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Numerical Simulations of a Micro-Channel Wall-Tube Condenser for Domestic Refrigerators*

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Abstract: In recent years, microchannel heat exchangers have begun to be used in refrigeration and air conditioning systems. This paper introduces a microchannel condenser for domestic refrigerators with a theoretical model to evaluate its performance. The model was used to obtain the optimal design parameters for different numbers of tubes and tube lengths. The results show that the needed tube height of the downward section decreases with the number of tubes and the tube diameter. Compared with the original condenser, the present optimal design parameters can reduce the total metal mass by 48.6% for the two wall two side design and by 26% for the two wall one side design. Thus, the present condenser is much better than the condensers usually used in actual domestic refrigerators.

Key words: refrigerator; micro-tube; condenser; numerical simulation

Introduction

With the development of technology and people's increasing desire for aesthetics, most wire-and-tube condensers have been replaced by wall-tube condensers in domestic refrigerators. The wall-tube condensers are often installed on the left and right sides of the refrigerator or on the back in some cases. The serpentine tubes are fixed by aluminum foil tightly to the inner side of the wall. The condenser design is often based on the designer's experience with few studies published in the literature. Bansal and Chin[1] simulated the heat transfer characteristics of a wall-tube condenser and found that the predicated heat transfer rate disagreed with the experimental data by 10%. Gupat and Gopal[2] developed a mathematical model of hot-wall condensers and found that their predictions agreed well with published experimental data. They found that the aluminum tape used to fix the condensing tube to the outer wall plays a significant role in the heat transfer from the condenser to the environment. However, the condensing tubes used in their work were conventional serpentine tubes with the refrigerant flowing horizontally.

Many studies of heat transfer in micro-channels were conducted starting in the 1980s with various practical micro-channel heat exchanger designs developed, such as for automobile air conditioners and electronic cooling. Their use in domestic air conditioners has also been evaluated. Li and Li[3] evaluated the use of micro-channel heat exchangers using multi-port aluminum flat tubes for domestic air conditioners to show that the micro-scale heat transfer enhancement improved the heat transfer, especially as the price of copper increased. Apart from the augmented heat

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transfer, the total material consumption was also reduced. In addition, many companies have begun to do research on micro-channel heat exchangers for air conditioners.

Li et al.[4] developed a micro-channel wall-tube condenser to adopt the micro-scale heat transfer technology to domestic refrigerator designs and reduce the material usage. Their condenser design used small diameter tubes with diameters of about 1 mm or even less that were straight rather than the serpentine tubes used in most domestic refrigerators now. A sketch of this kind of condenser is shown in Fig. 1. The small diameter tubes increase the heat transfer coefficients on the tube side while reducing the pressure drop by using multiple tubes installed in parallel with a larger total flow cross sectional area which reduces the flow velocity. The reduced diameter also reduces the total volume and mass of the material.

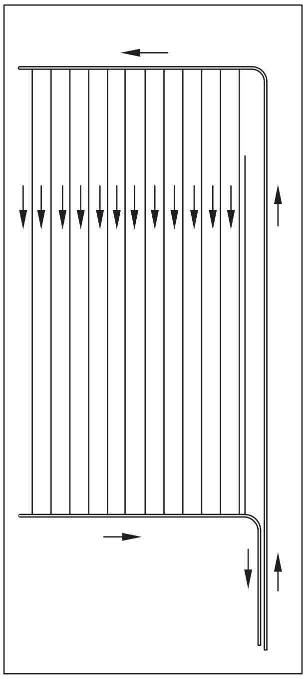

Fig. 1 Sketch of the micro-channel wall-tube condenser

This micro-channel condenser was studied experimentally by Song[5] by installing the micro-channel condenser directly in place of the old condenser in the original refrigerator which was still popular and typical in the market. He found that the pressure drop on the tube side was reduced and the heat transfer coefficients in the saturated section were increased greatly. He also found that the COP of the refrigeration cycle was improved by more than 5%.

In the present paper, the refrigerant flow and heat transfer processes of the condenser developed by Li and Li[3] are analyzed numerically for the same working conditions as for the condenser used in the original refrigerator. The effects of tube diameter and the numbers of tubes on the tube height of the downward section and the pressure drop are analyzed. Finally, the best parameters for practical designs are selected based on the present calculated results.

