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NUMERICAL INVESTIGATION OF FLOW AND HEAT TRANSFER IN CORRUGATED PARALLEL CHANNEL WITH SINUSOIDAL WAVE SURFACE

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ABSTRACT

Detailed numerical analysis is presented for flow and heat transfer in sinusoidal-corrugated parallel channel with six discrete heat sources placed under the bottom surface. Three dimensional numerical model are applied for simulating the flow and heat transfer process and the Colburn *j* factor is applied to evaluate the overall performance of the corrugated liquid cooled channel. The results show that the maximum temperature in the middle section decreases and the pressure loss increases as the wavelength of sinusoidal surface on the bottom decreases, while the increasing wave amplitude of corrugated surface can enhance the heat transfer rate in the ranges of inlet velocity from 1.5m/s to 3.5m/s. In addition, the corrugated channel have helped to improve heat transfer rate when it is compared with the traditional parallel channel.

Keywords: *sinusoidal wave, parallel channel, j f*actor

1. INTRODUCTION

The stability and reliability of electronic devices are facing severe challenges because of the rapid development of integrated electronic devices. The Liquid cooled plate presents some remarkable advantages such as compact structure and excellent heat transfer performance. Many scholars (Leszek, 2019; Choi, 1995; Steinke and Kandlikar, 2004) have done a lot of research on the geometric structure of the cold plate. Jeng et al. (2011) studied the effects of the number of channel channels, the number of fins and the inclination angle of fins on the heat dissipation ability. Their result showed that different geometric structures have a significant impact on the heat dissipation performance. Zhou (2020) investigated the the effect of modified double-inlet in a central-type parallel channel on flow distribution. The results showed that the modified double-inlet can reduce the flow maldistribution by 66.36% and pressure drop by 29.62% at the mean time. Gaurav (2020) investigated numerically and experimentally study the flow and heat transfer in parallel-microchannel. The result showed that the decrease of the number of channels leads to the increase of the surface temperature when other parameters are constant. Zhang et al. (2014) investigated the effect of hydraulic diameter, aspect ratio and relative roughness on the flow and heat transfer characteristics in parallel rectangular microchannels.

Recently, corrugated channels are regarded as one of the passive flow control techniques for improving the heat transfer rate significantly. The corrugated surface can not only increase heat transfer area, but also increase fluid disturbance and cut off the development of boundary layer, thus improving heat transfer efficiency. Kanaris et al. (2004) performed CFD studies on a plate heat exchanger comprising of corrugated walls with herringbone design. They studied the complex swirling flow in the furrows of the corrugations the Nusselt number and the friction factor. The results showed that the adding of corrugations increase the heat

transfer rate while the pressure losses also increase at the same time. Zhang and Che (2011) investigated the friction factor f, Colburn j and local Nusselt number of the flow and heat transfer in cross-corrugated channel by three dimensional model. Maryam (2013) studied the turbulent flow field in corrugated channel at different Prantal numbers by large eddy simulation. Sparrow and Comb (1983) investigated the effect of inlet velocity and wave height on flow behaviors. Their results showed that the Nusselt number increases with the increasing wave height on the bottom of channel. Wang (2002) analyzed the heat transfer rate in cooling process for sinusoidal converging-diverging channel. Faizal (2012) experimentally studied the heat transfer and pressure drop of the corrugated plate heat exchanger with variable spacing and variable flow rate by measuring the inlet and outlet temperature of the corrugated plate heat exchanger. For a given plate spacing, the average heat transfer rate increases due to high turbulence at higher velocities. The overall heat transfer coefficient and pressure losses can be enhanced by the increasing hot water flow rate Gao (2014) numerically studied the characteristics of three-dimensional turbulent flow and heat transfer in the channel with one corrugated wall heated with constant temperature by means of large eddy simulation. The results show that flow separation bubbles appear and near-wall vortices are generated with larger population in the upslope region of the bottom wall as wave amplitude increases. Ahmed et al. (2012) numerically investigated the convective heat transfer performance of a copper–water nanofluid in a trapezoidal corrugated channel under a range of Reynolds numbers from 100 to 700. The above studies indicated that the corrugated surface on the bottom of parallel channel can enhance the heat transfer rate obviously when it is compared with the case of the cooling channel with flat plate. Most of the authors focus on the flow and heat transfer in a single corrugated channel or double corrugated channels. However, few reports have been reported on flow and heat transfer in a more complex domain, such as multiple parallel channels with sinusoidal wave surface.

