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Flow and Thermal Performances of Pin Fin Heat Sink

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Abstract

The current work bolds hexagonal and cylindrical pin fin arrays at 50°C on a horizontal base plate. This, along with the transfer of heat of fiction characters of cylindrical fin, was studied experimentally with the help of a rectangular wind tunnel. Inline and staggeredarrangements are to be made with pin fins. Finding the impact of fin distance in both span and stream directions must be done. Maintaining constant values for both the relative span pitch (Sx/d=1.2) and reactive stream wise pitch (Sy/d = 1.2, 2.4, and 3.6) is recommended. The research uses varying air mass flow rates (Re varies between 2000 and 280000). The clearance rate (C/H = 0.0) greatly affects the pin's performance, which makes the hexagonal fin conduct heat more rapidly than the cylindrical fin.The experiment's findings demonstrate that, for a given pumping power, the optimal inter-fin pitch, Sx/d=1.2 and Sy/d=1.2, are determined by looking for the greatest Nusselt number.

Keywords: Wind Tunnel, pin-fin, Nusselt number, Reynolds number, Pitch and tip clearance, Hexagonal Fin

Introduction

The mechanism of many engineering systems ends with heat generation, which leads to too many heating problems. Even system failure heat transfer components that cost less and have smaller weights can be used to create the solution.The dissolution of heat produced by various appliances such as transformers, air-cooled engines, compressors, boilers, condenser coils, superheated tubes, and electronic components is necessary for the system to function well at its operating temperature. Qinggang Qiu et al. (1) study analysis offered numerous approaches. Active techniques requiring external power include churning a liquid or shaking a solid surface. Fins, on the other hand, don't require this power. Increasing the fins' surface area can improve heat transfer at the solid-fluid contact. Armstrong has used fins in many ways for heat transmission, from more basic shapes like square, rectangular, cylindrical, tapered and annular pin-fins to a combination of many geometries.Owing to the inexpensive cost of production and high thermal efficiency of these fins, it is important and popular to study how geometric parameters like fin length, spacing, and height affect heat dissipation.Kadir et al. (2) and That et al. (3)).The performance of heat transmission of an array of cylindrical fins mounted to a staggered wall was initially studied by Agaro & comini et al. (4). Metzger et al. (5) examined the characteristics of a staggered cylindrical pin-fin array's heat transfer. Heat transfers from staggered arrays of cylindrical pin-fins are also studied by Simoneau et al. (6). Armstrong et al. (7) recommended investigating the heat transfer from dispersed array pin fins to cool turbines.Jubran et al. (8) conducted tests to investigate the impact of the optimal interfin spacing and shroud clearances (both span- and stream-wise). Forced convection in exposing the characters of heat transfer was investigated by Ashish of Anil (9) with a grooved flat pin type. It shows the grooved type's ability to enhanceheat transfer characteristics over conventionally employed fins.Tariq et al. (10) used a duct equipped with various ribs of pentagonal, trapezium, and square forms of truncated prismatic forms to explore improved heat transfer of flow characteristics to determine the fundamental heat transfer process.Mechanism of efficient engineering applications: Hamdi et al. (11), heat exchangers, solar collectors, electronics cooling, automotive applications, etc. Karami et al. (12) conducted general convention experiments with three-fin tubular exchangers and a square fin array.Fin spacing ranges from 5 to 14 millimetres. Based on the experiment's results, free convection contributes approximately 80% of the total energy input and 10% of the temperature distribution from ambient radiations to the heat source. To a certain extent, the heat transfer coefficient rises with fin spacing. Mesut Abuşka and Vahit Çorumlu [13] This study investigated the thermal and hydraulic performance of five heat sink models (CPFHSmst, CPFHSst, CCPFHSpar, CCPFHSperp, and FHS) using forced convection. The studies were conducted at 33 W, 66 W, and 99 W, with seven airflow velocities ranging from 1 m/s to 7 m/s (Re range from 2000 to 16,000) and a constant ambient temperature of 30 °C±0.6. The effects of airflow rate, heating power, fin geometry, and fin arrangement on the cooling performance of heat sinks have been investigated thoroughly. Chieh and Chou[14] one of the goals of this numerical analysis was to see how the flow configuration impacts heat transfer in a staggered cylindrical array. Because the stagger angle exceeds 35°, the mean Nusselt numbers of array systems are greater than those of a single cylinder. The magnitude of Nu/Cd is proposed as a parameter to assess the efficiency of heat transfer in an array system.

