Proceedings

Enhanced, Compact and Ultra-Compact Heat Exchangers: Science, Engineering and Technology

Engineering Conferences International *Year* 2005

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Proceedings of Fifth International Conference on **Enhanced, Compact and Ultra-Compact Heat Exchangers: Science, Engineering and Technology**, Eds. R.K. Shah, M. Ishizuka, T.M. Rudy, and V.V. Wadekar, Engineering Conferences International, Hoboken, NJ, USA, September 2005.

CHE2005 – 15

EXPERIMENTAL AND 3D NUMERICAL STUDY OFAIR SIDE HEAT TRANSFER AND PRESSURE DROP OF SLOTTED FIN SURFACE

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ABSTRACT

Experimental and numerical investigation of air side heat transfer and fluid flow performance of a typical slotted fin surface with "X" arrangement of strips is presented. Three-dimensional numerical computation was made in which the tube configuration is simulated with step-wise approximation, and the fin efficiency is also calculated with conjugated computation. The numerical results are verified by the experimental data and the experimental correlation of friction factor and heat transfer coefficient are provided. The performances of the slotted and plain plate fin surfaces are compared. It is found that the slotted fin-and-tube heat transfer surfaces has excellent performance compared to the plain plate fin heat transfer surfaces. The numerical results are further analyzed with the field synergy principle.

INTRODUCTION

Plate fin-and-tube heat exchangers are widely used in various engineering fields, such as HVAC&R (heating, ventilating, air-conditioning and refrigeration) , automobiles, in which heat transfer enhancement techniques is a key issue to reduce the equipment volume. Slotted fin surface is an effective way to reduce the air-side thermal resistance which is often the major part of the overall thermal resistance of many industrial heat exchangers. According to the traditional understanding of heat transfer enhancement mechanism, the major idea adopted in this kind of fin surfaces is to interrupt the flow by repeated geometries or to reduce the thickness of the thermal boundary layers (Kays and London, 1980; Incropera and DeWitt, 1996; Yang and Tao, 1998).

Experimental study is very useful but very expensive because of the high cost of the tools needed to produce a wide rage of geometric variations of fin surface. For example, to investigate the influence of design parameters on the heat transfer and friction characteristics of slit fin surfaces, eighteen kinds of scaled-up models are specially made by Kang and Kim(1999). Experimental studies for different types of plate fins have been extensively performed, and to name a few publications, Kang et al. (1994a), Kang et al.(1994b), Xin et al.(1994), Horuz and Kurem(1998), Yun and Lee (1999), Yun and Lee(2000), Kang and Kim (2001), Wang et al.(2001), Wang et al. (2002) may be consulted.

Numerical modeling, once validated by some test data, on the other hand, provides a cost-effective means for such a parametric study. Great deals of efforts have been paid in this regard. Shah et al.(2000) made a comprehensive review of numerical investigations on heat transfer and pressure drop characteristics of plate fin-and-tube surfaces published before 2000. Most previous numerical simulations have studied the characteristics of the wavy fin, louver fin, offset strip fin on the whole fin surface or for a representative unit of the periodic fully developed region. To the authors' knowledge, no numerical modeling has been done for the strip fin with X-arrangement where the strips are positioned on the fin surface in the shape of "X" when viewed from the top of the fin surface. The X-arrangement of the strips was first put forward by Hiroaki et al.(1989) and is probably one of the most widely used fin geometries nowadays in the air-conditioning engineering.

The reasons why the interrupted geometries can enhance heat transfer are usually attributed to the decrease in the thermal boundary layer and the increase of the disturbance in the fluid . Recently Guo et al.(1998) and Wang et al.(1998) proposed a novel concept of enhancing convective heat transfer. It is indicated that the heat transfer can be enhanced by decreasing the intersection angle between velocity and temperature gradient for the boundary layer flow. Later Tao et al.(2002a) extended this principle to elliptic flow.This principle is now called as the field synergy (coordination) principle. Tao et al.(2002b) made a unified analysis for the existing explanations of heat transfer enhancement techniques with field synergy principle.

