Indian Journal of Science and Technology Vol. 4 No. 10 (Oct 2011) ISSN: 0974- 6846

Effect of geometric and flow parameters on the performance of pin-fin arrays

R. Karthikeyan and R. Rathnasamy

[1]Mechanical Engineering, Annamalai University, Annamalainagar-608002, India
rkyuha@yahoo.co.in; mecratz@yahoo.co.uk

Abstract

This investigation pertains to an experimental study of forced convective heat transfer from the array of circular pin-fins attached on a rectangular-horizontal base plate maintained at 50°C placed in a rectangular channel formed the passage for flow of air. The study involved the effect of pin-fins spacing in both stream and span wise directions on the performance of heat sink. The tip clearance effect was also studied. The positions of cylindrical fins arranged in array either in-line or staggered manner. The parameters, i) Reynolds number ranged between 2000 to 25000 ii) inter fin distance varied from 12 mm to 228 mm in both the stream wise (s_y) and span wise(s_x) directions and iii) clearance ratio (C/H) 0.0, 0.5 and 1.0 were used in the study. The dependence of Nusselt number and friction factor on the above parameters is accomplished and the results were matching with previous reports.

Keywords: Forced convection, Pin-fin, heat exchanger, Channel flow, pressure drop, pitch and tip clearance

Introduction

The phenomenal internal heat generation is inevitable in many industrial processes and that too in most of the electronic components causing system failure due to overheating and insufficient heat removal mechanism. According to a U.S. Air Force study (Reynell, 1990) the four primary sources of stresses that cause failures in avionics systems are temperature (~55%), vibration (~20%), excessive humidity (~19%) and dust (6%). Humidity is also a temperature related phenomenon. Therefore, a total of ~74% of break down resulted from thermal over stressing. Therefore, removal of heat by an effective cooling technique is often required. Cooling for electronic system is usually needed to maintain the component temperature lower than 50ºC in order to achieve prolonged mean life between failure and replacements. Fins are frequently used in heat exchanging devices to increase the heat transfer rate from surface and the surrounding fluid. Various types of fin geometry employed include rectangular, cylindrical, square, elliptical, diamond and tapered pin-fin configurations (Chyu *et al.* 1998; Tahat *et al.,* 1990; Sara, 2003).

The pin-fin array system's heat transfer characteristics have been the subject of extensive investigation because of its importance. Commonly used heat sink is pin-fin type. A pin-fin is an element attached perpendicular to a wall against the fluid flow. There are various parameters characterizing the pin-fins are, by shape, height, diameter and height to diameter ratio. In addition to the physical geometry, pin-fins are positioned in arrays either inline or staggered with respect to the flow direction. The pin-fin finds variety of engineering applications like compact heat exchangers and the cooling of advanced gas turbine blades and electronic devices.

The heat transfer from a pin-fin assembly to the surrounding environment occurs by convection and radiation. The rate of heat dissipation depends on the (i) temperature distribution in pin-fins and its assembly (ii) pin-fin geometry (iii) Pin-fin arrangement (iv) Fluid flow rate and (v) Tip clearance. For a given base plate temperature, heat transfer rate could be enhanced based on the Newton's law of cooling ($Q=h\ A\ \Delta T$) by altering the values of heat transfer coefficient or surface area (A) and while limiting the temperature difference. An increase in heat transfer co-efficient can be achieved via forced convection or changing fluid (not practicable always). The alternate is by changing the geometry of the heat sink to enhance the heat transfer.

Jubran *et al.* (1993) investigated experimentally the effects of shroud clearance, optimum inter-fin spacing in both span wise and stream-wise directions and missing pin upon heat transfer, and they correlated their heat transfer data in their study. Tahat *et al.* (2000) performed using a channel with rectangular cross section equipped with cylindrical fins. The clearance ratio used in their study is C/H=0.0 and the optimum spacing in the span-wise direction of 7.6 mm was achieved. Kadir *et al.* (2000) conducted the experiments in a channel having the rectangular cross-section and they reported that the Reynolds number increases with increase in Nusselt number and the maximum heat transfer occurred at $S_y/d=2.97$, the Reynolds number covered in the study is 3700-30,000. Numerous works have been conducted on long pin-fins. Zukauskas (1987) presented famous correlations between heat transfer and pressure drop for in-line and staggered banks of tubes over wide ranges of Reynolds number and relative transverse and longitudinal pitches. Armstrong and Winstanley (1988) reviewed low pin-fin height and inter fin pitch affect heat transfer and flow friction, as well as the effect of accelerating flow in covering pin-fin channels. Babus'Haq *et al.* (1995) reported that the optimal ratio of the inter-fin pitch to the pin-fin diameter in the transverse direction was 2.04 for all pin-fin systems. From the literatures it is observed that the researchers contributed more for lower aspect ratio (H/d) and results available for higher H/d is less. Hence it is necessary to investigate the pin-fin array performance with higher aspect ratio.