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The objective of this article is to numerically simulate flow and heat transfer in sinusoidal-corrugated parallel channel with six discrete heat sources placed under the bottom surface. The three-dimensional numerical model will be adopted and the Colburn j factor will be defined to evaluate the overall performance of the corrugated liquid cooled channel. A series of numerical calculations will be done in order to analyze the effect of geometric parameters such as wave length and height on the heat transfer rate and j factor.

2. PHYSICAL MODEL

A schematic diagram of the physical model under consideration is shown in Figure 1. There are three parts including substrate, cover plate and heat sources. The flow channel formed by milling on the base material and the cover plate constitute a closed cooled flow channel. The corrugated surfaces in present study instead of the flat bottom surface of traditional parallel-channel. The size of the liquid cooled plate is 600×260×35mm. There are six flow channels in the milling on the substrate, and the straight part size of each flow channel is 500×20×20mm. The inlet and outlet of the flow passage is a cylindrical drain pipe with a radius of 9 mm and a length of 25mm. For the convenience of calculation and analysis, the fluid region is assumed to be in a constant physical property, incompressible, stable state and without considering the viscous dissipation.
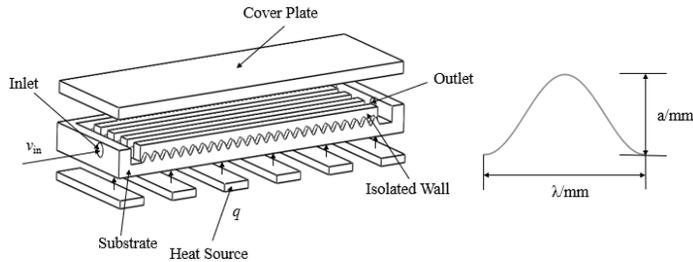

Fig. 1 Physical Model

3. NUMERICAL APPROACH

The problem is considered to be three-dimensional and coupled heat transfer between isolated walls and fluid in flow passage of parallel channel. The way of heat transfer in isolated is conduction while the cooling fluid flow is forced convection. The equations governing the cooling fluid flow and heat transfer in parallel channels can be written in the following form:
Continuity Equation:

$$\nabla \cdot \vec{v} = 0 \tag{1}$$

Momentum Conservation Equation:

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot (\bar{\bar{\tau}}) + \vec{F} \tag{2}$$

where p is the static pressure and $\bar{\bar{\tau}}$ is the stress tensor. $\vec{F}$ is the source term due to other forces such as friction. The stress tensor is defined as:

$$\tau = \mu \left[\left(\nabla \vec{v} + \nabla \vec{v}^T \right) - \frac{2}{3} \nabla \cdot \vec{v} I \right] \tag{3}$$

where, μ is the dynamic viscosity (molecular viscosity). I is the unit tensor.
Energy Conservation Equation:

$$\frac{\partial}{\partial t}(\rho h) + \nabla \cdot (\rho h \vec{v}) = \nabla \cdot \left[(k + k_t) \nabla T \right] + S_h \tag{4}$$

where, k is the molecular heat conductivity, k_t is the heat conductivity caused by turbulence $k_t = (c_p \mu_t / \text{Pr}_f)$ and S_h is the body heat source term, including all defined heat source terms.
For the solid region, the heat flow including conduction and body heat source satisfies the heat conduction equation:

$$\frac{\partial}{\partial t}(\rho h) = \nabla \cdot (k \nabla T) + S_h \tag{5}$$