Overall, a literature study reveals how fins have been altered to improve heat flow by adding holes, slots, and struts. However, it is not advised to use rib struts.This could make the current fin heavier. Employing fins with various cross-sectional shapes, such as hexagonal and cylindrical fins, can affect flow resistance and heat transmission differently. Because pin fins are relatively easy to make, circular and hexagonal fins are frequently employed in applications.The current experiment compares the behaviour of hexagonal-type fins to that of cylindrical fins by examining the impact of heat transfer on fluid flow parameters.

EXPERIMENTAL SETUP

Fig.1displays the experimental apparatus's schematic

diagram.This experimental setup includes a heating unit, a data unit, a pin fin assembly, and a wind tunnel . A working open circuit suction mode wind tunnel was utilised for this experiment.

Fin configurations

Each cylindrical fin measured 90 mm in height and 10 mm in diameter, hexagonal fin measured 77.7 mm in height and 6 mm side diameter. Then, at the base of each fin, a rectangular aluminium plate measuring 250 mm, 145 mm 25.4 mm is fixed, allowing for electrical heater heating. A fin is positioned inside the test section so that the air passageways travel over the fin surfaces during the test run. Figure 2 depicts the geometries of cylindrical fins and perforated fins (a and b)

A. Wind tunnel:

The 2 m long wind tunnel has an internal width of 150 mm and is constructed of plywood that is 19 mm thick. Air is supplied to the testing area at the required flow rate by a centrifugal blower controlled by an operative gate valve.The electricity could only flow at a maximum rate of 0.242 kg/sec. Two cardboard honeycombs were used to mix the heated air from the finned tube assembly. One honeycomb was placed crosswise to the flow stream and had a relatively low porosity, while the second honeycomb had a larger porosity and was placed through an isolated chamber.The pressure drop over a calibrated orifice plate was measured with a differential manometer to calculate the air's mass flow rate. It indicates the air flow rate using a conventional orifice plate. In order to keep the temperature consistent, the heat exchanger base was heated evenly.

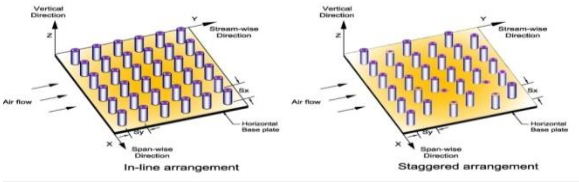

Fig.1 Pin-fin assembly

A well-fitting open-topped hardwood box with enough thermal insulation protects the pin-fin assembly base and heaters. Altering the voltage and variac position would allow one to control how much power was delivered to the main heater.Nine thermocouples were used to measure the test module's base plate temperature.

