Three Dimensional Numerical Investigation on the Effect of Wedge Angle of a Passage with Pin-fin Arrays

Changming LING **Chunhua MIN**
Thermophysics Research Center, Zhanjiang Ocean University, Zhanjiang, Guangdong 524025, China
Yixian XIAO
China Aviation Power Plant Research Institute, Zhuzhou, Hunan 412002, China

Heat transfer in passage with pin-fin arrays for cooling blade trailing edge was studied numerically. Three-dimensional numerical simulations were carried out for steady laminar flow in passages with different wedge angles between pressure surface and suction surface of cooling blade trailing edge to study the effect of different wedge angles (from 0° to 30°) on heat transfer and pressure losses. Research was carried out for both in-line array and staggered array. From this investigation, wedge angle 10° gives the best heat transfer performance.

Keywords: trailing edge of turbine blade, pin-fin array, numerical heat transfer, wedge angle of passage.
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Introduction

It is well known from the thermodynamic theory that the performance of a gas turbine engine is strongly influenced by the temperature at the inlet of turbine. Thus to use higher turbine inlet temperature is a growing tendency to meet the development of modern gas turbine engines. To maintain acceptable life and safety standards, cooling techniques must be used. Pin-fin arrays are often used as heat transfer augmentation devices for the cooling of turbine blade for internal cooling passage near the blade trailing edge. Heat transfer and flow in passage with pin-fin arrays are related to type of array, the geometrical dimension, the distance between pin-fins, the Reynolds number and the angle between the pressure surface and suction surface of blade. Much attention has been paid to studies of heat transfer in the aspect mentioned above except the effect of angle between the pressure surface and suction surface of blade. For example, Jacob[1] studied the heat transfer in passage with high value of height-to-diameter ratio of pin-fin, and more studies related to lower H/D of pin-fin, such as Van Fossen[2] studied the heat transfer in passage with staggered pin-fin arrays for H/D=0.5 and 2 , Metzger[3] (1982) studied the effect of the distance between pin-fins on heat transfer while H/D is low. There were also many studies for the problems of pin-fins using numerical method, such as Tzong-Shyan Wung and Ching Jen Chen[4,5], for low Reynolds number while the pin-fin arrays were staggered and in line respectively.

Heat transfer in passage with pin-fin arrays was usually treated as heat convection between two horizontal plates with pin-fin arrays in many papers. It cannot describe the real phenomena. Many researchers like to make this kind of simple model in their studies when they met engineering items. What is the difference? Can we use this simplification instead of the passage with wedge angle? Report in this field has not been found. In this paper, the heat transfer and fluid flow in passage with pin-fin arrays between two parallel plates and between two plates not parallel (i.e. the wedge passage) have been investigated numerically, both staggered pin-fin arrays and in-line pin-fin arrays. The three-dimensional numerical simulations have been completed under the isotherm boundary conditions for the pressure surface and suction surface of blade.

Physical Model and Governing Equations

A steady incompressible Newtonian fluid flow in passage with pin-fin arrays in trailing edge of cooling blade has been considered. A schematic diagram of

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staggered pin-fin arrays and its cross section view are shown in Fig.1, in which θ is wedge angle of the passage. Three dimensional model for numerical simulation has been devised as shown in Fig.2, and upper surface of computational region is also a symmetric surface. The computational conditions shown in Fig.1 and Fig.2 are $d=16$ mm, $s_1=2d$, $s_2=1.75d$, $L=168$ mm, $W=16$ mm and $b=3.2$ mm. The assumptions used for heat transfer simulation are as follows:

1. The flow is laminar and the buoyancy effect is neglected for air.

2. There is no slip happened.

3. The boundary conditions for outer surface of the upper wall and lower wall are of isotherm boundary approximation to make the research more typical.

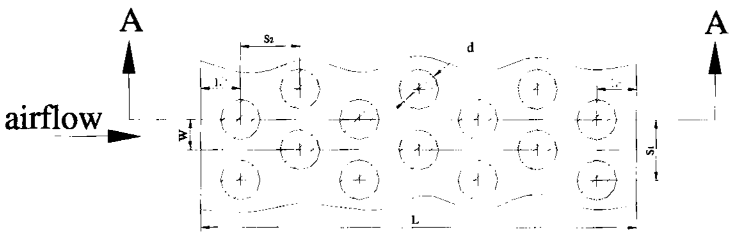
4. The mass flow is assumed the same for all the situations with different passage angle, the inlet temperature is assumed, and the outlet condition is dealt with one-way coordinate.

