Experimental research on heat transfer of natural convection in vertical rectangular channels with large aspect ratio

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ABSTRACT

This work presents the experimental research on the steady laminar natural convection heat transfer of air in three vertical thin rectangular channels with different gap clearance. The much higher ratio of width to gap clearance (60–24) and the ratio of length to gap clearance (800–320) make the rectangular channels similar with the coolant flow passage in plate type fuel reactors. The vertical rectangular channels were composed of two stainless steel plates and were heated by electrical heating rods. The wall temperatures were detected with the K-type thermocouples which were inserted into the blind holes drilled in the steel plates. Also the air temperatures at the inlet and outlet of the channel were detected. The wall heat fluxes were calculated by the Fourier heat conduction law. The heat transfer characteristics were analyzed, and the average Nusselt numbers in all the three channels could be well correlated with the Rayleigh number or the modified Rayleigh number in a uniform correlation. Furthermore, the maximum wall temperatures were investigated, which is a key parameter for the fuel's integrity during some accidents. It was found that even the wall heat flux was up to 1500 W/m², the maximum wall temperature was lower than 350 °C. All this work is valuable for the plate type reactor’s design and safety analysis.

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1. Introduction

Natural convective flows are widely encountered in nature and engineering systems. Because of their significant potential, such as the simplicity, absence of moving parts and reliability, these flows have been employed in the material processing, electronic equipment cooling, building heat transfer, solar collectors and the nuclear reactors, etc. These applications have motivated a substantial body of the research on natural convection heat transfer characteristics.

Elenbaas [1] conducted the first comprehensive experimental study of the natural convection heat transfer in vertical parallel plates, which has served as a benchmark for the most of the following works. However, Elenbaas’s experiments were conducted without the consideration of the open edge effects. Sparrow and Bahrami [2] performed some experimental study on the natural convection heat transfer from vertical parallel plates with either open or closed edges. The naphthalene sublimation technique, which is a mass transfer measurement technique, was adopted to eliminate the radiation effects, the extraneous convective/conductive heat losses, and the variable property effects. It was discovered that at low Rayleigh number, the results did not consist with Elenbaas’ experiments, which demonstrated the uncertainties in Elenbaas’ results owing to the large fluid property variations. However, at high Rayleigh number, their experimental data, along with results obtained by Levy et al. [3], were consistent with each other. Hung and Perng [4] experimentally investigated the natural convection heat transfer between two vertical parallel plates with the gap between 7 mm and 160 mm. One plate was heated up with constant heat flux between 40 W/m² and 270 W/m², and the other plate is insulated. The results indicated that both the heating power and the gap between these two plates affected the heat transfer, and the former had a greater impact. Moreover, the heat transfer coefficient decreased slightly with the gap. Sparrow et al. [5] analyzed the water natural convection heat transfer in a vertical rectangular channel both experimentally and numerically. One of the wider walls of the channel was maintained at a uniform temperature, while the other one was unheated. The visualization revealed a recirculating flow pocket adjacent to the unheated wall at the upper part of the channel, when the Rayleigh number exceeded a threshold value. Average Nusselt numbers of the heated wall were measured and correlated with the modified Rayleigh number: \( Nu = 0.688Ra^{0.249} \). The Nusselt numbers were found unaffected by the presence of the recirculation zone. Numerical solutions obtained via a parabolic finite difference scheme yielded...
that \( Nu = 0.655 Ra^{0.245} \), which was in good agreement with those of experiments. Ramakrishna et al. [6] numerically analyzed the natural convection heat transfer in a vertical square duct with the vorticity-stream function method. Under the constant heat flux condition of the wall, \( Nu = 0.53 Ra^{0.3} \) was obtained.

Krishnan et al. [7] gave out the results of an experimental and semi-experimental investigation of the natural steady laminar convection and surface radiation between three parallel vertical plates with air as the working fluid. Experiments were done for six plate gap clearance ranging from 12.66 to 52.20 mm and for an order of the magnitude range of wall-to-ambient temperature difference. The analysis brought out the significance of the radiation heat transfer rate at even a low temperature of 310 K.