where ρ is the density, k is thermal conductivity, T is the temperature, S_h is the source term of body heat. In the region of the flow of the above two equations to solve at the same time, in order to solve the conduction/convection coupled heat transfer process.
Since turbulent flow is simulated in this paper, zero equation is selected for model setting.
The zero-equation model (mixed length model), also known as the algebraic model, calculates the eddy viscosity by the following relational expression:

$$\mu_t = \rho l^2 S \tag{6}$$

where l is the mixing length:

$$l = \min(k' d, 0.09 d_{\max}) \tag{7}$$

where d is the distance from the point to the wall, k' is Karman's constant, $k' = 0.419$.
The average strain rate tensor modulus can be writtern as:

$$S = \sqrt{2 S_{ij} S_{ij}} \tag{8}$$

The average strain rate：

$$S_{ij} = \frac{1}{2} \left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right) \tag{9}$$

To represent and analyze the heat transfer and the flow in the corrugated channel, the relevant parameters need to be defined. The friction coefficient based on the hydraulic diameter, the Colburn j factor and the Nusselt number are calculated as follows:
The average convective heat transfer coefficient of the channel is defined as:

$$h = \frac{\Phi}{A_s \Delta T} \tag{10}$$

where A_s is the area of the fluid-solid interface in the channel; ΔT is the logarithmic mean temperature difference between the channel wall and the cooling water. Φ is the heat transfer rate and it can be written as:

$$\Phi = q_m c_p \Delta T \tag{11}$$

The average Nusselt number is expressed by the equation:

$$Nu = \frac{D_k h}{k} \tag{12}$$

The friction coefficient f is computed by using the formula:

$$f = \frac{2 \Delta P D_h}{\rho V^2 L} \tag{13}$$

where ΔP is the pressure difference between inlet and outlet of channel; ρ is the density and V is the average fluid velocity. L is the length of the channel.
The Colburn j factor is defined as follow:

$$j = \frac{Nu}{Nu_o} \left(\frac{f}{f_o} \right)^{-1/3} \tag{14}$$

where the subscript o represents the corresponding values of plane channel, and Nu are f the values in the corrugated channel. The

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Colburn j factor comprehensively considers the changes of heat transfer performance and fluid pressure drop of the corrugated channel compared with the straight channel.

The wave surface on the bottom of cooling channel is a sine function as follows:

$$y = h \sin(2\pi x / \lambda) \tag{15}$$

The algorithm is evaluated to verify the reliability of the calculation model and simplified method. The result shows that the maximum error of frictional drag factor is 9.3% and the maximum error of average Nusselt number is 4.4% (Amitav et al., 2019). Within the error allowable range, it is considered that the calculation model and simplified method used in this paper are reasonable and reliable.

4. RESULTS AND DISCUSSION

4.1 Flow and heat transfer characteristics of corrugated channels

The flow and heat transfer in flat parallel channels with discrete heat sources placed under the bottom surface were simulated for the case of a=14 mm. Figure 2 shows the comparison of average Nusselt number for the case of corrugated and flat cooling channels with different velocity. It can be seen from Fig. 2 that the value of Nu numbers increases with the increasing of inlet velocity. The heat transfer capacity of the corrugated cooling channel with sinusoidal wave surface is significantly better than that of the channels without wave surface for all the same parameters. The numerical results indicated that the corrugated surface on the bottom can effectively improve the heat transfer capacity of the cooling channel because of the increasing heat transfer area.
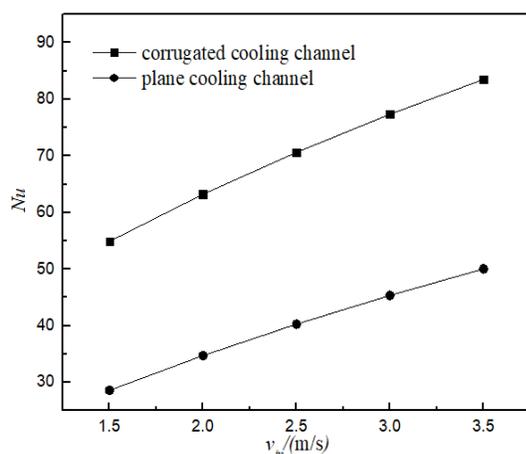

