Effect of inclined angle of pin arrays on flow and heat transfer characteristics in flow channel

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Abstract. The aim of this research is to study effect of inclined angle of pin arrangement on flow and heat transfer characteristics in a rectangular flow channel with array of pins. In this study, the cylindrical of pins which having diameter of D=10 mm was mounted in the flow channel with staggered arrangement. The pin-to-pin distance was fixed at Sy=2D in spanwise direction and at Sx=2.5 in streamwise direction. The effect of pin inclined angle was investigated at θ = 60°, 90°, 120°, and 135° for constant Reynolds number at Re=32,860. The flow and heat transfer were investigated using ANSYS ver.15.0 (Fluent) to simulate 3-D steady turbulence flow. The realizable k-ε turbulence model was applied in this study. The results show that the pin inclined angle with θ = 120° and 135° can enhance the heat transfer on the upstream and downstream of the pins when compared to the case of pin angle θ = 90°. While, the pin inclined angle with θ = 60° gives the lowest heat transfer rate overall surface. The mechanism of heat transfer enhancement for inclined pin can be explained with increasing velocity and reduced wake flow region behind the pin.

1. Introduction
The pin array is normally used for increasing heat transfer on surface in engineering applications, such the cooling of gas turbine blades and electronic devices and enhancing heat transfer in compact heat exchangers. The heat transfer enhancement for pin array can be divided in two factors. The first factor is increasing of heat transfer surface of pin. The second factor is increasing of heat transfer coefficient on surface due to turbulence promotion near endwall of pins [2]. The heat transfer enhancement for pin arrays with long height is dominated by the enhancement of pin surface area while the endwall effect is not significant [3]. However, the mechanism of heat transfer enhancement of short pins due to thermal boundary disturbance on endwall is larger than addition of heat transfer surface on pin surface. Pins are commonly cylindrical shape protruding from the heat transfer surface. The heat transfers fluid is forced to flow pass the array of pins for cooling or heating. Figure 1 shows the flow pattern near the endwall of the pin. The horseshoe vortex that promotes the turbulent was generated on endwall in front of the pin. A Pair of wake flow and separate flow is observed behind the pin. The separate flow will disturb/destroy the boundary layer and will promote the turbulent flow behind the pin. This will affect to the heat transfer enhancement in region in front of the pin and behind the pin.
Chyu [4] studied the effect of pin shape on heat transfer enhancement focusing only for case of mounting pin normal to the heat transfer surface. Inline and staggered arrays with and without base fillet are studied. The heat transfer performance of straight cylinder (without fillet cylinder) with staggered arrays was higher than that of a fillet-cylinder. Pandit et al. [5] demonstrated three-types of pin fin arrays with staggered arrangement of circular, triangular, hexagonal and diamond shapes on increasing heat transfer in flow channel. The result shows that heat transfer from the diamond pin fin is higher than that all shape. Rao et al. [6] studied on heat transfer characteristics of pin arrays attached on surface with dimples in a channel flow. The pin arrays with dimple surface can enhance heat transfer performance by up to 19% when compared to pin arrays on surface with no dimples. Lawson et al. [7] studied the effects of spanwise and streamwise spacing on heat transfer with mounting pin arrays for a range of Reynolds number between 5,000 and 30,000. As a result, the reduction of spanwise and streamwise pin spacing increases heat transfer enhancement due to flow acceleration. The heat transfer enhancement with low streamwise pin spacing was highly significant than low spanwise pin spacing. Choi et al. [8] studied the influence of inclined short single cylinder on heat transfer surface and flow field around a cylinder. The heat transfer on the upper endwall is enhanced due to the interaction between the horseshoe vortex and the jet-like flow. However, the heat transfer on the lower endwall is reduced due to the large wake region behind the inclined cylinder. However, the effect of angle of inclined for short cylinders on flow and heat transfer characteristic is still not clearly understood for inclined pin arrays.

The objective of this study is investigating the flow and heat transfer characteristics of inclined pin arrays on heat transfer surface with simulation. The effects of pin inclined angle to heat transfer surface were investigated for case of θ = 60°, 90°, 120°, and 135° at constant Reynolds number of airflow at Re=32,860 for comparison.