1 Micro-Channel Wall-Tube Condenser

Figure 1 shows a sketch of the micro-channel wall-tube condenser in which the refrigerant flow in the condenser is labeled with solid arrows. The condenser contains upper and lower manifolds. The superheated refrigerant vapor from the compressor's outlet first flows in the upward section to the upper manifold and then flows into the downward section. The downward section is made of a bundle of micro-tubes fixed on the wall in parallel connected by aluminum foil with uniform spacing in between. The condensation begins when the vapor enters the condenser. In the upward section, the vapor will be partially condensed. The energy from the condensing vapor is transferred to the inner wall, then to the outer wall of the micro-tubes through conduction, and then the heat is transferred to the environment by natural convection and radiation. The heat transfer processes are shown in Fig. 2 and labeled with solid arrows. At the condenser outlet, the vapor has condensed to sub-cooled liquid and flows to the throttling valve or capillary tube.

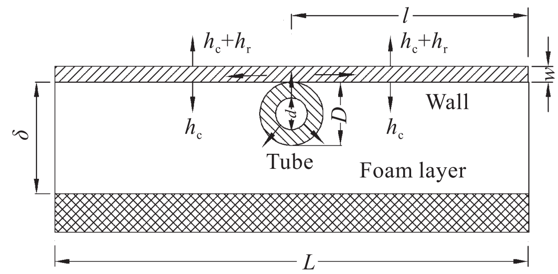

Fig. 2 Computational element cross section

To increase the heat transfer area, the condenser could be installed with a small gap between the wall and the inner foam layer, as shown in Fig. 2 and labeled with δ. In this way, the heat could be transfer by natural convection in the gap from the inner side of the wall and by natural convection and radiation from the outer side of the wall, increasing the heat transfer area. Theoretically, some heat will be also dissipated by radiation through the top and bottom ends of the gap, but because the radiation angle from the wall to the top and bottom ends is very small, the heat dissipation by radiation through the gaps can be neglected. Two wall-tube condensers are usually needed for a refrigerator and are installed on the left and right sides of the refrigerator. However, with a micro-channel wall-tube condenser, only one condenser may be needed for the heat dissipation if the gap is appropriately sized because of the additional heat transfer due to natural convection in the gap.

There are four installation modes for this condenser which are referred to as two wall two side (TWTS), two wall one side (TWOS), one wall two side (OWTS), and one wall one side (OWOS). TWTS means there are two walls and two gaps between the wall and the inner foam layer, with the heat transferred from two walls (left and right) and two sides (outside and inside) of the two walls. TWOS means that the width gap, δ, is zero and the wall is adjacent to the foam layer so there is no space for natural convection heat transfer between the wall and the foam layer. The other two modes are similar to these modes but with only one condenser installed on one side (left or right) of the refrigerator. Since the heat transfer area for the OWOS mode is too small and definitely cannot satisfy the heat transfer need, this installation mode is not discussed in the present work.

2 Calculational Models

2.1 Heat transfer model

The heat transfer involves the tube side flow, the conduction in the tube wall, and the air side flow. On the tube side, the refrigerant flow in the micro-tubes can be divided into a single-phase flow section (superheated vapor or sub-cooled liquid) and a two-phase flow section. For the single-phase flow, the heat transfer coefficients were calculated using the Dittus-Boelter correlation[6]. For the two-phase flow, the heat transfer coefficients were calculated using the Shah[7] correlation:

$$h_{\mathrm{tp}} = h_{\mathrm{l}} \frac{(1-x)^{0.8} + 3.8x^{0.76}(1-x)^{0.04}}{0.38 \ln p_{\mathrm{r}}} \tag{1}$$

where h_{l} was also calculated using the Dittus-Boelter[6] correlation as for the single-phase section assuming all the refrigerants are single-phase liquids.

Equation (1) has been widely used to predict the heat transfer coefficients in conventional large diameter tubes. However, many studies have reported that Eq. (1) underpredicts the condensation heat transfer coefficients in small or micro-tubes, so this analysis may underpredict the heat transfer which would result in a larger tube than necessary. Since there are no other better general correlations for condensation heat transfer in micro-tubes, the Shah[7] correlation is used here and the error is very small because the thermal resistance on the refrigerant side is very small.

The conduction in the tube wall was assumed to be one dimensional, because the temperature gradient in the refrigerant in the axial direction was very small, especially in the two-phase flow section and the high thermal conductivity of tube wall gives a very low thermal resistance.