5. To make the research comparable with each other, same cross sectional area of the central place between inlet and outlet is kept for the passages with different angles.

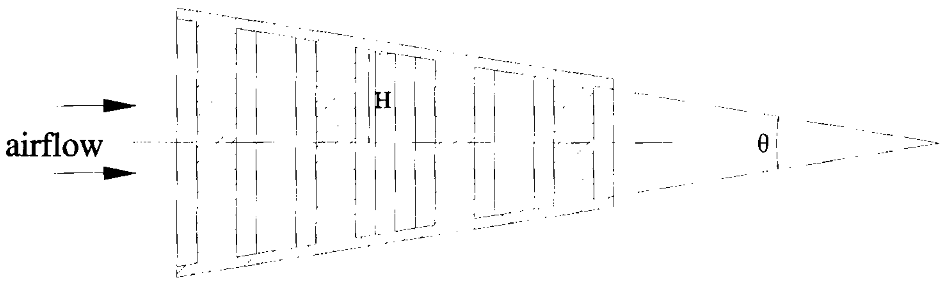
With these assumptions the governing equations can be expressed as:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

$$-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)=0 \tag{2}$$

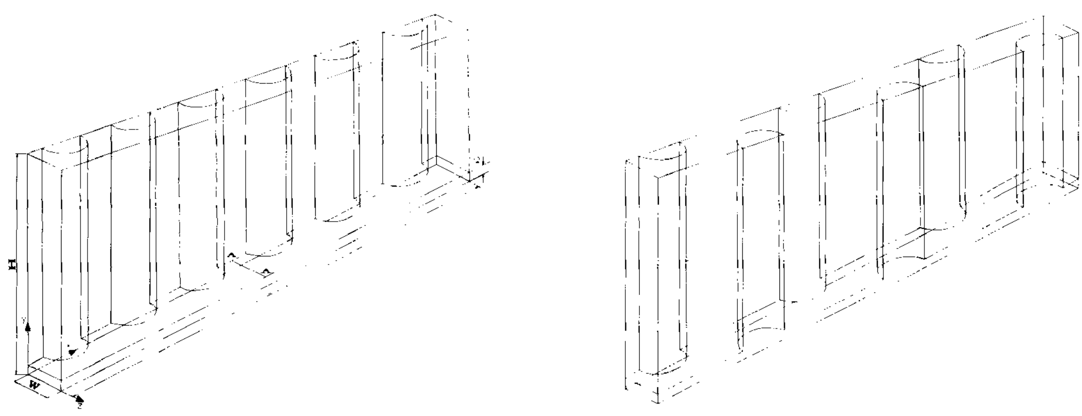


(a) Planform



(b) A-A cross section

Fig.1 Schematic diagrams of construction



(a) In-line array (b) Staggered array

Fig.2 Schematic diagrams of computational region in the passage

$$-\frac{\partial p}{\partial y}+\mu(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2})=0 \tag{3}$$

$$-\frac{\partial p}{\partial z}+\mu(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2})=0 \tag{4}$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\frac{k}{\rho c_p}(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}) \tag{5}$$

where ρ is density (kg / m^3), u,v,w are velocities in direction of x,y and z respectively (m / s), p is pressure (Pa), μ is viscosity of air (kg / (m • s)), k is thermal conductivity for flow (W / (m • K)), T is temperature (K).