2. Numerical Method

2.1. Geometric Model

Figure 2 shows computation domains and mesh arrangement for prediction of inclined pin arrays in flow channel. The computational domains with a width of 40 mm and a height of 32mm were divided into three parts; upstream of test section, test section, and downstream of test section. First part of domain (upstream of test section) with a length of 1,480 mm (25.6Dh) had a sufficient length to provide a fully developed flow through the test section. Second part of domain (test section) with a length of 370 mm (6.4Dh) was mounted inclined pin arrays and third part of domain (downstream of test section) with a length of 400 mm (6.9Dh) was connected to flow outlet. The hydraulic diameter of Dh = 57.8 mm
is determined based on the rectangular of the wind tunnel. The cylinder pins which having diameter of $D = 10$ mm were mounted with staggered arrangement in flow channel. The pin-to-pin distance was fixed at $S_y=2D$ in spanwise direction and at $S_x=2.5$ in streamwise direction as shown in Fig. 3(a). The size of heat transfer surface was with length of $32D$ and width of $4D$. In the simulation, the effects of inclination angle between pin centreline to mounted surface were investigated in the range of $\theta = 60^\circ$, $90^\circ$, $120^\circ$, and $135^\circ$. The Reynolds number of internal flow was fixed at $Re = 32,860$. For all computations, the $y^+$ values on heat transfer surface are approximately 1.0 for all case investigated.

**Figure 2.** Computational domain and grids

**Figure 3.** Configuration of inclined Pins arrangement and cross-section of grids
2.2. Solution methodology
For the boundary conditions, the inlet was identified as uniform velocity profile and outlet was specified as outflow boundary condition (zero gradient in flow direction). The two side of computational domain in spanwise direction were identified as symmetry boundary condition. The top and bottom of computation domain were set wall boundary condition. A non-slip boundary condition was applied over the wall surface. The turbulence intensity was specified as 5% at the inlet section, and a uniform heat flux was fixed on the top surface.

The Numerical simulation was used to solve Reynolds Averaged continuity equation, Navier-Stokes equations and energy equation for obtaining three-dimensional flow field and heat transfer on the surface. The model of fluid flow and heat transfer were analyzed using the realizable $k - \varepsilon$ turbulence model. The solution method was based on Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) algorithm with second order upwind for all spatial discretization. The root mean square (RMS) residual of continuity and energy equation is less than $10^{-5}$ and that of momentum equation is less than $10^{-4}$, the computed domain is considered convergent.

2.3. Data reduction
The Reynolds number was calculated by maximum velocity ($V_C$) at centre of channel and the equivalent hydraulic diameter ($D_h$) at the entrance, defined as:

$$Re = \frac{V_C D_h}{v}$$

(1)

The local heat transfer coefficient of the test channel is defined by:

$$h = \frac{\dot{Q}_{net}}{(T_w - T_m)}$$

(2)

This $\dot{Q}_{net}$ is the net heat flux on heated surface. $T_w$ is local temperature on heat transfer surface. $T_m$ is mean temperature of the airflow. To obtain the mean temperature of the airflow, $T_m$, the following equation is used

$$T_m = \frac{(T_i + T_o)}{2}$$

(3)

This $T_i$ is the fluid temperature at inlet. $T_o$ is the fluid temperature at outlet.

The local Nusselt number on heat transfer surface is defined by:

$$Nu = \frac{hD_h}{k}$$

(4)

The average Nusselt number of the smooth channel with fully-developed flow was calculated by Dittus-Boelter correlation as follow:
\[ Nu_0 = 0.023 \text{Re}^{0.8} Pr^{0.4} \]  

Where, \( Pr \) is Prandtl number of air.

2.4. Turbulence models

To examine the effect of pin inclined angle on flow and heat transfer characteristics of pin arrays in flow channel, commercial software ANSYS FLUENT 15.0 has been utilized. For a turbulence closure, the realizable \( k - \varepsilon \) turbulence model combined with a standard wall function from Launder and Spalding [9] was used as a turbulence closure. The realizable \( k - \varepsilon \) turbulence model was used to predict complex turbulent flows with flow swirling and separation, and was used to simulate the flow and heat transfer in the channel with pin fins [10, 11]. Figure 4 shows a comparison of the numerical data with the experimental data for the inclined pin at \( \theta = 90^\circ \). The overall average Nusselt number, \( \overline{Nu} \), was defined as follows:

\[ \overline{Nu} = \sum_{i=1}^{n} \overline{Nu}_i / n \]  

Here \( \overline{Nu}_i \) is spanwise average Nusselt number. It can be seen that the numerical data of the average Nusselt number of the inclined pin arrays in flow channel is in good agreement with experimental data. At \( \text{Re} = 20,400 \), the experimental \( \overline{Nu} \) value of inclined pin at \( \theta = 90^\circ \) is about 2.82\% higher than the case of numerical. The numerical results showed the same trend as the experiments obtained.