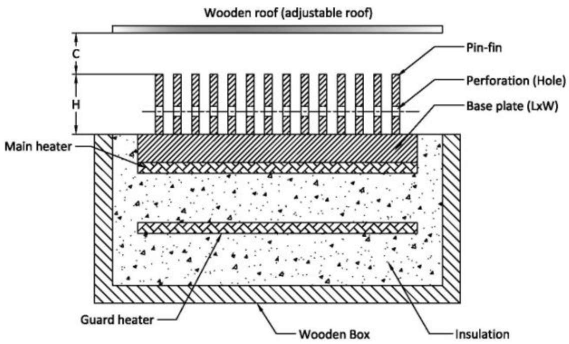

Fig.2 Sectional elevation of pin-fin assembly

Eight additional thermocouples were used to monitor the temperature of air entering and leaving the system.There were four additional positioned downstream of the pin fin array and four more across from the pin-fin assembly's entrance. The steady state reached in the experimental run is when all of the thermocouples were linked to the temperature indicators and those that indicate the ambient air temperature. Eight additional thermocouples were used to monitor the air temperature coming in and going out of the system.Eight were placed, four downstream of the pin fin array and four in front of the pin-fin assembly's entrance.The steady state was obtained around two and a half hours into the experiment. The trials were run for an additional hour or so using a second thermocouple, representing the ambient air temperature attached to the temperature indicators to verify the steady state result. An orifice meter calibrated against the hotwire anemometer measured the air flow rate. A gate valve on the blower's suction side can be adjusted to open to control the airflow rate. The blower forces air through the test portion, which escapes to the outside for the whole experiment.

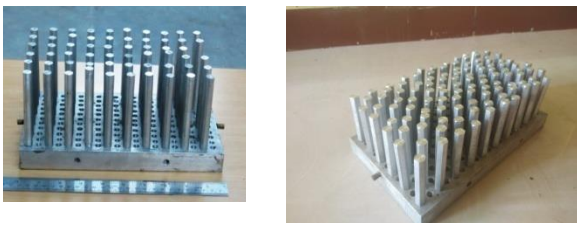

Fig.3 Photographic view of cylinder and hexagonal arrays

5.2 DATA REDUCTION

The calculated data set helps to measure the friction factor (FF) and heat transfer rate (HTR). From this finned surface, HTR in steady state condition is about,

$$Q_{tot} = Q_{conv} + Q_{rad} + Q_{loss} \quad (1)$$

There are related data reduction cases studied by**Naik *et al.* (15),Jubran *et al.*(8), Tahat*et al,*(3)**.They stated the fins arrays are comparable, and the assembly heat loss is controlled to below 5%.Because of the existing operating conditions, the test segment is well insulated, and it is assumed that the losses are minimal, and then the Equation. (1) is rephrased as,

$$Q_{conv} = mc_p (t_{out} - t_{in}) \quad (2)$$

The base plate and convective heat transfer fin surface are provided by

$$Q_{conv} = hA_s \left[t_b - \left(\frac{t_{in} + t_{out}}{2} \right) \right] \quad (3)$$

Where are the air flow temperatures, the average temperature at the base assembly's center, and the total test surface area, which can be written as,

$$A_s = W\,L + \pi\,d\,H\,N_{xy} - \frac{\pi\,d^2\,N_{xy}}{4} \quad (4)$$

The average heat transfer coefficient for the heated pin-fin assembly can be obtained by combining the equations. (2) and (3):

$$h = \frac{m\,c_p\,(t_{out} - t_{in})}{A_s \left[t_b - \left(\frac{t_{in} + t_{out}}{2} \right) \right]} \quad (5)$$

Given the current operating circumstances and the well-insulated test section, the free flow area is computed as follows:

$$A_{ff} = W(H+C) - N_x H\,d \quad (6)$$

The Nusselt number is calculated using the previous heat transfer coefficient value as conventionally,

Nu=hDh/ (7)

The Reynolds number (Re) is defined as

$$Re = \frac{G\,d}{\mu} \quad (8)$$

Results And Discussions

The following outcomes for each parametric influence on heat transmission are examined independently or in combination: Pin-fin spacing, pin-fin clearing ratio, pin-fin array, pin-fin shape (cross-section), and pin-fin size (number of fins).

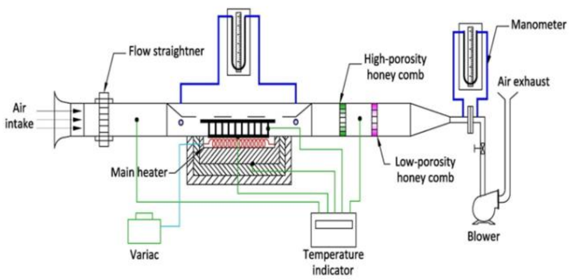

Fig.4 Experimental setup

Heat Transfer Rate

Figure 5 shows the Reynolds number and average heat transfer rate in the finned surface for various constants (Sx=1.2; Sy=1.2, 2.4, and 3.6). The hexagonal fin array has a greater maximum heat transfer rate than the cylindrical fin at Sy/d=1.2.