The present work involves experimental study on the heat transfer and friction loss characteristics of a surface with cylindrical fins in a channel having rectangular cross section with larger fin diameter and different channel geometry. The experiment was performed for in-line arrangement. Keeping the constant distance between fins in the span wise direction to the flow, the effect of the distance between the fins in the flow direction on the heat transfer was investigated. Furthermore, friction loss was determined by measuring overall pressure drop between test module ends. The experimental parameters were chosen as Reynolds number, fin arrangement and fin distance in the flow direction.

Experimental test rig

The experiments were conducted in a wind tunnel with the test section having 180 mm height, 150 mm width and 2 m length. The wall thickness of the channel is 20 mm. The test module (pin-fin assembly) was so mounted at the bottom side of the channel such that the midpoint and or the end walls of the channel are symmetric with reference to centre plane in the stream-wise direction. The pin-fin assembly used was a duralumin plate; the dimensions are of 250 mm length, 145 mm width and 25.4 mm (1 in) in thickness.

A plate heater of approximately the same dimensions as the test module was located just below to supply constant heat input to maintain constant surface temperature. This is possible with such a thick duralumin plate. The maximum power of the heater is about 1500 W. The amount of heat supplied by the heater is controlled with auto transformer (Variac). To reduce contact resistance to heat flow, a heat sink compound was applied, both between the heater and test plate and the test plate and the fins. Heat loss to surrounding is minimized to zero by insulating at the bottom side of the