![Figure 4. Comparison of the computational data with the experimental data, \( \overline{Nu} \) varying with \( \text{Re} \)](image)

3. Results and discussion

3.1. Heat transfer characteristics

Nusselt number contour on heat transfer surface obtained from simulation are shown in Fig. 5. The results show for four different pin inclined angle at constant \( \text{Re} = 32,860 \). In the Fig.5, the air flows from left hand to right hand in X-axis direction. There are 10 rows of inclined pins in the test section for simulation. As can be seen from the contours, the local Nusselt number for case of \( \theta = 90^\circ \) is high for small area in upstream region in front of pin. This is due to the effect of horse shoe vortex around the pin while, the Nusselt number in downstream region is more distributed with large area. This agrees with the results obtained from other studies [7, 12 and 13]. The local Nusselt number in downstream region tends to increase from third rows to seventh rows but it tends to decrease next to eighth rows. For pin inclination to upstream, for case of \( \theta = 60^\circ \), the local Nusselt number in upstream of pin tends to decrease when reduce the pin inclined angle from \( \theta = 90^\circ \) to \( 60^\circ \). The local Nusselt number...
number in downstream of pin increased in small area at first row, second row and third row when reduce the pin angle from $\theta = 90^\circ$ to $60^\circ$. However, the overall Nusselt number in downstream of pin deceases and smaller than the case of $\theta = 90^\circ$ when the pin inclined angle at $\theta = 60^\circ$. For pin inclination to downstream, for case of $\theta = 120^\circ$ and $135^\circ$, the local Nusselt number in upstream of pin tends to increase at the third row onwards and the local Nusselt number in downstream of pin tends to significantly increase with large area, when increase the pin inclined angle from $\theta = 90^\circ$ to $120^\circ$ and $135^\circ$. The inclined pin for case of $\theta = 135^\circ$ give the largest heat transfer enhancement in upstream and downstream of pin row.

Figure 6 shows local Nusselt number along the X-axis at $Y/D = 0$ obtained from Fig. 5. The local Nusselt number was not shown for region of heat transfer surface under the pins. For case of pin angle $\theta = 90^\circ$, the local Nusselt number in upstream region tends to increase as going close to the pin in each of the row of pin and it tends to decrease when increasing X.D. In downstream region, the local Nusselt number tends to increase from pin in each of the row of pin and it tends to decrease as increasing X.D. For pin inclined angle to upstream at $\theta = 60^\circ$, the local Nusselt number in upstream of pin tends to decrease in each of the row of pin when compared to case of $\theta = 90^\circ$. For case of pin inclined to downstream, for case of $\theta = 120^\circ$ and $135^\circ$, the local Nusselt number in upstream and downstream of pin tends to increase in each of the row of pin when compared to case of $\theta = 90^\circ$.

The overall average Nusselt number for all of pin inclined angle are plotted in Fig. 7. The overall average Nusselt number is calculated by averaging the local Nusselt number obtained from Fig. 5. The pins array with pin inclined angle $\theta = 120^\circ$ and $135^\circ$ can enhance the overall average Nusselt number about 28.24% and 34.33% when compared to pin with angle $\theta = 90^\circ$, respectively.
3.2. Flow characteristics

Figure 8 compares the streamline on X-Y plane above the heat transfer surface about 1 mm obtained from CFD results for different pin inclined angles. The air flows from left hand to right hand passing pin arrays. For case of pin inclined angle $\theta=60^\circ$, there are also footprints two counter rotating vortices between the downstream of pins. For case of pin angle $\theta=90^\circ$, there are footprint of two wake flows behind each pin. The velocity near the heat transfer surface becomes small for large area behind the pins. For case of pin inclined angle $\theta=120^\circ$ and $135^\circ$, the footprint of wake flows can be also found behind the pin. However, the area of low velocity region is smaller when compare to case of pin angle $\theta=90^\circ$. The area of wake flow behind the pin was reduced with increasing pin inclined angle from $\theta=90^\circ$ to $120^\circ$ and $135^\circ$. The enhanced velocity in promote the heat transfer on surface with increasing pin inclined angle from $\theta=90^\circ$ to $120^\circ$ and $135^\circ$. 

![Figure 6](image1.png)

**Figure 6.** Local Nusselt number along the X-axis at Y/D=0

![Figure 7](image2.png)

**Figure 7.** Overall average Nusselt number for different pin inclined angles
Figure 8. Streamlines near the heat transfer surface (1.0-mm under heat transfer surface) of inclined pins on the X-Y plane

Figure 9 shows the velocity contour on Z-X plane passing the pin centre obtained from CFD results compared with different pin inclined angle. The region with low velocity behind the pin can be obviously detected for case of pin angle $\theta = 90^\circ$. This region corresponds to wake flow region behind the pin. However, the region with low velocity cannot be seen for case of $\theta = 135^\circ$. This flow characteristic will promote the heat transfer behind the pins.