One of the major objectives of this paper is to provide a three dimensional numerical model on the slotted fin surface of "X" arrangement of strips, verify the numerical results by experiment, and compare the performance of plain plate fin and slotted fin surfaces. The second objective of this paper is to apply the field synergy principle to analyze the numerical results to deepen our understanding of the heat transfer enhancement essence.

In the following presentation, the numerical implementation process of the slotted fin surface of "X" arrangement of strips is presented, experimental measurement process is briefly introduced , and the numerical and experimental results are compared. The heat transfer performance comparison of the slotted fin surface and plain plate fin are discussed. Finally the numerical results are analyzed with the field synergy principle.

PHYSICAL MODEL

Figure 1 show the schematic diagram of a typical plain plate fin and tube heat exchanger. The slotted fin is alike except that some pieces of strips are punched from the base sheet. The tube is usually made of copper and the fin surfaces are made of aluminum. It is assumed that the fluid is incompressible with constant property and the flow is laminar and in steady state. Some hot fluid, say freon refrigerant, goes through the tubes and heat is transmitted through the tube wall and the fin surfaces to the air. The heat transfer and pressure drop characteristics of the airside are to be solved by numerical modeling. Because of the relatively high heat transfer coefficient of the inner fluid and the high thermal conductivity of the tube wall, the tube temperature is assumed to be constant. However, the fin surface temperature distribution is to be solved, hence, the problem is of conjugated type in which both the temperatures in the fin solid surface and in the fluid are to be determined simultaneously (Tao, 2001). In order to implement the computation simply, the inclined angle of strip corner is omitted. The actual and modeled fin shapes are shown in Figure 2(a) and (b), respectively. The value of parameter are listed in Table 1.

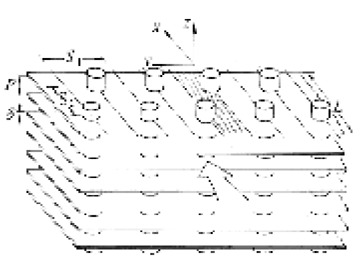
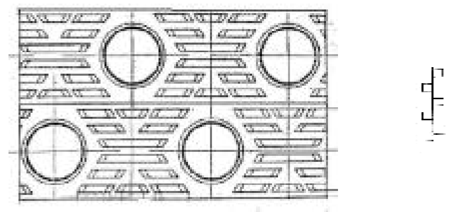
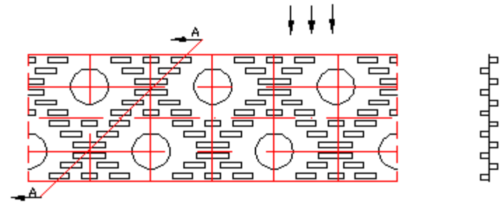


Fig. 1 Pictorial view of a plain plate tube-and-fin surface.



(a) Actual X-arrangement of strips



(b)Modeled X-arrangement of strips

Fig.2 Diagram of slotted fin geometry

Table 1 Fin geometry parameters

Tube outside diameter	7mm
Longitudinal tube pitch	21mm
Transverse tube pitch	12.97mm
Fin thickness	0.12mm
Fin pitch	1.4mm
Strip height	0.7mm

COMPUTATIONAL DOMAIN

The cell between two neighboring fin surfaces is studied according to the geometry character of symmetry and periodicity(Fig. 3). In the numerical simulation the tube wall temperature is specified, while the fin temperature distribution is solved during the iterative process. We take the whole region outlined by the dashed lines in Fig.3 as the computational domain, and use three dimensional Cartesian coordinates with sufficient fine grid system to simulate different parts of materials (air, tube and fin surface).

In the computational domain shown in Figure 3, X,Y,Z stands for the stream-wise coordinate, span-wise coordinate and the fin pitch direction, respectively. The fluid flows across the fin surface from the left to the right. We extend 1.5 times of stream-wise fin length for the inlet domain and 5 time in the outlet domain for the execution of boundary conditions which is to be addressed later. This implies that the whole stream-wise length of the computation domain is 7.5 times of the real fin length. For saving space, in Figure 3 the extended domain is not presented in scale.

