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Numerical study and performance analyses of the mini-channel with discrete double-inclined ribs

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ABSTRACT

A 3-D numerical study is carried out to investigate the laminar flow and heat transfer performance of the rectangular mini-channel where the discrete double-inclined ribs are worked as the longitudinal vortex generators. The effects of the Reynolds number, the height of the ribs and the number of double-inclined ribs along the mainstream on the heat transfer and flow performance of the mini-channel are examined and analyzed from the field synergy perspective and the entropy generation. The results show that the heat transfer performance is enhanced effectively by the double-inclined ribs which cause the generation of the longitudinal vortexes in the mini-channel. The heat transfer performance increase with the increasing height or number of the double-inclined ribs, but the flow resistance will increase at the same time. In order to obtain the best overall performance of the mini-channel, the height of the ribs should be reduced with the increase of the Reynolds, and the overall performance would be improved with the increase of the ribs number in the mini-channel. The heat transfer performance has a direct relation to the field synergy characteristic of the mini-channel. The entropy generation rate dues to heat transfer irreversibility and fluid frictional irreversibility can be used for the evaluation of the heat transfer and the flow performance of the mini-channel well respectively, while the total entropy generation rate cannot be used as a criterion for the overall performance.

1. Introduction

With the rapid development of microelectronics technology, the corresponding problem of heat dissipation with high heat-flux in a restricted space has become a focus of the heat transfer research nowadays. Since the concept of micro-channel cooling system was firstly proposed in 1980s [1], micro-channel heat sink shows great superiority in the cooling of the modern integrated electronic devices due to its high performance in heat transfer and the compact structure. Therefore, much attention has been attracted to the related study. In order to get higher performance, the structure of the macro-channel has been optimized by many researchers [2–4], and many new channel structures were proposed, such as the fractal tree-like structure [5], the diamond-shaped interrupted fins structure [6], the multilayer staggered honeycomb structure [7], etc.

The traditional heat transfer enhancement technologies, such as to increase the Reynolds number, to strengthen the flow disturbance, etc., were developed from experience rather than under the guidance of proper theories [8]. Generally, they always cause great increase in power consumption with the enhancement of the heat transfer. The longitudinal vortex heat transfer enhancement technology can improve the convection heat transfer performance effectively, and now is defined as the third generation heat transfer technology [8]. The typical longitudinal vortex generators are generally divided into four types, the delta wing, rectangular wing longitudinal vortex generator, the delta winglet and the rectangular winglet longitudinal generator. Researches show that performance of the delta winglet and the rectangular winglet longitudinal vortex generators is much better than the delta wing and the rectangular longitudinal vortex generators [9], and performance of the delta winglet longitudinal vortex generators is better than the rectangular winglet longitudinal vortex generator [10]. Research also shows that longitudinal vortexes can enhance the global heat transfer of channel, while transverse vortexes can only enhance the local heat transfer of channel [11]. However, look over the existing discussions on the longitudinal vortex generator, the wings or winglets are with a certain gap between the practical application since the thickness of the vortex generator are often

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neglected in most of the researches. And the number of the vortex generators in channels also seldom studied.

The field synergy theory which was proposed by Guo et al. [12] is committed to enhance convective heat transfer under constant power consumption, and is developed as a guidance theory for the convective heat transfer in recent years. In this theory, they proposed a concept that the physical nature of convective heat transfer is up to the synergetic relation between its velocity field and heat-flux field. Under the same boundary conditions of velocity and temperature, the better the synergy between velocity field and heat-flux field is, the higher the heat transfer intensity will be. Then based on the field synergy principle for heat transfer enhancement, the concept of physical quantity synergy in the laminar flow field was proposed by Liu [13]. The physical nature of enhancing heat transfer and reducing flow resistance, which is directly associated with synergy angles α, β, γ, ϕ and ψ, is also explained. It provides a basis to develop new heat transfer technologies or to evaluate the flow and heat transfer performance. The discrete double-inclined ribs tube [14] which was developed based on the convective heat transfer field synergy theory is a new technology for heat transfer enhancement, and its main advantage is that multi-longitudinal vortexes would generate spontaneously when fluid flow through the discrete double inclined ribs, so the discrete double-inclined tube can achieve heat transfer augmentation under the same power consumption and has a positive effect in energy-saving.

In addition, the minimum entropy generation principle has been widely adopted to evaluate the thermal systems for determining the optimal design recently [15,16], and the optimum system were designed and evaluated from the perspective of the second law of thermodynamics, which focus on the entropy generation due to the irreversible heat transfer and fluid flow. So in present study, the entropy generation is also considered to evaluate the performance of the mini-channel.