Auletta et al. [8] experimentally analyzed the effect of the adding downstream adiabatic extensions of a vertical isoflux symmetrically heated channel. Optimal configurations were identified through measuring the wall temperature profiles. Conspicuous maximum wall temperature reductions have been achieved in these optimal configurations. Also the percentage increases of the heat transfer rate were in the order of 10–20%, depending on the channel’s aspect ratio and the imposed wall heat flux. Manca et al. [9] experimentally investigated the natural convection heat transfer of air in an asymmetrically heated channel with unheated extensions. The local and maximum wall temperatures and heat transfer coefficients were presented for the different process parameters. Optimal configurations in terms of the minimum values of the maximum wall temperatures had been obtained.

As reviewed above, the natural convection heat transfer in the vertical parallel channels, which is several centimeters or larger in gap clearance, has received considerable attention. In spite of this, air natural convection heat transfer in vertical rectangular channels with large aspect ratio and narrow gap has rarely been investigated experimentally or theoretically. However, this type of case does can be encountered taking into account of some extreme conditions of a plate type reactor. For example, when the core is exposed to the atmosphere during the accident of completely loss of the coolant because of the rupture of an experimental beam channel; or when the fuel assembly is stuck and hung up in the air during the process of refueling, because of the mechanical problems of the refueling machine. So, it is necessary and important to discover the heat transfer characteristics under these extreme situations from the view point of the nuclear reactor’s design and safety analysis.

Considering the lack of existing investigations, natural convective heat transfer of air in three vertical thin rectangular channels has been experimentally conducted to evaluate the heat remove capacity, the wall temperature distribution, the heat flux distribution and the heat transfer correlation between \( Nu \) and \( Ra \).

### 2. The experiments

#### 2.1. Experimental apparatus

A schematic drawing of the experimental apparatus is shown in Fig. 1. It was designed and constructed to investigate natural convection heat transfer of air in vertical rectangular channels, and help to understand whether the fuel plate could be well cooled during some extreme conditions mentioned above. The apparatus

![Fig. 1. Schematic representation of experimental setup.](image)
consisted of the rectangular test section, the heating system and the data detecting system. The test section was vertically fixed and placed in a large room, the cold air flow into it at the bottom because of the buoyancy force and then the heated air exhausted to the atmosphere at the top. The channels were made similar with that applied in plate type fuel reactors. Of the heating system, the stainless heating rod was powered with alternating current (AC) adjusted by an electrical transformer. In this way the decay heat of the fuel plate were simulated and the possible power range was covered. Of the data detecting system, temperatures were detected by the Industrial Personal Computer (IPC) with three IMP35951C data acquisition boards.

Fig. 2 shows the schemes of the experimental test section. As sketched in Fig. 2a, the channels were 800 mm in length, and 60 mm in width, and 1.0 mm, 1.8 mm and 2.5 mm in gap clearance, respectively. The test section was composed of two stainless steal plates and welding together with different gap clearance ducts. And the test section was heated by the stainless steel heating rod with alternating current. There were 20 heating rods on each side of the test sections fixed by two metal splints on the outer side. The device was wrapped in the aluminia-silicate fibre to insulate the test section.

The measuring junctions embedded in grooves in the wall normal to the channel axis as shown in Fig. 2. The thermocouples were fixed by drilling forty blind holes along the channel wall in ten sections with 76 mm apart from each other. The holes were 1.2 mm in diameter and 30 mm in depth as shown in Fig. 2b. The nickel-chromium–nickelsilicon (K-type) thermocouples of 1.0 mm in diameter were adopted to detect the temperatures. The inlet and outlet bulk air temperatures were also measured by the K-type thermocouples at the inlet and outlet part of the test section, respectively. All the thermocouples had been calibrated in a bath of constant temperature, and their maximum measurement uncertainty was 0.75%.

