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NUMERICAL STUDY ON HEAT TRANSFER CHARACTERISTICS OF A CHANNEL WITH DENSELY-MOUNTED CUT-FINS

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ABSTRACT

The present study reports the results of the three-dimensional numerical simulation on heat transfer characteristics of a channel having densely-mounted ‘cut-fins’. The effects of the length, depth and geometrical phase shift of the notch of the fins on heat transfer and friction loss have been examined. From the results, it is shown that by applying a notch to the fin, both friction factor and thermal resistance decreases compared to a continuous plain-fin. This tendency appears more clearly for larger notch length and depth. Furthermore, higher heat transfer performance is obtained for the cut-fins with obliquely arranged notches due to the induced secondary flow motion.

1. INTRODUCTION

Micro fin array is an effective cooling technique and is widely used in various small applications such as compact heat exchangers and electronic devise cooling. In such micro fin arrays, high heat transfer coefficient is available, whereas unfavorable pressure loss takes place. On the other hand, when a clearance exists between the fin top edge and the channel top wall, improvement of the pressure loss is obtained at the cost of a considerable decrease of the heat transfer coefficient at the fin sidewalls due to the reduction of the mass flow rate in those areas. Thus, a serious trade-off problem takes place (Sparrow, 1978, Matsubara, 1996, Min, 2004). Considering this background, a fin with square shape notches, referred to as ‘cut-fin’, is proposed and investigated in the present study from a three-dimensional numerical approach. The expected effectiveness of the cut-fin is threefold: to reduce the pressure loss penalty; to increase heat transfer rate at the fin sidewalls by increasing the mass flow in between the fins; to enhance the mixing between the fluid passing the fin tip clearance and the fluid in between the fins.

2. COMPUTATIONAL PROCEDURE

2.1 Numerical conditions

The computational domain is illustrated in Fig. 1. Periodical boundary conditions are applied in the streamwise and spanwise boundaries representing a continuous fin arrangement. Constant heat flux condition and adiabatic wall condition are applied at the channel bottom and top walls, respectively. At the fin surface, harmonic mean value of the fluid and solid thermal conductivity is used to solve the heat flux at the interface. The working fluid is air and the material of the fin is stainless steel.

In the present computation, the fin height H_f, fin pitch W_P, fin width w and notch pitch L are kept constant, i.e. $H_f/H=0.75$, $W_P/H=0.075$, $w/H=0.025$, $L/H=3.125$. The notch length L_n and depth H_n, streamwise geometrical phase shift ΔL are varied whose actual values are tabularized in Table 1. The Reynolds number, which is based on the channel hydraulic diameter and the streamwise mean velocity, is fixed at a constant value as Re=200.

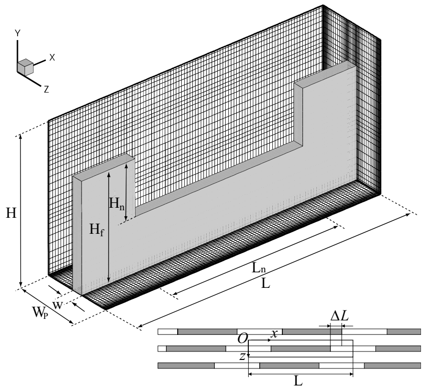


Fig. 1 Computational domain.

2.2 Numerical method

The governing equations solved in the present computation are three-dimensional, incompressible, time-dependent continuous, Navier-Stokes and energy equations. The present numerical method is based on Finite Volume Method. The Navier-Stokes equation is discretized by applying the 5th-order upwind scheme and 4th-order central-difference scheme to the convection and diffusion terms, respectively. Fully implicit method is adopted for the temporal discretization and SIMPLE is used to solve the pressure correction.

Table 1 Geometric parameters

Notation	H_n/H_f	L_n/L	$\Delta L/L$
Plain	-	-	-
Cut-a	0.5	0.6	0
Cut-b	0.5	0.32	0
Cut-c	0.75	0.6	0
Inline	1.0	0.6	0
O-Cut	0.5	0.6	0.1

3. RESULTS AND DISCUSSION

Figure 2 shows the friction factor f and the thermal resistance R. R is calculated as $(\overline{T_w}-\overline{T_{in}})/q$ where $\overline{T_w}$, $\overline{T_{in}}$ and q are the averaged channel bottom wall temperature, bulk mean temperature of the inlet flow and the heat flux provided at the bottom wall, respectively. Both values are normalized by the value of the Plain case, a fin without notches. In all cases, f/f_p is smaller than unity which indicates that adding a notch to the fin reduces the pressure loss. Comparing cases Cut-a~c, we can see that larger pressure loss reduction is attained by increasing the notch length L_n, and also that the effect of the notch depth H_n is virtually negligible. For O-Cut case, f/f_p shows a much smaller value compared to the other cases. This is due to the fact that large amount of flow passes in the oblique direction along the notch.

