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NUMERICAL STUDY ON HEAT TRANSFER CHARACTERISTICS IN RECTANGULAR DUCTS WITH PIN-FINS

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ABSTRACT

By using the CFX software, three-dimensional flow and heat transfer characteristics in rectangular cooling ducts with in-line and staggered array pin-fins of gas turbine blade trailing edge were numerically simulated. The effects of in-line and staggered arrays of pin-fins, flow Reynolds number as well as density of cylindrical pin-fins in flow direction on heat transfer characteristics were analyzed. Both in the cases of in-line and staggered arrays of pin-fins, the results show that the pin-fin surface averaged Nusselt number increases with the increasing of Reynolds number. In the case of the same Reynolds number, the mean Nusselt number of pin-fin surface decreased with the increasing of X/D (the ratio of streamwise pin-pitch to pin-fin diameter) value. The Nusselt number increases gradually before the first pin-fin row and then reached the fully developed value at fourth or fifth row. The pin-fin Nusselt number at flow direction is larger than that at back flow direction. Along the height direction of pin-fin, the Nusselt number in middle area is larger.

Key words: blade trailing edge; pin-fin cooling; rectangular ducts; numerical simulation

1. INTRODUCTION

Gas turbine has been widely used in the aviation, power generation, shipbuilding and many other fields. The efficiency of gas turbine can be significantly improved by increasing the inlet gas temperature, but high gas temperature could cause the high temperature corrosion or damage of blades, so the measures must be adopted to protect turbine blades. At present, there are two major measures to be taken: the first is to develop new materials for improving blade's alloy performance and the second is to use effective structure and technology for cooling blades. In the trailing edge region of gas turbine blade, ribs can't be used due to restriction of blade thickness, so that the installation of pin-fins is a good choice for intensive convection in a limited space. Pin-fins are almost cylindrical objects extending from the heat transfer surface of blades to the cooling channel, and perpendicular to the flow direction. Because of the pin-fins, the cooling flow is disturbed and then the heat transfer between the pin-fins and the blade surfaces is enhanced. Multiple pin-fins in way of in-line and staggered arrays are often applied in practice [1-2]. The cooling channel with pin-fin could increase the heat transfer coefficient but also increase the pressure loss. There are several types of pin-fins, such as cylindrical pin-fin used most frequently, square pin-fin and diamond pin-fin. The heat transfer coefficient of cooling channel with square pin-fin is the highest while cylindrical pin-fin is the lowest, but the pressure loss of cooling channel with square or diamond pin-fin is larger than that with cylindrical pin-fins [3].

The cross-section area of the gas turbine blade trailing edge region is gradually reduced and the flow of the cooling channel is accelerated, the flow and heat transfer characteristics in different positions of cooling channel are also different. The heat transfer performance of pin-fins is calculated generally in average of pin-fins row. Arora and Abdel-Messeh[4] investigated the effects of the half-pins on heat transfer characteristics. There is a tip-spacing between the half-pin and top surface of the flow channel. The friction factor of the half-pins is less than that of the full pins. Under the condition of high Reynolds numbers, the local Nusselt number of the half-pins is a little higher than that of the full pins, but the local Nusselt number of the top surface is less than that of the bottom surface with the half-pins. On the whole, the array average heat transfer coefficient decreases linearly with increasing of gap. S.W. Chang et al. [8] experimental investigated the effects of the gap between the pin-tips and the rectangular channel end wall, Reynolds numbers on the heat transfer and pressure loss characteristics. The C/D is 0, 1/4, 1/2 and 3/4 respective and the Reynolds numbers is 10,000, 15,000, 20,000, 25,000 and 30,000 respective. Their conclusions are that the area averaged end wall Nusselt numbers decrease with substantial reductions in channel inlet-to-exit pressure drops by way of increasing C/D ratio from 0 to 3/4. The end wall heat transfer level with detached pin-fins at C/D=1/4 is the highest thermal performance factor among the four comparative cases in the Re range examined in their study. N. K. C. Selvarasu et al. [9] studied the effect of pin-fin density on

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friction and heat transfer at low Reynolds numbers from 10 to 600.

So far, plenty of experimental studies on the heat transfer characteristics of rectangular duct with pin-fins have been conducted, but numerical methods used to study are relatively small. It is necessary to validate the result numerically, and we can get every detail we want from the numerically generated flow domain. The present study was initiated with the objective of investigating the cooling flow and heat transfer characteristics of gas turbine blade trailing edge with the pin-fins. Both the cases of staggered and in-line arrays of pin-fins in the rectangular ducts, the effects of flow Reynolds number and density of cylindrical pin-fins at flow direction on flow and heat transfer characteristics were analyzed.

2. THE COMPUTATIONAL MODEL AND METHOD

2.1 GOVERNING EQUATIONS AND TURBULENT MODEL

In turbulent flows, the local physical variable can be expressed as the sum of time-averaged component and a time-dependent fluctuating component, as:

$$\phi = \phi' + \overline{\phi} \tag{1}$$

The general form of governing equations is given by:

$$\frac{\partial(\rho\phi)}{\partial t} + div(\rho\vec{V}\phi) = div(\Gamma_\phi grad\phi) + S_\phi \tag{2}$$

In which, the expressions of ϕ, Γ_ϕ and S_ϕ are shown in Table 1.