Outside the tube, the heat was assumed to be dissipated by natural convection and radiation from the wall. Since the refrigerator was set up in a large room which could be thought of as an infinite space, the heat transfer coefficients were calculated using a correlation for natural convection along a vertical flat surface. If the natural convection was laminar, the heat transfer coefficients could be calculated using[6]

$$Nu = 0.59(GrPr)^{0.25} \tag{2}$$

If the natural convection was between laminar and turbulent, the heat transfer coefficients was calculated using[6]

$$Nu = 0.0292(GrPr)^{0.39} \tag{3}$$

When the condenser was installed with a gap (for the TWTS or OWTS modes) between the wall and the foam layer, the natural convection in the gap should also be considered. The space between the wall and foam layer was assumed to be so large that the gap could be thought of as an infinite space. Thus, the heat transfer coefficients for natural convection in the gap could also be found using Eq. (2).

In general, the wall is coated with a gray lacquer whose emissivity is about 0.9. The radiation heat transfer coefficients were determined based on the average wall temperature and the average air temperature in the surrounding surface calculated as

$$h_{\mathrm{r}} = \varepsilon\sigma \frac{T_{\mathrm{w}}^4 - T_{\mathrm{inf}}^4}{T_{\mathrm{w}} - T_{\mathrm{inf}}} \tag{4}$$

with T_{w} calculated as

$$T_{\mathrm{w}} = \eta_{\mathrm{f}}(T_{\mathrm{o}} - T_{\mathrm{inf}}) + T_{\mathrm{inf}} \tag{5}$$

The fin efficiency, η_{f}, was calculated for the micro-tubes having uniform fin spacing, so that the plates adjacent to the tubes could be modeled as rectangular fins on the left and right sides as shown in Fig. 2 with the fin height equal to half the distance between tubes and the edge being adiabatic due to the symmetry. Thus, the fin efficiency was calculated as:

$$\eta_{\mathrm{f}} = \tanh\left(\frac{mL}{2}\right) \tag{6}$$

$$m = \sqrt{\frac{h_{\mathrm{o}}}{\lambda_{\mathrm{w}} w}} \tag{7}$$

where *m* depends on the gap between the wall and the foam layer. With the gap, the heated perimeter is doubled, so the numerator under the radical sign is doubled.

2.2 Thermal resistance model

The calculation analyzed the tube in steps from the condenser inlet to the outlet with the lengths of the computational elements, d, being 0.01 m for the upward section and 0.02 m for the downward section. For each element, d, the inner thermal resistance on the tube side was

$$R_{\mathrm{i}} = \frac{1}{h_{\mathrm{i}} A_{\mathrm{i}}} = \frac{1}{h_{\mathrm{i}} \pi d_{\mathrm{i}} dl} \tag{8}$$

The thermal conduction resistance in the tube wall was

$$R_{\mathrm{c}} = \frac{\ln(d_{\mathrm{o}} / d_{\mathrm{i}})}{2\pi\lambda dl} \tag{9}$$

The thermal resistance on the air side needed to calculate the heat transfer from the element is based on a fin analysis. The heat transfer through the left or right half wall was calculated assuming one dimensional conduction model.

$$\frac{\partial^2 T}{\partial x^2} - \frac{hP}{\lambda_{\mathrm{w}} A_{\mathrm{c}}}(T - T_{\mathrm{inf}}) = 0 \tag{10}$$

The total heat transfer from the element is then given by

$$Q = \sqrt{h_{\mathrm{o}} P \lambda_{\mathrm{w}} A_{\mathrm{c}}} \tanh\left(\frac{mL}{2}\right) \tag{11}$$

Therefore, the thermal resistance on one side is

$$R_{\mathrm{o1}} = \frac{1}{\sqrt{h_{\mathrm{o}} P \lambda_{\mathrm{w}} A_{\mathrm{c}}} \tanh\left(\frac{mL}{2}\right)} \tag{12}$$

with the symmetry of the walls on the left and right sides of the tube, the heat transfer is dissipated equally to both sides from the central tube. The thermal resistances on both sides are identical and in parallel, so the total resistance on the air side is

$$R_{\mathrm{o}} = \frac{1}{2\sqrt{h_{\mathrm{o}} P \lambda_{\mathrm{w}} A_{\mathrm{c}}} \tanh\left(\frac{mL}{2}\right)} \tag{13}$$