Figure 10 shows the velocity streamline on Y-Z plane at different location X-D-10, 11.25 and 12.5. For case of pin inclined angle $\theta = 60^\circ$, there are a pair of counter-rotating vortices behind each pin on Y-Z plane at X-D-10, 11.25 and 12.5. The counter-rotating vortices in downstream of this inclined pin may destroy the wake flow behind the pins. For case of pin angle $\theta = 90^\circ$, the counter-rotating vortices cannot be found. For case of pin inclined angle $\theta = 120^\circ$ and $135^\circ$, the counter-rotating vortices can be found behind the pin on Y-Z plane at X-D-12.5. The vortices locate near the bottom surface, which is far from the heat transfer surface.
Figure 9. Comparisons of the streamwise velocity on Z-X plane passing the centre of wind tunnel for various pin inclined angles

Figure 10. Comparisons of the streamlines on Y-Z plane with at some X/D for different pin inclined angles
4. Conclusions
In this study, the flow and heat transfer for inclined pin arrays attached in wind tunnel was studied with numerical simulation. The effects of pin inclined angle were investigated in the range of $\theta = 0^\circ$, 60°, 90°, 120°, and 135° at constant Reynolds number $Re = 32,860$ for comparison. The main results obtained from this study can be concluded as follows:

- The pin inclined angle affects strongly on the heat transfer characteristics on surface with pin arrays. The pin inclined angle at $\theta = 120^\circ$ and $135^\circ$ can improve the heat transfer in region in front of pin and can enhance the heat transfer behind pin. Particularly for the case of $\theta = 135^\circ$, the overall average Nusselt number is higher than the case of $\theta = 90^\circ$ about 34.33%.
- For case of pin inclined angle $\theta = 60^\circ$, $120^\circ$ and $135^\circ$, a pair of counter-rotating vortices was generated behind each pin. The counter-rotating vortices may reduce the wake flow behind the pins.
- For case of pin angle $\theta = 90^\circ$, the wake with low velocity region can be detected behind each pin. This results in low heat transfer region on the surface behind the pin. When increase the pin angle from $\theta = 90^\circ$ to $120^\circ$ and $135^\circ$, the region of wake flow become smaller and start to increase strongly velocity behind the pins at near the heat transfer surface. Therefore, the heat transfer rate is higher than the case of $\theta = 90^\circ$.

References
[1] Lau S C, Kim Y S and Han J C 1985 Effects of Fin Configuration and Entrance Length on Local Endwall Heat/Mass Transfer in a Pin Fin Channel ASME Paper 85-WA-HT-62
[2] Chyu M K and Natarajan V 1996 Heat Transfer on the Base Surface of Three-Dimensional Protruding Elements Int. J. Heat Mass Transfer 39 2925-2935
[3] Zukauskas A A 1972 Heat Transfer from Tube in Cross Flow Adv. Heat Transfer 8 116-133
[4] Chyu M K 1990 Heat Transfer and Pressure Drop for Short Pin-Fin Arrays with Pin-Endwall Fillet ASME J. Heat Transfer 112 926-932
[5] Pandit J, Thompson M, Ekkad S V and Huxtable S T 2014 Effect of pin fin to channel height ratio and pin fin geometry on heat transfer performance for flow in rectangular channels Int. J. Heat Mass Transfer 77 359-368
[6] Rao Y, Wan C, and Xu Y 2012 An experimental study of Pressure loss and heat transfer in the pin fin-dimple channels with various dimple depths Int. J. Heat Mass Transfer 55 6723-6733
[7] Lawson S A, Thrift A A, Thole K A and Kohli A 2011 Heat transfer from multiple row arrays of low aspect ratio pin fin Int. J. Heat Mass Transfer 54 4099-4109
[8] Choi I K, Kim T, Song S J and Lu T J 2007 Endwall heat transfer and fluid flow around an inclined short cylinder Int. J. Heat Mass Transfer 50 919-930
[9] Launder B E and Spalding DB 1974 the numerical computation of turbulent flows Comput.Meth.Appl.Mech.Eng. 3 269-389
[10] Xie G, Sunden B and Zhang W 2011 Comparisons of pins/dimples/protrusions cooling concepts for a turbine blade tip-wall at high Reynolds number J.Heat Transfer 133 1-9
[11] Rao Y, Wan C, Xu Y and Zang S 2011 Spatially-resolved heat transfer characteristics in channels with pin fin and pin fin-dimple arrays Int. J. Therm. Sci. 50 2277-2289
[12] Wang F, Zhang J and Wang S 2012 Investigation on flow and heat transfer characteristic in rectangular channel with drop-shaped pin fins Propulsion and Power Research 1 64-70
[13] Chang S W, Yang T L, Huang C C and Chiang K F 2008 Endwall heat transfer and Pressure drop in rectangular channels with attached and detached circular pin-fin array Int. J. Heat Mass Transfer 51 5247-525