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Performance enhancement of cabinet cooling system by utilizing cross-flow plate heat exchanger

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ABSTRACT

Gas-gas plate heat exchanger is an important component to remove heat generated from electronic devices in the cabinet cooling system. The counter-flow plate heat exchanger is usually used due to its higher heat transfer performance than the cross-flow plate heat exchanger. However, in the cabinet cooling system, the overall dimensions for the heat exchanger is limited. Therefore, it is necessary to consider the overall dimensions of the system during heat exchanger design, but the traditional thermal design method of heat exchanger doesn't consider the effect of system parameters. In this paper, a cross-flow plate heat exchanger is proposed to improve the cooling performance of cabinet cooling system, and is compared with the counter-flow plate heat exchanger. The ε-NTU method and effectiveness-thermal resistance method are applied to evaluate the performance. It is found that for large width of cabinet cooling system, the system with cross-flow plate heat exchanger has higher cooling performance and lower thermal resistance than the system with counter-flow plate heat exchanger. When the width is 700 mm, the cooling capacity of the two systems are 176.13 W/K and 138.95 W/K, respectively. The dimensionless thermal resistance can characterize the irreversibility of the heat transfer at constant mass flow rate.

1. Introduction

Plate heat exchangers are widely used in energy consuming and handling industries [1]. Generally, plate heat exchangers are used for liquid–liquid heat exchange [2]. Imran et al. [3] optimized liquid–liquid plate heat exchanger. The overall dimensions of the heat exchanger were independently considered in their research. Miao et al. [4] numerically studied liquid–liquid plate heat exchangers. The heat exchanger was counter-flow plate heat exchanger. Two methods were utilized to get simulation results, respectively. The results of grey-box method were better. Kumar et al. [5] experimentally studied the liquid–liquid plate heat exchangers and obtained correlation based on their experimental data, in which the overall dimensions of the measured heat exchangers were constant. Yang et al. [6] obtained correlation based on their experimental data and empirical correlations in open literature. The overall dimensions of heat exchangers in their work were varied. Counter-flow plate heat exchanger also could be applied for two-phase heat exchange [7] and the heat exchangers in organic Rankine cycle could be counter-flow plate heat exchangers for better performance [8]. However, due to the primary heat transfer surfaces and light weight the plate heat exchanger was developed for gas–gas heat exchange [9] and even for liquid–gas heat exchange [10]. Cross-flow arrangement was convenient for liquid–gas plate heat exchanger [11]. High temperature gas–gas heat exchangers were mainly applied for the microturbine recuperated cycle system [12] and air preheater of the fossil fuel power system [13]. Recuperators for micro gas turbine was reviewed by Xiao et al. [14] recently. Wang et al. [15] improved the performance of air preheater by optimizing geometrical parameters of primary surface. Vafajoo et al. [16] utilized a two-dimensional mathematical model to simulate thermal–hydraulic performance of plate heat exchanger. This model was developed to gas–gas heat exchange under compressible and turbulent flow conditions. Two different chevron-type plates were used to replace the flat plate to enhance heat transfer. Li and Gao [17] numerically studied heat transfer and turbulent flow in a gas–gas plate heat exchanger, and the heat exchanger was a cross-corrugated (CC) primary surface heat exchanger. They added baffles to improve the heat transfer. Ma et al. [18] evaluated heat transfer and flow resistance of a gas–gas plate heat exchanger, and the heat exchanger was a cross-wavy (CW) primary surface heat exchanger. They found that the entrance area had small effect and similar thermal–hydraulic performance was obtained for channels with different equivalent diameters. Wang et al. [19] studied heat transfer and flow performances of gas–gas cross-flow CC primary surface heat exchangers experimentally, in their experiment the fluids on

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Nomenclature

a	heat transfer area of one plate, mm^2
A_i	cross sectional area of channel inlet mm^2
A	overall heat transfer area, mm^2
c_p	specific heat, J/(kg K)
C	heat capacity rate, W/K
C_r	heat capacity ratio
d	plate pitch, mm
D_f	fan diameter, mm
f	friction factor
h	heat transfer coefficient, W/(m^2 K)
H	cabinet cooling system height, mm
H_{co}	counter-flow plate heat exchanger height, mm
H_{cr}	cross-flow plate heat exchanger height, mm
H_f	fan height, mm
L	cabinet cooling system length, mm
L_{co}	counter-flow plate heat exchanger length, mm
L_{cr}	cross-flow plate heat exchanger length, mm
m	mass airflow rate, kg/s
n	number of heat exchange channels
n_c	number of cold channels
n_h	number of hot channels
NTU	number of heat transfer units
Nu	Nusselt number
Pr	Prandtl number
q	heat transfer rate, W
q_p	cooling capacity, W/K
Re	Reynolds number
R_{ex}	equivalent thermal resistance, K/W
R^*	dimensionless thermal resistance
T	temperature, K
u	averaged inlet velocity, m/s
U	overall heat transfer coefficient, W/(m^2 K)
W	cabinet cooling system width, mm
W_{co}	counter-flow plate heat exchanger width, mm
W_{cr}	cross-flow plate heat exchanger width, mm

Greek

ΔE	entransy dissipation rate, W K
ε	heat exchanger effectiveness
λ	thermal conductivity, W/(m K)
ν	kinematic viscosity, m^2/s
ρ	density, kg/m^3

Subscripts

co	counter flow
cr	cross flow
c	cold side
h	hot side
i	inlet
o	outlet

both sides were air. The hydrophilic polymer membrane gas–gas plate heat exchanger was utilized in air ventilation [20], the plate heat exchanger could transfer the heat and moisture simultaneously. Lu et al. [21] experimentally and theoretically analyzed a plastic film cross-flow plate heat exchanger. The film vibrated when the airflow passed through the channels and the vibration improved the heat transfer. Bermejo-Busto et al. [22] connected several gas–gas cross-flow plate heat exchangers in a cascade layout and installed the heat exchangers in two different locations. This new configuration of heat exchangers, which saved energy, had been simulated with computational fluid dynamics (CFD) software.

