before each test, the thermal steady states of all tests were difficult to maintain complete consistency. Especially for large volume cabinet, it was more difficult to maintain the stability and consistency of thermal steady states. Therefore, next step was to study the heat leakage of gaskets with small volume refrigerator as the experimental box.

3.2. 133 L refrigerator as the test box

3.2.1. Characteristic heat flux for 133 L refrigerator

According to the distribution of thermal resistance, the wall of 133 L refrigerator was divided into 48 test zones, as shown in Fig. 9. Characteristic heat fluxes are located at zones A3, A7, B4, B9, C3, C6, D5, D5, E3, F2 and F4 in Fig. 9, which correspond to zones A'1, A'2, B'1, B'2, C'1, C'2, D'1, D'2, E'1, F'1 and F'2, in Fig. 10, respectively.

3.2.2. Repeated testing for gasket with structure 3

Five tests were carried out in the light of test zones location of Fig. 10. The experimental results were listed in Table 8.

Based on the result of test 1, relative errors of other three test results were at range of ±5%. The valid value was the average of result at test 1, test 2, test 3 and test 5. It could be concluded that the heat transfer test of gasket with a small volume refrigerator as the test box had a relatively high repeatability. The average relative uncertainty of five single tests was ±10.96% with eleven test zones, so the probability of test results deviation within ±5% was increased. The way to minimize the relative error was to keep only one characteristic heat flux zone on the walls, but considering the measurement accuracy, at least one characteristic zone should be kept on each wall.

3.3. 108 L refrigerator as the test box

3.3.1. Characteristic heat flux for 108 L refrigerator

133 L refrigerator cannot install gaskets with different structures because the assembly structure among door and box is special. In order to compare the heat transfer performance of gaskets with different structures, a 108 L freezer/refrigerating refrigerator was selected as the test box which can adapt to different structures. As shown in Fig. 11, zone A2, B7, B10, C13, D4, E5, F2 and F6 were characteristic zone of each wall.

3.3.2. Repeated testing for gasket for structure 4~structure 7

As shown in Table 9 taking the average value of results within relative error of ±5% as the valid value, the heat transfer coefficients of structure 4, structure 5, structure 6 and structure 7 gaskets were 0.04472 W/m·°C, 0.04550 W/m·°C, 0.04900 W/m·°C and 0.04403 W/m·°C, respectively. When the number of test zone was simplified to eight, the relative uncertainty of single test was about ±7.5%. As a result, the instrument uncertainty had less effect on the repeatability (within ±5%). The test results of single gasket with five times repeated testing were nearly constant. Good repeatability is the goal of this study. A series of measures were performed to achieve relatively high repeatability. Additionally, the heat transfer coefficients of gaskets with different structures were also nearly constant. The mainly reason for this phenomenon is that the basis structural characteristics of the structure 4~7 gasket are similar. And the material of all the gaskets is same. This meant that the sizes (δ_i) of gaskets are similar and the material thermal conductivities (λ_i) of gaskets are same. And the thermal steady states of all tests were kept to be as consistent as possible. It was could be inferred that the internal surfaces thermal resistance ($1/h_{i,in}$) and external surfaces thermal resistance ($1/h_{i,out}$) were close. Thus, the overall thermal resistances ($\frac{1}{h_{i,in}} + \frac{\delta_i}{\lambda_i} + \frac{1}{h_{i,out}}$) of these serval gaskets were close. As a result, the tested heat transfer coefficients were close.

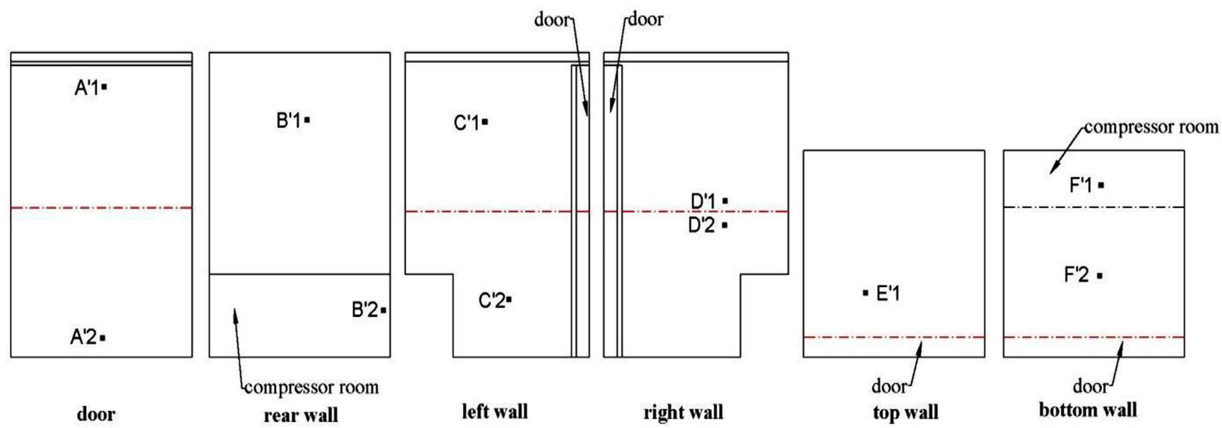

Fig. 10. Exp.5 test zones division.