The total thermal resistance is then the sum of the inner, conduction, and outer resistances:

$$R_{\mathrm{t}} = R_{\mathrm{i}} + R_{\mathrm{c}} + R_{\mathrm{o}} \tag{14}$$

The temperature difference for the heat transfer is the difference between the inner temperature of the refrigerant and the air in the room. In the two-phase section, the inner temperature is the saturation temperature corresponding to the local pressure. Thus, the heat transfer in each element is

$$Q_{\mathrm{e}} = \frac{T_{\mathrm{i}} - T_{\mathrm{inf}}}{R_{\mathrm{t}}} \tag{15}$$

2.3 Pressure drop model

As with the heat transfer coefficients, the pressure drop analysis was divided into two single-phase flow sectors and the two-phase flow section. For the single-phase flow, superheated vapor or sub-cooled liquid, the flow resistance factor coefficient for laminar or turbulent flow was calculated as:

$$f = \frac{64}{Re_{\mathrm{f}}}, \ Re_{\mathrm{f}} \leqslant 2300 \tag{16}$$

$$f = 0.3164 Re_{\mathrm{f}}^{-0.25}, \ Re_{\mathrm{f}} > 2300 \tag{17}$$

For the two-phase flow, the flow decreases as condensation occurs, leading to a pressure rise, which is called the deceleration pressure rise, Δp_{a}. The deceleration pressure rise was determined by the momentum change between the tube inlet and outlet calculated as:

$$\Delta p_{\mathrm{a}} = G^2 \left\{ \left[\frac{x^2}{\rho_v \alpha} + \frac{(1-x)^2}{\rho_l \alpha} \right]_{\mathrm{inlet}} - \left[\frac{x^2}{\rho_v \alpha} + \frac{(1-x)^2}{\rho_l \alpha} \right]_{\mathrm{outlet}} \right\} \tag{18}$$

The additional pressure drop due to gravity, Δp_{g}, depends on the flow direction. If the flow is upward, the gravitational effect is negative and leads to an additional pressure drop, but if downward, the effect will be positive and will lead to an additional pressure rise which is very large for low vapor qualities. The gravitational pressure drop or rise was calculated using the homogenous model:

$$\Delta p_{\mathrm{g}} = [\rho_v \alpha + \rho_l (1-\alpha)] g dl \tag{19}$$

where the void fraction was calculated using the model developed by Kawaji and Chung[8].

The flow friction resistance predominates in the upward section with the friction pressure drop calculated using the separated phase model. The modified L-M model by Chisholm[9] was used to calculate the two-phase friction factor, f_{th}, where the L-M parameter is

$$X = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_l}\right)^{0.1} \tag{20}$$

and the two-phase flow friction factor is

$$f_{\mathrm{th}} = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{21}$$

where C is a constant recommended by Bansal and Chin[1] as C=8 for the present working conditions. The pressure drop in the upper manifold was neglected and the mass flow rates in the various micro-tubes were assumed to be identical.

3 Computational Procedure

The software Matlab was used to simulate the heat transfer in the condenser. The total condenser width was limited to 0.45 m. The temperature and pressure at the condenser inlet and the refrigerant mass flow rate were based on measured values from standard tests of the compressor used in the original refrigerator tests.

The analysis first defined the constants and parameters at the condenser inlet with the outer wall temperature of the tube assumed for the characteristic temperature and thermodynamic properties of the air. Then the heat transfer coefficients and resistances were calculated and the heat transfer in one element was determined. Then, a new outer wall temperature was calculated. The computational process was repeated until the temperature changes were small enough. The pressure drop was then calculated from the friction, gravitational, and deceleration effects. Finally, the parameters for the current element outlet were used as the input parameters to the next element. The total tube height in the downward section was calculated to be the sum of the element lengths of every step until the total heat transfer was equal to that of the condenser in the original refrigerator.

All the thermodynamic properties were calculated using REPROF Version 7.01[10]. The upward section was the same for all cases with only the number of tubes and the tube height of the downward section changed, so the three sets of parameters for the micro-tubes inlet for the three designs, TWTS, TWOS, and OWTS were the same for the various downward tube designs. The effects of the number of tubes in the downward section on the tube height, pressure drop, and total mass of the tubes were analyzed to optimize the design.

4 Results and Discussion

4.1 Upward section

The tube used in the upward section is the same as the connecting tube with an inner diameter of 2.98 mm and an outer diameter of 4 mm. The refrigerant state at the end of the upward section was constant but differs for the TWTS, TWOS, and OWTS designs. The states at the end of the upward section are given in Table 1 for the three designs.