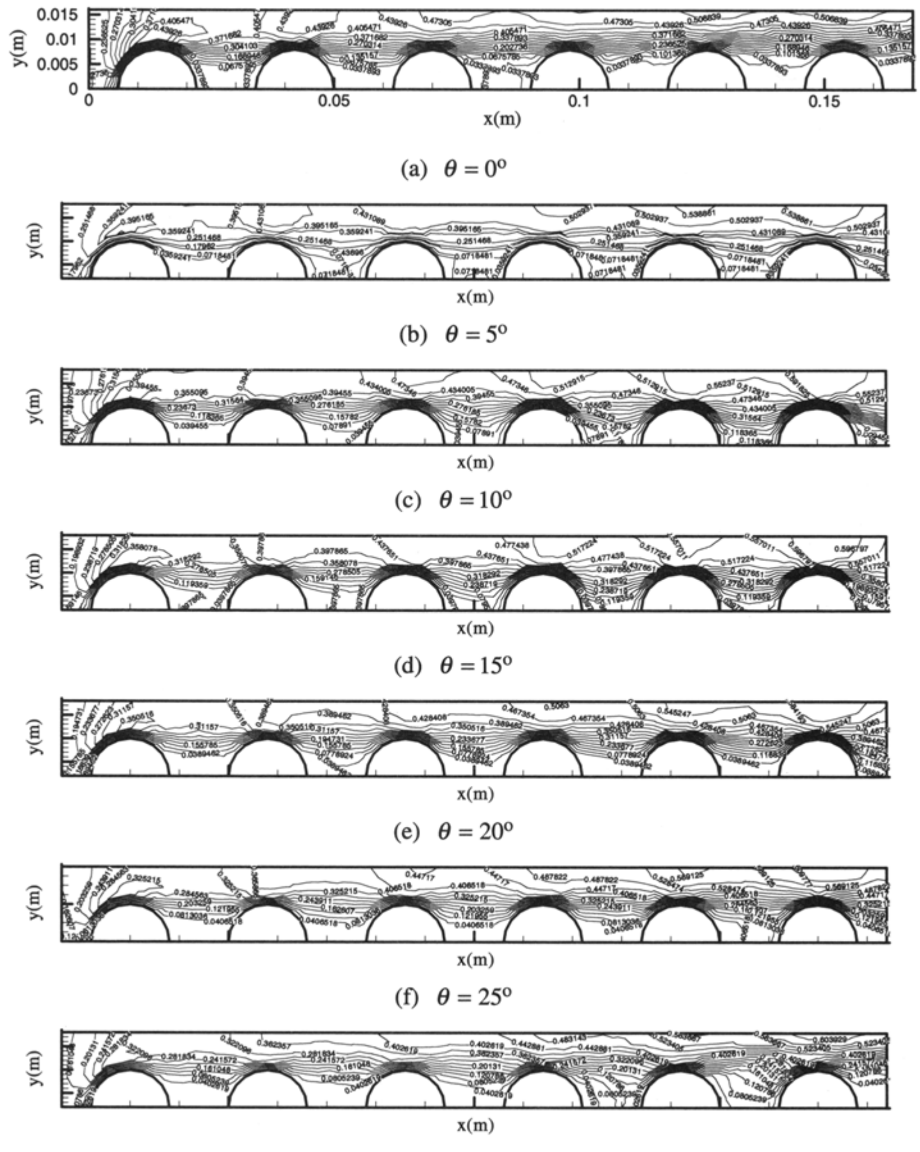
The effect of dissipation of viscosity is neglected for the energy equation.

Method of Solution

The numerical method is based on a finite difference approximation of the steady incompressible Navier-Stokes equations. The SIMPLE algorithm is adopted to deal with the connection between pressure and velocity. A finite volume approach is used to discretize the governing equations. A staggered grid is employed, whereby the scalar variables such as pressure and temperature are located at the cell center and velocity components are located at the cell boundaries.

Results and Discussions

The computational results presented as follows are



(a) $\theta=0°$

(b) $\theta=5°$

(c) $\theta=10°$

(d) $\theta=15°$

(e) $\theta=20°$

(f) $\theta=25°$

(g) $\theta=30°$

Fig.3 Velocity distribution for passage with in-line pin-fin array for a typical case of

$$T_w = 400 \text{ K}, \ G = 0.00025 \text{ kg/s}, \ T_\infty = 300 \text{ K}$$

$$Re = \frac{ud}{\nu} = \frac{Gd}{\rho HW\nu} = 224$$

Where, d is the diameter of pin-fin, W is the width of the part of the passage for computation and H is the mean height of passage.

Fluid velocity distribution

Fig.3 shows the velocity distribution at the plane of symmetry between upper wall and lower wall for wedge angles of $0°$, $5°$, $10°$, $15°$, $20°$, $25°$, $30°$ respectively with in-line pin-fin array.

