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Experimental Analysis of Convective Heat Transfer Performance of Pin Fin Heat Exchangers

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Abstract

The research work summarized in this thesis presents a combined experimental & analytical investigation of various aspects of single-phase convective heat transfer enhancement by the use of pin fins is presented. After a brief review of the basic methods used to enhance the heat transfer by simultaneous increase of heat transfer surface area as well as the heat transfer coefficient, a simple Experimental method to assess the heat transfer enhancement is presented. The method is demonstrated on pin fins as elements for the heat transfer enhancement, experimental investigations of a double-pipe pin fin heat exchanger were carried out. The order of the magnitude of heat transfer enhancement obtained experimentally was similar to that obtained analytically. The heat transfer and pressure drop results for the pin fin heat ex-changer were compared with the results for a smooth-pipe heat exchanger. It was found that by a direct comparison of *Nu* and *Eu*, no conclusion regarding the relative performances could be made. This is because the dimensionless variables are introduced for the scaling of heat transfer and pressure drop results. *St/d* ratio is varied from *1.8 to 4.5*, where St is Spacing between pins and *d* is dia. of pin. Heat transfer depends on numbers of Pins and spacing between pins. In this study Heat transfer enhancement is found to be about *5-15%.*

Keywords:*Pin Fin Heat Exchanger, Nusselt Number, Eulers Number, Convection*

1. Introduction

Heat exchangers are widely used in various industrial, transportation, or domestic applications such as thermal power plants, means of transport, heating and air conditioning systems, electronic equipment and space vehicles. In all these applications, improvements in the efficiency of heat exchangers can lead to substantial cost, space and materials savings. Therefore, considerable research work has been done in the past to seek effective ways to increase the efficiency of heat exchangers. The referred investigations include the selection of working fluids with high thermal conductivity, selection of their flow arrangement and high effective heat transfer surfaces made from high-conductivity materials. For both single-phase and two-phase heat transfer, effective heat transfer enhancement techniques have been reported. However, in the present work only the single-phase forced convection enhancement techniques have been considered. Over 8000 technical papers and reports have been published in various bibliographic reports, reviews, monographs and edited texts with yearly growth tendencies (Bergles, 1985; Manglick, 2003). The heat transfer enhancement methods reported in publications be systemized in many forms but primarily they may be grouped as passive and active enhancement methods.

Over the years pin fins have been used as elements for effective heat transfer in various applications where space and weight are important constraints such as in the power plant industry for cooling of gas turbine blades, in the electronics industry for cooling of electronic components and recently also in the hot water boilers of central heating systems. Thus the thermal and fluid dynamic characteristics of flow over pin fins have been the interest of numerous investigators.

2. Literature Survey

[1]Norris and Spofford (1942) derived the first basic heat transfer and flow-friction data for pin fin surfaces. The experiments were carried out with the aim of the derivation of basic parameters of forced convection heat transfer for continuous, corrugated, strip and pin fins. By use of the perimeter as the length scale, they could approximately represent the heat transfer data with a single curve for a single plane, single cylinder, various strip fins and pin fins. They also included in their tests an in-line pin fin arrangement with pin diameters of 0.5 mm and 1 mm and a pin length of ~ 19 mm.
[2]Kays (1955) performed probably the most extensive study of pin fins as elements for heat transfer enhancement. He presented test data for four in-line pin arrangements and one staggered arrangement. It was demonstrated that owing to a high area to perimeter ratio, pin fins provide one method for obtaining very high heat transfer coefficients while at the same time maintaining high fin effectiveness. He concluded that despite high friction factors of pin fin surfaces, it is possible to design heat exchangers that are competitive, from volume and weight points of view, with heat exchangers having continuous or

louvered fins. With some problems which may be encountered with pin fin heat exchangers such as the pin vibration and the tendency of the flow to become completely laminar if the pins are in an in-line arrangement and too close to each other.
[3]Theoclitus (1966), who performed a limited parametric study of pin fins with an in-line arrangement. He investigated nine different geometries of in-line pin fins with circular cross-section with length to diameter ratios in the range $4 \leq l/d \leq 12$. Further, he investigated the acoustic and vibrational characteristics of the flow over the pins and concluded that these phenomena are basically influenced by the fluid velocity and heat exchanger configuration. In general, the average heat transfer rates reported by him were lower for short than for longer cylinders. Some papers related to pin fins deal with the influence of the clearance between the fin tip and the upper surface of the channel in the thermal and pressure drop characteristics of the pin fin assembly. This kind of flow over pin fins is characteristic of cooling of various electronic components where the pins might not occupy the whole surface between the bottom and covering plate.