One of the classical methods for the thermal design of heat exchangers is ε-NTU (Effectiveness-Number of Transfer Units) method [23]. Dizaji et al. [24] measured the coiled tube heat exchanger. They found that the air bubble injection strongly increased NTU and ε. Sammeta et al. [25] simulated a 9-plate counter-flow corrugated plate heat exchanger using CFD software. The ε-NTU chart of the heat exchanger was obtained from simulated data. They found that effectiveness was increased with increment of NTU and decrement of heat capacity ratio. When the effectiveness was less than 0.4, the effects of heat capacity ratio on effectiveness was small. Rogiers et al. [26] optimized the counter-flow plate heat exchanger by using ε-NTU based thermal design. They found that down sized the plate heat exchanger at a constant pressure drop, the effectiveness exhibited a maximum value because of longitudinal heat conduction. The pure counter-flow heat transfer was considered for the whole channel of the plate heat exchanger. It was not possible for real plate heat exchanger due to the overlap of two fluids inlets and outlets. In their later work [27] similar investigation was applied on the cross-flow plate heat exchanger. Pure cross-flow heat transfer could be obtained in gas–gas plate heat exchanger. Gherasim et al. [28] calculated effectiveness of a multi-passage plate heat exchanger with a one-dimensional heat transfer model. The model considered dissipation and viscosity. They found that computational results were identical to the ε-NTU relations for low viscosity fluid. Fernández-Torrijos et al. [29] used the ε-NTU method to obtain a general expression of the effectiveness of plate heat exchanger. The expression was generalized for any heat exchanger with series–parallel configurations.

In the ε-NTU method, the effectiveness of heat exchanger reflects the thermal performance of the heat exchanger. The higher the effectiveness is, the greater the thermal performance is. An effectiveness-thermal resistance method for heat exchanger design and analysis was carried out by Guo et al. [30]. In the method the thermal resistance of a heat exchanger was defined based on the concept of the entransy dissipation rate [31]. Cheng and Liang [32] analyzed two-stream heat exchanger networks. They pointed out that with increment of effectiveness the thermal resistance always decreased. Rodríguez et al. [33] evaluated the ecological impact of shell-and-tube heat exchangers. In their work the entransy dissipation was combined with the ecological function.

The electronic devices are often housing by a telecommunication cabinet. The cabinet is usually cooled by fans directly [34]. A cabinet cooling system comprises a gas–gas plate heat exchanger, an internal fan and an external fan. Gas-gas plate heat exchanger is applied to remove the heat generated in cabinet and prevent water and dust from entering the cabinet. Here, internal air (hot fluid) and external air (cold fluid) are forced by internal and external fans, respectively, and convective heat transfer occurs in the gas–gas plate heat exchanger. At the same NTU and heat capacity ratio, the effectiveness of the counter-flow heat exchanger is higher than that of cross-flow heat exchanger, while the effectiveness of the parallel-flow heat exchanger is the lowest [35]. Similarly, at the same geometrical parameter and flow rates, the counter-flow heat exchanger has the highest heat transfer rate [36]. It is common sense that the counter-flow heat exchanger is better than the cross-flow heat exchanger, so the counter-flow gas–gas plate heat exchanger is widely used in the cabinet cooling system [37].

Generally, taken the heat exchanger into consideration independently, at the same dimensions, the counter-flow heat exchanger has better thermal performance than the cross-flow heat exchanger. However, both the counter-flow and cross-flow arrangements are used in some applications due to the restrictions of the heat transfer channel structures. CW primary surface heat exchanger only performs in the counter-flow arrangement, but CC primary surface heat exchanger performs in both counter-flow and cross-flow arrangements [38]. Most of the liquid–liquid plate heat exchangers are counter-flow [39].

The special feature of the cabinet cooling system is that it has restrictions in size. The geometrical parameters of the gas–gas plate heat exchanger are highly depended on the system's size. For the cabinet cooling system, the arrangements of the counter-flow and cross-flow plate heat exchangers are significantly different. Cross-flow heat transfer occurs for whole channel of cross-flow plate heat exchanger in cabinet cooling system. But for counter-flow plate heat exchanger in cabinet cooling system, the counter-flow heat transfer is not occurred for whole channel, because both-side air flow directions vary in outlet area of the heat exchange channels. The variation of flow direction also increases resistance of heat exchange channel. So the cross-flow heat exchanger may have better thermal performance than the counter-flow heat exchanger in the cabinet cooling system.

The thermal design of heat exchanger should be performed in the system level and the thermal performance of the heat exchangers should be compared under same overall dimensions of the cabinet cooling system. To the best of our knowledge, this work has not been reported in open literature. In this paper, a cross-flow plate heat exchanger is proposed to improve the performance of the cabinet cooling system. The ε-NTU method and effectiveness-thermal resistance method are used for thermal design of plate heat exchangers in the cabinet cooling system. The cooling performance of the systems with cross-flow plate heat exchanger and counter-flow plate heat exchanger are compared. The effects of overall dimensions are studied. The effect of mass flow rate on thermal resistances are also studied.