Mini-channel can be applied in various kinds of applications, such as applied in the mini-channel heat sink for the cooling of the electronic devices with high heat flux [17,18], applied in the proton exchange membrane fuel cell to enhance the heat and mass transfer in the gas channel or to enhance the heat transfer in the coolant channel [19,4], etc. In the present paper, the discrete double-inclined ribs are applied to mini-channel to give a new kind of mini-channel for the applications above. According to a typical mini-channel structure used in the mini-channel heat sink, the size of the new type mini-channel is determined, and the discrete double-inclined ribs with a certain thickness are arranged in the rectangular channel on the wall which is corresponding to the heated wall. Three dimensional models are established to investigate the performance of the mini-channel with the discrete double-inclined ribs.

2. Physical model and mathematical description

Fig. 1 shows the physical model of the mini-channel analyzed in the present work, where some pairs of the discrete double-inclined ribs with a certain thickness are arranged in the rectangular mini-channel. As shown in the figure, the ribs pairs are numbered from 1 to 10 in sequence along the flow direction. The effects of the ribs height and the ribs number on the performance are the main objects to investigate. In the present work, heights of ribs are taken as H = 0.25, 0.50, 0.75, 1.00, 1.25 mm, and compared with the plain channel (H = 0); the numbers of the ribs pairs have three cases: (1) as shown in Fig. 1, 10 pairs of ribs in the mini-channel (N = 10), (2) removing the even-numbered ribs pair, 5 pair of ribs remained in the mini-channel (N = 5), (3) only remain the ribs pair numbered 1 in the mini-channel (N = 1). The other main geometry parameters are shown in Table 1.

According the model, the average Nusselt number (Nu) and the average flow resistance coefficient (f) for the current problem are defined as

$$h = \frac{Q}{A_{\mathrm{w}}\Delta T_{\mathrm{m}}} \tag{1}$$

$$Nu = \frac{hD_{\mathrm{h}}}{k} \tag{2}$$

$$f = \frac{\Delta P}{\rho u_{\mathrm{a}}^2/2}\frac{D_{\mathrm{h}}}{L} \tag{3}$$

where Q is the total heat flux of the heated wall, $A_{\mathrm{w}} = W \times L$ is the nominal heat transfer area, ΔT_{m} is the average temperature difference of the heated wall and the fluid field, h is the heat transfer coefficient, k is the thermal conductivity of the fluid medium, ΔP is the pressure drop of the fluid medium from the inlet to the outlet, ρ is the density of the fluid medium, u_{a} is the average flow velocity of the fluid medium in the mini-channel, D_{h} is the hydraulic diameter of the mini-channel, which is defined as $D_{\mathrm{h}} = 2w \times h/(w+h)$.

The present problem is about the three-dimensional flow and heat transfer processes with laminar and steady condition. The relevant control equations are continuity equation, momentum equation and energy equation. The general form of these equations is as follow

$$\frac{\partial(\rho u\varphi)}{\partial x} + \frac{\partial(\rho v\varphi)}{\partial y} + \frac{\partial(\rho w\varphi)}{\partial z} = \frac{\partial}{\partial x}\left(\Gamma_\varphi \frac{\partial \varphi}{\partial x}\right) + \frac{\partial}{\partial y}\left(\Gamma_\varphi \frac{\partial \varphi}{\partial y}\right) + \frac{\partial}{\partial z}\left(\Gamma_\varphi \frac{\partial \varphi}{\partial z}\right) + S_\varphi \tag{4}$$

where u, v and w refer to the fluid velocity components in x, y and z directions, respectively; φ refers to the generalized variable to be displaced by $\varphi = 1$ for continuity equation, $\varphi = u$, v and w for momentum equation, and $\varphi = T$ for energy equation; Γ_φ refers to generalized diffusion coefficient defined in Ref. [20], S_φ refers to source term with different meanings in different equations.

According to the velocity and temperature fields solved by using the above equations and the given boundary conditions, the synergy angle β' between velocity $\boldsymbol{U}$ and temperature gradient ∇T for any element in the fluid field can be calculated as [13]

$$\beta' = \arccos\frac{\boldsymbol{U}\cdot\nabla T}{|\boldsymbol{U}||\nabla T|} \tag{5}$$

The average synergy angle of the whole fluid field β can be calculated by the volume weighted average method through the synergy angle of all the elements.

The local volumetric entropy generation due to the heat transfer irreversibility (S_{T}) and the fluid frictional irreversibility (S_{P}) can be calculated by the following equations [16]

$$S_{\mathrm{T}} = \frac{k}{T^2}(|\nabla T|)^2 \tag{6}$$

$$S_{\mathrm{P}} = \frac{\mu}{T}\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j} \tag{7}$$

where μ is the kinetic viscosity of the fluid medium. Entropy generation due to heat transfer irreversibility and fluid frictional irreversibility are the measures of the irreversibility of the practical heat transfer process and fluid flow process respectively, and it will be used for the evaluation of the heat transfer and fluid flow performance.