[4]Sparrow and Ramsey (1978) reported excellent experimental work on the influence of tip clearance for a staggered wall-attached array of cylinders. They obtained data on heat transfer coefficients by applying the analogy between heat and mass transfer via the naphthalene sublimation technique. They found that the heat transfer coefficient increases moderately as the length of the cylinder increases and the tip clearance between the pin and the shroud decreases. On the other hand, the array pressure drop increases markedly with increasing cylinder length. This behaviour was explained with inter-cylinder velocities for short pins which are less than the mean velocity, whereas for taller cylinders the inter-cylinder velocities tend to approach the mean value.
[5]Sara et al. (2001) and by Dogruoz et al. (2002) during their investigations of the heat transfer and pressure drop characteristics of a square pin fin in a rectangular channel with tip clearance. Moores and Joshi (2003) discovered for a small amount of clearance (<10%) they reported higher mean heat transfer and lower overall pressure drop compared with the case with no tip clearance. The authors attributed the heat transfer increase to the additional surface area that is exposed to the fluid when clearance is provided. The observations made on the effect of the tip clearance on the thermal and hydraulic characteristics of pin fin arrays is important for the application of pin fins in heat sinks with the fan situated on one of the sink sides. The fan fixed opposite to the heat sink is another fan-heat sink arrangement used widely in the electronics industry.

[6]Maveety and Jung (2000) investigated the increase in thermal performance of heat sinks due to air impingement in the sink has been. They performed computational and experimental estimations of the heat transfer from a heat sink with air impingement cooling and observed a large pressure gradient within the heat sink centerline channels that induces better mixing of the air and results in a higher heat transfer coefficient compared with channels away from the heat sink centerline. Further, they found the cooling performance to be greatly affected by minor changes in fin dimensions.

[7]Issa and Ortega (2002), in their investigations of heat transfer by air impingement in a square pin fin array reported a weak dependence of sink thermal resistance on pin length. They hypothesized that these phenomena occur due to much higher heat transfer coefficients in cross-flow than parallel to the pin axis. Further, the authors observed that the pressure drop increases with increase in pin density and pin hydraulic diameter and decreases with increase in pin length. Furthermore, they noted a little dependence of pressure loss coefficient on *Re* and a decrease in thermal resistance with increase in *Re*, pin density and pin diameter.

[8]Babus'Haq et al. (1995) studied the influences of the pin fin distance and the pin fin material on thermal performance of the inlineand staggered pin fin assembly. They determined the optimal fin distance in the streamwise direction for a uniform spanwise distance and noted that the optimal spacing increases as the thermal conductivity of the pin fin material increases. Further, they noted that the overall pressure drop for all tested configurations increases steadily with increasing mean inlet velocity and with decreasing uniform pin fin spacing. There have also been some contributions on the influence of the cross-sectional shape of the fin on their thermal and fluid dynamic characteristics.

[9]Li et al. (1996) have investigated the convective heat transfer and pressure drop for arrays of long drop-shaped cylinders in cross flow. They showed that mean heat transfer coefficients of drop-shaped cylinder arrays are about 8-29% higher than those of the corresponding circular cylinder arrays, and the pressure drop of the former is only about half of the latter.

[10]Chen et al. (1997) also carried out experiments on heat transfer and pressure drop coefficients in a rectangular duct with drop-shaped pin fins. They reported Nusselt numbers for channel with drop-shaped pin fins which are slightly higher than those of circular ones. However, they found that the pressure drop of drop-shaped pin fins is 42-51% less than that of round ones. In another study,
[11]Li et al. (1998) observed heat transfer and pressure drop characteristics of elliptical pin fins in a rectangular channel. They measured higher heat transfer coefficients for elliptical fins than those measured by other workers for circular pin fins. Furthermore, they reported a lower pressure drop for elliptical pins in the range of 44-58%.

[11]Camci and Uzol (2001) compared experimentally the heat transfer and pressure drop of two geometries of elliptical pin fins with circular pins. By using liquid

crystal thermography, they showed that the endwall heat transfer enhancement of the circular pin fin array is about 25-30% higher than that of the elliptical pin fin array. However, they observed that the circular pins induce 120-185% more pressure losses than elliptical pins and heat transfer enhancement on the elliptical fin itself due to the increased wetted surface area, which is 35-85% more than that in the circular fins. Hence they found that elliptical pins offer a very attractive alternative to circular pin fins for turbine blade cooling purposes. In general, the intensity of heat transfer from the pin wall differs from that from the endwall. However, observations have shown that for short pin fins, such as those used in turbine blade cooling, this difference is small and sometimes may be neglected. Some papers dealing with pin fin array performance in blade cooling also report such differences.