2. Methods

The outdoor electronic devices are often housing by a cabinet. Instead of cooling by a fan directly, the heat generated from electronic devices in the cabinet is removed by the cabinet cooling system. In this section, the cabinet cooling system with counter-flow plate heat exchanger and cross-flow plate heat exchanger are introduced. Then the methods and assumptions for thermal design of plate heat exchanger in the cabinet cooling system are introduced. At last, the thermal design of plate heat exchanger is validated.

2.1. Counter-flow cabinet cooling system and cross-flow cabinet cooling system

The cabinet cooling system is installed on the front surface of the telecommunication cabinet. The cabinet cooling system includes a gas–gas plate heat exchanger, an internal fan and an external fan. The internal and external fans are the same centrifugal fan. The shape of the fan is cylindrical, with a diameter of 175 mm and a height of 68 mm so that the fan needs a rectangular space of 200 mm × 200 mm × 70 mm. The cabinet cooling system height H is influenced by the fan height. Here, the variation of H is from 70 mm to 140 mm.

The cabinet cooling system with counter-flow plate heat exchanger is called counter-flow cabinet cooling system, as shown in Fig. 1(a), while that with cross-flow plate heat exchanger is called cross-flow cabinet cooling system, as shown in Fig. 1(b). In the counter-flow cabinet cooling system, the internal and external fans are installed at two ends of the heat exchanger, the length of the heat exchanger and two centrifugal fans determine the system length L. The length of the heat exchanger must be long enough to ensure the effective counter-flow heat transfer. Here, the value of L is at least 700 mm.

In the system, the hot and cold channels of the counter-flow plate heat exchanger are the same. Here, the length and width of the cross-flow plate heat exchanger are equal. So the hot and cold channels are the same in cross-flow cabinet cooling system. One diagonal line of the heat exchanger is parallel to the length of the system. Length and width of the cross-flow plate heat exchanger are not limited by L, when the value is 700 mm. However, the length and width of the cross-flow plate heat exchanger are strongly limited by the cabinet cooling system width W. At small W, the cross-flow plate heat exchanger length and width are so small, which leads to a bad thermal performance. Different from the counter-flow cabinet cooling system, the internal and external fans are not installed at two ends of the heat exchanger and can be installed beside each other. So the value of W is at least 400 mm.

Another important parameter of the plate heat exchanger is plate pitch d. The plates of the cross-flow plate heat exchanger are parallel to the cabinet surface so that number of the heat exchange channels is determined by heat exchanger height and plate pitch. For the counter-flow cabinet cooling system, the heat transfer plates are perpendicular to the cabinet surface, the number of heat exchange channels is determined by heat exchanger width and plate pitch. Heat transfer area of one plate is determined by L_{co}, H_{co} for counter-flow plate heat exchanger and by L_{cr}, W_{cr} for the cross-flow plate heat exchanger. The geometrical parameters of the heat exchangers are summarized in Table 1.

In this paper, flat plate heat exchangers are applied in the cabinet cooling systems. Filonenko and Gnielinski correlations are used to calculate the Fanning friction coefficient f and Nusselt number Nu, respectively (characteristic length is $2d$).

$$f = (1.58\ln Re - 3.28)^{-2},\ (2300 < Re < 10^{5}) \tag{1}$$

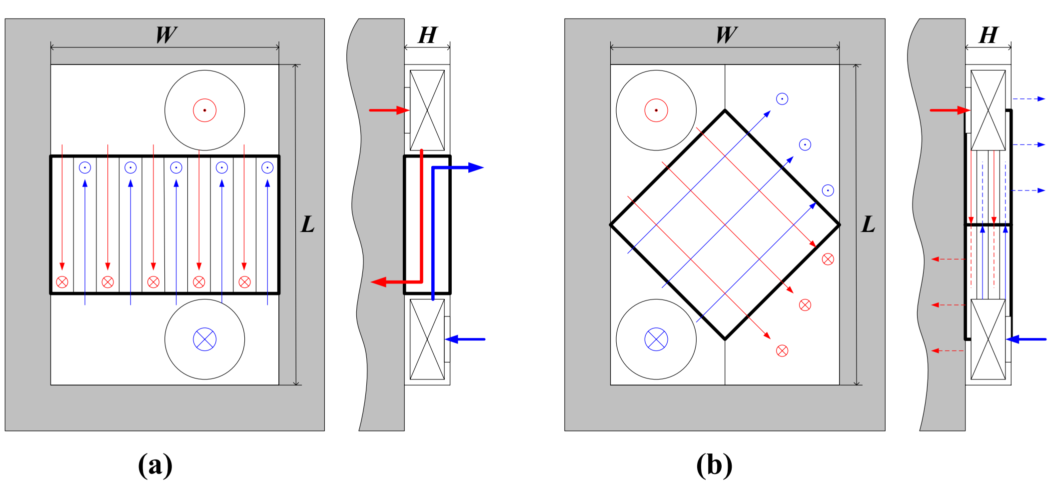

Fig. 1. Cabinet cooling system.

Table 1
Parameters of counter-flow and cross-flow plate heat exchangers.