Table1. Parameter expressions

ϕ	Γ_ϕ	S_ϕ
u	$\mu_{eff} = \mu + \mu_t$	$-\frac{\partial p}{\partial z} + \frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial y}\right) + \frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial z}\right)$
v	$\mu_{eff} = \mu + \mu_t$	$-\frac{\partial p}{\partial z} + \frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial y}\right) + \frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial z}\right)$
w	$\mu_{eff} = \mu + \mu_t$	$-\frac{\partial p}{\partial z} + \frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial y}\right) + \frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial z}\right)$
T	$\frac{\mu}{\Pr} + \frac{\mu}{\sigma_t}$	0

When the Reynolds time-averaged equation is used to solve the turbulent flow and heat transfer, an additional equation must be added to make the equations being closed because of the introduced fluctuating variable. In method of Reynolds time-averaged, the introduced new fluctuating variable is $-\overline{\rho u_j'\phi'}$. When ϕ' is u_i', the item represents Reynolds stress.

Boussinesq assumes that the additional stress caused by turbulence is correlated with strain rate as same as laminar stress. The following equation can be derived from the constitutive equation of the stress and strain:

$$-\rho\overline{u_i'u_j'} = \left(\tau_{i,j}\right)_t = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\mu_t\delta_{i,j}div\vec{V} \tag{3}$$

Being different with molecular viscosity, μ_t is not a physical parameter and correlated with the flow state.

Based on the Boussinesq's assumption, the key problem in solving turbulent flow is to ascertain the turbulent eddy viscosity μ_t. The two equations turbulence model is the most widely used for engineering calculations. The governing equations of k and ε are as follows:

$$\rho u_j\frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + \mu_t\frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \rho\varepsilon \tag{4}$$

$$\rho u_k\frac{\partial \varepsilon}{\partial x_k} = \frac{\partial}{\partial x_k}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_k}\right] + \frac{c_1\varepsilon}{k}\mu_t\frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - c_2\rho\frac{\varepsilon^2}{k} \tag{5}$$

Turbulent eddy viscosity μ_t is defined as:

$$\mu_t = c_\mu\rho k^2/\varepsilon \tag{6}$$

In which, the empirical coefficients are shown in Table 2.

Table2. Empirical coefficients value of $k-\varepsilon$ model

σ_T	c_1	c_2	c_μ	σ_k	σ_ε
0.9-1.0	1.44	1.92	0.09	1.0	1.3

In equations (4) and (5), standard $k-\varepsilon$ model is a high Reynolds number turbulence model, and can only be applied to solve turbulent flow in the fully developed region. For the wall area, the wall function method was adopted to solve this problem [6].

The basic idea of the wall function method is that, $k-\varepsilon$ model is applied to solve the flow in the core area of turbulent flow, while the semi-experience relationship equation is employed directly to combine the physical variables on wall area with that in the core area of turbulent flow. The standard wall function is given as follow:

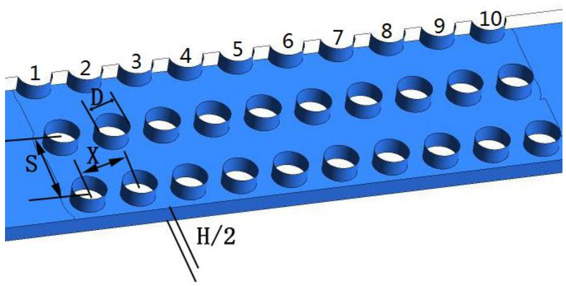
$$\frac{U_P C_\mu^{1/4} k_P^{1/2}}{\tau_w/\rho} = \frac{1}{\kappa}\ln\left(Ey_P\frac{\rho C_\mu^{1/4} k_P^{1/2}}{\mu}\right) \tag{7}$$

Control volume method is applied to discrete governing equations, and the discretization scheme is second-order upwind difference. In addition, SIMPLE algorithm is adopted to solve the discrete algebraic equations.

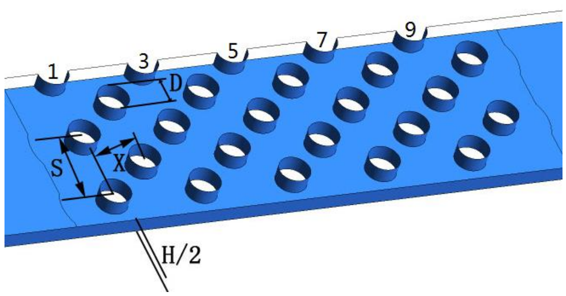
2.2 THE COMPUTATIONAL MODEL AND GRID

There are ten rows of cylindrical pin-fin mounted on the heat transfer surface inside the blade trailing edge of gas turbine, which is simplified as the rectangular cooling channel with cylindrical pin-fins in way of in-line and staggered array. To save computing resources, a quarter of the real model was taken in calculation. In the study, the streamwise pin-pitch is a constant 12.7 mm. S/D and H/D of every channel is 2.5 and 1 respective. The length of inlet and outlet section of each channel is the same as the experimental section, which is decided by X/D varying from 1.5 to 3.5, and the width of each channel is 5 times S. Figure 1 is the numerical computation model.

Figure 2 is the numerical computation grids of the rectangular channel with staggered arrays of cylindrical pin-fin. The O-grid mesh is adopted around the pin-fin and the structured mesh is adopted on the heat transfer surface. Distance from the heat transfer wall to the first layer of the grid is 0.001 mm, the increasing ratio is 1.5 and the y plus is 0.45.



a) in-line arrangement



b) Staggered arrangement

Fig.1 Numerical computation model

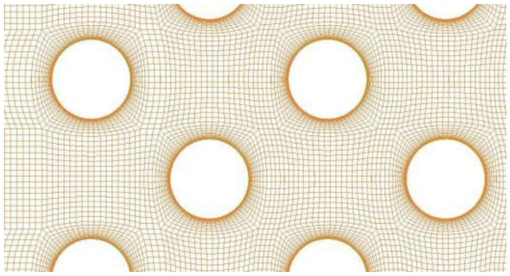
The top and bottom surfaces of test section are no-slip and at a constant temperature of 306K. The other surfaces in the channel are adiabatic. The inlet turbulence intensity is 5% and the temperature of the cooling fluid is 300K, and the velocity is calculated by the maximum Reynolds number. The definition of Reynolds number is as follows:

$$\mathrm{Re} = \frac{\rho V_{\max} D}{\mu} \tag{8}$$

The outlet static pressure is 1 atm. The cooling medium is dry air. The other boundary conditions are presented in the Fig.3. The flow field has reached the convergence requirements if the residual error is less than 1×10^{-5}.