GOVERNING EQUAIONS AND BOUNDARY CONDITIONS

The governing equations for continuity, momentum and energy in the computational domain can be expressed as follows:

Continuity equation:

$$\frac{\partial}{\partial x_i}(r u_i) = 0 \qquad (1)$$

Momentum equations:

$$\frac{\partial}{\partial x_i}(r u_i u_k) = \frac{\partial}{\partial x_i}\left(m \frac{\partial u_k}{\partial x_i}\right) - \frac{\partial p}{\partial x_k} \qquad (2)$$

Energy equation:

$$\frac{\partial}{\partial x_i}(r u_i T) = \frac{\partial}{\partial x_i}\left(\Gamma \frac{\partial T}{\partial x_i}\right) \qquad (3)$$

where

$$\Gamma = \frac{l}{c_p} \qquad (4)$$

The boundary conditions are required for all boundaries of the computation domain due to the elliptic form of governing equations in the Cartesian coordinates.

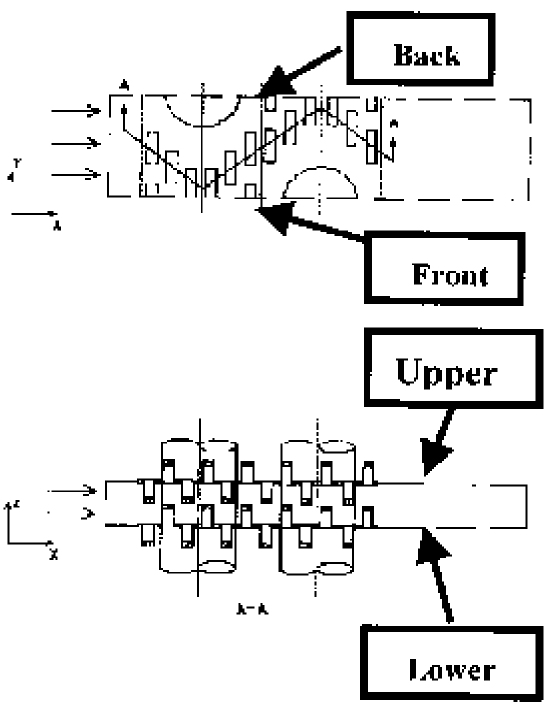


Fig. 3 Computational domain

For the studied conjugated type of the problem, the fin surfaces are considered as part of the solution domain and will be treated as a special type of fluid. The required conditions are described for the three regions (upstream extended region, fin coil region, and down stream extended region), which are listed in Table 2 for the slotted surface. In the table USED means upstream extended domain, and DSED as downstream extended domain. With the extended region added, it is conveniently to adopt the so-called one-way coordinate assumption for the treatment of the outlet boundary condition. For the plain plate fin surface, the boundary conditions are much simpler and can be treated alike. They are not shown in the table for simplicity.

NUMERICAL METHODS

For conjugated computation the solid regions including fin, strip, and tube should be also meshed and treated as some special fluids. The code should be able to identify which is which in the field. To this regard, we introduce a special array called FLAG to identify different regions composed of fluid, strip fin and tube, and define the value of FLAG is 1 for fluid, and 2,3,4 for fin, tube, and strip, respectively. After the computation domain is meshed with staggered grid, the appropriate values of the elements in FLAG have to be input according to the fin geometry such that the above four regions can be identified correctly.

To guarantee the continuity of the flux rate at the interface, the thermal conductivity of fin and fluid adopts individual value, while the heat capacity of solid takes the value of the fluid (Tao,2001) and a very large value of the thermal conductivity is assigned to the tube region to guarantee the tube temperature to be constant.

The computational domain is discretized into non-uniform grids, with the grids of fin coils region being fine, and that in the extension domain being coarse. Governing equations are discretized by finite volume method, and the coupling of pressure and velocity is implemented by CLEAR algorithm in the staggered grids (Tao et al, 2004a, 2004b).