	Counter-flow	Cross-flow
Length (mm)	$L_{co} = L - 400$ mm	$L_{cr} = 0.707\ W$
Width (mm)	$W_{co} = W$	$W_{cr} = 0.707\ W$
Height (mm)	$H_{co} = H$	$H_{cr} = H$
Number of heat exchange channels	$n = W_{co}/d$	$n = H_{cr}/d$
Cross sectional area of channel inlet (mm^2)	$A_i = dH_{co}$	$A_i = dL_{cr}$
Heat transfer area of one plate (mm^2)	$a = L_{co}H_{co}$	$a = L_{cr}W_{cr}$
Overall heat transfer area (mm^2)	$A = (2n - 1)a$	$A = (2n - 1)a$

$$Nu = \frac{(f/2)(Re - 1000)Pr}{1.07 + 12.7\sqrt{f/2}\,(Pr^{2/3} - 1)},\ (2300 < Re < 10^5) \tag{2}$$

where Pr is Prandtl number, Re is Reynolds number defined as below:

$$Re = \frac{2ud}{\nu} \tag{3}$$

where d is plate pitch, ν is the kinematic viscosity, u is averaged inlet velocity defined as below:

$$u = \frac{2m}{n\rho A_i} \tag{4}$$

where ρ is density, A_i is cross sectional area of channel inlet, m is mass flow rate and n is number of heat exchange channels. Here, the number of hot channels n_h is assumed to be equal to the number of cold channels n_c, which is

$$n_h = n_c = \frac{n}{2} \tag{5}$$

2.2. ε-NTU method and effectiveness-thermal resistance method

The thermal design based on ε-NTU method is widely used for various kinds of heat exchangers. For a given heat exchanger, the method is utilized to evaluate the heat transfer performance. Mass flow rates and inlet temperatures are usually known. The heat capacity rates of both fluids are calculated as:

$$C_h = (mc_p)_h \tag{6}$$

$$C_c = (mc_p)_c \tag{7}$$

where c_p is specific heat capacity and m is mass flow rate. Then the heat capacity ratio C_r can be calculated:

$$C_r = \frac{C_{min}}{C_{max}} \tag{8}$$

where $C_{min} = \min(C_c, C_h)$, and $C_{max} = \max(C_c, C_h)$.

The NTU is defined as follows:

$$\mathrm{NTU} = \frac{UA}{C_{min}} \tag{9}$$

where A is the overall heat transfer area and U is the overall heat transfer coefficient. Here, the plate thickness is negligible, so U is calculated by:

$$U = (h_h^{-1} + h_c^{-1})^{-1} \tag{10}$$

where h is the heat transfer coefficient, which calculated as:

$$h = \frac{Nu\lambda}{2d} \tag{11}$$

where λ is the thermal conductivity, d is plate pitch and Nu is Nusselt number.

The ε-NTU curves of both counter-flow and cross-flow heat exchangers are given by Kays and London [23]. Here, the effectiveness is obtained from ε-NTU curves.

Then the heat transfer rate q and outlet temperatures of the heat exchanger are obtained as:

$$q = \varepsilon C_{min}(T_{hi} - T_{ci}) \tag{12}$$

$$T_{ho} = T_{hi} - \frac{q}{C_h} \tag{13}$$

$$T_{co} = T_{ci} + \frac{q}{C_c} \tag{14}$$

where T is temperature, ε is effectiveness and C_{min} is the minimum heat capacity rate.

The effectiveness-thermal resistance method was proposed by Guo et al. [30]. The equivalent thermal resistance of a heat exchanger was defined based on the concept of the entransy dissipation rate. The entransy dissipation rate ΔE of the heat exchanger is defined as [31]:

$$\Delta E = \left(\frac{1}{2}, C_h, T_{hi}^2, +, \frac{1}{2}, C_c, T_{ci}^2\right) - \left(\frac{1}{2}, C_h, T_{ho}^2, +, \frac{1}{2}, C_c, T_{co}^2\right) \tag{15}$$

where T is temperature and C is heat capacity rate. And, the equivalent thermal resistance R_{ex} of the heat exchanger is defined as [30]:

$$R_{ex} = \frac{\Delta E}{q^2} \tag{16}$$

where ΔE is entransy dissipation rate and q is heat transfer rate.

Combining all the Equations above, the equivalent thermal resistance R_{ex} can be expressed as:

$$R_{ex} = \frac{1}{C_{min}}\left[\frac{1}{\varepsilon} - \frac{1}{2}(C_r + 1)\right] \tag{17}$$

where C_{min} is the minimum heat capacity rate, C_r is heat capacity ratio and ε is effectiveness. Then the dimensionless thermal resistance is calculated as follows [30]:

$$R^* = R_{ex}C_{min} = \frac{1}{\varepsilon} - \frac{1}{2}(C_r + 1) \tag{18}$$

Entransy dissipation rate reflects the irreversibility of heat transfer process. The thermal resistance hinders the irreversibility of heat transfer, the higher the thermal resistance of a heat exchanger is, the lower the heat transfer rate is. So the thermal resistance could be used to analyze the irreversibility of heat transfer process of the heat exchanger.

2.3. Assumptions

For thermal design of both counter-flow plate heat exchanger and cross-flow plate heat exchanger in the cabinet cooling system, the following assumptions are made to simplify the model:

(1) Both internal air and external air are incompressible gases with constant thermal physical properties and characteristic temperatures are the same because the maximum temperature difference is small.
(2) The width of the cross-flow plate heat exchanger equals to the length of the cross-flow plate heat exchanger, $W_{cr} = L_{cr}$.
(3) The plate thickness is negligible and plate pitch is constant at 2.5 mm.
(4) Mass flow rates of both-side airs are the same. The number of hot channels n_h and cold channels n_c are the same for both counter-flow plate heat exchanger and cross-flow plate heat exchanger.
(5) Internal air and external air are uniformly distributed for every hot channel and cold channel, respectively.
(6) The inlet temperatures of the internal air T_{hi} and external air T_{ci} are fixed.
(7) The cabinet surfaces are considered to be heat insulation. So, for the cabinet cooling system, the cooling capacity q_p is defined as:

$$q_p = \frac{q}{T_{hi} - T_{ci}} = \varepsilon C_{min} \tag{19}$$

where q is heat transfer rate, T is temperature, ε is effectiveness and $C_{\min}$ is the minimum heat capacity rate. The cooling performance of the system is only reflected by the thermal performance of the plate heat exchangers.