2.2. Experimental procedure

Maintain a certain electric supply, the heat transfer in the test section will reach a stable condition, which indicates that the temperatures will not changed with time. Here, after an electrical power adjustment, if the temperature reading of each thermocouple does not change by more than 0.5 K during half an hour, just as which depicted in Fig. 3, the heat transfer was considered to be stable. Once the steady state condition is established, the readings of all the thermocouples were recorded in an Industrial Personal Computer (IPC) with three IMP35951C data acquisition boards. Generally the duration of a data run was about 20 h. It was much longer than the test time in some other experiments [5,10], since these test sections were much larger than those in their experiments.

2.3. Data reduction

The heat transfer results are presented in terms of the average Nusselt number and Rayleigh number which are, respectively, defined as:

\[
Nu = \frac{h_a D_e}{k}
\]

\[
Ra = Gr Pr
\]

\[
Gr = \frac{g a b^4 q_{aw}}{v^2 k}
\]

where \(D_e\) is the equivalent diameter, \(4A/P = 2ab/(a + b)\); \(h_a\) is the average heat transfer coefficient; \(k\) is the thermal conductivity of the air; \(Gr\) is the Grashof number; \(Pr\) is the Prantl number. \(g\) is the gravity acceleration; \(b\) is the clearance of the gap; \(v\) is the kinetic viscosity; \(\alpha\) is the cubical expansion coefficient of the air.

The average heat transfer coefficient is:

\[
h_a = \frac{q_{aw}}{T_{aw} - T_{in}}
\]

where \(T_{aw}\) is the temperature of the atmosphere.

The average wall heat flux is:

\[
q_{aw} = \frac{1}{H} \int_0^H q_w dz
\]

where \(q_w\) is the wall heat flux. It supposed that \(q_w = 0.5(q_{aw} + q_{wr})\), where \(f\) and \(r\) indicates the left and the right side of the channel. As shown in Fig. 2b, the heat flux from the left side (\(q_{wl}\)) and the right side (\(q_{wr}\)) of each section can be deduced by the Fourier heat conduction law using the detected temperatures:

\[
q_{wl} = \frac{\lambda}{\delta_{wl}} (T_{wl} - T_{in})
\]

where \(T_{out}\) and \(T_{in}\) are the outer and inner temperatures of the test section on corresponding side of the channel (Fig. 2b); \(\lambda\) is the thermal conductivity of the test section; \(\delta_{wl}\) is the dimension between the two thermocouples on the same side (the left or the right side).
of the test section. The subscript i is l or r, which represents the left side or the right side of the test section, respectively.

The $T_{\text{ave}}$ is the average inner wall temperature, which is:

$$T_{\text{ave}} = \frac{1}{H} \int_0^H T_{\text{wi}} \, dz$$

(7)

where $T_{\text{wi}}$ is the inner wall temperature of the test section, which can be calculated by

$$T_{\text{wi}} = T_{\text{in}} - q_{\text{wi}} \delta_{\text{in}} / \lambda$$

(8)

where $\delta_{\text{in}}$ is the dimension between the inner thermocouple and the inner wall.

All the thermo-physical properties of the air are evaluated at the average air temperature, $T_{\text{ave}}$, which is defined as:

$$T_{\text{ave}} = 0.5 (T_{\text{inlet}} + T_{\text{outlet}})$$

(9)

where $T_{\text{inlet}}$, $T_{\text{outlet}}$ are the air temperatures at the inlet and outlet of the channel, respectively.

The modified Rayleigh number is $Ra' = Ra(b/H)$, where $H$ is the length of the channel.