Then the local volumetric entropy generation (S_{g}) can be obtained by

$$S_{\mathrm{g}} = S_{\mathrm{T}} + S_{\mathrm{P}} \tag{8}$$

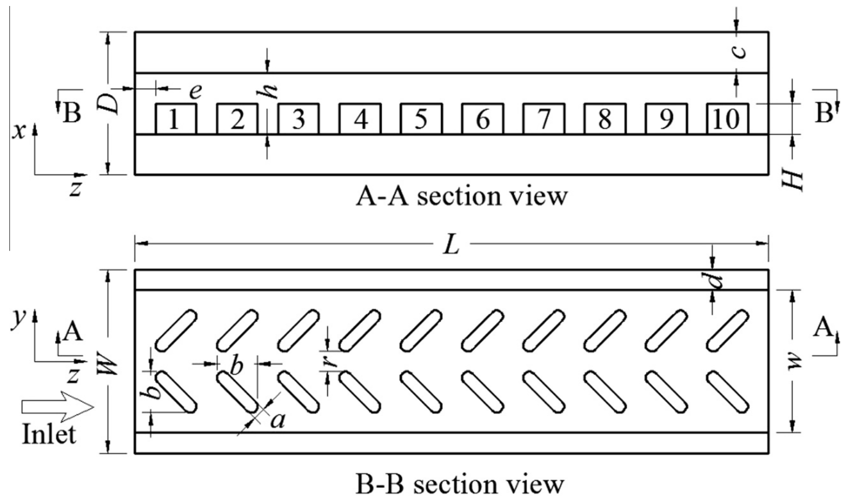

Fig. 1. Schematic view of the mini-channel with discrete double-inclined ribs.

Table 1
Main geometry parameters of the present model.

L/mm	W/mm	D/mm	H/mm	a/mm	b/mm
15.5	4.5	3.5	0.25–1.25	0.3	1.0
w/mm	h/mm	c/mm	d/mm	r/mm	e/mm
3.5	1.5	1.0	0.5	0.5	0.5

In order to make further analysis and comparison, the non-dimensional entropy generation rate (S^*_T, S^*_P and S^*_g) are calculated as below [16]

$$S^*_T = \frac{\int_V S_T dV}{\dot{m}c_p}, \quad S^*_P = \frac{\int_V S_P dV}{\dot{m}c_p}, \quad S^*_g = \frac{\int_V S_g dV}{\dot{m}c_p} \tag{9}$$

where $\dot{m}$ is the mass flow rate of the fluid medium, c_p is the specific heat capacity of the fluid medium, V refers to the volume of the fluid medium.

3. Numerical method and boundary conditions

In this study, the software FLUENT 6.3 which is based on the finite volume method is applied for the numerical computation, and the SIMPLE algorithm is used for the solution of the coupling between the pressure and the velocity. The model is built and meshed with the software Gambit, and unstructured grids are used. In addition, grids in the solid–fluid coupling region are refined to improve the computation accuracy. The result of the grid-independent tests shows that the suitable grid density is about 250,000 cells in total. Copper was selected for the material of the solid zone (include the wall of the mini-channel and the ribs), and the fluid medium is deionized water. The heat flux given on the heated wall is $q = Q/A_w = 40$ W/cm^2, and other external walls are adiabatic. The inlet and outlet conditions used in the calculation are velocity-inlet condition and pressure-outlet condition respectively, and the inlet fluid temperature is $T_\infty = 293.15$ K. The Reynolds number in the mini-channel range from 200 to 800, where the corresponding inlet velocity of the fluid medium range from 0.10 to 0.38 m/s. The convergence criterion is that all of the norms of the residuals for continuity, momentum and energy equations are less than 10^{-6}.

4. Results and analyses

4.1. Performance analyses with different ribs heights H

4.1.1. Effects of the ribs height H on heat transfer performance

Fig. 2 shows the variations of the heat transfer performance (Nu/Nu_0) and the average synergy angles β of the mini-channel ($N = 10$) with different ribs heights. It can be seen from the figure that the heat transfer performance of channels with double-inclined ribs are obviously higher than that of the plain channel, which means that the double-inclined ribs in mini-channel can improve the heat transfer performance effectively. In addition, the heat transfer performance of the mini-channel is increasing with the increase of the ribs height. From the field synergy theory, we know that the heat transfer performance of the mini-channel is mainly depend on the synergy characteristic between the velocity field and the heat-flux field, and height of the ribs is one of the important factors that can cause the changes of the synergy characteristic and result in the changes of the heat transfer performance. Furthermore, from the variation of the synergy angles, it can be seen that the average synergy angles of channels with double-inclined ribs are smaller than that for the plain channel, and the higher the ribs are, the smaller the synergy angles will be, which means that the better the heat transfer performance will be.