2.4. Validation

In order to validate the thermal design of plate heat exchanger, the heat transfer rates of gas–gas CC primary surface heat exchanger calculated by ε-NTU method and computational results were compared with experimental results of Wang et al. [19]. The overall dimensions of the gas–gas CC primary surface heat exchanger are 260 mm × 260 mm × 320 mm. The geometry of CC primary surface plate is shown in Fig. 2. The parameters of the CC primary surface plate are 11 mm in wavelength and 5.5 mm in corrugation height. The corrugation angle for hot and cold sides are 60° and 120°, respectively. The mass flow rates of the hot air and cold air are the same. For 26 different cases, the maximum, minimum and average deviations of computational results from experimental results were 7.67%, 4.32% and 6.50%, respectively. The deviations are as shown in Fig. 3. It is indicated that the ε-NTU method of this study could obtain accurate results for gas–gas plate heat exchangers.

3. Results and discussion

The parameters of both counter-flow plate heat exchanger and cross-flow plate heat exchanger are highly depended on overall dimensions of the cabinet cooling system. The effects of the overall dimensions of counter-flow plate heat exchanger and cross-flow plate heat exchanger on the thermal performance are obtained at constant mass flow rate of $m = 0.4$ kg/s. For a cabinet cooling system, whose overall dimensions are 700 mm × 400 mm × 90 mm, the effect of mass flow rate on thermal resistances of counter-flow plate heat exchanger is obtained. The cooling performance of the cabinet cooling system and thermal resistance of the plate heat exchangers are studied. The thermal performance of the plate heat exchanger directly reflects the cooling performance of the system. The thermal resistance of heat exchanger, which hinders the heat transfer, is used to analyze the irreversibility of the heat transfer.

3.1. Effect of cabinet cooling system width

The effects of system width on the cooling capacity of cabinet cooling system and dimensionless thermal resistance of plate heat exchanger are discussed in this section. It is obtained from the Table 1 that the counter-flow plate heat exchanger width increases with the increment of system width. The overall heat transfer area of counter-flow plate heat exchanger increases because the number of heat exchange channels increases. However, for the cross-flow plate heat exchanger, the overall heat transfer area greatly increases due to the significant increment in heat transfer area of heat exchanger plate.

The cabinet cooling system width is increased from 400 mm to 700 mm at constant system length of $L = 700$ mm and system height of $H = 90$ mm. The overall heat transfer area and coefficient of plate heat exchanger with different system widths are summarized in Table 2. The variations of cooling capacity of the cabinet cooling system and dimensionless thermal resistance of the plate heat exchanger with increment of cabinet cooling system width are illustrated in Figs. 4 and 5, respectively.

With the increment of the system width, the cooling capacity of the cabinet cooling system with counter-flow plate heat exchanger is nearly constant and the dimensionless thermal resistance of the counter-flow plate heat exchanger is almost constant. The inappreciable change of thermal performance of the counter-flow plate heat exchanger also can be measured by dimensionless thermal resistance. It is indicated that in these cases the irreversibility of heat transfer process is nearly constant. The thermal performance of counter-flow plate heat exchanger is not increased with increment of heat transfer area, because the mass flow rate for every channel decreases which causes a decrement in the heat transfer coefficient. In these cases, the overall heat transfer area of counter-flow plate heat exchanger increases by 75% and overall heat transfer coefficient of counter-flow plate heat exchanger decreases by 41%. The irreversibility of heat transfer process is also influenced by increment of heat transfer area and decrement of heat transfer coefficient.

However, for the cross-flow plate heat exchanger the cooling capacity increases with the increment of the cabinet cooling system width. In these cases, the overall heat transfer area increases by 206% and the overall heat transfer coefficient decreases by 38%. The increment in thermal performance of the cross-flow plate heat exchanger is mainly caused by the great increment of the overall heat transfer area.

The dimensionless thermal resistance of the cross-flow plate heat exchanger is decreased monotonically. It means the irreversibility of the heat transfer process is increased. It also indicates that the thermal resistance of the heat exchanger considers heat transfer area and coefficient simultaneously.

It is proved that the cross-flow cabinet cooling system has great performance for larger cabinet cooling system width. At the width of 700 mm, the cooling capacity of the two systems are 176.13 W/K and 138.95 W/K. The cooling capacity of the cross-flow cabinet cooling system is 1.26 times larger. The dimensionless thermal resistances are 1.28 and 1.89, respectively. The dimensionless thermal resistance of cross-flow plate heat exchanger is 0.68 of that value of counter-flow plate heat exchanger.

3.2. Effect of cabinet cooling system length

The parameters of the cross-flow plate heat exchanger are not changed with variation of cabinet cooling system length. So a cross-flow cabinet cooling system, whose overall dimensions are 700 mm × 500 mm × 90 mm, is carried out to compare with the counter-flow cabinet cooling system. The cooling capacity of the cabinet cooling system and dimensionless thermal resistance of the cross-flow plate heat exchanger are 147 W/K and 1.73, respectively. In this case the overall heat transfer area and overall heat transfer coefficient of cross-flow plate heat exchanger are 4.38 m^2 and 53.12 W/(m^2 K), respectively.