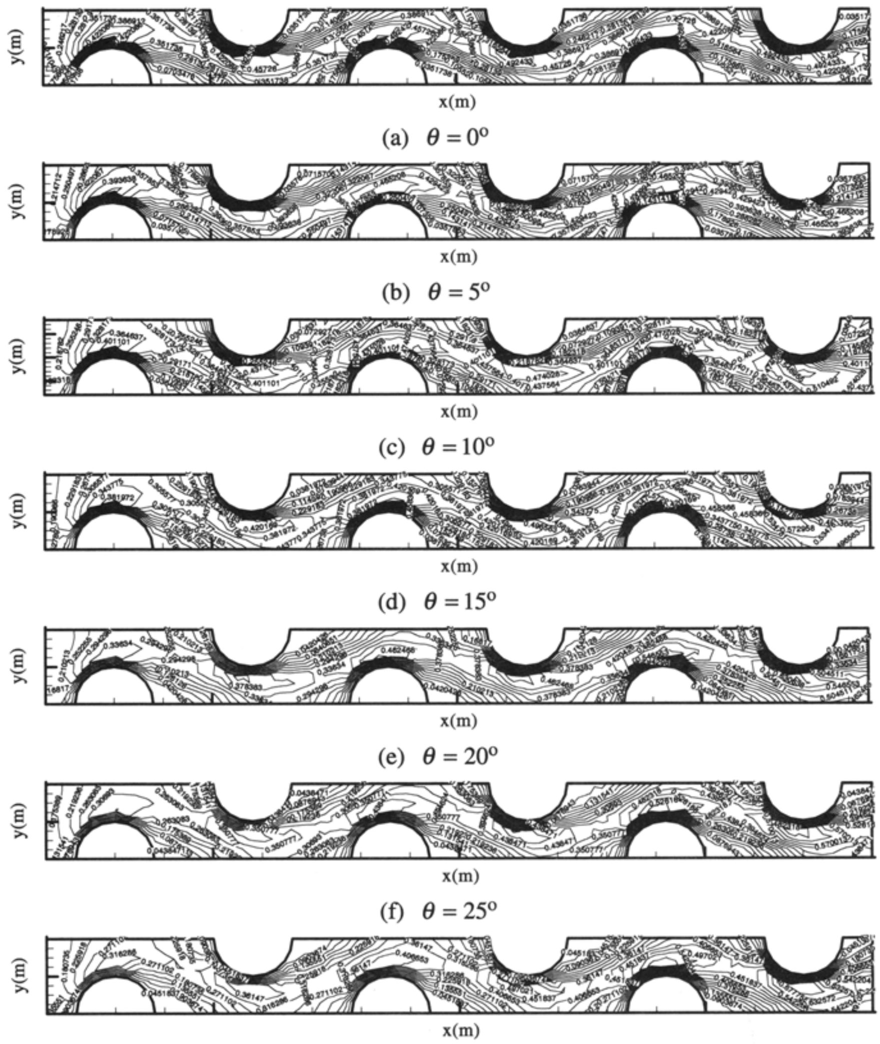
Fig.4 shows the velocity distribution at the plane of symmetry between upper wall and lower wall for wedge angle of $0°$, $5°$, $10°$, $15°$, $20°$, $25°$, $30°$ respectively with staggered pin-fin array.

Characteristics of pressure drop for pin-fin passage

Characteristics of pressure drop for the pin-fin passage in Fig.5 shows that resistance in passage of staggered array is obviously higher than that of in-line array; resistance rises while the wedge angle of the passage increases.

Average Nusselt number for different angles

Attention is now turned to the effects of different passage angles on average Nusselt number of the whole passage. Nusselt number for the passage with angle Θ is expressed as Nu_Θ, especially, Nu_0 is for parallel plate passage Nu_s and Nu_c represent respectively the Nusselt number for the passage with in-line pin-fin array and that for the passage with staggered pin-fin array.