Fig.1. Ensemble of pin-fin with shroud

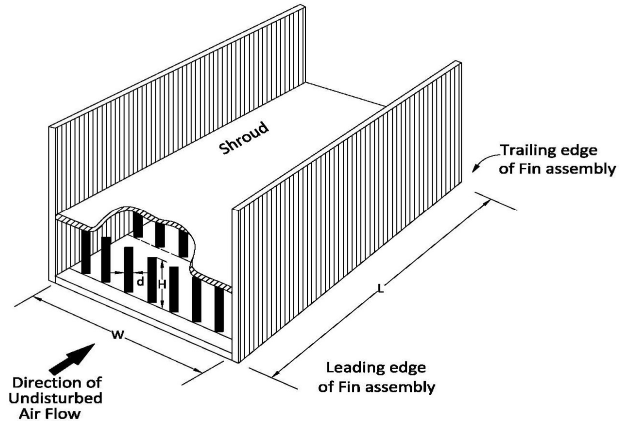


Fig. 2. Pin-fin assembly (a) Inline (b) Staggered

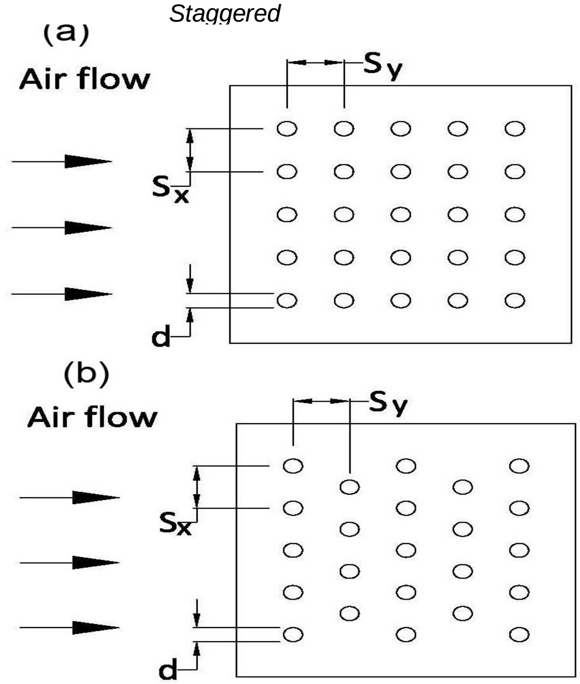
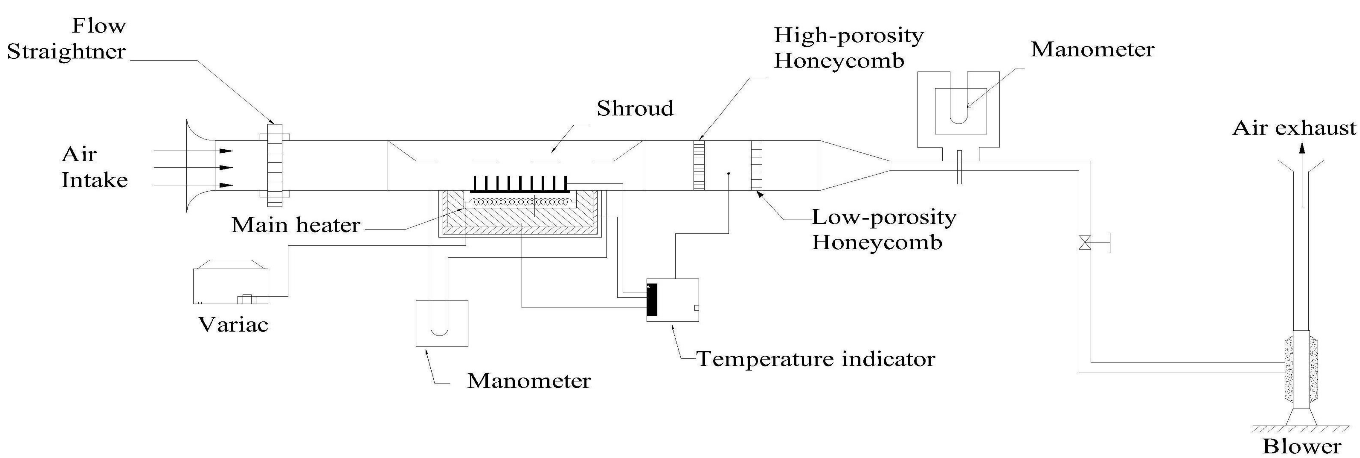


Fig. 3. Schematic arrangement of experimental test rig



heater and side of the test module. This is achieved by means of high thermal resistance packing, glass wool of 25 cm thickness first and then wood of 2 cm thickness below the former. Moreover, all the outside surfaces of the channel, downstream from the beginning of the test section, were completely insulated by wrapping with 5 cm of common glass wool. To place easily the pin-fin assembly on the heating surface, the upper surface of the channel was equipped with an adjustable top cover. Figure 1 shows the pin fin assembly with shroud. Figure 2. illustrates in-line and staggered arrangement and the experimental test rig is schematically shown in Fig.3.

Data Measurement

Nine copper-constantan (type-T) thermocouples are symmetrically distributed to measure the base plate temperature of the test module. These thermocouples are connected to a digital temperature indicator having resolution ±0.1 °C. The average of these values was taken as the steady state temperature of the test surface. The experiments were performed at a constant average surface temperature of 50±0.5 °C. The inlet temperature of the air stream was taken as the average reading of four RTDs' located after the flow straightener. Similarly, the outlet temperature of the air stream was taken as the average reading of four RTDs' located in the downstream region of the insulated channel. One thermocouple each for the outer surface temperature of the heating section and the ambient are employed. All the temperature sensors are calibrated against a Pt100 standard one using a temperature bath. Each trial of experiments was continued 25 min even after the steady state which is about 1½ h. An orifice meter is incorporated in the downstream side of the test section in order to measure the flow rate of air through the tunnel. Air flow in the wind tunnel was established by installing an air blower that runs in induced draft mode. The cylindrical fins used have the dimensions of 10 mm diameter and 90 mm height.