On the other hand, it is clearly seen from the Table 1 that the counter-flow plate heat exchanger length increases with the increment of the cabinet cooling system length. The heat transfer area of every plate is increased. The variations of the cooling capacity of the cabinet cooling system and dimensionless thermal resistance of counter-flow plate heat exchanger with increment of cabinet cooling system length are illustrated in Fig. 6. The cabinet cooling system length is increased from 650 mm to 800 mm at constant system width of $W = 500$ mm and system height of $H = 90$ mm. The overall heat transfer area is increased

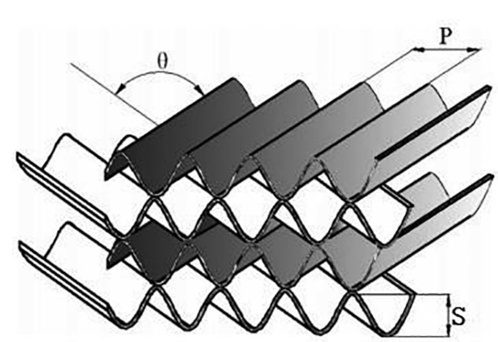

Fig. 2. Sketch of CC primary surface plate [19]: P – Wavelength, S – Corrugation height, θ – Corrugation angle.

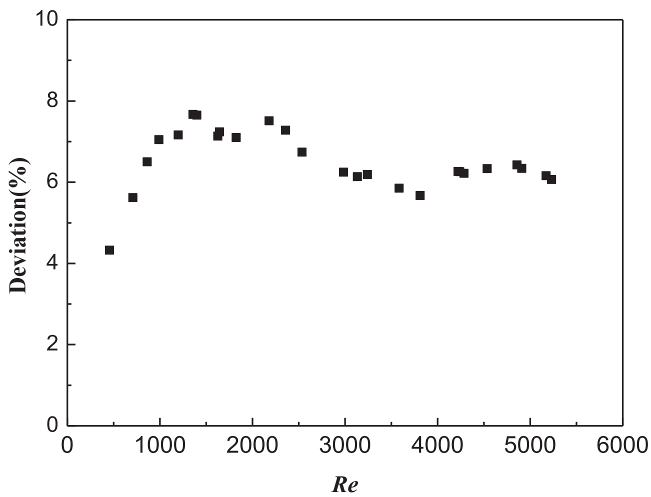

Fig. 3. Deviations data.

Table 2
Overall heat transfer area and overall heat transfer coefficient of plate heat exchanger with different system widths.

W (mm)	A (m^2) Cross-flow	Counter-flow	U ($W/(m^2\ K)$) Cross-flow	Counter-flow
400	2.8	4.29	63.97	47.78
450	3.54	4.83	58.04	43.08
500	4.38	5.37	53.12	39.16
550	5.29	5.91	48.96	35.82
600	6.3	6.45	45.39	32.95
650	7.39	6.99	42.27	30.42
700	8.58	7.53	39.52	28.19

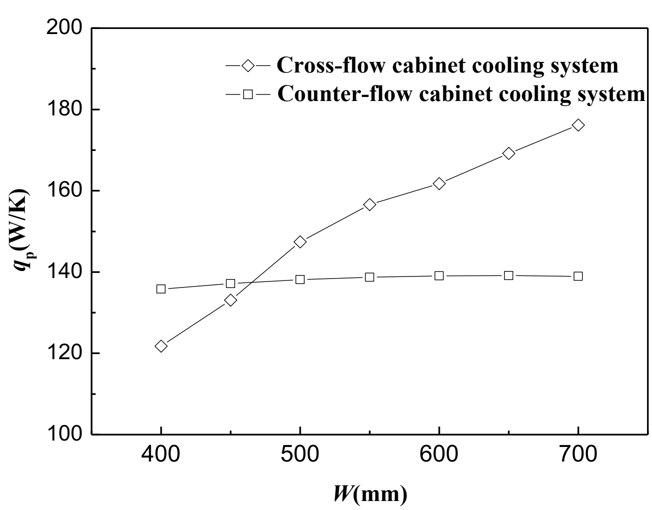

Fig. 4. Effect of cabinet cooling system width on cooling capacity.

from 3.58 m^2 to 7.72 m^2.

The cooling capacity of the cabinet cooling system with counter-flow plate heat exchanger increases by 35.7% with the increment of the system length because of the overall heat transfer area increment. The overall heat transfer coefficient is fixed in these cases. The dimensionless thermal resistance decreases by 37.5%. It indicates that the irreversibility of the heat transfer process increases with the increment of heat transfer area at constant overall heat transfer coefficient. If the system is limited in width, the counter-flow heat transfer is recommended to improve the cooling performance.

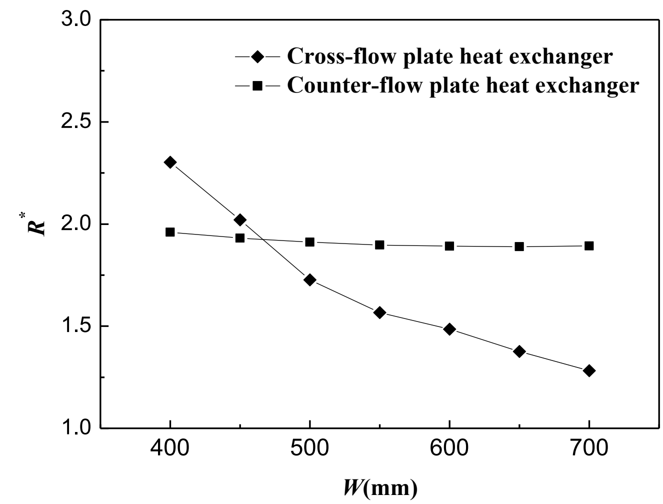

Fig. 5. Effect of cabinet cooling system width on dimensionless thermal resistance.