(a) $\theta = 0°$

(b) $\theta = 5°$

(c) $\theta = 10°$

(d) $\theta = 15°$

(e) $\theta = 20°$

(f) $\theta = 25°$

(g) $\theta = 30°$

Fig.4 Velocity distribution for passage with staggered pin-fin array

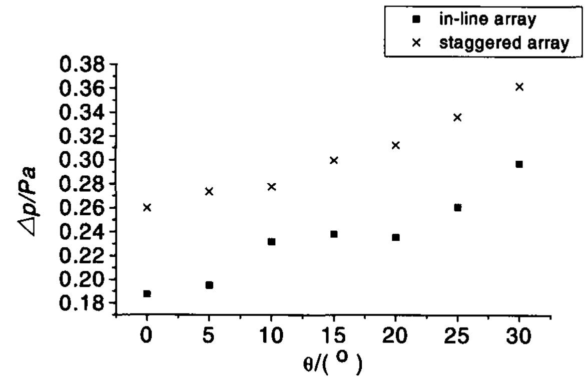


Fig.5 Pressure drop vs. angle of the passage

Fig.6 shows that for both in-line array and staggered array Nusselt number reaches the maximum while the angle is 10°. For in-line array, Nusselt number of the parallel plate passage is minimum, and the Nusselt number of the passage with 10° angle is 1.128 times of that of parallel plate passage. However, for staggered array, the Nusselt number of the passage with 10° angle is 1.012 times of that with 5° angle, at this angle Nusselt number reaches the minimum. This indicate that the variation of the angle affects much more for in-line array, and the effect is less for staggered array.

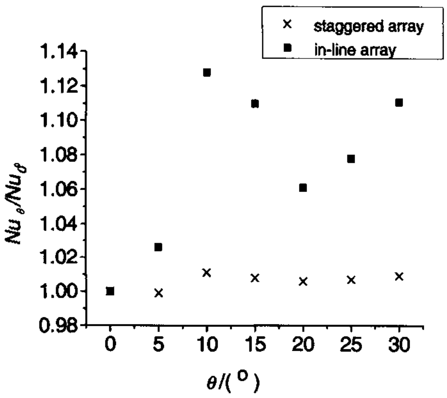


Fig.6 Effects of passage angles to average Nusselt number

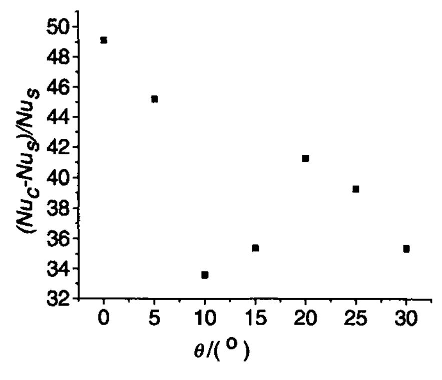


Fig.7 Comparison between the two kinds of arrays

It is obvious from Fig.7 that Nusselt number for staggered array is higher than that for in-line array. For example, the Nusselt number for staggered array is 49. 1% higher than that for in-line array for parallel plate passage (zero degree wedge angle), the Nusselt number for staggered array is 33. 6% higher than that for in-line array while the passage angle is 10°. So, the heat transfer advantage of staggered array compared with in-line array is most obvious while the passage is parallel plate.

Conclusions

1. Heat transfer of passage with staggered pin-fin array is stronger than that with in-line pin-fin array, variation of passage angle affects much more to heat transfer for in-line array, but for staggered array this kind of effect is less; heat transfer reaches the maximum while the angle is 10° for both in-line array and staggered array.

2. Resistance of passage for staggered array is obviously higher than that for in-line array; resistance rises while the angle of the passage increases.

3. Starting from the fifth array of pin-fins for both in-line array and staggered array, heat transfer reaches the periodic steady state. For the conditions mentioned in this paper, 10° angle of passage is the best choice for heat transfer performance even though the resistance rising slightly. The resistance rises more quickly while the angle of passage is larger than 10°.

Acknowledgment

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References

[1] Jacob, M. Heat Transfer and Flow Resistance in Crossflow of Gases over Tube Banks. Trans. ASME, 1938, 60: 384—386

[2] Van Fossen, G J. Heat Transfer Coefficients of Staggered Arrays of Short Pin Fins. ASME Paper No. 81-GT-75, 1981

[3] Metzger, D E, Berry, R A, Bronson, J.P. Developing Heat Transfer in Rectangular Ducts With Staggered Arrays of Short Pin Fins. Trans. of ASME, 1982, 104: 700—706

[4] Tzong-Shyan Wung, Ching Jen Chen. Finite Analytic Solution of Convective Heat Transfer for Tube Arrays in Crossflow: Part 1 — Flow Field Analysis. J. of Heat Transfer, 1989, 111: 636—640

[5] Tzong-Shyan Wung, Ching Jen Chen. Finite Analytic Solution of Convective Heat Transfer for Tube Arrays in Crossflow: Part 2 — Heat Transfer Analysis. J. of Heat Transfer, 1989, 111: 641—648