The local heat transfer coefficient and the Nusselt number are expressed respectively by:

$$h = \frac{q}{T_s - T_\infty}, \quad Nu = \frac{hD}{K} \tag{9}$$



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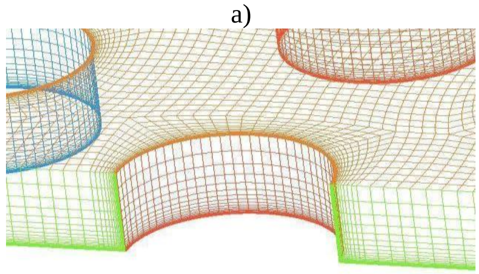


b)

Fig.2 Numerical computation grid

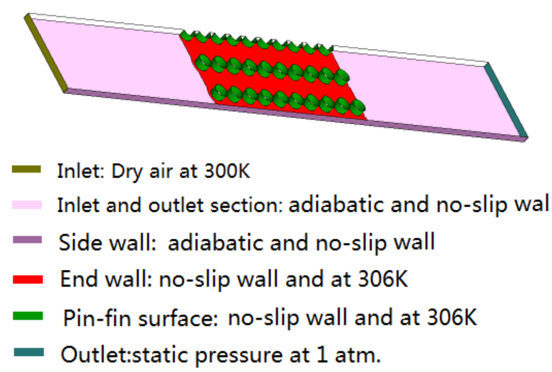


Fig.3 Boundary conditions

2.3 VALIDATION OF CALCULATION MODEL AND METHOD

The calculation model and boundary conditions are the same as the experimental conditions of Metzger[5]. Figure 4 shows the effects of the mesh number on the average Nusselt number of pin-fin surface in staggered arrangement under the condition of X/D=1.5 and Re=3980. The results show that the mesh number has little influence on calculation results when the mesh number is greater than 1.66 million. The mesh number of 1.6 million for in-line arrangement and 1.66 million for staggered arrangement were adopted in the paper.

Fig.5 shows the comparison curves between the experimental data and computational result under the condition of X/D=1.5. It can be seen that the varying trend of the pin-fin row averaged Nusselt number is the same as that of the experimental data. Under both conditions, the computational results are in good agreement with experimental data, meaning that the standard $k-\varepsilon$ turbulence model is applicable to investigate this problem.