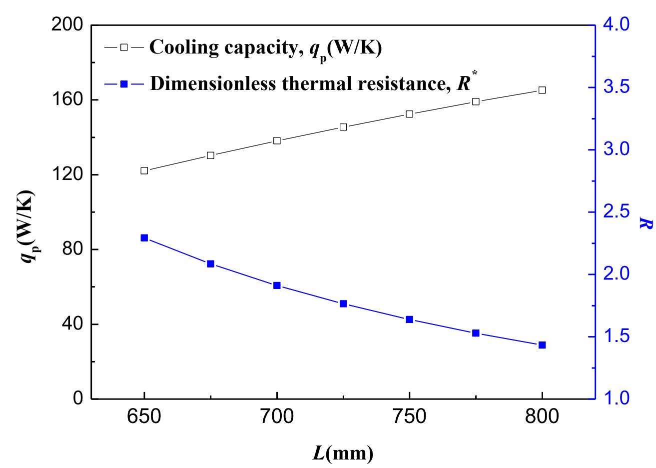

Fig. 6. Effect of cabinet cooling system length on cooling capacity and dimensionless thermal resistance.

3.3. Effect of cabinet cooling system height

The effects of system height on cooling capacity of the cabinet cooling system and dimensionless thermal resistance of plate heat exchanger are discussed in this section. It is obtained from the Table 1 that for the counter-flow plate heat exchanger, with the increment of the system height the overall heat transfer area increases because of the increment in every plate heat transfer area. For cross-flow plate heat exchanger the overall heat transfer area increases due to the increment of number of heat exchange channels.

The variations of the cooling capacity of the cabinet cooling system and dimensionless thermal resistance of plate heat exchanger with increment of cabinet cooling system height are illustrated in Figs. 7 and 8, respectively. The cabinet cooling system height is increased from 70 mm to 140 mm at constant system length of L = 700 mm and system width of W = 400 mm. The overall heat transfer area and overall heat transfer coefficient of plate heat exchanger with different system heights are summarized in Table 3.

With the increment of the system height, the cooling capacity of the counter-flow cabinet cooling system increases at first and then decreases due to the comprehensive effects of heat transfer area increment and heat transfer coefficient decrement. The dimensionless thermal resistance of the counter-flow plate heat exchanger decreases at first and then increases. It is indicated that dimensionless thermal resistance

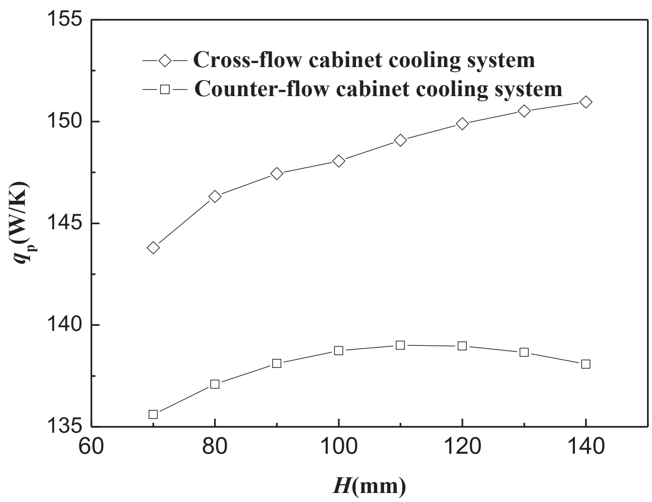

Fig. 7. Effect of cabinet cooling system height on cooling capacity.

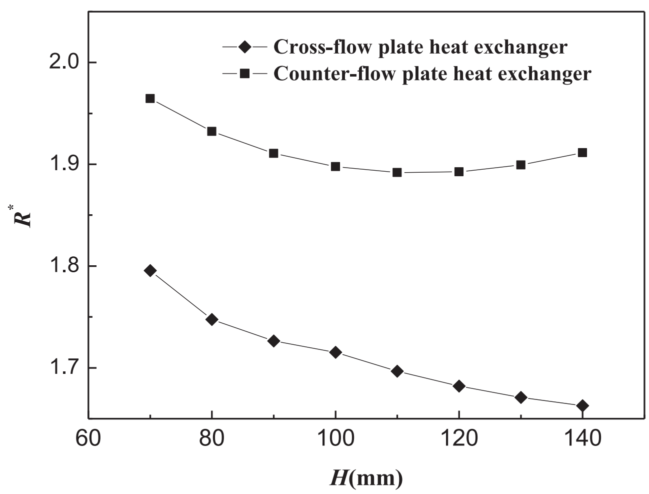

Fig. 8. Effect of cabinet cooling system height on dimensionless thermal resistance.

Table 3
Overall heat transfer area and overall heat transfer coefficient of plate heat exchanger with different system heights.

H (mm)	A (m^2) Cross-flow	Counter-flow	U (W/(m^2 K)) Cross-flow	Counter-flow
70	3.38	4.18	65.46	48.96
80	3.88	4.78	58.64	43.56
90	4.38	5.37	53.12	39.16
100	4.88	5.97	48.54	35.48
110	5.38	6.57	44.66	32.36
120	5.88	7.16	41.32	29.65
130	6.38	7.76	38.40	27.27
140	6.88	8.36	35.83	25.16

successfully measures the comprehensive effects of heat transfer area and heat transfer coefficient, so the irreversibility of the heat transfer keeps in synch with thermal performance of heat exchanger. The overall heat transfer area of the counter-flow plate heat exchanger increases by 100% and the overall heat transfer coefficient decreases by 49%.