[12]VanFossen (1982) investigated the heat transfer from short pin fin arrays taking into account the heat transfer from the pin fin surfaces and from endwalls. He found that heat transfer from short pins with a length to diameter ratio l/d = 2 and 0.5 was lower than those of long pins based on the available data for long pins. Further he found that heat transfer from the pin surface was 35% higher than from the end walls. Similar results were reported by Yeh and Chuy (1998) in their experimental investigations of overall heat transfer and heat transfer coefficients of both the pins and end walls in a channel embedded with a staggered pin fin array. Thus for l/d = 1.0 they reported a 0-10% higher heat transfer coefficient of the pins than the end walls, whereas for l/d = 2 this difference was 10-20% in favor of pin fin heat transfer coefficient.

[13]Metzger et al. (1984), in their investigations of short pin fin arrays with l/d = 1, found the pin surface heat transfer coefficients to be approximately twice as large as those acting on the endwalls. The main objective of the investigations was the influence of the array orientation with respect to the mean flow direction on the heat transfer rates and the associated pressure losses for circular and oblong pin fin arrays. It was reported that with circular pin fins rotated two-thirds of the way towards a in-line orientation from a staggered orientation, a 9% increase in heat transfer and an 18% decrease in pressure loss were observed. For the oblong pins there was 20 % increase in heat transfer compared with the circular pin fin arrays but this increase was offset by an approximately 100% increase in pressure loss.

In a recent study,[14] Short et al. (2003a, b) conducted an experimental study to derive basic heat transfer and friction data for cast pin fins for cold walls used in electronic applications. They carried out experiments on a staggered pin arrangement in a wide range of geometric configurations and Reynolds numbers. The data were derived for various stream wise and transverse pin spacings and for various pin length to pin diameter ratios. By use of correlation analysis, the authors provided corresponding equations for heat transfer factor j and friction factor f for the considered geometric parameters and Re.

[15] Sahiti (2005) investigated performance of pin fin for inline and staggered arrangement. Considerable enhancements of heat transfer were demonstrated by using small cylindrical pins on surface of heat exchanger pipe. With staggered pin fin arrangement heat transfer rate is more than in-line pin fin arrangement and considerable low velocity occurred in staggered arrangement. This study also gives the optimized spacing and diameter of pin fin.

While seeking heat transfer enhancement, apart from the utilization of various surface enhancement elements, efforts have also been made to select an optimal flow arrangement within the heat exchanger in order to obtain the maximum advantage for a given heat exchanger configuration. It may be noted that in a counter-flow arrangement of a heat exchanger, the outlets of fluid streams lie at opposite ends and this enables the outlet temperature of cold fluid to rise above the outlet temperature of the warm fluid. This is not possible in a parallel flow arrangement, and hence as far as the heat transfer rate is concerned, the counter-flow heat exchangers are superior to the parallel type [15]. Nevertheless, it is not possible to use a counter-flow heat exchanger in all practical situations. Therefore, a common arrangement in practice is a heat exchanger with a cross-flow arrangement and this is characterized by a better temperature distribution compared with the parallel-flow heat exchanger, but the temperature distribution in such a heat exchanger is not as good as in the case of a counter-flow arrangement.
Both effective surface enhancement elements and the optimal flow arrangement were employed during the experimental investigation of the pin fin heat exchanger described in the present work.

3. Estimation of heat transfer enhancements

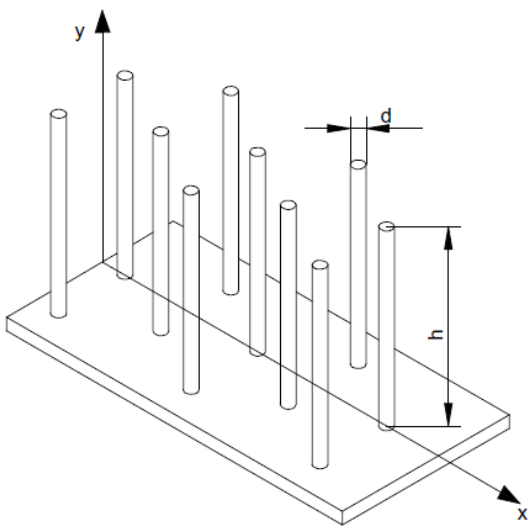

Fig.1: Heat transfer area covered by pins

First for some approximate theoretical considerations of the heat transfer from a surface, the molecularly conducted heat from a plate without any heat transfer enhancement element (bare plate) can be given as

$$q_b = -k_a \left(\frac{dT}{dy}\right)_{a,y=0} \qquad (1)$$

Where k_a thermal conductivity of the air is, $\left(\frac{dT}{dy}\right)_{a,y=0}$ is the temperature gradient at the air side of the wall–air interface and q_b is the heat transfer rate per unit area of bare plate.