2.4. Uncertainty analysis

The accuracy of experimental results depends upon the accuracy of the individual measuring instruments and the manufacturing accuracy of the test section. Based on the error theory, the total uncertainty $U$ comprises of uncertainties of many experimental

Table 1

<table>
<thead>
<tr>
<th>Parameter</th>
<th>$x_i$ (mm)</th>
<th>$\delta_x$</th>
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</thead>
<tbody>
<tr>
<td>a</td>
<td>60.0</td>
<td>±0.02</td>
</tr>
<tr>
<td>b</td>
<td>1.0–2.5</td>
<td>±0.02</td>
</tr>
<tr>
<td>$\delta_{\text{in}}$</td>
<td>3.0</td>
<td>±0.02</td>
</tr>
<tr>
<td>$\delta_{\text{out}}$</td>
<td>15.0</td>
<td>±0.02</td>
</tr>
<tr>
<td>$T_{\text{ave}}$ (K)</td>
<td>285.55–294.15</td>
<td>±0.1</td>
</tr>
<tr>
<td>$T_{\text{in}}$ (K)</td>
<td>318.75–632.25</td>
<td>±0.1</td>
</tr>
<tr>
<td>$T_{\text{inlet}}$ (K)</td>
<td>315.45–573.05</td>
<td>±0.1</td>
</tr>
<tr>
<td>$T_{\text{outlet}}$ (K)</td>
<td>318.95–633.95</td>
<td>±0.1</td>
</tr>
<tr>
<td>$T_{\text{out}}$ (K)</td>
<td>317.15–595.85</td>
<td>±0.1</td>
</tr>
</tbody>
</table>

Table 2

<table>
<thead>
<tr>
<th>Researchers</th>
<th>Correlations and description</th>
</tr>
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| Rohsenow et al. [15]     | $Nu_{\text{ff}} = 0.29 (Ra)^{1/2}$, $Ra' < 5$
|                         | $Nu_{\text{ff}} = H/(Ra')^{1/2}$, $10^4 < Ra' < 10^6$
|                         | $H_{\text{ff}} = 2 \left[ \frac{10^3 \cdot (Ra')^{1/2}}{3 \cdot 10^4} \right]^{1/5}$
| Ramanathan and Kumar [16] | $Nu = \left( \frac{H}{b} \right)^{1/3} \left( \frac{Ra' - 506}{506} \right) + 7.06 \left( \frac{1}{10^3} \right)$, $1 < H/b < 15$
|                         | $10^5 < Ra' < 3 \times 10^5$, $Pr = 0.7$
| Lu [17]                  | $Nu = \left( \frac{12}{10^3 \cdot (Ra')^{1/2}} \right)^{1/2}$, for $Ra'$ range
| Bar-Cohen and Rohsenow [18] | $Nu = \left( \frac{12}{10^3 \cdot (Ra')^{1/2}} \right)^{1/2}$, all $Ra'$ range

Fig. 4. Variation of Nusselt number with Rayleigh number.
parameters, which influence on the experiment. For a value of \( M \), whose results depend on uncorrelated input estimates \( x_1, x_2, \ldots, x_N \), the standard measurement uncertainty is obtained by appropriately combining the standard uncertainties of these input estimates. The combined standard uncertainties of the estimate \( M \) denoted by \( U(M) \) is calculated from the following equations [11,12]:

\[
M = f(x_1, x_2, \ldots, x_N) \quad (10)
\]

\[
U(M) = \left( \sum_{i=1}^{N} \left( \frac{\partial f}{\partial x_i} U(x_i) \right)^2 \right)^{1/2} \quad (11)
\]

where \( f \) is the function of \( M \) in terms of input estimates \( x_1, x_2, \ldots, x_N \), and each \( U(x_i) \) is a standard input uncertainty. Table 1 lists the experimental values of the directly measured parameters and the associated uncertainties. Thus, the uncertainties of the average Nusselt number expressed in Eq. (1) can be deduced to be 4.57–7.51% for 1.0 mm gap channel and 4.19–7.28% for 2.5 mm gap channel. For all the three channels, it was 4.19–7.51%.