Table 8
Heat transfer parameters of gasket at Exp.5.

Heat transfer parameters	Heat leakage /W	Temperature difference between cabinet and ambient/°C	Heat transfer coefficient/(W/m·°C)	Relative error (based on test 1)
Test 1	2.390	21.20	0.04546	
Test 2	2.367	20.89	0.04568	0.48%
Test 3	2.354	21.05	0.04511	−0.77%
Test 4	2.164	21.31	0.04096	−9.90%
Test 5	2.401	20.82	0.04649	2.26%
Valid value(Test 1,2,3,5)	2.378	20.99	0.04569	

Table 9
Heat transfer parameter of structure 4~ structure 7 gasket at Exp.8–Exp.12.

Shapes	Heat transfer parameters	Heat leakage/W	Temperature difference between cabinet and chamber/°C	Heat transfer coefficient/(W/m·°C)	Relative error (based on test 1)
Structure 4	Test 1	2.591	20.90	0.04460	
	Test 2	2.636	20.82	0.04555	2.14%
	Test 3	2.569	20.86	0.04431	−0.64%
	Test 4	2.573	20.83	0.04444	−0.34%
	Test 5	2.570	20.69	0.04468	0.20%
	Valid value (Test 1,2,3,4,5)	2.588	20.82	0.04472	
Structure 5	Test 1	2.630	20.69	0.04573	
	Test 2	2.476	20.71	0.04300	−5.97%
	Test 3	2.591	20.69	0.04505	−1.47%
	Test 4	2.628	20.69	0.04569	−0.09%
	Test 5	2.629	20.78	0.04552	−0.45%
	Valid value (Test 1,3,4,5)	2.620	20.71	0.04550	
Structure 6	Test 1	2.837	21.00	0.04860	
	Test 2	2.876	20.94	0.04940	1.65%
	Test 3	2.667	21.00	0.04569	−5.99%
	Test 4	3.080	20.97	0.05283	8.70%
	Test 5	2.622	20.89	0.04514	−7.12%
	Valid value (Test 1,2)	2.857	20.97	0.04900	
Structure 7	Test 1	2.505	20.87	0.04317	
	Test 2	2.715	20.84	0.04687	8.58%
	Test 3	2.527	20.83	0.04363	1.06%
	Test 4	2.596	20.85	0.04479	3.74%
	Test 5	2.582	20.85	0.04454	3.16%
	Valid value (Test 1,3,4,5)	2.553	20.85	0.04403	

The structural characteristics are shown in Fig. 12 and the installation state of structure 7 gasket is shown in Fig. 13. With the energy conservation design of multi-air cells, auxiliary air cell, auxiliary edge, stiffener, buckle edge and long edge, the heat transfer coefficient of structure 7 gasket was smallest. The thermal conductivities of rubber strip, magnet strip and air were 0.20 W/m·°C, 10 W/m·°C, 0.02 W/m·°C, respectively. If only heat conduction was considered, the volume of air should be increased, while the volume of rubber strip and magnetic strip should be reduced. It could be explicated the insulation performance of the structure 5 gasket was worse than that of the structure 4 gasket. However, the natural convective heat transfer coefficient of single air cell is increased with the volume increasing. Thus, the large volume cavity divided into three uniform air cells to reduce that in the inner cell, such as structure 4 gasket, structure 5 gasket and structure 7 gasket. Meanwhile, it can maintain the shape of gasket unchanged. The stiffener plays the same role. As shown in Fig. 12, the auxiliary air cell can prevent the hot air from contacting the gasket directly. Thus, the