Fig. 3 shows the secondary velocity field and temperature field of the cross-section in the fluid zone of the mini-channel, where

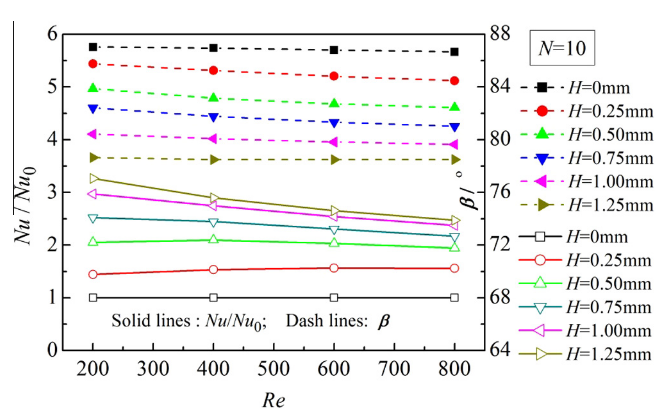

Fig. 2. Variation of the heat transfer and field synergy performance under various ribs heights with Reynolds number, $N = 10$.

$N = 10$, $z/L = 0.5$ and $H = 0$ mm (plain channel), 0.5 and 1 mm, respectively. It can be found that a strong longitudinal vortexes pair is generated in the channels with double-inclined ribs, and the vortexes centers are close to the top of the ribs. Vortexes improve the temperature distribution in the flow field at the same time, as we can see from Fig. 3(d) and (f), the temperature gradient near the wall become larger, especially at the area where the velocity vector pointed to, as a result, the temperature distribution in the middle area of the cross-section is becoming more even. According to the field synergy theory, such kind of the temperature distribution is very helpful to the enhancement of the convective heat transfer. Furthermore, the vortexes can become stronger with higher ribs, and so the synergy characteristic between the velocity and heat-flux field is much better. For the plain channel, as shown in Fig. 3(a), there is no secondary flow generated in the flow field, thus it has an inferior synergy characteristic between the velocity and the temperature field. The phenomenon analyzed above is the essential reason that the double-inclined ribs enhance the mini-channel heat transfer performance.

4.1.2. Effects of the ribs height H on the flow performance

The variation of the average flow resistance coefficient with different ribs heights in the mini-channel ($N = 10$) is shown in Fig. 4. It is obvious that accompanied with the heat transfer enhancement, the flow resistance of the mini-channel increases significantly when the double-inclined ribs are put into mini-channel, and the flow resistance increases with the increasing ribs height. Since both the heat transfer coefficient and the flow resistance are increase with the increase of the ribs height, it is necessary to analysis the overall performance to obtain the optimum ribs height in mini-channel.

4.1.3. Effect of the ribs height H on the overall performance

Fig. 5 shows the performance evaluation criterion (*PEC*) value variation with the ribs height in mini-channel ($N = 10$), where *PEC* is an evaluation coefficient represents the overall performance of a heat transfer unit, which is commonly defined as

$$PEC = (Nu/Nu_0)/(f/f_0)^{1/3} \tag{10}$$

where Nu_0 and f_0 refer to the Nusselt number and the fluid resistance coefficient in plain channel, respectively.

As shown in the figure, the optimum ribs height is not a constant within the whole Reynolds number range, and it shows a trend of decrease with the increase of the Reynolds number. Combined with Figs. 2 and 4, it can be found that Nu/Nu_0 has a rapid increase speed while f/f_0 has a relative lower increase speed with the increase of ribs height when the Reynolds number is low, but it shows a contrary phenomenon when the Reynolds number is higher. In other words, the effects of the ribs height on the flow and heat transfer performance is different under different Reynolds numbers, so the optimum ribs height is different under different Reynolds numbers. This conclusion may make some practical significance for the practical design and application of mini-channel heat sink with double-inclined ribs.

4.2. Performance analyses with different ribs numbers N

4.2.1. Effect of the ribs number N on the heat transfer performance

Fig. 6 shows the comparison of the heat transfer performance of mini-channels between the ribs numbers $N = 1, 5, 10$ and the plain channel ($H = 0$). It is clear that the channels with double-inclined ribs always show better heat transfer performance when compared with the plain channel no matter how many ribs are arranged in