PARAMETER DEFINITIONS

Some parameters are defined as follows:

$$\mathrm{Re} = \frac{r u_m D_e}{m} \qquad (5)$$

$$Q_{real} = \overset{g}{m} c_P (T_{out} - T_{in}) \qquad (6)$$

Table 2 Boundary condition of slotted surface

			USED	**DSED**
inlet	Velocity	U	$u=0$	
		V	$v=0$	
		W	$w=0$	
	Temperature		$T=T_{in}$	
Upper and Lower sides	Velocity	U	$\frac{\partial u}{\partial z}=0$	$\frac{\partial u}{\partial z}=0$
		V	$\frac{\partial v}{\partial z}=0$	$\frac{\partial v}{\partial z}=0$
		W	$w=0$	$w=0$
	Temperature		$\frac{\partial T}{\partial z}=0$	$\frac{\partial T}{\partial z}=0$
Front and back sides	Velocity	U	$\frac{\partial u}{\partial y}=0$	$\frac{\partial u}{\partial y}=0$
		V	$v=0$	$v=0$
		w	$\frac{\partial w}{\partial y}=0$	$\frac{\partial w}{\partial y}=0$
	Temperature		$\frac{\partial T}{\partial y}=0$	$\frac{\partial T}{\partial y}=0$
Outlet	Velocity	U		$\frac{\partial u}{\partial x}=0$
		V		$\frac{\partial v}{\partial x}=0$
		w		$\frac{\partial w}{\partial x}=0$
	Temperature			$\frac{\partial T}{\partial x}=0$

$$\Delta T=\frac{T_{\max}-T_{\min}}{\log\left(T_{\max}/T_{\min}\right)} \tag{7}$$

$$h=\frac{Q_{real}}{A\Delta Th_0} \tag{8}$$

$$Nu=\frac{hD_e}{l} \tag{9}$$

$$f=\frac{\Delta P}{\frac{1}{2}ru_m^2}\bullet\frac{D_e}{L} \tag{10}$$

Table 2 Boundary condition of slotted surface (Continued)

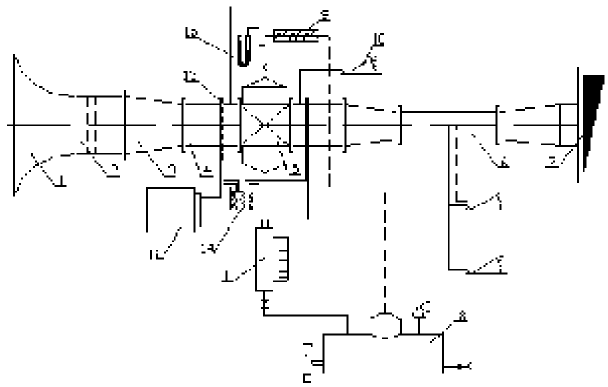
			Fin	**Tube**	**Fluid**
inlet	Velocity	U			
		V			
		W			
	Temperature				
Upper and Lower sides	Velocity	U	$u=0$	$u=0$	periodic
		V	$v=0$	$v=0$	periodic
		W	$w=0$	$w=0$	periodic
	Temperature		periodic	$T=T_w$	periodic
Front and back sides	Velocity	U	$u=0$	$u=0$	$\frac{\partial u}{\partial y}=0$
		V	$v=0$	$v=0$	$v=0$
		w	$w=0$	$w=0$	$\frac{\partial w}{\partial y}=0$
	Temperature		$\frac{\partial T}{\partial y}=0$	$T=T_w$	$\frac{\partial T}{\partial y}=0$
Outlet	Velocity	U			
		V			
		w			
	Temperature				

In the above equations, u_m is the mean velocity of the minimum transverse area, D_e is the outside tube diameter, T_{in}, T_{out}、 are the fluid bulk temperature of inlet and outlet of the fin surface, respectively. T_{max} and T_{min} are the maximum and minimum temperature differences between wall and fluid , respectively.