Fig. 2 Comparison of average Nusselt numbers for the case of plane and corrugated cooling channels

However, the pressure loss of corrugated cooling channel is greater than that of plane cooling channel, as is shown in Figure. 3. The pressure loss increases with the increasing inlet velocity of fluid in cooling process of heat sources. The corrugated surface on the bottom surface of the parallel cooling channel can effectively improve the enhanced heat transfer capacity of the cooling channel, but it also leads to the larger pressure loss in cooling process. Fig. 4 shows the comparison of Colburn j factor for the case of plane and corrugated cooling channels. The value of j factor for different inlet velocities are all above 1 and the maximum of the value of j factor appears at $v_{in} = 1.5$ m/s. Although the value of j factor decreases with the increasing value of inlet velocity, the value of j factor at v_{in}=3.5 m/s is still over 1.4. The results indicated that the increase in pressure drop is acceptable.

While the corrugated channel improves the heat exchange capacity of the channel, it also brings a greater pressure loss. Therefore, it is necessary to evaluate the comprehensive heat transfer performance of the corrugated cooling channel through the comprehensive heat transfer factor. The integrated heat transfer factor of the plane channels is set to 1. Fig.4 shows that the heat transfer factor of the corrugated cooling channel decreases with the increase of the inlet flow rate of the coolant and the values are all greater than 1. When the coolant inlet velocity is 1.5 m/s, the heat transfer factor of the corrugated cooling channel has a maximum value of 1.73.
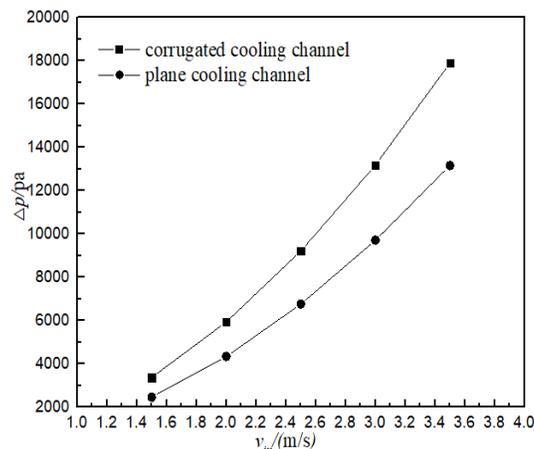

Fig. 3 Comparison of pressure losses for the case of plane and corrugated cooling channel
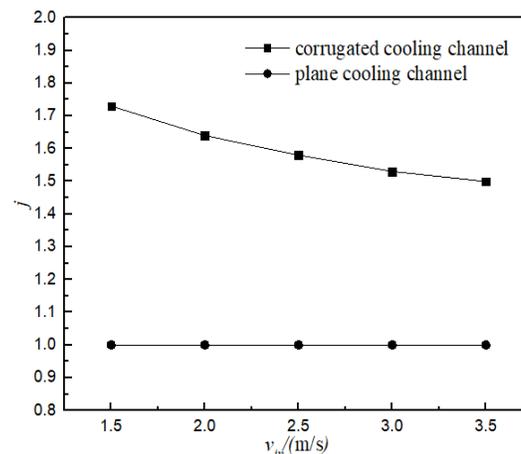

Fig. 4 Comparison of j factor for the case of plane and corrugated cooling channel

4.2 Influence of the sinusoidal wave amplitude

The effect of geometric parameters is investigated by varying wave amplitudes from 6mm to 14mm by a step of 2mm. Figure.5 shows velocity vector fields along the direction of the fluid flow in the corrugated channels for the case of different wave amplitudes. It can be seen that when the wave amplitude is larger than 6mm, one small vortex appears at the bottom of corrugated channel. The vortex moves up and becomes stronger with the increasing of the value of wave amplitude. The corrugated surface increases the flow disturbance and the area of heat transfer surface. Therefore, the heat transfer performance of the corrugated channel can be improved.