Often the specification of a heat exchanger will state the maximum pressure drop that can be accommodated in the system. In the current investigation, it was necessary to take into account the pressure drop at the entrance and pressure rise at the exit that would occur due to change in flow-area. Pressure loss occurs due to the flow through the pin-fin assembly. Pressure measurements using the static pressure tapings installed in the pin-fin assembly (Figure 3) were observed and the overall pressure drops were evaluated for all the tests conducted. After acquiring the steady state data for each trial with necessary parametric variations, the setup re-configured to next clearance ratio (C/H). Table 1. shows minimum and maximum values of various experimental conditions used in the present investigation.

Table 1. Experimental conditions

Parameters	Minimum value	Maximum value
x-direction pin-fin spacing (S_x) [mm]	12	12
y-direction pin-fin spacing (S_y) [mm]	12	228
Mass flow rate [kg/s]	0.069	0.143
Clearance ratio (C/H)	0.0	1.0
Number of fins	22	220
Reynolds number	2000	25000

Data reduction

The measured data are used to evaluate the heat transfer rate and friction factor in the setup. The steady state heat transfer from the finned surface is,

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

For the present case the data reduction is similar to that of Naik *et al.* (1987) and Jubran *et al.* (1993) conducted experimental tests and fin arrays alike and reported that the total heat loss from the assembly ensue less than 5%. Under the present operating conditions together with the fact that the test section was well insulated, eqn. (1) is rewritten as,

$$Q_{conv} = mCp(t_{out} - t_{in}) \qquad (2)$$

The heat transfer by convection from fin surface including base plate is given by,

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

where, t_{in} and t_{out} are the temperatures of air flow, t_b is the average temperature at certain designated locations on the base assembly and A_{ts} is the total surface area of base assembly and fins, which can be written as,

$$A_{ts} = W\ L + \pi\ d\ H\ N_{tot} - \frac{\pi\ d^2\ N_{tot}}{4} \qquad (4)$$

The average heat transfer coefficient can be calculated by the combinations of eqn. (2) and (3)

$$h = \frac{m\ Cp\ (t_{out} - t_{in})}{A_{ts}\left[t_b - \left(\frac{t_{in} + t_{out}}{2}\right)\right]} \qquad (5)$$

The free- flow area A_{ff} is calculated as

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

Expressions for the specific heat, dynamic viscosity and thermal conductivity of air at atmospheric pressure (Tahat *et al.* 1994) are given in the range $250 \le (T_{in} + T_{out})/2 \le 400$ respectively:

$$Cp = \left[9.8185 + 7.7 \times 10^{-4}\ \frac{(T_{in} + T_{out})}{2}\right] \times 10^2 \qquad (7)$$

$$\mu = \left[4.9934 + 4.483\text{x}10^{-2} \frac{(T_{in} + T_{out})}{2} \right] \text{x } 10^{-6} \qquad (8)$$

$$k = \left[3.7415 + 7.495\text{x}10^{-2} \frac{(T_{in} + T_{out})}{2} \right] \text{x } 10^{-3} \qquad (9)$$

The Reynolds number (Re) is defined in the conventional way as,

$$\text{Re} = \frac{G \quad d}{\mu_a} \qquad (10)$$

Where, $G = m/A_{ff}$ is the mass flux.

The pressure loss due to flow through the pin-fin array is represented non-dimensionally by a friction factor (f), is based on the D'Arcy relationship, $\Delta p/\rho = f\,(L/d)\,(v^2/2)$ rewritten as,

$$f = \frac{2\,\Delta p}{(L/d)\,(m/A_{ff})^2\,(1/\rho)} \qquad (11)$$

Fig. 4. Variation of Nusselt number with the stream-wise tin-pin spacing for C/H=0.0