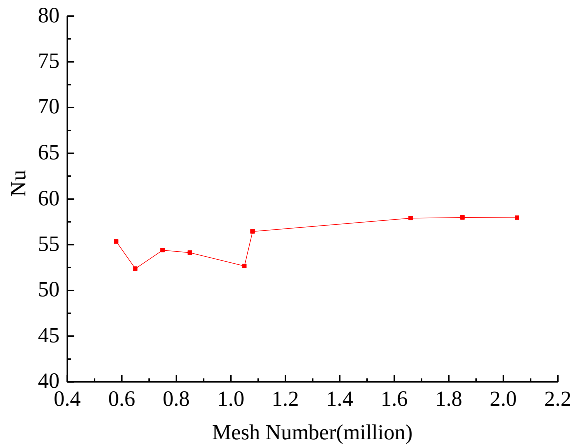


Fig.4 Effect of mesh number on Nu distribution

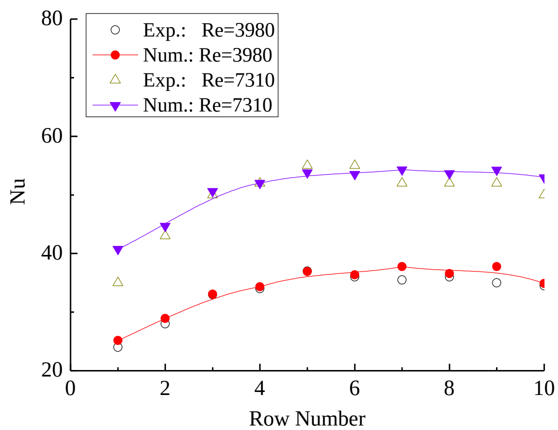


Fig.5 Comparison of computational result with experimental data

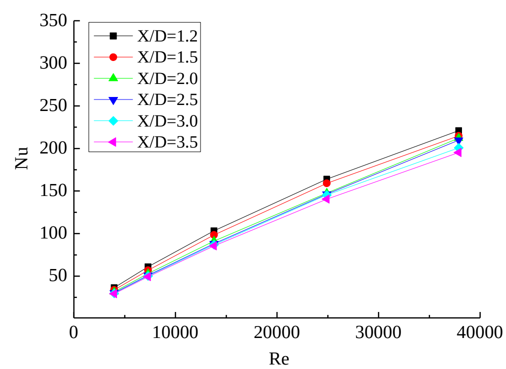
3. CALCULATION RESULTS AND DISCUSSIONS

3.1 THE EFFECT OF REYNOLDS NUMBER ON PIN SURFACE AVERAGED NUSSELT NUMBER

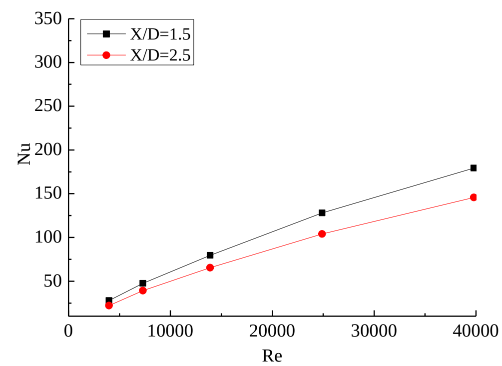
Fig.6 shows the effect of the Reynolds number on the mean Nusselt number of pin-fin surface in both cases of staggered and in-line arrangement. It can be seen that the mean Nusselt number of pin-fin surface increases approximate linearly with the increasing of the Reynolds number in two arrangement forms of pin-fin and different X/D. The reason is that, with the increasing of the Reynolds number, the inlet velocity and the turbulence of working fluid increase, which reduce the thickness of temperature boundary on the bottom and pin-fin surface, and then the heat transfer is obviously enhanced, so the mean Nusselt number of pin-fin surface increases [7].

In the cooling channel with staggered pin-fins and X/D=1.5(as shown in Fig.6a), the mean Nusselt number when the Reynolds number being 37900 increases 6.45 times than that when Reynolds number being 3980, which means that the Reynolds number has larger effect on the heat transfer. When the X/D varies from 3.5 to 1.2, with the increasing of Reynolds number, the varying amplitude of the absolute Nusselt number increases gradually while the percentage decreasing. When the Reynolds number is 3980, the mean Nusselt number of pin-fin surface with X/D=1.2 increases about 35% than that of X/D=3.5. While the Reynolds number is 37900, the mean Nusselt number of pin-fin surface with X/D=1.2 increases about 13% than that of X/D=3.5. It is noticed that the Nusselt number values have little difference in the Re range of 20000 to 35000 for X/D = 2, 2.5 and 3. The reason is that the interaction between the upstream and downstream pin-fin row is greatly reduced when X/D>2. The increment of streamwise pin-pitch when X/D>2 has little influence on heat transfer characteristics.

In the cooling channel with in-line pin-fins (as shown in Fig.6b), the mean Nusselt number of pin-fin surface with X/D=1.5 increases about 26% than that of X/D=2.5 when the Reynolds number is 3980. When the Reynolds number is 37900, the mean Nusselt number of pin-fin surface increases about 23% than that of X/D=2.5.



a) Staggered arrangement

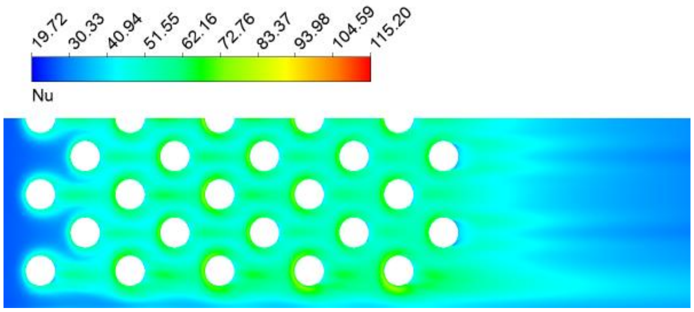


b) In-line arrangement

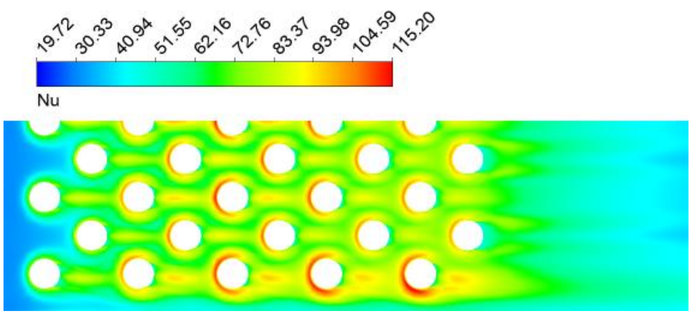
Fig.6 The average Nu number varying with Re

3.2 THE EFFECT OF REYNOLDS NUMBER ON ENDWALL NUSSELT NUMBER

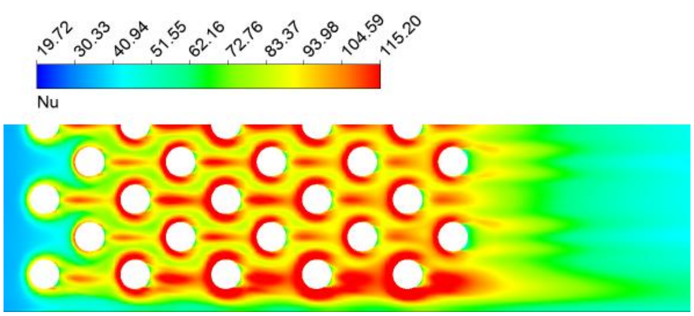
Fig.7 and Fig.8 show the effect of the Reynolds number on the contours of the Nusselt number on the cooling channel bottom surface under the conditions of staggered and in-line arrangements respectively.