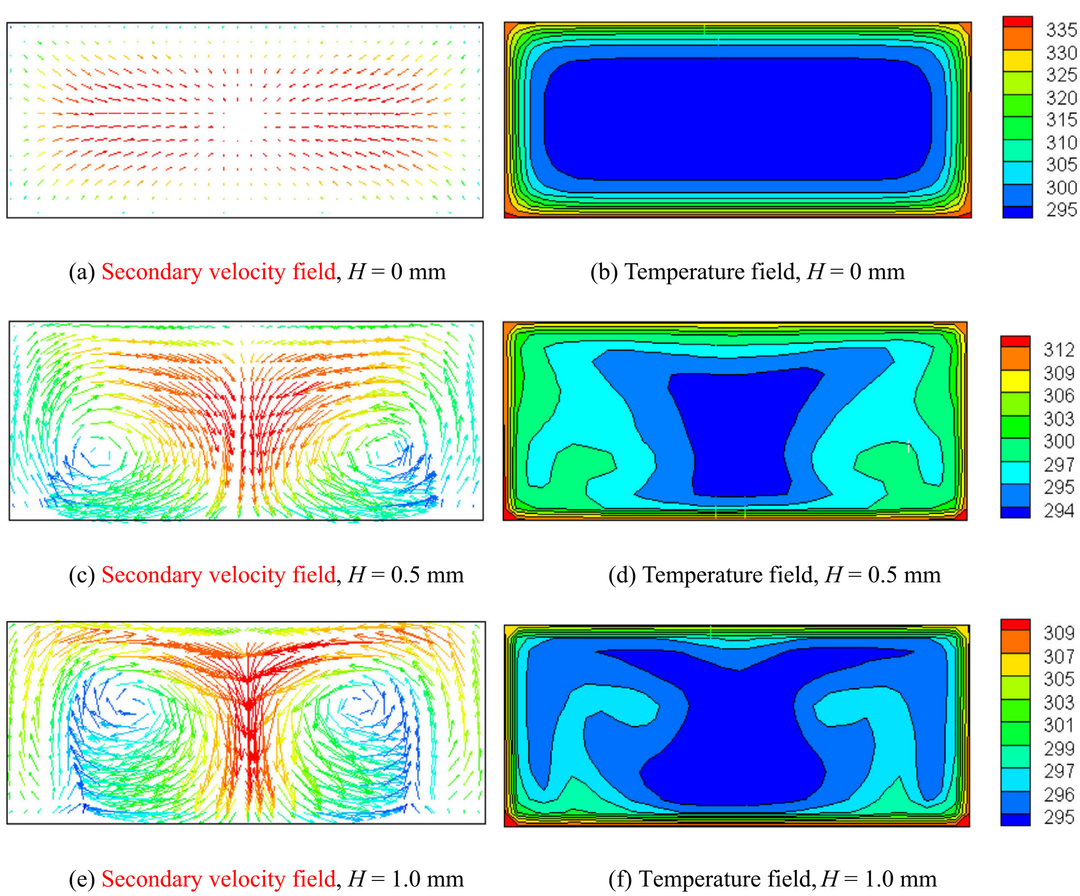

Fig. 3. Tangential velocity and temperature fields at the cross-section ($z/L = 0.5$), $Re = 600$.

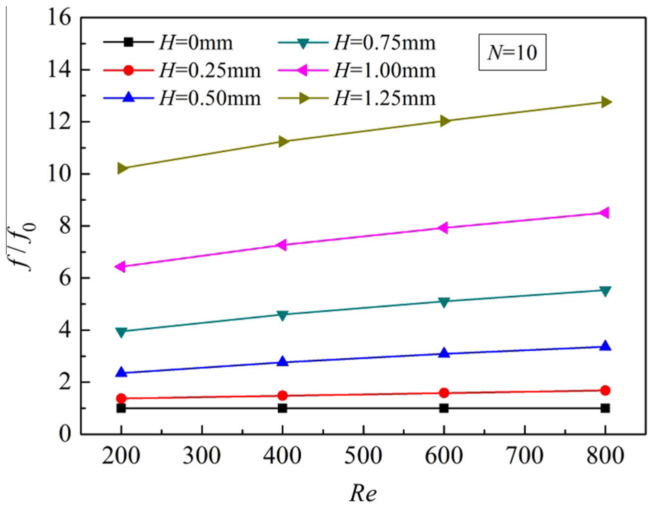

Fig. 4. Variations of the flow performance under various ribs heights with Reynolds number, $N = 10$.

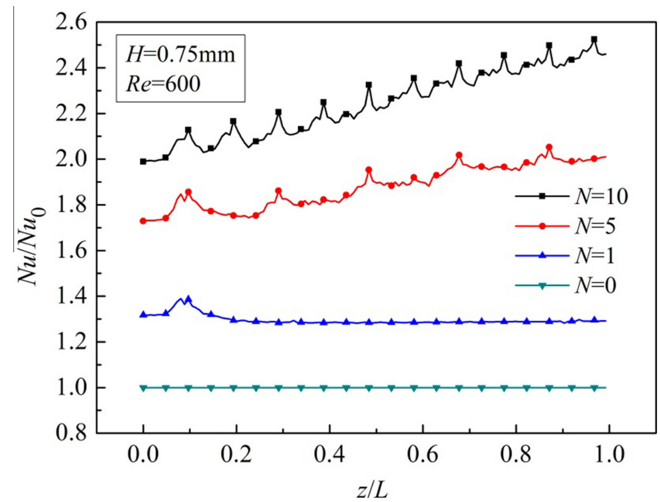

Fig. 7. Variations of average Nu/Nu_0 under different ribs numbers along the flow direction.