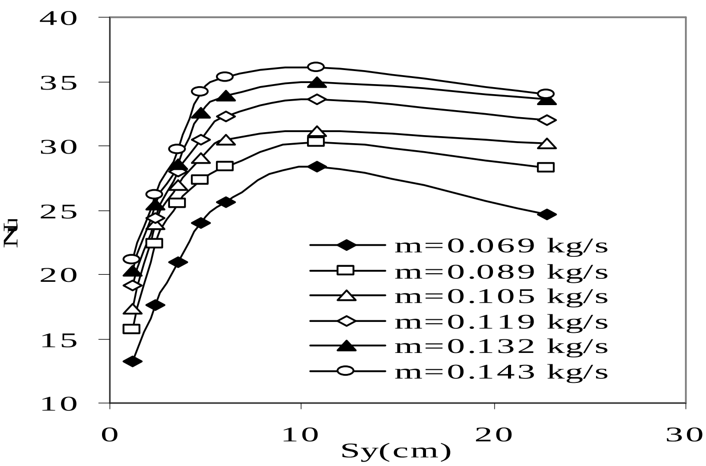


Fig. 5. Variation of Nusselt number with the stream-wise fin-pin spacing for C/H=0.5

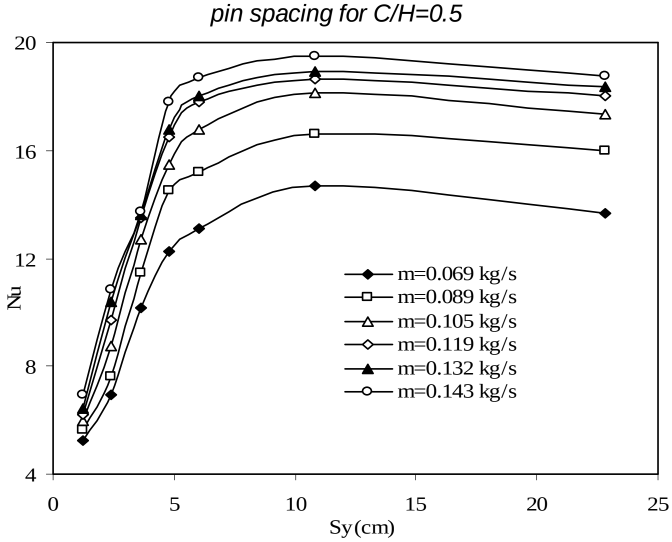


Fig.6. Variation of Nusselt number with the stream-wise fin-pin spacing for C/H=1.0

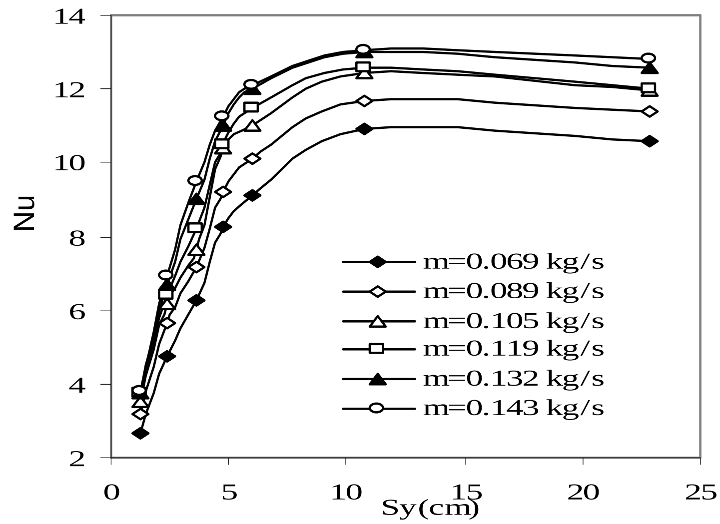


Fig.7. Variation of the heat transfer with Reynolds number

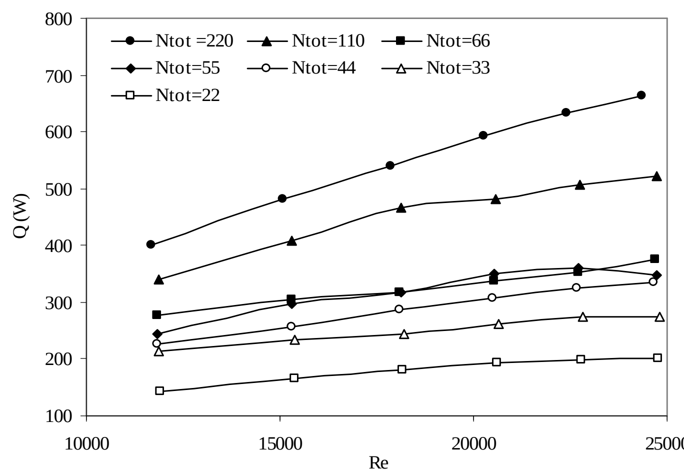


Fig. 8. Plot of heat transfer versus Reynolds number for S_y/d =1.2 and S_x/d=1.2