a) Re=3980; X/D=1.5



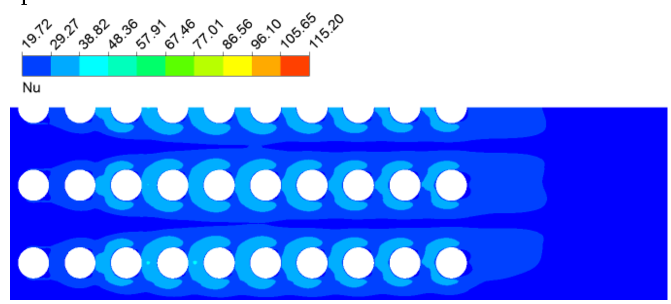
b) Re= 7310; X/D=1.5



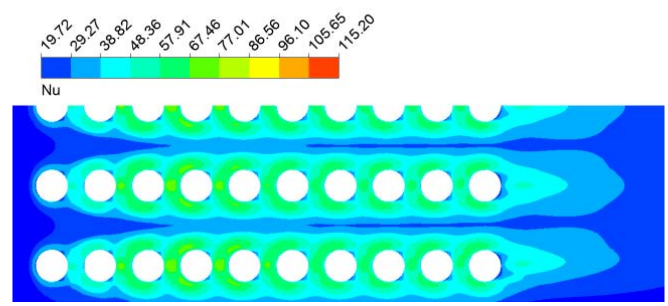
c) Re=13800; X/D=1.5

Fig.7 The contours of Nu on channel bottom surface with staggered pin-fin

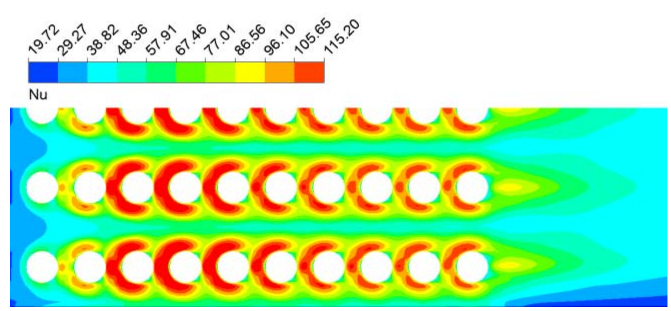
It can be seen that the endwall Nusselt number, which reached the fully developed value at fourth or fifth row, increases gradually before the first pin-fin row. The horse shoe vortex and wake flow, which greatly disturbed the flow, were formed when the fluid passed by the pin-fin and the heat transfer was greatly enhanced in the area. In front of the pin-fin area, the Nu value is higher than that of the back of the pin-fin area because of the interaction between the horse shoe vortex and the wake flow from upstream. When X/D=1.5, the distributions of the Nusselt number on the channel bottom surfaces are basically identical, and the mean Nusselt number on the bottom surface increases with the increasing of the Reynolds number. For example, when the Reynolds number increases from 3980 to 7310 and 13800, the mean Nusselt number of the channel bottom surface with staggered pin-fin increases by 71.7% and 203.6% respectively (as shown in Fig.7a, b and c), and the mean Nusselt number of the channel bottom surface with in-line pin-fin increases by 83% and 232.2% respectively (as shown in Fig.8a, b and c). Compared with the mean Nusselt number of the channel bottom surface with in-line pin-fin, the mean Nusselt number of the channel bottom surface with staggered pin-fin is larger 68.8% when Re=3980, 58.4% when Re=7310 and 54.3% when Re=13800. In addition, the cylindrical pin-fin near the side surface of the channel heat transfer is better. The reason is that the side surface of the channel is solid plane and the fluid can't pass through. The flow in this region differs from the flow among the cylindrical arrangements and the heat transfer is improved.



a) Re=3980; X/D=1.5



b) Re=7310; X/D=1.5



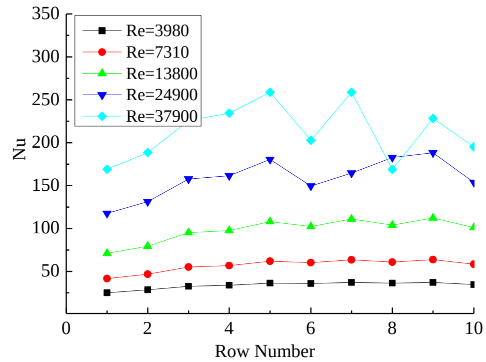
c) Re=13800; X/D=1.5

Fig.8 The contours of Nu on channel bottom surface with in-line pin-fin

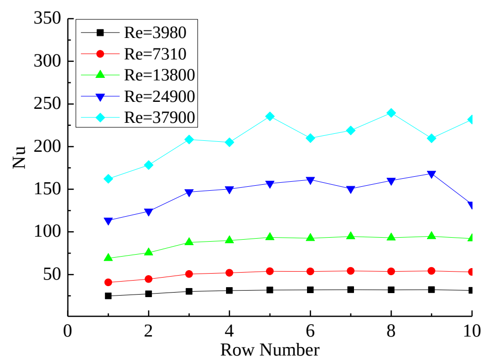
3.3 THE EFFECT OF REYNOLDS NUMBER ON PIN-FINS ROW AVERAGED NUSSELT NUMBER

Fig.9 shows the variation curves of the mean Nusselt number of pin-fins with the Reynolds number in both cases of staggered and in-line arrangements when X/D=1.5 and X/D=2.5. It can be seen that, the varying trends that the mean Nusselt number of pin-fins increases with the Reynolds number are basically identical for different X/D. The Nusselt number near the channel inlet increases gradually, and then reaches the fully developed value. But in the smooth channel, it is the starting point of thermal boundary that reaches the maximum Nusselt number, and then reduces to the fully developed value. In the channel of the cylindrical pin-fin, the gain of the developing region Nusselt number is due to the interaction between upstream and downstream pin-fin. The wake flow produced by each pin-fin increases the turbulence of the free flow, which disturbs the boundary layer of the end wall surface installed with pin-fins, and improves the property of heat transfer. In addition, the pin-fin itself has also extended the area for heat transfer. All of above make the heat transfer been enhanced significantly. The wake flow of the front makes the heat transfer of the rear enhanced [8-11]. After the third row of pin-fin, the flow is self-similar and the Nusselt number has reached the fully developed value. This is consistent with the experimental observations by Metzger[5] and others.