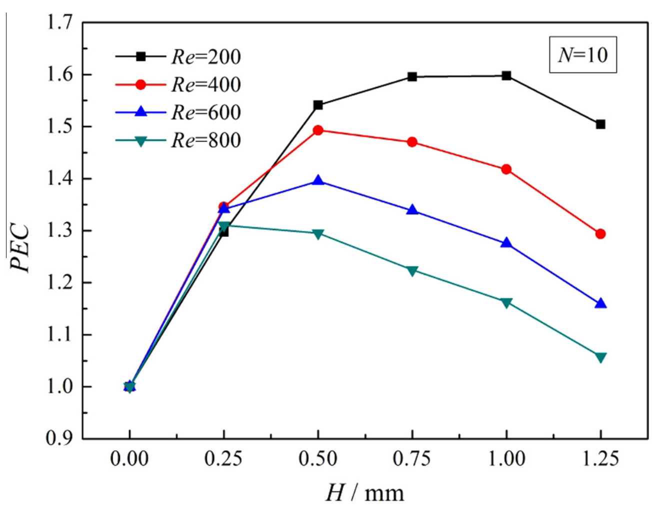

Fig. 5. Variations of the *PEC* under different Reynolds numbers with ribs height, $N = 10$.

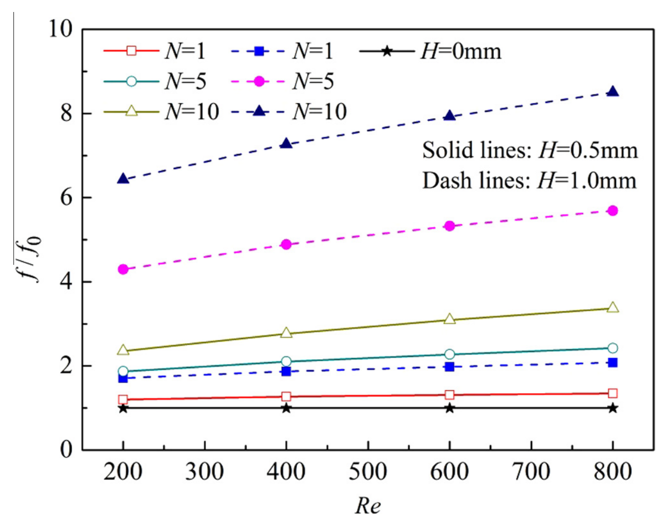

Fig. 8. Variations of the flow performance under various ribs numbers with Reynolds number.

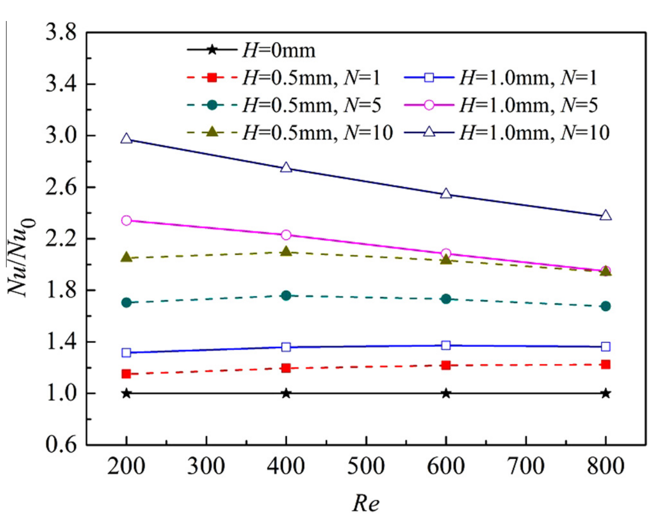

Fig. 6. Variations of the heat transfer performance under various ribs numbers with Reynolds number.