NUMERICAL RESULTS AND EXPERIMENTAL VERIFICATION

Experimental System

For the verification of numerical results, special experimental measurements were conducted in the wind tunnel of the authors' heat transfer laboratory. The air frontal velocities covered a range of 1~9.5m/s with the corresponding Reynolds number ranging from $.80\times10^2 < Re < 6.84\times10^3$. The measurements were conducted for the overall heat transfer coefficients and the air-side heat transfer coefficients were obtained via the Wilson plot techniques. The experimental system is shown in Fig.4.



1.entrance 2.straighting section 3. contraction section 4. stabilizer 5. test section 6. flow metering duct 7. blower 8. boiler 9. steam overheater 10.incliend tube draft gauge 11.volumetric meter 12. digital volt meter 13. thermocouple grids 14.mixture of water and ice 15. pressure difference meter

Fig.4 Experimental system

Heat Transfer Coefficient Comparison

The heat transfer coefficient results obtained from experiment and the corresponding numerical results are compared in the Figure 5. It is to be noted that the ordinate is the production of heat transfer coefficient and the overall fin efficiency. The frontal velocity ranges from 1~9.5m/s. It can be seen from Fig. 5 that at the velocity of V=3.6m/s the two results meet each other. When the frontal velocity is lower than this one, the numerical result is higher than experiment while when the frontal velocity is higher than this value, the situation is the opposite. The deviation between experimental and numerical results ranges from -8.8% to $+9.26\%$. General speaking, the numerical results are in good agreement with experimental data, and it can be accepted from the view of engineering.

Fin Efficiency Calculation

As far as fin efficiency is concerned, only for some fin shapes whose structure are quiet regular formulae can

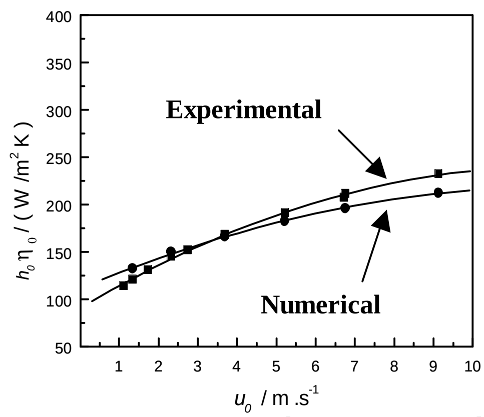


Fig. 5 Comparison between tested and numerical results

be obtained under some special conditions. For example Gray and Webb (1980) provided fin efficiency with theoretical analysis; Chien and Jiin (2002) studied fin efficiency for two dimensional situation. However, it is difficult to gain the fin efficiency of slotted fin surface except for numerical methods. It is to be noted that Tao and Lu (1994) proposed some numerical methods for determining slotted fin efficiency in which the air side heat transfer coefficient is assumed to be constant.

In this paper, we used a numerical method to calculate slotted fin efficiency from the basic definition of fin efficiency which can take the variation of air side heat transfer coefficient into account. In our numerical simulation the efficiency of the slotted fin is computed as follows. According to heat transfer theory (Incropera and DeWitt, 1996; Yang and Tao, 1998), the fin efficiency is defined as follows:

$$h_f = \frac{Q_{real}}{Q_{ideal}} \quad (11)$$

where Q_{real} is the actual heat transfer rate between air and the fin surface; Q_{ideal} is the ideal heat transfer rate when the fin temperature is equal to the tube temperature $T_{W.}$ To implement the ideal situation, we just artificially give the fin surface a very large value of thermal conductivity, say $1.0\text{x}103^0$, which leads to the results of uniform temperature of the fin surface equal to the value of the tube wall. The overall efficiency of fin surface is defined by

$$h_0 = (A_{tube} + A_{fin} h_{fin}) / A \quad (12)$$

The fin efficiency of the studied slotted fin surface is presented in Figure 6. It can be seen that the fin efficiency decrease with the increase of Re number or frontal velocity which is consistent with the heat transfer theory.