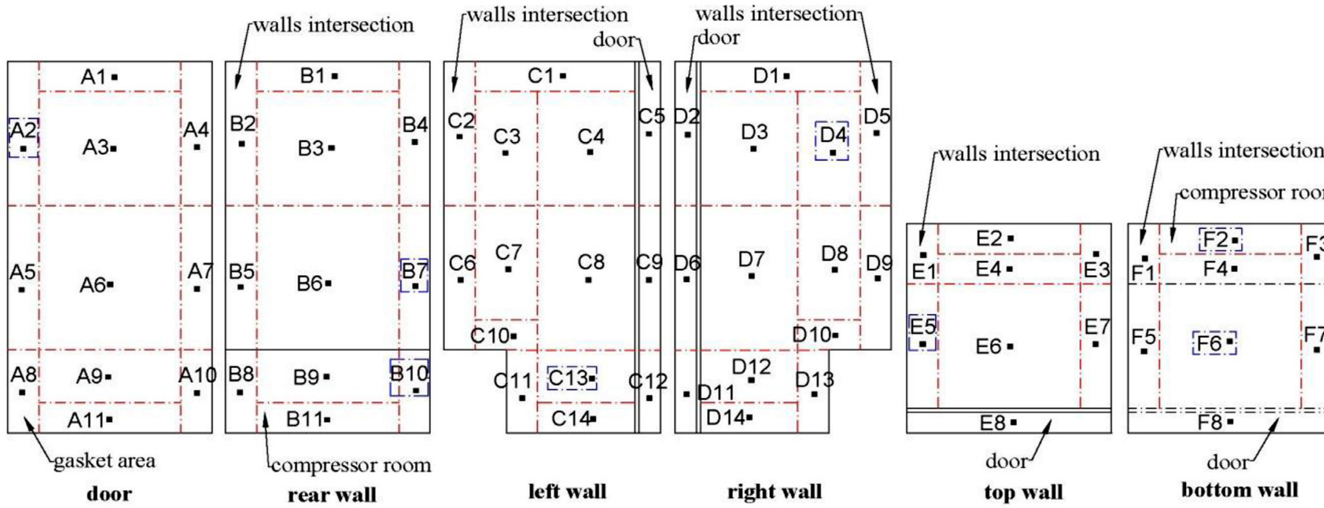

Fig. 11. Exp.7 test zones division.

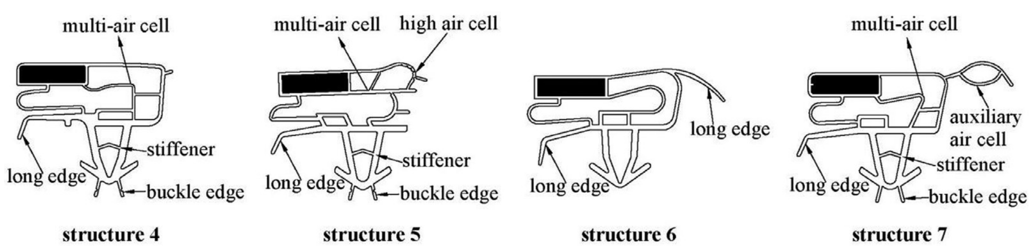

Fig. 12. Characteristics of different shape gasket.

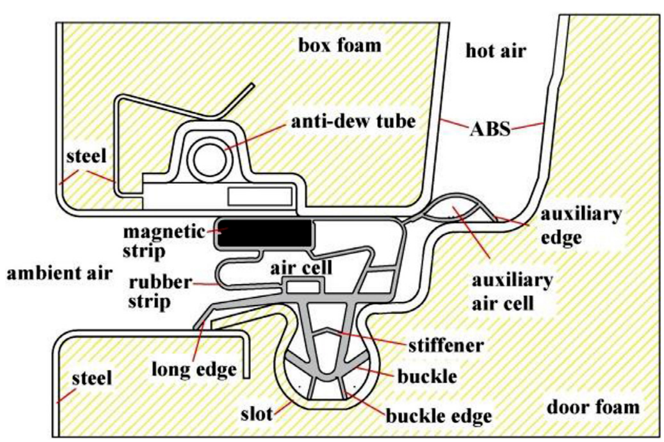

Fig. 13. Installation diagram of structure 7 gasket.

heat transfer coefficient of structure 4 gasket is a little larger than that of structure 7 gasket due to lack of auxiliary air cell. Additionally, the functions of the long edge, the buckle edge and the auxiliary edge are to form a new auxiliary air cell with the door, which can not only increase the thermal resistance of the gasket, but also prevent the air leakage. The insulation performance of structure 6 gasket was the worst because of only long edge design.

3.4. Influence of fan on heat transfer

A DC 12 V axial flow fan which is usually applied to forest-free refrigerator was selected in this experiment. And the installation position of the fan in the cabinet imitated the actual situation of a forest-free refrigerator. In order to supply power to the fan, the switching power supply converts 220 V AC voltage to 12 V DC voltage. The total power (P_{fan}) of the fan consists of the actual running power (W_{fan}) and heat loss (Q_{fan}). The total power (P_{fan}) of the fan was measured by the DC power meter, and the power difference of the system when the fan was on and off was regarded as the total power (P_{fan}). The actual running power (W_{fan}) was the product of the actual running DC voltage and electric current, which were all measured by a Fluke multimeter. The heat loss (Q_{fan}) of the fan can be calculated by the following formula Eqs. (16) and (17).