Table 1 Thermodynamic states at the end of the upward section

Design	p/kPa	T/K	Q/W	D_{P}/Pa
TWTS	761.10	327.52	15.21	−998.31
TWOS	761.30	327.53	9.49	−798.23
OWTS	760.37	327.48	15.92	−1728.84

Figure 3 shows the changes of the pressure drop (D_{P}), heat transfer, and average outer wall temperature for the TWTS case. Since the refrigerant state is very close to saturation, the wall temperature, which depends on the condensation heat transfer coefficients, increases quickly to a maximum and then decreases with decreasing vapor mass quality. As the condensation continues, the heat from the refrigerant to the environment and the total pressure drop increase nearly linearly.

4. 2 Downward section

Since the tube height and the total tube mass differ for the different numbers of tubes, the heat transfer rate and pressure drop were calculated for various numbers

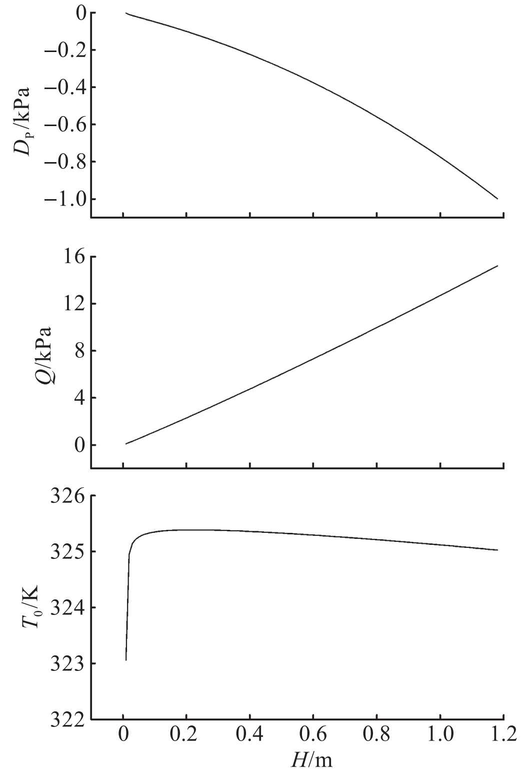

Fig. 3 Variations of the pressure drop, heat transfer, and outer wall temperature along the upward section for the TWTS case

of tubes for the three designs (TWTS, TWOS, and OTWS). The best parameters for the optimal practical design were selected based on the geometry, pressure drop, and total tube mass and compared with the original condenser.

4.2.1 Tube height

Figure 4 shows the tube height needed for the downward section for the various tube diameters and numbers of tubes. The total tube height was calculated so that the heat transfer from the tubes was equal to the heat transfer in the original condenser. As expected, the tube height decreases with the number of tubes and the tube diameter, because smaller tube diameters give higher heat transfer coefficients for the same condition and higher average wall temperatures, improving the heat transfer efficiency. However, in the smaller diameter tubes, the pressure drop is higher which lead to lower average wall temperatures for saturated conditions and fin efficiencies, so smaller diameter tubes results in larger tube heights. The TWTS design has the largest heat transfer area which is the predominant factor affecting the tube height.

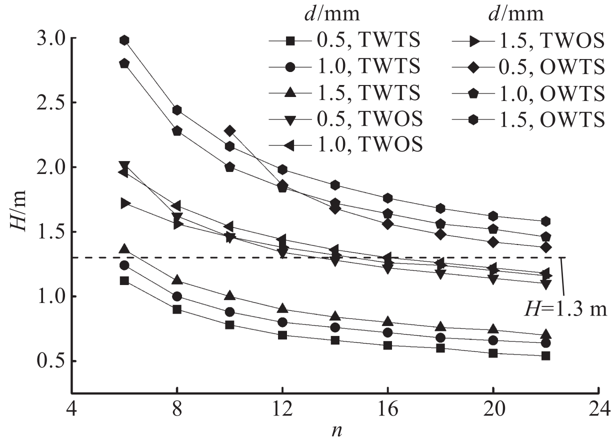

Fig. 4 Tube height needed for the downward section for various tube diameters and numbers of tubes

If the tube height is larger than 1.3 m, other devices installed on the refrigerator will be affected, so the tube height must be less than 1.3 m to replace the original condenser. Therefore, tube heights greater than 1.3 m (above the dashed line in Fig. 3) must be excluded. Only the small diameter tubes in the TWOS designs and all the results for the TWTS designs satisfy this condition.