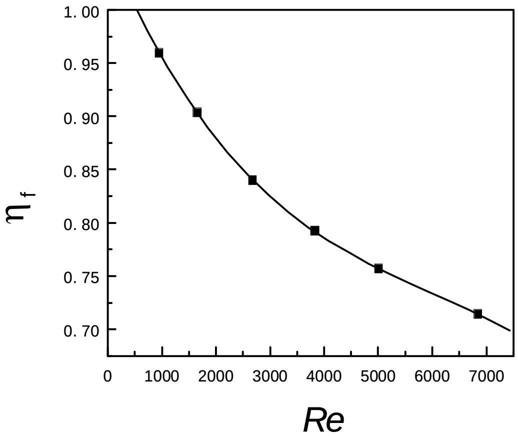


Fig. 6 Predicted fin efficiency

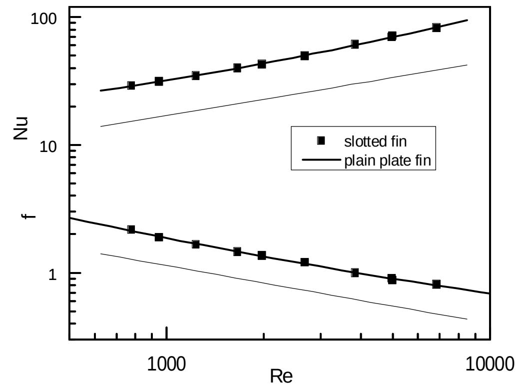


Fig. 7 Comparison of slotted and plain plate fin surfaces

Correlation of Heat Transfer and Friction Factors

Based on the fin efficiency, we can get the pure heat transfer coefficient and Nu number from the experimental results. The experimental data for pressure drop and heat transfer are then correlated and following results are obtained:

$$Nu = 10^{1.1974 - 0.2078(\lg Re) + 0.1034(\lg Re)^2}$$

$$f = 10^{2.4249 - 0.9307(\lg Re) + 0.0711(\lg Re)^2}$$

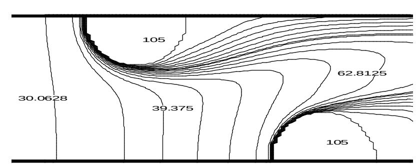
The Reynolds number ranges from 7.80×10^2 to 6.84×10^3. The maximum deviation of the curve fitting ranges from＋1.75％ to -1.57％ and＋2.42％to-2.94％ for heat transfer and friction factor, respectively.

Comparison between Plain Plate and Slotted Fin Surfaces

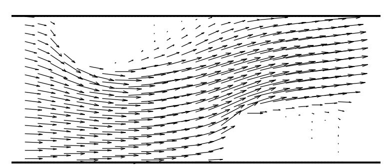
The heat transfer and friction factor of plain plate fin are also provided for comparison with slotted fin surface. Figure 7 shows the comparison of the test results. The slotted fin can averagely improve heat transfer 97.3% compared to plain plate fin, and friction factor is 63.4% higher than that of plain plate fin. These results revealed the advantage of "X" arrangement of strips in the slotted fin surface.

HEAT TRANSFER ENHANCEMENT ANALYSIS

As indicated above, from the traditional viewpoint, strips in the fin can enhance convective heat transfer because it can interrupt the flow boundary layer to reduce its thickness by repeatedly recreation of new thermal boundary layers, or can increase the disturbance in the flow field. In the following we try to analyze the results based on the field synergy principle. The basic idea of the field synergy principle can be stated as follows: to enhance convective heat transfer the most fundamental mechanism is to reduce the intersection angle between velocity and the temperature gradient. In this study a three dimensional numerical simulation is also made on plain plate fin which have the same geometry except being without strips. To have deeper understanding of this point of view, we examined some computational results of the velocity vector and isotherm distributions in the middle plan between the two adjacent fin surfaces for the plain plate fin and slotted fin situation. These results are presented in Figs. 8 and 9.