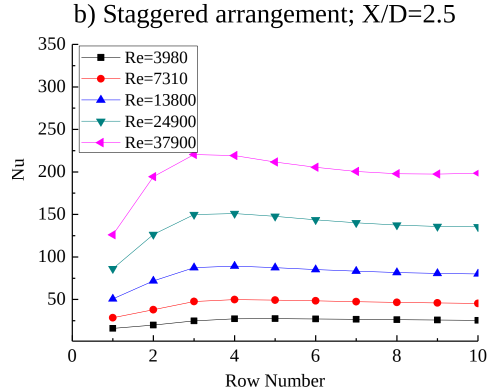
For example of Re=13800, when X/D=1.5, the Nusselt number of staggered pin-fins reaches the maximum in the fifth row, which is larger 50% than that of in first row and 6% than that of in the tenth row; the Nusselt number of in-line pin-fins achieves the maximum in the fourth row, which is larger 77.6% than that of in the first row and larger 10.8% than that of in the tenth row. When X/D=2.5, the Nusselt number of staggered pin-fins reaches the maximum in the fifth row, which is larger 34.9% than that of in the first row and larger 2.5% than that of in the tenth row; the Nusselt number of in-line pin-fins achieves the maximum in the fourth row, which is larger 44% than that of in the first row and larger 11% than that of in the tenth row. In addition, the mean Nusselt number of staggered pin-fins is larger than that of in-line pin-fins. For X/D=1.5, the mean Nusselt number of staggered pin-fins is larger 40.6% than that of in-line pin-fins in the first row, 22.8% in the row where the Nusselt number achieves the maximum and 28.6% in the last row. For X/D=2.5, the mean Nusselt number of staggered pin-fins is 3.1 times as large as that of in-line pin-fins in the first row, 3 times in the row where the Nusselt number reaches the maximum and 3.2 times in the last row. Relative to in-line pin-fins, the staggered pin-fin enlarges the disturbance of the fluid and then increases turbulence of the flow, which leads to the result discussed above. It also can be seen that, when the Reynolds number is larger than 13800, the added value of the Nusselt number from the first row to the row of the maximum Nusselt number increases substantially with the increasing of the Reynolds number. Therefore, the effect of the Reynolds number on the heat transfer is comparatively significant.



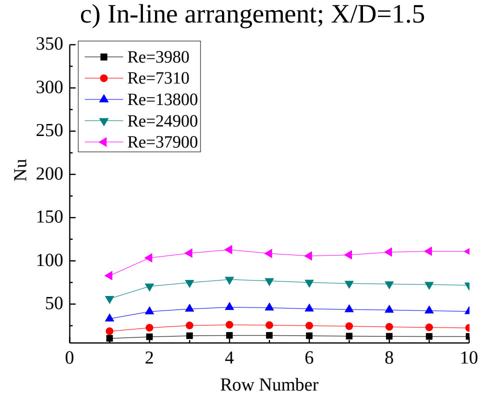
a) Staggered arrangement; X/D=1.5



b) Staggered arrangement; X/D=2.5



c) In-line arrangement; X/D=1.5



d) In-line arrangement; X/D=2.5

Fig.9 The average Nu number of pin-fin varying with Re

It is noticed that when X/D = 1.5, Re = 37900, row 5 and 7 has peak and valley at 6. Again when X/D = 2.5 has peaks and valley of different kind for same Reynolds number in Fig.9 (a) and Fig.9 (b). But no such things found for lower Reynolds number. The reason is that the fluid domain is very complicated when X/D is relatively small at higher Reynolds number. At the condition of X/D=1.5 and Re>24900, the fluctuations of the first five pin-fin rows averaged Nusselt number are smaller than that of row 6 and the later rows. Because the influences of heat transfer between adjacent rows are highly reduced by the large temperature difference between the fluid and the pin-fin surfaces. After row 5, the fluid temperature has reached the maximum value, and the influences between adjacent rows are significant. Fig.9 (a) and Fig.9 (b) also present that the fluctuation value is decreasing with the increasing of X/D value. For example, the mean Nusselt number at row 5 and row 7 is 27.52% and 27.56% higher than that of row 6 respective when X/D=1.5 and Re=37900, while the mean Nusselt number at row 5 and row 7 is 12.10% and 4.34% higher than that of row 6 respective when X/D=2.5 and Re=37900.

3.4 EFFECT OF STREAMWISE PIN DENSITY ON PIN SURFACE AVERAGED NUSSELT NUMBER

In cooling channel with staggered pin-fins, the effect of X/D on the mean Nu number of pin-fin surface is shown in Fig.10. Under the condition of the same Reynolds number, with increasing of X/D, the mean Nusselt number of pin fin surface slightly decreases. As the Re number increases, the reduction of the Nu number become more distinct. With the decline of X/D the Nu number increases. When Re number is 24900, the value of pin-fin surface averaged Nu number at X/D=1.5 is about 11.1 higher than that at X/D=2.5. When Re number is 3980, it is about 2.3 higher than that at X/D=2.5.