3. Results and discussion

Air natural convection heat transfer have been conducted in three vertical rectangular channels with different gap clearances of 1.0 mm (\( a/b = 60, H/b = 800 \)), 1.8 mm (\( a/b = 33, H/b = 444 \)) and 2.5 mm (\( a/b = 24, H/b = 320 \)). The range of heat flux was from 100 W/m\(^2\) to 1500 W/m\(^2\), the working temperature of the air was varied from 315.45 K to 595.85 K.

3.1. Average Nusselt number

Fig. 4 gives out the variation of the average Nusselt number related to the Rayleigh number. In this figure the characteristic length in the Nusselt number and the Rayleigh number was the channel clearance, which was chosen according to the work in literatures [15–18]. The dots represent the experimental data. The regression equations of \( \text{Nu} \) as a function of \( \text{Ra} \) were founded with the method of the least square, which were represented with solid lines. From Fig. 4a–c, it can be clearly seen that all the experimental data in the three different test channels could be nicely correlated into the \( \text{Nu} = \text{C} \text{Ra}^a \) form:
For the 1.0 mm channel, in the $0.14 \leq Ra \leq 0.33$ range:

$$Nu = 0.205Ra^{0.143}$$  \hspace{1cm} (12)

For the 1.8 mm channel, in the $1.80 \leq Ra \leq 4.55$ range:

$$Nu = 0.316Ra^{0.133}$$  \hspace{1cm} (13)

For the 2.5 mm channel, in the $8.07 \leq Ra \leq 16.82$ range:

$$Nu = 0.268Ra^{0.302}$$  \hspace{1cm} (14)

As mentioned above, all the heat transfer data of the three channels can be well correlated in the same form, it can be inferred that the heat transfer characteristics of natural convection heat transfer with air in the three channels were the same. Taking all the experimental data of the three channels in the $0.14 \leq Ra \leq 16.82$ range, they could be well fitted (Fig. 4d) with

$$Nu = 0.263Ra^{0.308}$$  \hspace{1cm} (15)

Many correlations of natural convection with constant heat flux in symmetrically heated vertical parallel plates have been suggested in the literatures. Since Nusselt numbers were generally correlated with $Ra^*$ in these researches, the data of our experiments were also re-correlated with $Ra^*$ and compared with the results predicted with correlations in [15–18] (see Table 2). Fig. 5 shows that our experimental data can be well correlated in the range of $Ra^*$ from $1.75 \times 10^{-4}$ to $5.26 \times 10^{-2}$ with the following correlation:

$$Nu = 1.299Ra^{0.253}$$  \hspace{1cm} (16)

The $Ra^*$ was very low because of the large ratio of the length to the gap clearance, which is between 800 and 320. The Nusselt numbers predicted by the correlations of Rohsenow et al. [15], Lu [17] and Bar-Cohen and Rohsenow [18] were almost the same, but lower than those of Ramanathan and Kumar [16]. Because correlations of Rohsenow et al. [15], Lu [17] and Bar-Cohen and Rohsenow [18] were arranged in the same way, which were developed by combining correlating equations for boundary type and fully developed free convection heat transfer, while only Ramanathan and Kumar [16] considered the axial heat conduction of air at the channel inlet in their correlation. Moreover, in this experiment, the working temperature was much higher than that in the literature, thus the effect of radiation and air conduction was larger. Furthermore, the Nusselt numbers of our experiments include the impact of free convection, radiation and air conduction, thus the Nusselt numbers were much larger than any of those in [15–18] (see Fig. 5).

The channel clearance was taken as the characteristic length in Fig. 5 for the comparation with results from literature. However, for natural convective heat transfer in channels, it is valuable to analyze the experimental data with the channel length as the
characteristics length, which can be seen in Fig. 6. From Fig. 6b and c, it can be seen that the experimental data can be well correlated with the follow equations:

\[ \text{Nu} = 0.142 \text{Ra}^{0.332} (b/H)^{0.236} (b/a)^{-0.006} \]  

(17)

or

\[ \text{Nu} = 0.145 \text{Ra}^{0.332} (b/H)^{0.236} \]  

(18)

From the Eq. (17), it can be seen that the exponential part of \((b/a)\) is \(-0.006\), which means that the impact of the channel width on the heat transfer is small. This can also be revealed from Fig. 6b and c.