The maximum temperature in the middle section is presented for different values of the wave amplitudes in Table 1. It can be seen from Table 1 that the maximum temperature value in the middle section of corrugated channel decreases as the increasing values of wave amplitude and inlet velocity. When the value of inlet velocity is 2.5m/s, the maximum value of temperature is 41.0ºC for the case of a=6mm and that

of temperature for the case of a=14mm is 38.3°C. The difference between them is approximate to 5.5%.

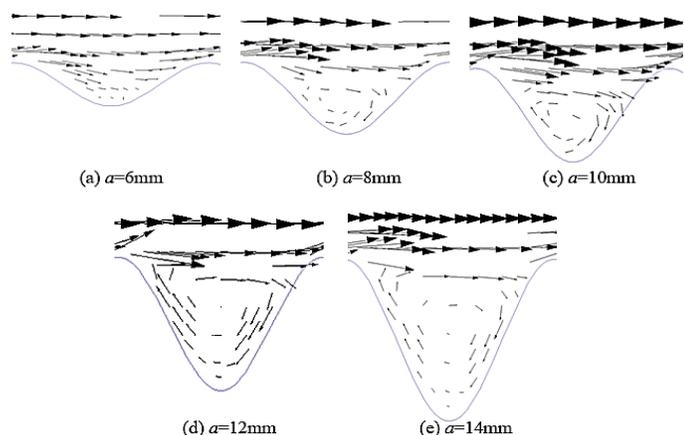(a) a=6mm (b) a=8mm (c) a=10mm

(d) a=12mm (e) a=14mm

Fig. 5 Velocity vector fields in middle section for the case of different wave amplitudes

Table 1 The value of maximum temperature in the middle section for different wave amplitudes and inlet velocities

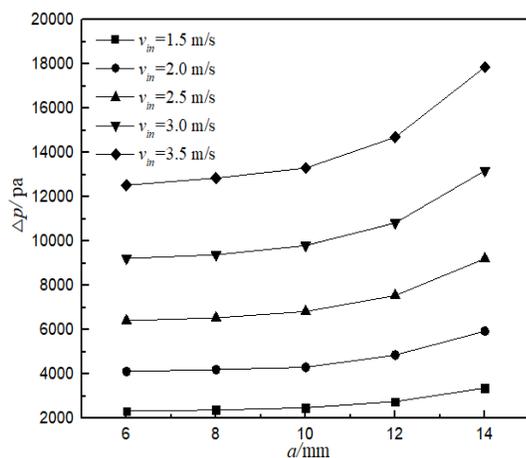|	a=6mm	a=8mm	a=10mm	a=12mm	a=14mm
v_{in}=1.5m/s	44.7°C	43.9°C	43.4°C	42.4°C	41.7°C
v_{in}=2.0m/s	43.1°C	42.3°C	41.8°C	40.8°C	40.2°C
v_{in}=2.5m/s	41.9°C	41.1°C	40.7°C	39.7°C	39.1°C
v_{in}=3.0m/s	41.0°C	40.2°C	39.8°C	38.8°C	38.3°C
v_{in}=3.5m/s	40.3°C	39.5°C	39.1°C	38.2°C	37.7°C