When elements for heat transfer augmentation are placed on the surface to cover an areaφA_b, the area for the heat transfer from the solid surface to the fluid

(Air in the present paper) decreases to (1-φ)A_b, where A_b denotes the surface area of the bare plate. Hence, to estimate the heat transfer enhancement by augmentation elements, we may write

$$q_s = q_b + q_{bp}$$
$$= -(1-\varphi)k_a \left(\frac{dT}{dy}\right)_{a,y=o} - \varphi k_s \left(\frac{dT}{dy}\right)_{a,y=o} \qquad (2)$$

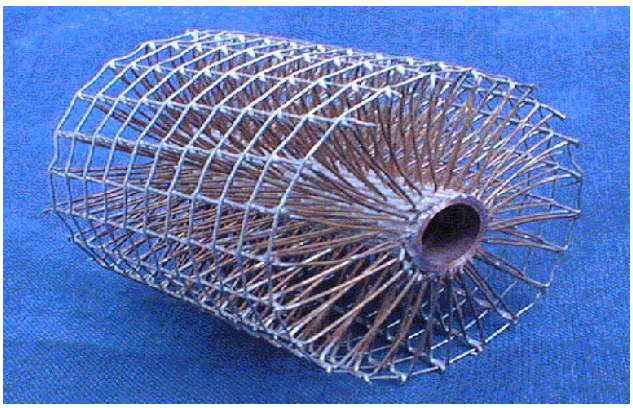

Fig.2: Core part of the heat exchanger.

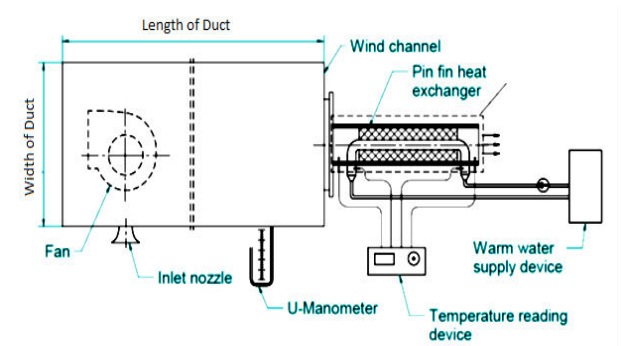

Fig.3: Schematic of the pin fin heat exchanger measurement setup

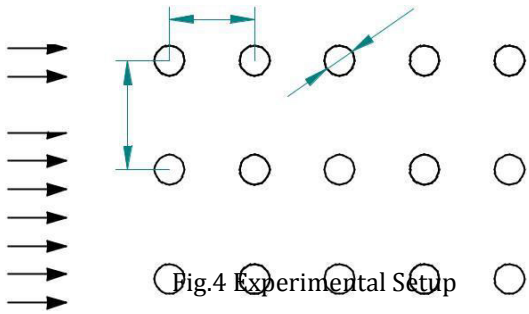

Fig.4 Experimental Setup

1. Calming tube
2. Test section
3. Blower
4. Manometer
5. Venturimeter
6. Flow control valve
7. Control panel
8. Tempereture indicator
9. Dimmer stat
10. Delivery pipe

Uu∞ *S_l d*

Fig.5: Basic pin fin arrangements: in-line arrangement

Analogous to the tube banks, it can be expected that the pin arrangement will have a strong influence on the performance characteristics of flat-tube and pin fin heat exchangers. Keeping this in consideration, numerical computations were performed for both in-line and staggered pin arrangements, as depicted in Fig.5.

4. Experimentation

Experimentation is carried put on following configurations. Variable dimensions of S_L *and d* are given in table 1. Heat is supplied with the help of Heater and is controlled by control panel as shown in fig. 6. Heat Supplied is *Q=750W.* And actual Qs is calculated by *Eq no.2.*

Fig.6:Control Panel

Table-1 Specifications of Samples

Sr. No.	Specimen	D	L	S_L(Longitudinal)	St(Transverse)
1	Case-I	6mm	500 mm	11mm	11mm
2	Case-II	6mm	500 mm	15mm	15mm
3	Case-III	5mm	500 mm	18mm	18mm
4	Case-IV	5mm	500 mm	24mm	24mm

Here $S_L$and S_t are the same values for given cases.