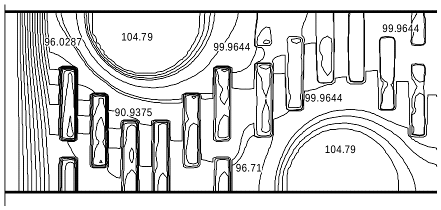
(a) Streamlines



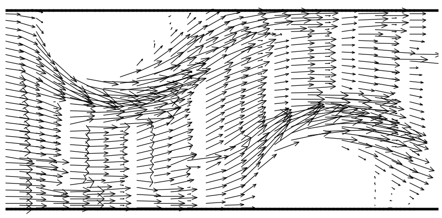
(b) Isothermals

Fig. 8 Streamlines and isothermals of plain plate fin surface (Re = 1656).

It can be seen from Figs. 8(a),(b) that in the upstream part of the fin, the temperature contours are almost perpendicular to the velocity vector, this implies that the fluid temperature gradient are almost in the same direction as the velocity, i.e, the synergy between velocity and temperature gradient is very good.



(a) Streamlines



(b) Isothermals

Fig.9 Streamlines and isothermals of slotted fin surface (Re = 1656)

Hence, in this part of the fin, the heat transfer augmentation by using strips is not very worthy. However, in the downstream part of the fin, the temperature contours are almost parallel to the velocity vector in the major part of the region. It is this place that enhancement techniques are needed in order to improve the synergy between velocity and the temperature gradient and the existing strips contribute more to the heat transfer enhancement than the strips in the upper stream part. From Fig. 9we can find that the strips clearly improve the synergy between velocity and pressure.

Figure 10 shows the numerically predicted heat transfer coefficient distribution of plain plate fin and slotted fin along the flow direction. It can be found that for the plain plate fin the heat transfer coefficient in the down stream part is lower than upstream part for plaint plate fin, on the other hand for the slotted fin the heat transfer coefficient in the upstream part and downstream

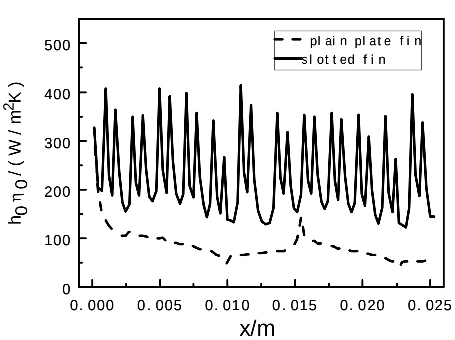


Fig. 10 Heat transfer coefficient distribution (Re = 1656)

part are almost the same which proves the correctness of the above analysis.

CONCLUSONS

In this paper, three dimensional numerical study on air side heat transfer and fluid flow characteristics of slotted fin surface with "X" arrangement of strips are conducted by experimental and numerical methods. The heat transfer coefficients obtained from the numerical simulation are verified with experimental data and the two results are in good agreement. The fin efficiency of slotted fin is calculated with numerical method. The Nu number and friction factor correlations are provided and compared with the plain plate fin. Finally the numerical results are analyzed from view of field synergy principle. It is revealed that the essence of heat transfer enhancement is the improvement of the synergy between velocity and temperature field due to the existing of strips and the strips in the downstream part contributes more to heat transfer enhancement than the one in the upstream part of the fin surface.

ACKNOWLEDGMENT

The work was supported by the following funds: (1) The National Natural Science Foundation of China (50476046); (2) The Natural Science Fund for the Distinguished Young Scholars from the National Natural Science Foundation of China (50425620); (3) A Foundation for the Author of National Excellent Doctorial Dissertation of PR China..

NOMEMCLATURES

A	Heat transfer area
c_p	Specific heat at constant pressure
D_e	Outer tube diameter,
f	Friction factor
h	Heat transfer coefficient
L	Fin depth in air flow direction
ΔP	Pressure drop
Q	Heat transfer rate
Re	Reynolds number
T	Temperature
u , v	Velocity in x,y- direction

Greek Symbols

h	Fin efficiency

l Thermal conductivity
m Dynamic viscosity
r Air density
Γ Nominal diffusion coefficient

Subscripts

in inlet
out outlet

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