4.2.2 Total mass

The total mass of metal (the copper tubes used in the upward and downward section) in the original condenser was 1.517 kg. The new condenser should have less total mass to be an improvement over the original condenser design. Figure 5 shows the total masses of the various condensers analyzed in this work. The original condenser's total mass is shown by the dashed line in Fig. 5. The result shows that only the TWTS design with more than 22 tubes has more mass than the original condenser. Therefore, this type of micro-channel condenser can greatly reduce the material consumption.

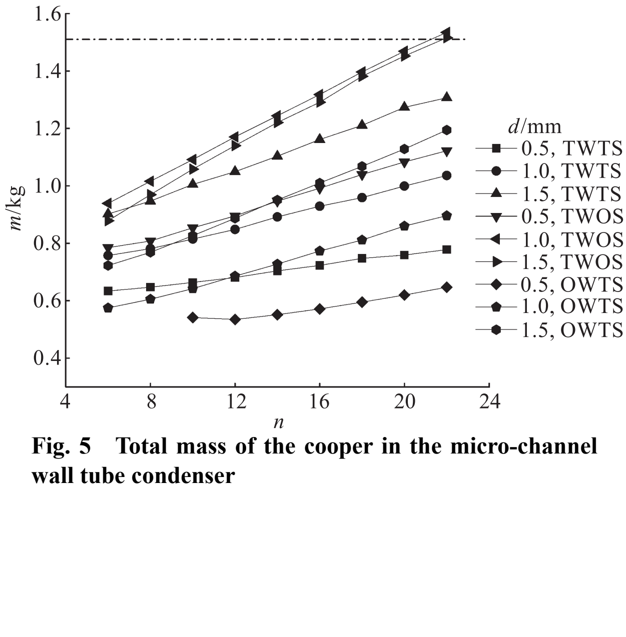

Fig. 5 Total mass of the cooper in the micro-channel wall tube condenser

4.2.3 Pressure drop

Apart from the geometry and lower total mass of metal limits, the total pressure drop is also important, because lower pressure drops mean higher average condensation temperatures and higher heat dissipation rates from the wall. Figure 6 shows the pressure drop for various numbers of tubes in the downward section for the TWTS design. The pressure change decreases with the number of tubes due to the increasing flow cross sectional area which reduces the average flow velocity in the tubes. For diameters less than 0.5 mm, the pressure drop was negative as expected, but positive for diameters of 1.0 or 1.5 mm with the pressure increasing downstream as the gravitational pressure rise exceeded the friction pressure drop. If the total pressure drop in the upward and downward sections is positive, the pressure at the outlet will be larger than at the inlet, which is not true in practice. If this condenser design were installed in a refrigerator, the refrigeration cycle will change so that the pressure decreases along the tubes. Our aim is to replace the original condenser with a micro-channel condenser having lower pressure drop and mass without affecting the refrigeration cycle. Therefore, the flow conditions resulting in a pressure rise were not considered in the optimization analysis. Figure 6 shows that the pressures decreased for the 0.5 mm diameter for the TWTS design which was also true for the other two designs. For the TWOS design with less than 14 tubes, the tube height exceeded the geometry limits, so that data was also excluded in Fig. 7.