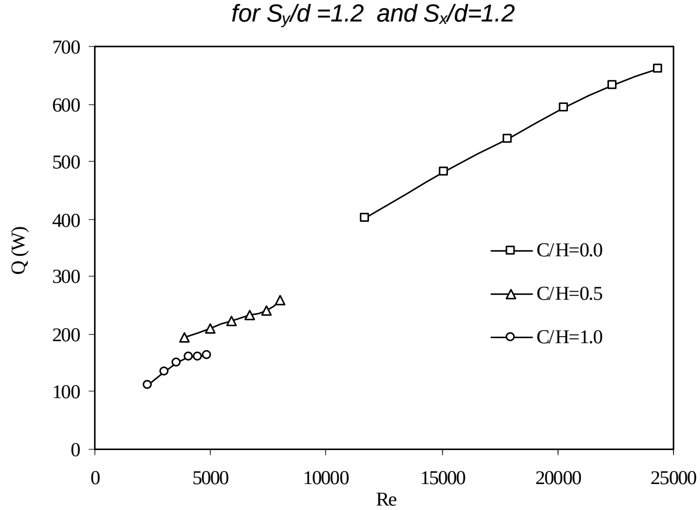


Fig. 9. Plot of heat transfer versus Reynolds number for S_y/d =1.2 and S_x/d=1.2

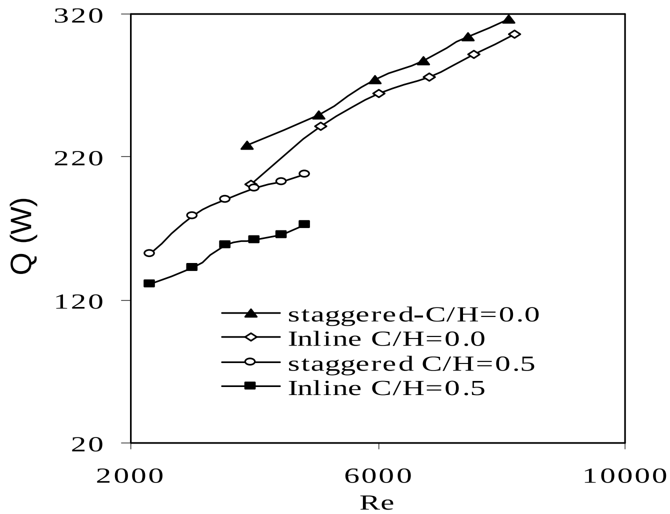


Fig. 10. Plot of friction factor versus Reynolds number for C/H=0.0 and S_x/d=1.2

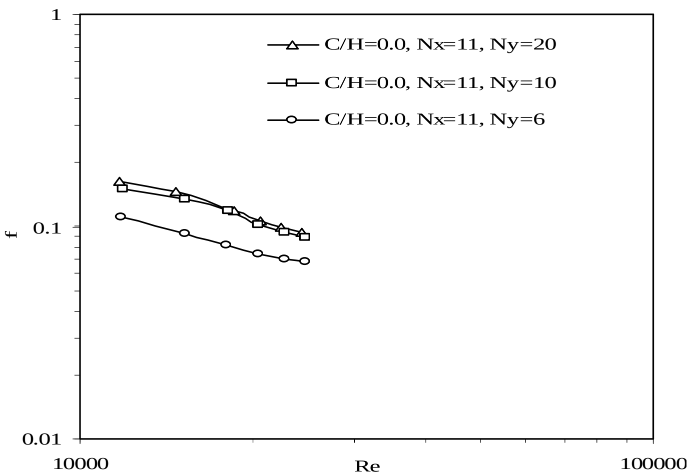
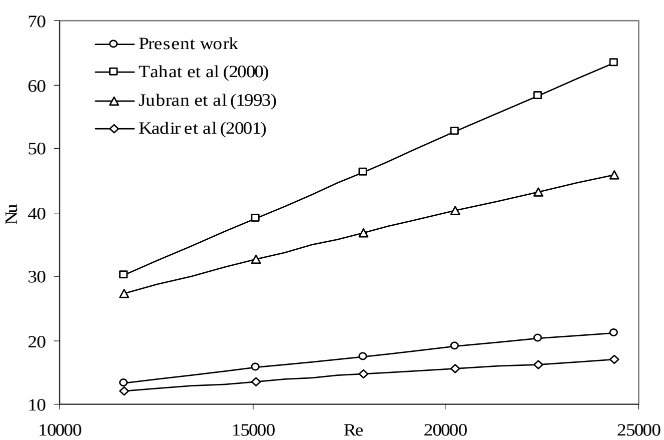