For the Inline case, in which the notch depth and fin height are equal, a much larger R is obtained compared to the other cases. This is ascribable to the salient increase of the fluid temperature observed in the area where no fin exists. On the other hand, only a small difference is found in R for cases Cut-a~c. This indicates that the influence of the notch length and depth on R is small within the examined cases and that equal heat transfer performance is obtained in spite of the reduced heat transfer area for the cut-fins compared to the Plain case.

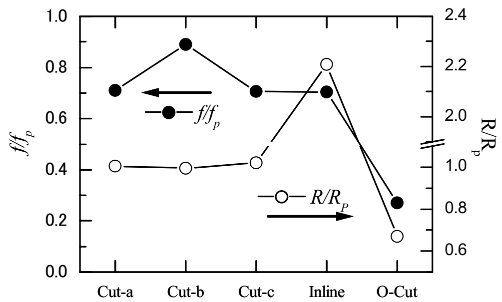
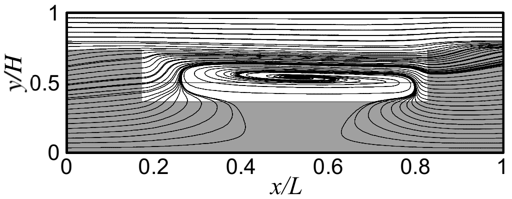
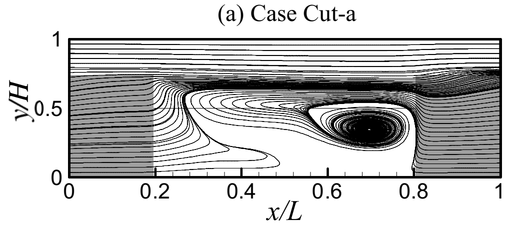


Fig. 2 Friction factor and thermal resistance.



(a) Case Cut-a



(b) Case Inline

Fig. 3 Fluid-particle tracking lines.

Figure 3 shows the side-view of the fluid-particle tracking lines of cases Cut-a and Inline. The initial locations of the tracking lines are z/W=1.0 at the domain inlet boundary. A circulation is formed in the region of $0.2 \leq x/L \leq 0.8$ and upward and downward flows are observed at $x/L=0.3$ and $x/L=0.8$, respectively. For case Inline, as mentioned before, the existence of this circulation and the lack of the fin in this area incur a temperature increase and impair the heat transfer performance at the bottom wall. At the downstream side of the notch, the downwash flow induces the fluid located at the upper side of the channel into the space between the fins. Although not shown here, the mass flow rate in between the fins increased by the rate of 2~5 time larger than the Plain case. Thus, thermal boundary layer is redeveloped and large heat transfer coefficient is obtained in this area, which is the reason why the cut fin case provides high heat transfer performance.

Figure 4 shows the fluid-particle tracking lines of case O-Cut in the same manner as Fig. 3 on which the contour maps of the spanwise velocity W at $z/W_P=0.5$ is superimposed. A strong spanwise flow motion exists inside the notch and no circulation is generated. This spanwise flow produces flow impingement and redevelopment of the thermal boundary layer at the leading and trailing edges of the fin entailing an increase of the heat transfer coefficient. Since the Reynolds number is based on the streamwise velocity without considering the spanwise flow, the total mass flow rate is not identical between case O-Cut and the other cases. Thus, direct comparison is difficult. However, from Figs. 2 and 4, it is believed that better heat transfer and pressure loss performances can be obtained by the O-Cut case.

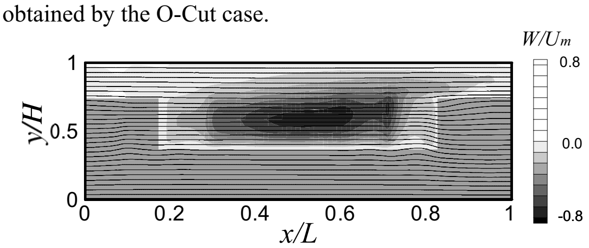


Fig. 4 Fluid-particle tracking lines and spanwise velocity contours of case O-Cut.

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