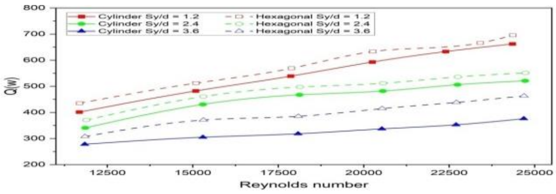

Fig. 5 Pin-shaped object's impact on Q out for Sx/d = 1.2

Effect of Nusselt Number as Function of Reynolds Number

Figure 6 illustrates the relationship between the Reynolds number and the Sx=1.2 and Sy=1.2, 2.4, and 3.6 constant values. There is a positive correlation between the Reynolds and Nusselt numbers. A low Sy/d correlates with a high Nusselt number because an increase in Sy/d reduces the pin-lowest fin's velocity.

Effect of clearance ratio (C/H)

The influence of the clearance ratio on heat transfer rate is shown in Figure 7, which covers the range of Reynolds numbers evaluated in the trials. For all C/H values, there is a monotonic increase in the heat transfer rate with Reynolds number.Additionally, the lowest C/H ratio is where the greatest heat transfer is observed; the heat exchanger's greater compactness makes this improvement in heat transmission possible (i.e., confining the flow of air freely). This pattern has been noted by prior researchers (Jubran et al. (8);Tahat et al. (3)). In Fig. 7, the impact of fin configurations on heat transport is also shown.

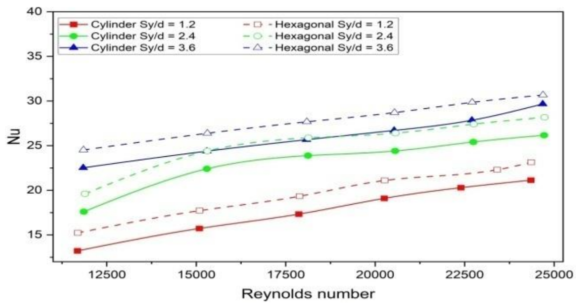

Fig. 6 Nu against Reynolds number plot for C/H = 0 and Sx/d = 1.2

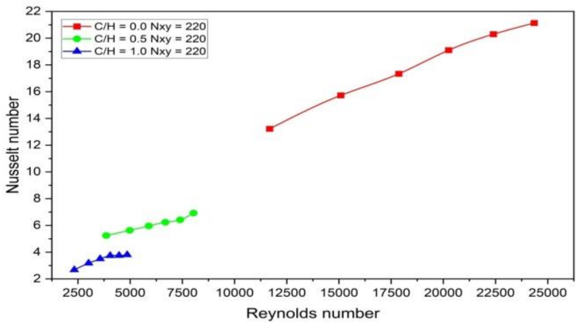

Fig. 7 Reynolds number against Nusselt number plot for Sx/d = Sy/d = 1.2 (cylinder)

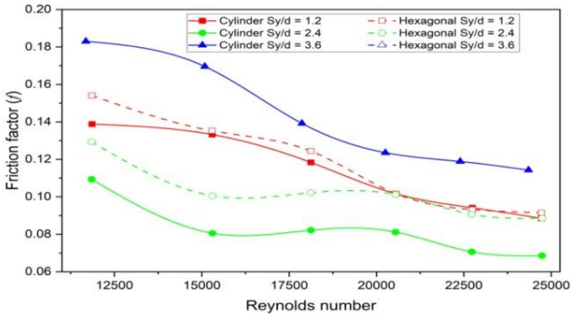

Fig. 8 Plot of "f" versus Reynolds number (cylinder) for the cases when Sx/d = 1. 2, Sy/d = 1.2, 2.4, 3.6

Friction and pumping power characteristics

Effect of Fin Number and Flow Rate on Pressure Drop: Often utilised as a constraint to have minimum pumping power, flow resistance is a vital component in heat transmission.The pressure decrease at different flow rates is shown in Fig. 8 for C/H=0.0 by plotting the friction factor against the Reynolds number. According to Jubran BA et al. (8) and Kodah et al. (3), staggered designs cause significant changes in Sx/d, although in-line layouts cause a modest variation in the friction factor on Sy/d change.