### 3.2. Wall temperature distribution

Fig. 7 shows the wall temperatures along the flow direction of these three channels under different heating power. It could be drawn that the temperature distribution in the three channels with different gap clearance almost the same, which is that the wall temperature was not always increased along the flow direction, but there is a decrease in the proximity of the exit section of the channel. The maximum temperature located at the point near to the outlet, but not at the exit point of the channel. These phenomena have also been discovered in [13,14]. The non-uniformity of the wall temperature increased with the clearance and the heating power. The shift of the maximum temperature point from the outlet to the upstream was due to the effects of the heat conduction, convection and radiation from the boundary to the ambient at the outlet of the channel, which is called the edge effects.

Furthermore, as expected, it can be discovered from Fig. 7a–c that at the same axial location with the same heating power, the wall temperature decreased when the channel gap clearance increased.

### 3.3. The maximum wall temperatures

For a plate type fuel reactor, during some accidents, the fuel plate can only be cooled by natural convection of air. Whether the residual heat can be removed out by air natural convection is
very important for the reactor safety. So the maximum wall temperatures were checked to investigate whether the fuel plate could maintain its integrity during that situation. From Fig. 8 it can be seen clearly that for all these three channels, even when the average wall heat flux was up to 1200 W/m², the maximum wall temperature was lower than 400 °C, which would not break the integrity of the fuel plate. Furthermore, as the heating power increased, the maximum wall temperature increased. Under the same heating power, the maximum temperature in the 1 mm channel was larger than that in the other two channels. Furthermore, the maximum temperature increased faster at lower heating power than that at higher heating power. The reason is that at higher heating power, the stable temperature is higher thus the radiation influence become significant.

3.4. Wall heat flux along the flow direction

The wall heat flux can be deduced from the correlation (6). In Fig. 9, it can be seen that the wall heat flux distributions in the axial direction of all the three channels were almost the same in the trend. The heat flux is larger at the inlet and outlet section, while almost the same in the middle. The reason is that the conduction and radiation heat transfer to the ambient on the two ends of the channel, and the flow recirculation at the outlet enhanced the heat transfer.

4. Conclusions

The characteristics of the air natural convection heat transfer in narrow vertical rectangular channels with large aspect ratio have been experimentally investigated here. The three channels were 800 mm in length, 60 mm in width, and 1.0 mm, 1.8 mm and 2.5 mm in gap clearance, respectively. Thus, the ratios of the width to the gap clearance were between 60 and 24, and the ratios of the length to the gap clearance were from 800 to 320, which were very high.

By heating electrically with stainless steel heating rods, the average heating power was 100–1500 W/m², the working temperature of the air was in a very high range of 315.45–595.85 K. Experimental results demonstrated that air natural convection heat transfer characteristics in these three channels were similar. The wall temperatures were not always increasing along the flow direction, but reached a maximum value at an upstream point near the channel exit. The maximum wall temperature was lower than 350 °C even the average wall heat flux was up to 1500 W/m². The wall heat flux was not uniform, but larger in the inlet and outlet part of the test section. The detected Nusselt numbers in the three channels could be well expressed with \( Nu = 0.263Ra^{0.308} \) or \( Nu = 1.299Ra^{0.253} \), which was higher than those in the literatures. Also with the channel length as the characteristic length, the experimental data could be well correlated as \( Nu = 0.145Ra^{0.332} \left( b/H \right)^{0.236} \) or \( Nu = 0.142Ra^{0.332} \left( b/H \right)^{0.236} \left( a/b \right)^{-0.006} \). These results are valuable for the design and safety analysis of plate type reactors.

References