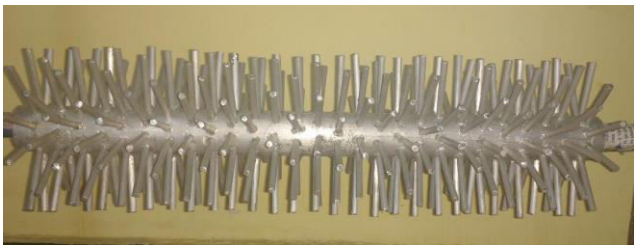

Fig.7: Actual Pin Fin arrangement.

5. Results and Discussion.

The quantitative heat transfer and pressure drop characteristics of pin fin heat ex-changers, are presented in terms of significant non-dimensional numbers in Fig.8 and Fig.9.

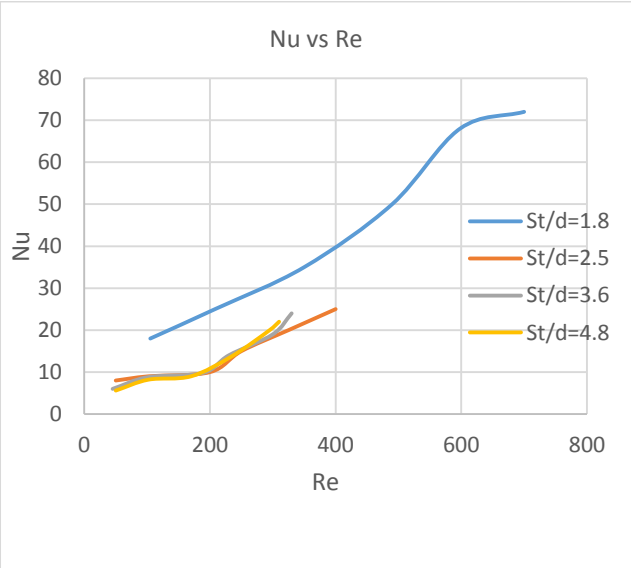

Fig.8:*Nu* of the pin fin heat exchanger, scattered pin arrangement

Fig.8 clearly suggests that with a reduction in transverse pin spacing, the Nu increases over the entire range of Re. The increase in Nu can be attributed to the increase in the wetted area, for smaller pin spacing's in the transverse direction. A similar behavior of Nu is observed also for variable

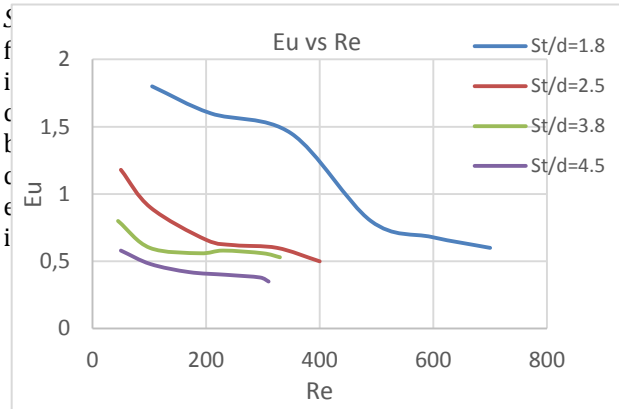

Fig.9:-*Eu* of pin fin heat exchanger, Scattered pin arrangement

The *Eu* variations presented in Fig.9 confirm such propositions.

$$Eu = \frac{2\Delta p}{\rho_a u_a^2 N} \qquad (3)$$

Where Pressure drop Δp is calculated by Manometers as shown in Fig.4 whereas mean air velocity of flow recorded by manometer. N is number of pin fins in a row.

However, Figs.9 show that the behavior of *Eu* in response to changes in Re is quite similar to the *Eu*behavior of the corresponding staggered arrangement. A completely different behavior can be observed regarding the variation of *Eu* with S_L/dratios. Whereas in the corresponding staggered arrangement Eu increases with a decrease in S_L/dratio, in the in-line arrangement the decrease in S_L/dratio is associated with a decrease in *Eu*. Keeping in mind that employing the same configuration an increase of Nu with decreases in S_L/dratio could be observed (Fig.8), this specific pin arrangement is likely to result in a very high heat exchanger performance.

6. Conclusions

1. Pin fin heat transfer surfaces are characterized by much higher heat transfer rates.
2. Heat transfer rate increases with number of Pin fins. But influenced by fin spacing and fin dia.
3. Although dimensionless parameters for heat transfer (Nu) and pressure drop (Δp) are very important for the scaling of the results of the heat exchanger with similar geometries, comparison of the performances of different heat transfer surfaces. Heat transfer and pressure drop com-pared with enhanced surfaces.
4. For performance evaluation of the heat transfer surfaces by consideration of both heat transfer and pressure drop.

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