Effect of pin-fin-arrangement

Fig. 9 shows how the hexagonal pin-fin and cylinder profile forms affect the enhancement of heat transmission. The cylinder outperforms the hexagonal fins despite the predicted four pin-fin profiles.The performance of staggered pin-fin arrays surpasses that of the other systems. This is because the wake flow impact becomes less pronounced once past the first row of pin fins.Between any two adjacent pin-fins of the upstream row, free flow fluid or air introduces new boundary layer growth into the downstream rows to support the heat transfer mechanism.A staggered array's primary flow path is longer or more convoluted, lengthening the fluid's residence time.

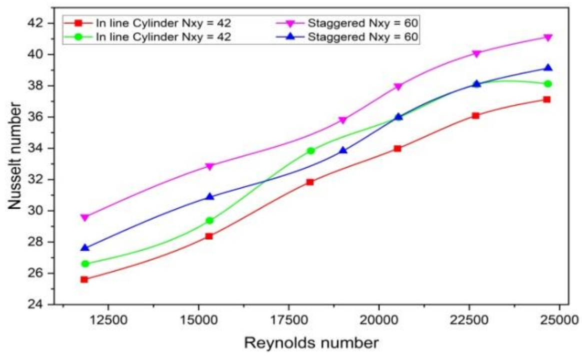

Fig. 9 Pin-fin shape's effect on Nu for C/H=0 and Sx/d= 1.2

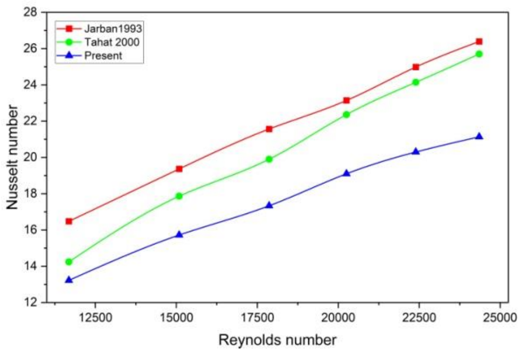

Fig. 10 Reynolds number vs. Nusselt number plot (with other correlation)

Comparison with other heat transfer correlations

Usually, heat transfer data are displayed as changes in Nusselt numbers relative to Reynolds numbers. For the current work, Nusselt numbers rise with Reynolds number, as seen in Fig. 10.it is being compared to earlier research to validate the data. Within the experimental range, it is discovered that the current results are greater and more in line with those of Kodah et al. [3] and Jubran BA et al. [8].

Conclusion

This work has experimentally investigated pin fin heat transfer under forced convection. There has been a performance comparison between the hexagonal and cylindrical fins. These inferences can be made based on the experimental results mentioned above.

It transfers heat more effectively when comparing hexagonal fin arrays with cylindrical fin arrays. 1.2 (Sy/d) is the ideal stream-wise ratio for enhancing heat transfer. Additionally, because more of the fluid is obstructed in hexagonal fin arrays than cylindrical fin arrays, there is a greater pressure drop and faster heat transfer. As Reynolds number decreases with decreasing streamwise ratio for all fin geometries, the Nusselt number increases. Whenever grooves are present, Nusselt numbers are significantly higher at all Reynolds numbers, regardless of the Reynolds number. A cylindrical fin and a hexagonal fin with Sy/d=2.4 and Sy/d=3.6 each have similar Nusselt numbers for different Reynolds numbers.

The enhancement factor is calculated by dividing the Nusselt number of hexagonal fins by the Nusselt number of cylindrical fins. Hexagonal fins have a relatively higher enhancement factor than cylindrical fins. A decrease in the inter-fin distance ratio led to an increase in friction factor. A hexagonal fin has a little different friction factor than a cylindrical fin.

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