Fig. 11. Plot of Nusselt number versus Reynolds number for C/H=0.0.



Results and discussion

For an in-line pin-fin arrangement and a fixed uniform spacing in span-wise direction (S_x), the steady state heat transfer on keeping the C/H value constant was evaluated for several values of spacing in the stream-wise direction, S_y and for different mass flow rate.

Effect of Fin Spacing and Clearance Ratio on Heat Transfer

Heat transfer measurements were carried out for three values of C/H, namely 0.0, 0.5, and 1.0. In each case, measurements were carried out for six values of mass flow rates, namely 0.069, 0.089, 0.105, 0.119, 0.132 and 0.143 kg/s. The Reynolds number used in the experiment ranges from 2000 to 25000. Under these specified values of C/H and air mass flow rates and for the in-line arrangement, a plot of Nusselt number (Nu) *versus* stream wise spacing (S_y) of fin-pin is depicted in Fig.4.

Corresponding figures for C/H =0.5 and 1.0 are shown in Fig.5 and 6 respectively. It can be seen from these plots that in general, the maximum heat transfer rate occurs at S_y=12 mm (S_y/d=1.2). The fact that the increase in C/H effects the heat transfer rate a decrease and is obvious as predicted by others (Jubran *et al.*, 1993). The maximum heat transfer occurs at lower inter fin distance in the span-wise and stream-wise direction at the same time, the pressure drop across the pin-fin array was so large that the flow was obstructed by the fins and however, optimum point is not seen in the covered range of spacing to take as the optimum spacing for the array.

Effect of Surface Area on Heat Transfer: Enhancement of heat transfer is normally achieved by many techniques and here it is attained by minimising inter-fin-distance in both directions. The effect of surface area on the pin fin performance is shown in Fig. 7. The surface area is varied by accommodating more number of pin-fins. Figure 7 indicates only for C/H=0.0. A similar observation is reported by Tahat *et al.* (2000) and Kadir *et al.* (2001).

Effect of Clearance Ratio on Heat Transfer: The effect of clearance ratio on heat transfer rate for the range of Reynolds number covered in the experiments is illustrated in the Fig. 8 It appears that the heat transfer rate increases monotonically with Reynolds number for all C/H values. Also, that for the smallest value of C/H the largest heat transfer is observed and this enhancement of heat transfer is achieved by increasing the compactness of the heat exchanger (ie; restricting the free flow of air). This trend is reported earlier investigators (Jubran *et al.,* 1993; Tahat *et al.,* 2000; Sara, 2003). The effect of fin arrangements on heat transfer is also given in Fig. 9.

Effect of Fin Number and Flow Rate on Pressure Drop: It is not uncommon that the flow resistance is an important aspect in heat transfer in order to have minimum pumping power as a constraint. The variation of pressure drop for various flow rate as a plot of friction factor versus Reynolds number is illustrated in Fig. 10 for C/H=0.0. A little variation in friction factor on S_y/d change is accomplished in in-line arrangement as compared to that of large variant on S_x/d and as well in case of staggered arrangement which was reported by Jubran *et al.* (1993) and Tahat *et al.* (2000).

Heat Transfer Results with other Correlations: It is customary to present heat transfer data in terms of Nusselt number variation against Reynolds number. For the present work one such plot is given in Fig. 11. It is clear that the Nusselt number arrived at are increasing with Reynolds number. The present results are closer to the result of Kadir *et al.* (2001) and much lesser than the results of Jubran *et al*. (1993) Tahat *et al*. (2000) within the range of experimental conditions.