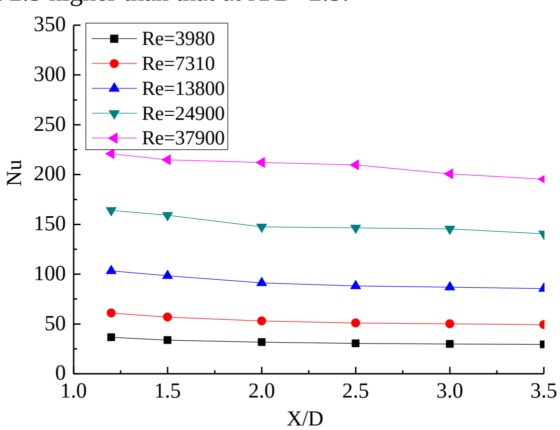


Fig.10 Variation of mean Nu of pin-fin surface with X/D

Fig.11 shows the effect of X/D on mean Nu number of staggered pin-fin array. It can be seen that with the decreasing of X/D the Nu number increases. When X/D increases over 2.5, the decreasing of Nu number becomes smaller. The reason is that when X/D increases over 2.5, the wake flow disturbance of the front pin-fin row on the back row is weaken, and the heat transfer of every pin-fin rows is approximate, so the curves are much approximate at high X/D value.

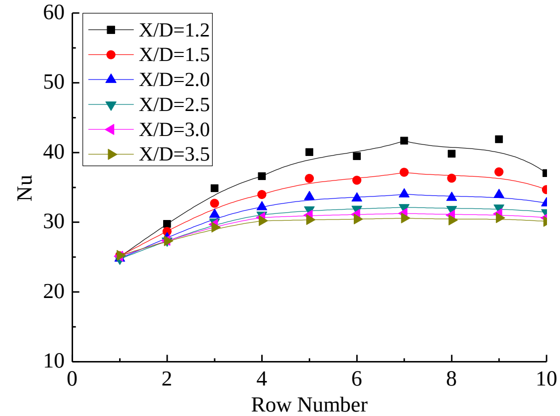


Fig.11 Effect of X/D on mean Nu number of pin-fin row

3.5 EFFECT OF STREAMWISE PIN DENSITY ON ENDWALL NUSSELT NUMBER

Figure 12 shows Nu number contours of the bottom surface in the flow channel with pin-fins in the cases of Re=3980 and different arrangement density at the flow direction. It can be seen that when X/D is smaller, the Nu number of the bottom surface is larger. The mean Nu number of the channel bottom surface with in-line arrangement of X/D=1.5 is twice as large as that of X/D=2.5; the mean Nu number of the channel bottom surface with staggered pin-fins arrangement of X/D=1.5 is 0.6 times larger than that of X/D=2.5. The mean Nu number of the channel bottom surface with staggered pin-fins arrangement is 1.7 times as large as that of in-line arrangement when X/D=1.5 and 1.4 times when X/D=2.5. As X/D=1.5 is more closely arranged than that of X/D=2.5, the channel cross-section area reduces and the fluid velocity increases in a certain extent. This would lead to the boundary layer on the cylindrical pin-fin and bottom surface becoming thinner or even separation, so the effect of heat transfer is enhanced. If X/D is larger, the heat transfer enhanced region caused by the different cylinder is no longer overlap. There is a significant heat transfer enhanced area around the pin-fins, and heat transfer in the back of pin-fin is relatively weak. This is because there is a significant back flow area back in the cylinder, and heat transfer in the region is weaker than in the other regions. It can be also seen clearly in Fig.12 that, Nu number increases gradually near the entrance and then reaches the fully developed value.

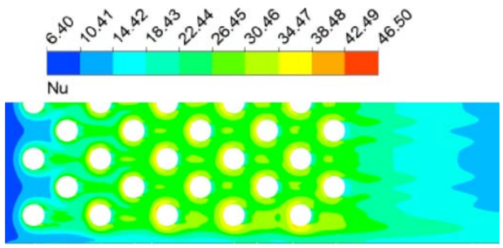
3.6 DISTRIBUTION OF NUSSELT NUMBER AROUND PIN-FINS

Figure 13 , in which arrowhead indicates the flow direction of cooling medium, shows the Nusselt number contours around staggered arrangement pin-fins when Reynolds number is 7310 and X/D is 1.5. The Nu number of pin-fins toward flow side is larger than that of pin fins at back flow direction. Along the height direction of pin-fins, the Nu number in middle area of pin-fins is larger. At the front edge of intersecting line between the cylinder pin-fins and the bottom surface, as well as the side surfaces of pin-fins, the Nu number is also larger. The Nu number reaches the fully developed value after the third row.

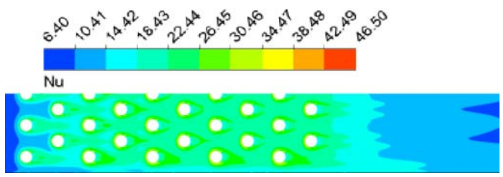
3.7 FLOW ANALYSIS

Fig.14 shows the distribution of velocity vector around pin-fin. Apparently, there is a significant back flow area back in the cylinder, and heat transfer in the region is weaker than in the other regions.

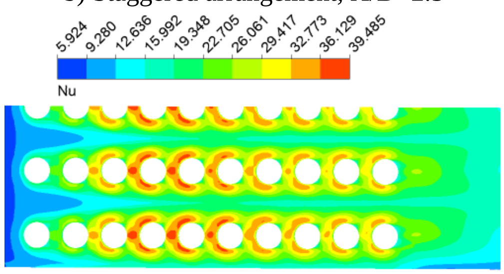
Fig.15 shows the streamlines of the test section with staggered pin-fins at X/D=1.5 and Re=3980. It can be seen that there is a stagnation region front in every pin-fin and a back flow area back in every cylinder. This is the reason that heat transfer of the back area of pin-fin is weaker than that of the front area of pin-fin.