Fig. 6 The pressure loss for the case of different wave amplitudes

Figure.6 shows that the pressure loss for the case of different values of wave amplitudes. When the inlet flow rate is 3.5m/s, the heat transfer performance of the channel is improved while the pressure loss reaches the maximum value of about 17000 Pa. It is equivalent to 1.4 times of the plane cooling channel under the same working condition. Therefore, the comprehensive performance of the channel is evaluated by the parameter j factor in Fig.7. As seen from Fig.7, the value of j factor in corrugated channel increases by the increasing value of wave amplitudes from 6mm to 12mm while it can decrease for the case of a=14mm. The maximum value of j factor appear at a=12mm. The results indicates that the increasing value of wave amplitudes of corrugated channel can enhance the heat transfer rate and at the same time increase the pressure

losses. Therefore, there is an optimal value of j jactor. The best performance of heat transfer can be arrived at a=12mm for all the cases of varying inlet velocities.
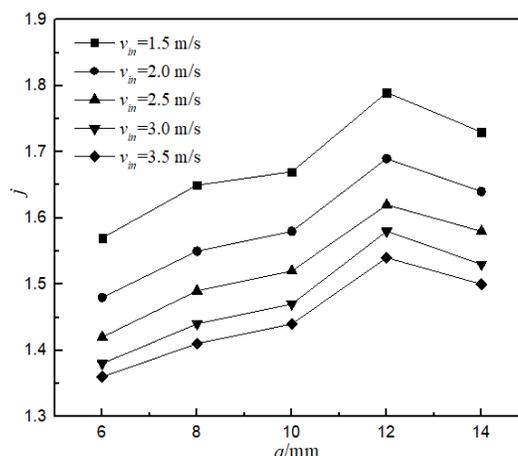

Fig. 7 The Colburn j factor for the case of different wave amplitudes

4.3 Influence of the sinusoidal wave length

The velocity vector of corrugated channels for the four wave lengths of 12.5mm, 14.3mm, 16.7mm and 25mm are shown in Figure.8, respectively. It can be seen that when the wave length is less than 25mm, one small vortex appears at the bottom of corrugated channel. The vortex moves up and becomes stronger with the decreasing of the value of wave length.

The change of the heat transfer performance of the channel is necessarily accompanied by the change of the pressure in the channel. Figure.9 shows that the pressure loss for the case of different values of wave lengths. When the wave length is 12.5mm, 14.3mm, 16.7mm, 25mm, the number of waves in the corrugated channel is 40, 35, 30 and 20, respectively. The corrugated channel with a large number of waves has a larger contact area with the cooling liquid in the channel. The friction force of the cooling liquid on the wall surface is also greater in the process of flowing. Therefore, the channel with smaller wave length has greater pressure loss.
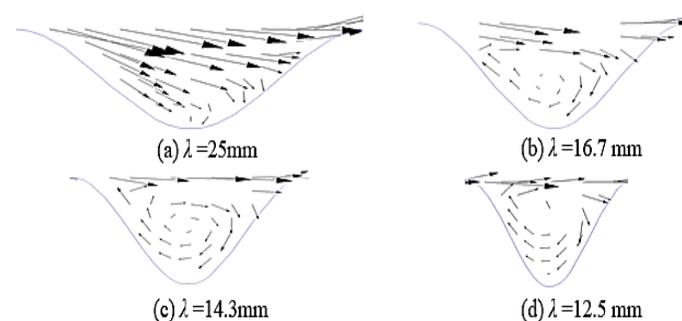

(a) λ=25mm (b) λ=16.7 mm

(c) λ=14.3mm (d) λ=12.5 mm

Fig. 8 Velocity vector fields in middle section for the case of different wave length

The maximum temperature in the middle section is presented for different values of the wave lengths in Table 2. It can be seen from Table 2 that the maximum temperature value in the middle section of corrugated channel decreases as the decreasing values of wave length and increasing values of inlet velocity. The larger the contact area between the cooling liquid and the channel in the smaller wavelength of corrugated channel, which effectively promoting the heat exchange between the fluid and the wall surface.