Experimental Uncertainties: In the present study, additional care was taken in constructing the experimental test rig and also in measuring the temperatures and electrical power supplied. Individual given parameters were limited to possible accuracies in each case. For example, fin dimension ±0.01 mm (i.e; 0.1%), temperature ±0.2 ^{o}C, watt meter ±1.0 W and manometer ±0.1 mm.

Conclusions

Experimental investigation on fin-pin assembly was carried out in a wind tunnel setup to study the heat transfer and pressure drop characteristics. The in-line pin-fin array significantly enhanced heat transfer as a result of increased heat transfer surface area and turbulence at the expense of higher pressure drop in the wind tunnel. The average Nusselt number increased with decreasing clearance ratio and inter-fin distance ratio. The average Nusselt number increased with increasing Reynolds number. For a given Reynolds number, the pin-fin array with smaller inter fin distance gives higher performance than those at higher inter fin distances.

The friction factor increased with decreasing clearance ratio and inter-fin distance ratio. A further study is required with staggered fin arrays in order to make some more specific conclusions.

Nomenclature

A: area, m^2; C: clearance between fin tip and the roof, mm; C_p: specific heat of air, J/kg K; d: diameter of the pin-fin; mm; f: friction factor; G: mass flux, kg/m^2 s; H: height of the pin-fin, mm; k:thermal conductivity, W/m K; L: length of the base plate, mm; m: mass flow rate of air, kg/s; N: number of pin-fin; Nu: Nusselt number; Q: heat transfer rate, W; Re: Reynolds number; S: spacing, mm; T: temperature, ^{o}C; T: temperature, K; W: width of the base plate, mm; x, y, z: set of Cartesian coordinates; Δp: overall pressure drop along the array, N/m^2; μ: dynamic viscosity, N s/m^2; ρ: density, kg/m^3

Subscripts

A: air; b: base plate; conv: convection; ff: free flow; in: inlet condition; loss: loss of heat; out: outlet condition; rad: radiation; tot: total; ts: total surface; x, y:directions.

References

1. Armstrong J and Winstanley DA (1988) A review of staggered array pin fin heat transfer for turbine cooling applications. *ASME J. Turbomach*. 110, 94-103.
2. Babus′Hag RF, Akintude K and Probert SD (1995) Thermal performance of a pin-fin assembly. *International of Journal of Heat and Fluid Flow*. 16(1), 50-55.
3. Chyu MK, Hsing YC and Natarajan V (1998) Convective heat transfer of cubic fin arrays in a narrow channel, ASME *J. Turbomachinery.* 120, 362-367.
4. Jurban BA, Hamdan MA and Abdullah RM (1993) Enhanced heat transfer, missing pin, and optimization for cylindrical pin fin arrays. *ASME J. Heat transfer.* 115, 576-583.
5. Kadir B Ugur A and Sinan Y (2001) Heat transfer and friction correlations and thermal performance analysis for a finned surface. *Energy Conversion & Management.* 42, 1071-1083.
6. Naik S, Probert SD and Shilston MJ (1987) Forced convective steady-state heat transfers from shrouded vertical fin arrays aligned parallel to an undisturbed air-stream. *Appl. Energy.* 26, 137-158.
7. Reynell M (1990) Advanced thermal analysis of packaged electronic systems using computational fluid dynamics techniques. *Computer Aided Eng. J.* 7, 104-106.
8. Sara ON (2003) Performance analysis of rectangular ducts with staggered square pin fins. *Energy Conversion & Management*. 44, 1787-1803.
9. Tahat MA, Babus′Hag RF and Probert SD (1990) Forced steady-state convection from pin-fin arrays. *Appl. Energy*. 48, 335-351.
10. Tahat MA, Babus′Hag RF and Probert SD (1994) Forced steady state convections from pin-fin arrays. *Appl. Energy.* 48, 335-351.
11. Tahat M, Kodah ZH, Jarrah BA and Probert SD (2000) Heat transfer from pin-fin arrays experiencing forced convection. *Appl. Energy.* 67, 419-442.
12. Zukauskas A (1987) Convective heat transfer in cross flow. In: Hand book of single-phase convective heat transfer. Wiley, NY.