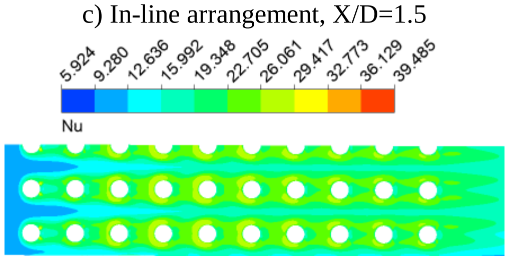
a) Staggered arrangement, X/D=1.5



b) Staggered arrangement, X/D=2.5

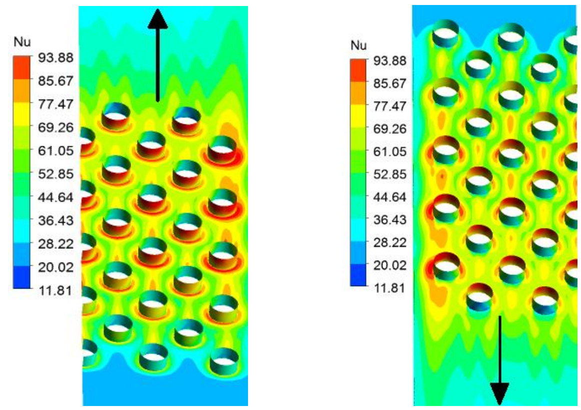


c) In-line arrangement, X/D=1.5

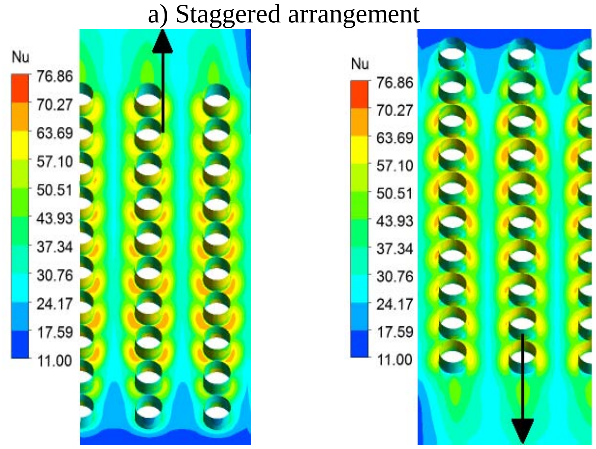


d) In-line arrangement, X/D=2.5

Fig.12 Nu number contours at same Reynolds number and different array density



a) Staggered arrangement



b) In-line arrangement

Fig.13 Distribution of Nu number around pin-fins

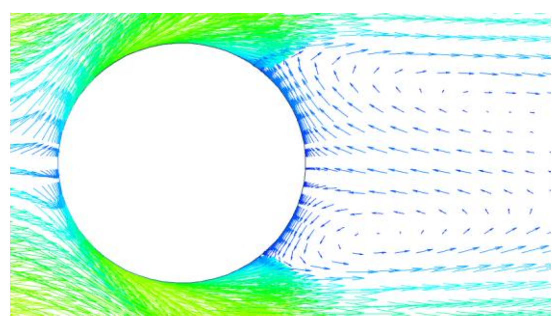


Fig.14 Distribution of velocity vector around pin-fin

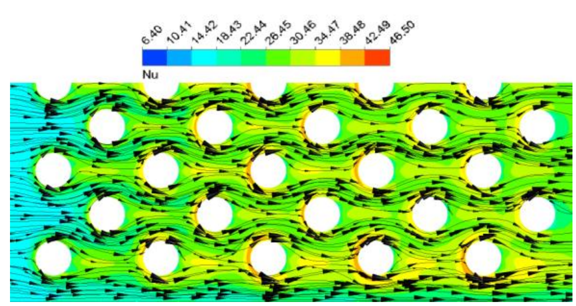


Fig.15 Streamlines of the test section

4. CONCLUSIONS

The heat transfer characteristics in the cooling channel of gas turbine blade trailing edge was simulated numerically. The effects of X/D and Reynolds number on the Nu number were analyzed. The conclusions are as follows:

1) The average Nusselt number of pin-fin surface increases linearly with the increasing of the Re number. At the same Reynolds number, the average Nusselt number of pin-fin surface decreases with the increasing of X/D.

2) The average Nu number of pin-fin row increases with the increasing of Reynolds number. The Nu number increases gradually near the entrance and then reached the fully developed value at fourth or fifth row. The average Nu number of staggered arrangement is larger than that of in-line arrangement.

3) The Nu number of pin fins at flow direction is larger than that of pin fins at back flow direction. Along the height direction of pin fin the Nu number in middle area is larger. At the front edge of intersecting line between the cylinder pin-fins and the bottom surface of cooling channel as well as the side surfaces of pin-fins, the Nu number are also larger.

ACKNOWLEDGMENTS

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NOMENCLATURE

C clearance between pin-tip and end wall
D pin-fin diameter
E empirical constant (9.81)
h local heat transfer coefficient;
H pin height
K conductivity coefficient of the medium;
K_p turbulent kinetic energy of node P
Nu local Nusselt number
Pr Prandtl number
q local heat flux density;
Re Reynolds number
S span-wise pin-pitch
S_ϕ source term;
T_s temperature of the heat transfer surface;
T_∞ temperature of the cooling medium;
u_i' turbulent fluctuating velocity component;
u_j' turbulent fluctuating velocity component;
U_p time-averaged rate of node P
$\vec{V}$ time-averaged velocity
$V_{\max}$ maximum velocity of the fluid;
X streamwise pin-pitch
y_p distance from node P to the wall;
ϕ general variable, such as u, v, w and T
ϕ' time-dependent fluctuating component;
$\bar{\phi}$ time-averaged component
μ fluid dynamic viscosity;
μ_t turbulent eddy viscosity
μ_t turbulent eddy viscosity
ρ density of the fluid;
τ_w wall shear stress
σ_T c_1 c_2 c_μ σ_k σ_ε empirical coefficients value;
Γ_ϕ diffusion coefficient;

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