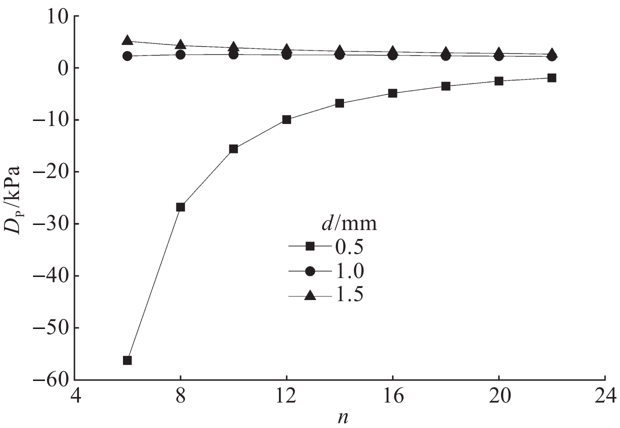

Fig. 6 Total pressure drop for various tube diameters for the TWTS design

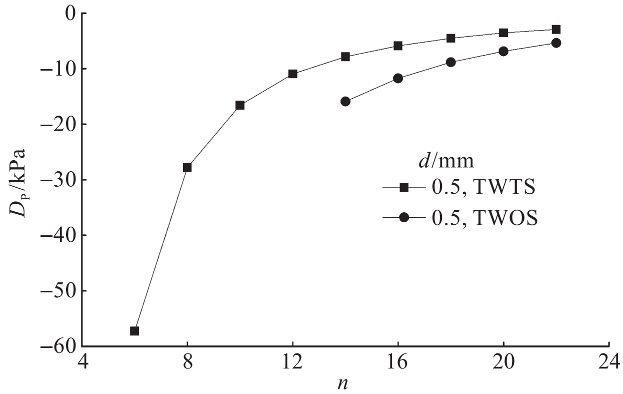

Fig. 7 Total pressure drops for designs that satisfy the design limits

5 Optimal Parameter Selection

The optimal design parameters for the micro-channel condenser were then based on these three limits. The designs that satisfy all three design limits are listed in Table 2. The TWTS design with 22 tubes has the lowest pressure drop but also has the highest total mass of the TWTS designs. Therefore, since the pressure drop probably has a larger impact on the operating costs, the design with the lowest pressure drop was selected as the best design. The best design with the minimum pressure drop had a total mass 48.6% less than the original condenser for the TWTS design and 26% less than the original condenser for the TWOS design.

Table 2 Designs that satisfy all three design limits

n	TWTS			TWOS		
	H/m	D_P/Pa	m/kg	H/m	D_P/Pa	m/kg
6	1.12	−57 254.00	0.63	N/A	N/A	N/A
8	0.90	−27 787.80	0.65	N/A	N/A	N/A
10	0.78	−16 584.30	0.66	N/A	N/A	N/A
12	0.70	−10 932.20	0.68	N/A	N/A	N/A
14	0.66	−7843.26	0.70	1.28	−15 910.00	0.95
16	0.62	−5890.54	0.72	1.22	−11 725.10	1.00
18	0.60	−4529.26	0.75	1.18	−8839.90	1.04
20	0.56	−3546.27	0.76	1.14	−6872.01	1.08
22	0.54	−2933.24	0.78	1.10	−5353.73	1.12

6 Conclusions

(1) The required tube height in the downward section of a refrigerator with a micro thermal condenser decreases with increasing number of tubes and tube diameter for a fixed total heat transfer rate, with the total mass increases with the number of tubes and the tube diameter.

(2) The OWTS design can not be used in practice because the gravitational pressure rise is higher than the friction pressure drop, which makes the pressure at the condenser outlet larger than at the inlet. In some cases, the TWOS design also had the same pressure increase problem as in the OWTS design, especially with less tubes.

(3) The present results show that the total mass of metal for the best design parameters for the TWTS design was 48.6% less than for the original condenser and 26% less for the TWOS design.

(4) The micro-channel wall-tube condenser for domestic refrigerators can significantly reduce the material usage in the condenser.

Nomenclature

A	Area, m^2
d	Diameter, mm
f	Friction factor
g	Gravitational acceleration, m/s^2
G	Mass flux, kg/(m$^2\cdot$s)
Gr	Grashof number, $g\alpha_v \Delta t l^3/\nu^2$
h	Heat transfer coefficients, W/(m$^2\cdot$K)
H	Height, m
L	Element length, m
l	Element half length, m
m	Mass, kg
n	Number of tubes
P	Perimeter, m
p	Pressure, Pa
Pr	Prandlt number, $\mu C_p/\lambda$
R	Thermal resistance in unit area, K/W
Re	Reynolds number, ul/ν
T	Temperature (K)
w	Wall thickness, m
x	Vapor mass quality

Greek symbols

η	Fin efficiency
μ	Viscosity, Pa$\cdot$s
δ	Gap width, m
ε	Radiation emissivity
ρ	Density
α	Void fraction
λ	Thermal conductivity, W/m
σ	Boltzmann constant, 5.67×10^{-8} W/(m$^2\cdot$K^4)

Subscripts

a	Air
c	Convection or conduction
i	Inner
inf	Infinite
f	Friction, fluid or fin
g	Gravitational acceleration
l	Liquid phase
o	Outer
r	Radiation or relative
t	Total
tp	Two phase
v	Vapor phase
w	Wall

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