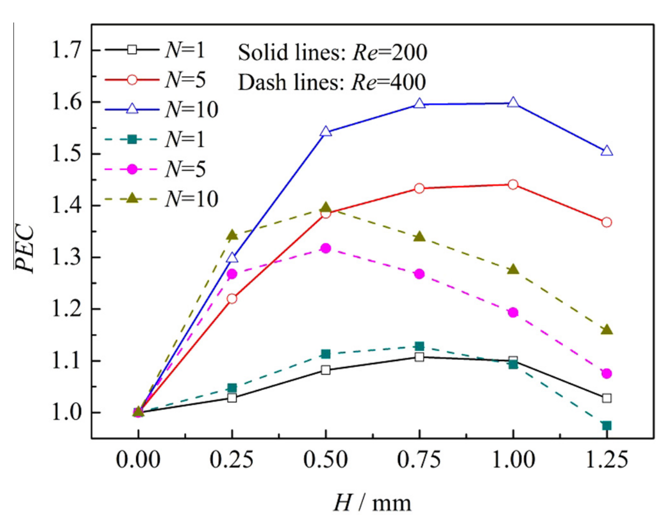

Fig. 9. Comparison of *PEC* values under different ribs numbers.

mini-channel, and the heat transfer enhancement effect become stronger with the increase of the ribs number when the ribs heights are fixed.

This can be explained from its internal mechanism. On the one hand, the vortexes generated by ribs at the upriver can only last for a certain distance along the flow direction because of the viscous dissipation of the fluid, but ribs at the downriver can strengthen the weakened vortexes. Therefore, the longitudinal vortexes can maintain strongly in the whole flow field if there are enough ribs in the channel, which means that characteristic between the velocity and heat-flux for the whole flow field can be improved effectively. Thus the heat transfer performance can be improved. On the other hand, the transverse vortexes also generated at the tip of the ribs when the fluid flow through the ribs. Since the transverse vortexes can enhance the local heat transfer, so more local areas are enhanced with more ribs number, and consequently the heat transfer performance of the entire field is improved.

Fig. 7 shows the evaluation of cross-section average Nu/Nu_0 in z-direction of a typical case ($H = 0.756$ mm, $Re = 600$). We can see from this figure that there will be a peak for the Nu/Nu_0 near each ribs pair. This is the reason for the increase of the heat transfer performance with the increasing ribs number.

4.2.2. Effect of the ribs number N on the flow performance

Fig. 8 shows the comparison of the average resistance coefficient of the mini-channel between different numbers of ribs under ribs heights $H = 0.5$ and 1.0 mm respectively. It is obviously that the flow resistance increase with the increasing ribs number under the same ribs height.

4.2.3. Effect of the ribs number N on the overall performance

Based on the results above, it is obvious that both the heat transfer performance and the flow resistance are increase with