Table 2 The value of maximum temperature in the middle section for different wavelength and inlet velocities

	λ=12.5mm	λ=14.3mm	λ=16.7mm	λ=25mm
v_{in}=1.5m/s	42.9°C	43.1°C	43.2°C	43.7°C
v_{in}=2.0m/s	41.3°C	41.4°C	41.6°C	42.1°C
v_{in}=2.5m/s	40.2°C	40.3°C	40.4°C	41.0°C
v_{in}=3.0m/s	39.3°C	39.4°C	39.6°C	40.1°C
v_{in}=3.5m/s	38.7°C	38.8°C	38.9°C	39.5°C

Figure.10 shows that the comprehensive heat transfer characteristic factors of corrugated channels at different wave lengths. It can be seen from Figure.10 that the value of j factor in corrugated channel shows a gradually increasing trend by the increasing value of wave lengths from 12.5mm to 25mm and the values are all greater than 1. The maximum value of j factor appear at λ=25mm. The results indicates that the best performance of heat transfer can be arrived at a=25mm for all the cases of varying inlet velocities within the four wave length.

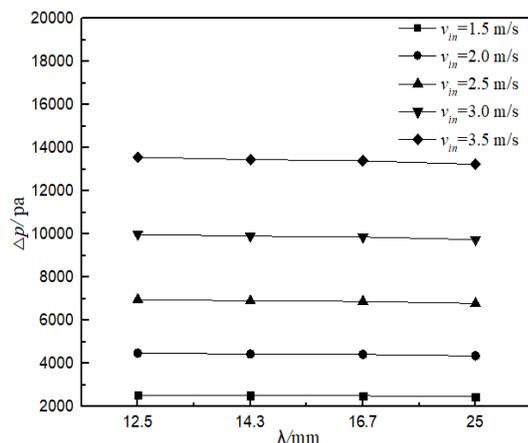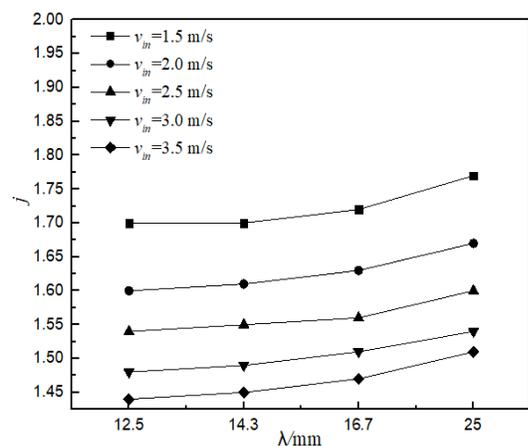**Fig. 9** The pressure loss varies with wave length

Fig. 10 The Colburn j factor varies with wave length

5. CONCLUSIONS

The flow and heat transfer characteristics of the corrugated cooling channel are analyzed. The results show that the maximum temperature in the middle section decreases and the pressure loss increases as the wavelength of sinusoidal surface on the bottom decreases, while the increasing wave amplitude of corrugated surface can enhance the heat

transfer rate in the ranges of inlet velocity from 1.5m/s to 3.5m/s. Although the corrugated surface can effectively improve the heat transfer performance of the cooling channel, the pressure loss is also increasing. Therefore, the j factor is applied to evaluate the comprehensive performance of flow and heat transfer in the channel. When the value of wave amplitude is 12mm of the same wave length, the j factor of the corrugated channel reaches its maximum value. The results indicated that the corrugated parallel-channel with an amplitude of 12mm has the best comprehensive performance than other channels at same condition. The results of this study are of great significance for the selection of corrugated cooling channel structures for the efficient design of water-cooling plates to improve the thermal performance.

NOMENCLATURE

a	wave amplitude, mm
A_s	the area of the fluid-solid interface in the channel, m²
C_p	specific heat, J/kg·K
D_h	hydraulic diameter, m
h	average convective heat transfer coefficient, $\text{W}\cdot\text{m}^{-2}\cdot\text{k}^{-1}$
j	the Colburn factor
k	thermal conductivity, W/m·K
Nu	Nusselt number
P	Pressure, pa
Φ	heat exchange, W
T	the temperature, K
V	the average fluid velocity, m/s
L	the length of the channel, m

Greek Symbols

ρ	Density, kg/m³
λ	wave length, mm

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