$$Q_{fan} = P_{fan} - W_{fan} \tag{16}$$

$$W_{fan} = U_{fan} I_{fan} \tag{17}$$

As shown in Fig. 14, with the fan turned on, spatial temperature dropped immediately due to the increase of internal surface convective heat transfer coefficient. Thus, total heat transfer coefficient increased with the improvement of internal surface convection heat transfer coefficient. The continuous type PID controller was implemented to control the cabinet inside temperature in this experiment. A T-type thermocouple connected to the PID controller and the difference between the measured temperature and the setting temperature determined the control action of the PID controller. Before using, the engineering experience method was used to tune the control parameters of the PID controller. The regulation deviation of used PID controller was ±0.2 °C and the setting temperature was 45 °C. When the fan turned on, the cabinet inside temperature dropped was lower than the 44.8 °C. The PID controller output 4–20 mA DC electric current signals to electronic voltage module, and then electronic voltage module added the voltage which supplied to electric heating wire. Thus, the power of the electric heating wire was also increased to heating the inside temperature to setting value. With the fan turned on and off, the heat transfer coefficients of structure 3 gasket were

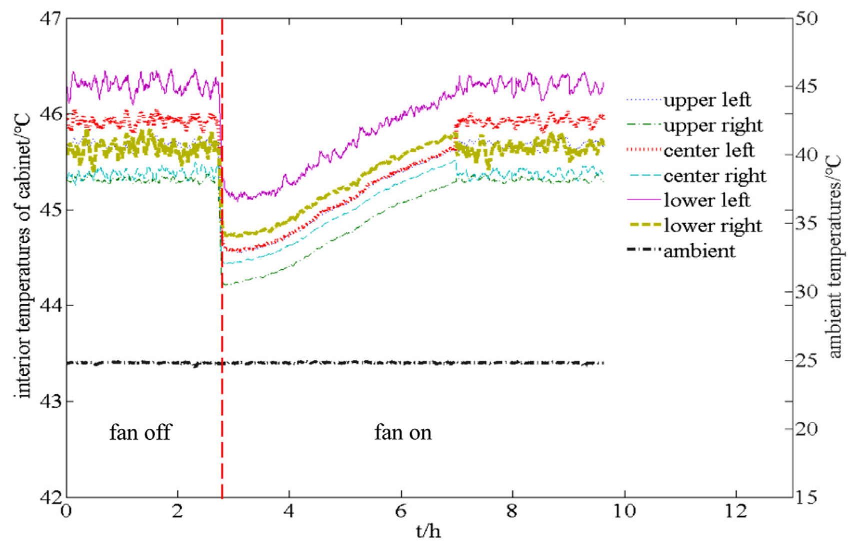

Fig. 14. Influence of fan on interior temperature (133 L refrigerator as an example).

0.04569 W/m·°C and 0.05358 W/m·°C, while structure 4 gasket were 0.04472 W/m·°C and 0.05264 W/m·°C. The heat transfer coefficients of structure 3 gasket and structure 4 gasket were increased by 17.27% and 17.71% respectively. This is very close to the simulation results in reference (Gao et al., 2017). The experimental platform can test the heat transfer of gasket for frost-free refrigerators and direct-refrigerating refrigerators, with the fan turned on and off.

4. Conclusions

Experimental research on test method of heat transfer coefficient for refrigerator gasket was carried out in this paper. A testing platform based on RHLM was established, and 535 L, 133 L and 108 L refrigerators were used as test boxes to explore the gasket heat transfer coefficient testing methods. Due to the difficulty to test the heat leakage of gasket directly, an indirect method by the difference between heater power and cabinet walls heat leakage was adopted. In order to obtain accurate heat leakage of cabinet walls, heat flux test zones were divided strictly in conformity with the thermal resistance distribution of cabinet walls. The calculated instrumentation uncertainties indicated that the number of heat flux test zones has large influence on experiment repeatability. Characteristic heat flux method was proposed to simplify test zones. The results also indicated that the deviations of tests are with ±5% by using small refrigerator as the test. Through testing the gaskets with different structures, the heat transfer coefficient of common gasket was about 0.045 W/m·°C. And it can be concluded that the auxiliary air cell, multi-air cells, stiffener and auxiliary edge structure can improve the insulation of gasket. The fan in the cabinet contributed to an increase of 17% to heat leakage of gasket. The experimental test method proposed in this paper can be used to evaluate the heat transfer coefficient of refrigerator and freezer. The test results have two applications. The refrigerator designer can use the test results to calculated the heat leakage of gaskets More accurate when estimate the total heat leakage of a refrigerator. In addition, the gasket designer can develop more energy-saving gasket based on the test results.

Declaration of Competing Interest

None.

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