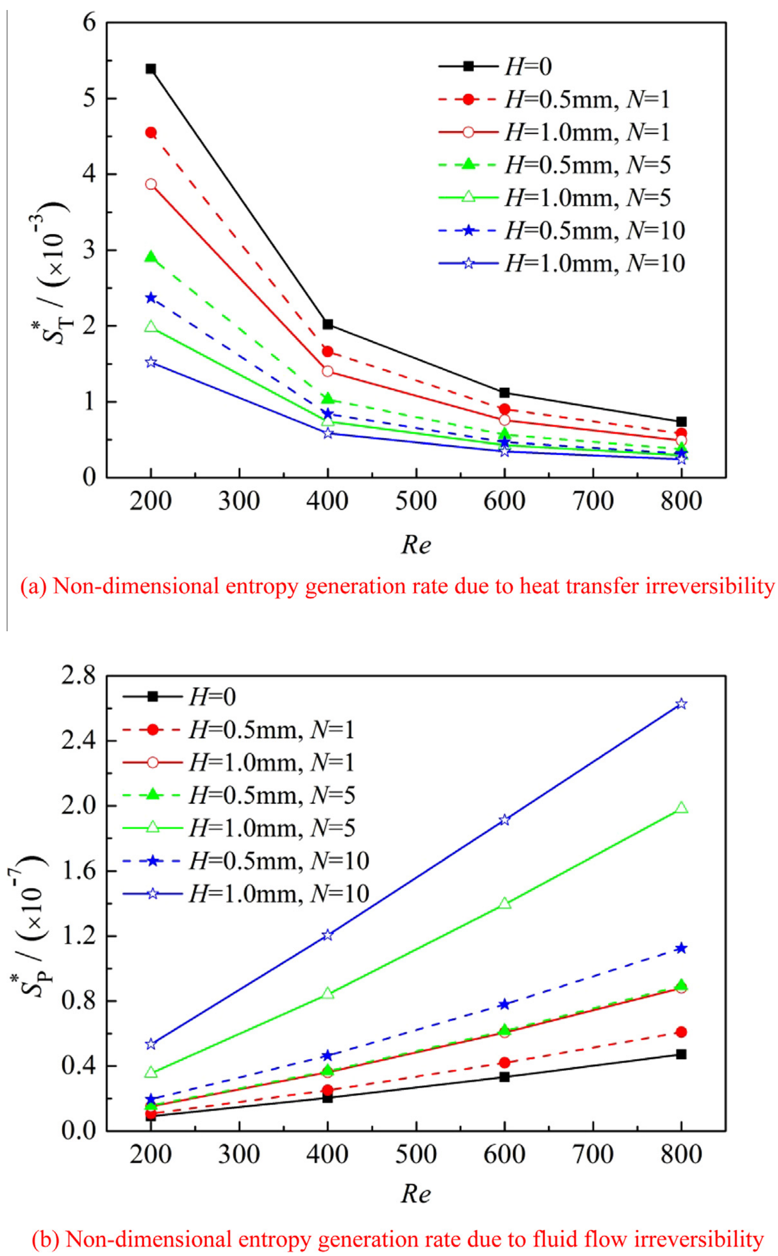

Fig. 10. Non-dimensional entropy generation rates in mini-channel.

the increasing ribs number, so it is also necessary to analysis the overall performance under different ribs numbers.

Fig. 9 shows the variation of the overall performance *PEC* with different ribs numbers for $Re = 200$ and 400. It can be seen that the overall performance increases with the increase of the ribs number when the ribs height and the Reynolds number are fixed. This is result from the fact that effect of the ribs number on the heat transfer performance is larger than that on the flow resistance. According to this conclusion, to increase the number of ribs in mini-channel is an effectively method to improve the comprehensive performance.

4.3. Entropy generation evaluation based on the second law of thermodynamics

The following analyses will be focused on the entropy generation which is based on the second law of thermodynamics. The entropy generation rates are used to analysis the heat transfer and flow performance.

Fig. 10(a) and (b) show the entropy generation in mini-channel with different ribs heights ($H = 0$, 0.5 and 1.0 mm) and different ribs numbers ($N = 1$, 5 and 10), where S_T^* and S_P^* are the non-dimensional entropy generation rate due to the irreversible heat transfer and fluid flow respectively. From this figure, it can be found that to increase any one of the three parameters of Reynolds number, ribs height H and ribs number N, will causes the decrease of S_T^* and the increase of S_P^*. This result is because that when the heat transfer performance is enhanced, the temperature gradients in the flow field become more smooth and so the values of S_T^* are reduced, and the flow irreversibility will become larger with the decrease of the flow resistance performance. So the entropy generation rate S_T^* and S_P^* are closely corresponding to the heat transfer and flow performance, and can be used for the evaluation of the heat transfer and flow performance respectively.

But from the comparison between Fig. 10(a) and (b), it is seen that the values of S_T^* is much larger than the S_P^*, and so the S_T^* takes a dominant place in the total entropy generation. Therefore, the total entropy generation rates S_g^* are almost the same as S_T^*. That is to say, if the total entropy generation rate is regarded as the criterion for the overall performance, the overall performance will increase with the increasing of the ribs height or ribs number. Compared with the *PEC* value analyzed above, it is not reasonable. Therefore, the total entropy generation rate S_g^* cannot be used to evaluate the overall performance, while the entropy generation rate S_T^* and S_P^* can be used to evaluate the heat transfer and the flow performance respectively.

Furthermore, from the perspective of the field synergy principle, we can see that a good field synergy characteristic between the velocity field and the heat flux field will lead a smaller temperature gradient in the flow region under the same heat flux, which will result in a smaller entropy generation rate due to heat transfer irreversibility, and vice versa. Therefore, the field synergy characteristic and the entropy generation rate due to heat transfer irreversibility are consistent in evaluating the heat transfer performance.

5. Conclusions

A three dimensional model of the mini-channel with discrete double-inclined ribs is established in the present paper. Effects of the ribs height H, ribs number N and Reynolds number on the mini-channel performance are numerically studied and analyzed from the field synergy theory and the entropy generation. The following conclusions are obtained.

(1) When the discrete double-inclined ribs are arranged in the mini-channel, strong longitudinal vortexes are generated in the channel and transverse vortexes generated near the tips of each rib. Therefore, the synergy characteristic between the velocity field and heat-flux field is improved, and the heat transfer performance is enhanced effectively. But the flow resistance has a large increase at the same time.

(2) According to the analysis of the comprehensive performance, the optimum height of the ribs is not a constant under different Reynolds numbers, and the height should be decreased under a larger Reynolds number to obtain the best overall performance. This is because that the effects of the ribs height on heat transfer performance are not the same as those on the flow performance. In addition, the overall heat transfer performance increases with the increasing of the ribs number along the flow direction when the Reynolds number and ribs height are prescribed.

(3) The entropy generation rate S_T^* and S_P^* can be used for the evaluation of the heat transfer and the flow performance of the mini-channel well respectively. However, the total entropy generation S_g^* rate cannot be used as a criterion for the overall performance. The field synergy characteristic and the entropy generation rate due to heat transfer irreversibility are consistent in evaluating the heat transfer performance.

Conflict of interest

None declared.

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