For the cross-flow plate heat exchanger, the overall heat transfer area increases by 104% and the overall heat transfer coefficient decreases by 45%. The increment rate of heat transfer area here is more than that of counter-flow plate heat exchanger and the decrement rate

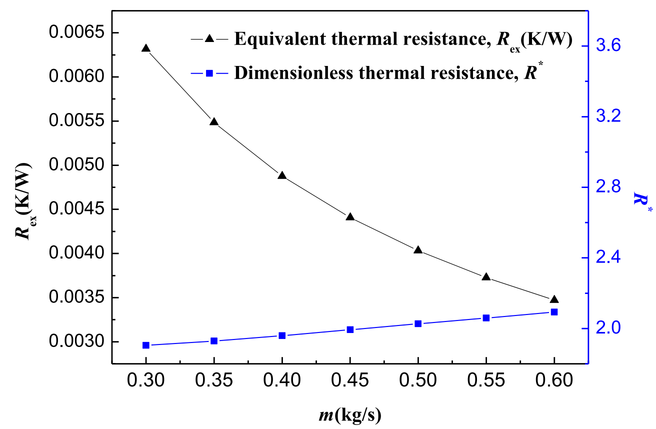

Fig. 9. Effect of mass flow rate on equivalent thermal resistance and dimensionless thermal resistance.

of heat transfer coefficient is less than that of counter-flow plate heat exchanger, when the cabinet cooling system height increases from 70 mm to 140 mm. The comprehensive effects of great increment of heat transfer area and small decrement of heat transfer coefficient cause a monotonic increment of cooling capacity and decrement of dimensionless thermal resistance.

3.4. Effect of mass flow rate

The mass flow rates of both fluids are assumed to be equal. The effects of mass flow rates on thermal resistance of a counter-flow plate heat exchanger are discussed in this section. The overall dimensions of the cabinet cooling system are 700 mm × 400 mm × 90 mm.

The variations of the equivalent thermal resistance and dimensionless thermal resistance of plate heat exchanger with increment of mass flow rate are illustrated in Fig. 9. The mass flow rate increases from 0.3 kg/s to 0.6 kg/s. The overall heat transfer coefficient increases from 36.88 W/(m^2 K) to 67.12 W/(m^2 K). The equivalent thermal resistance of plate heat exchanger decreases with mass flow rate. The equivalent thermal resistance at m = 0.3 kg/s is 1.82 times larger than that at m = 0.6 kg/s due to the mass flow rate. The dimensionless thermal resistance, which removes the heat capacity rate from the equivalent thermal resistance, increases slowly with increment of mass flow rate. The dimensionless thermal resistance increases by 9.88% when the mass flow rate increases from 0.3 kg/s to 0.6 kg/s. However, in every step change when the mass flow rate increases by 0.03 kg/s gradually, which equals to 10% of the 0.3 kg/s, the largest variation of the dimensionless thermal resistance is 1.00% and these variations are summarized in Table 4. It indicates that the irreversibility of the heat transfer is strongly influenced by the mass flow rate. The variation of dimensionless thermal resistance is negligible for small changes in mass flow rate and the dimensionless thermal resistance can be used to characterize the irreversibility of the heat transfer at constant mass flow rate.

Table 4
Variations of dimensionless thermal resistance.

Mass flow rate (kg/s)	Dimensionless thermal resistance	Variation (%)
0.33	1.92	0.72
0.36	1.93	0.86
0.39	1.95	0.94
0.42	1.97	0.98
0.45	1.99	1.00
0.48	2.01	1.00
0.51	2.03	0.99
0.54	2.05	0.98
0.57	2.07	0.96

4. Conclusions

The performance of the cabinet cooling systems with counter-flow plate heat exchanger and cross-flow plate heat exchanger are compared by using ε-NTU method and effectiveness-thermal resistance method at the same overall dimensions of the cabinet cooling system. The following conclusions can be made:

(1) With increment of cabinet cooling system width, the cross-flow plate heat exchanger thermal performance greatly increases and dimensionless thermal resistance greatly decreases, but the counter-flow plate heat exchanger performance varies a little. When the system width is 700 mm, the cooling capacity of cross-flow cabinet cooling system is 1.26 times larger than that of counter-flow cabinet cooling system. The dimensionless thermal resistance of the cross-flow plate heat exchanger is 0.68 of counter-flow plate heat exchanger dimensionless thermal resistance. The inappreciable change of thermal performance of the counter-flow plate heat exchanger can be measured by dimensionless thermal resistance.
(2) With the limitation of cabinet cooling system width, the counter-flow plate heat exchanger is recommended at large system length. When the system length increases from 650 mm to 800 mm, the cooling capacity increases by 35.7%, and the dimensionless thermal resistance decreases by 37.5%. The dimensionless thermal resistance decreases with increment of heat transfer area at constant heat transfer coefficient.
(3) The cross-flow cabinet cooling system has better cooling performance than counter-flow cabinet cooling system with increment of cabinet cooling system height. At these cases, cross-flow plate heat exchanger dimensionless thermal resistance is lower than that of counter-flow plate heat exchanger. For counter-flow plate heat exchanger the dimensionless thermal resistance varies synchronously with cooling capacity of the system.
(4) The dimensionless thermal resistance can be used to characterize the irreversibility of the heat transfer process at constant mass flow rate and the variation of dimensionless thermal resistance is negligible for small change in mass flow rate.

Credit authorship contribution statement

Saranmanduh Borjigin: Conceptualization, Methodology, Investigation, Writing - original draft. **Shuxiong Zhang:** Investigation, Methodology. **Ting Ma:** Methodology, Writing - review & editing. **Min Zeng:** Writing - review & editing. **Qiuwang Wang:** Supervision, Writing - review & editing.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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