Experimental Investigation of Heat Transfer Intensification of Pin Fins under Forced Convection (A Review)

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Abstract—*Recent development era in technology has huge requirement of high performance lightweight, and compact heat transfer equipment. To accomplish this demand fins are widely used as effective elements for heat transfer enhancement. One of the commonly used heat exchanger fins is the pin fin which offers an economical and trouble free solution in many situations. This is more important in cooling of air conditioning equipment, thermal power plants, gas turbine blade, aerospace industry, combustion chamber liners, and biomedical devices, electrical and electronic component. Therefore now a day's industries are utilizing thermal system with pin fins and analyse the various influencing parameters on performance of pin fin of different geometry under force convection. The turbulence occurred due to these techniques are good enough to enhance heat transfer rate. This article is focused on comprehensive review of work carried out in this technology.*

Keywords— Force convection, Heat exchanger, Heat transfer enhancement, Pin fin.

I. INTRODUCTION

Pin-fins installation on a heat exchanger surface can increase the surface area of heat transfer and cause turbulent mixing of flow, subsequently enhancing the heat transfer performance and protecting the reliability and life of devices [1, 5]. The development of high-performance thermal systems has been stimulated in many fields of new technologies. Conventional heat transfer devices have to be substantially improved to answer the needs of systems from the micro scale to large power plants. In the latter case, it is made possible by changing the structure of the heat exchanger or the properties of the heat exchange surface. The use of extended surfaces is a thermal management technique used to augment heat transfer by increasing the available surface area and therefore the total heat dissipation [1-8].

List of Symbol

A_s	Heat transfer surface area
D_h	Channel hydraulic diameter
$\hbar_{av}$	Average Convective heat transfer coefficient
k_{air}	Thermal conductivity of air
L_t	Length of test section
$N\bar{u}$	Nusselt number
Re	Reynolds number
T_s	Average Surface Temperature
f	Friction factor
L_t	Length of the test section
T_{in}	Inlet temperature
T_{out}	Outlet temperature
T_m	Bulk mean temperature
ΔP	Pressure drop across the test section
Q_{elec}	Electrical heat transfer
Q_{cond}	Conduction heat loss
Q_{conv}	Convection heat loss
Q_{rad}	Radiation heat loss
V	Mean velocity of air
v	Kinematics viscosity of air

In this perspective, convective heat transfer can be enhanced in several ways, by using active, passive or compound augmentation techniques. For active methods, some external power is needed to achieve the attempted heat transfer enhancement, usually from flowing fluid to heat exchanger wall. If the power is taken from the actual fluid flow, it is possible to drive with this flow instabilities to yield an increased heat transfer coefficient.

To increase the heat transfer coefficient, a common passive method is to employ turbulence promoters of different geometry. The passive enhancements methods are those which do not require external power to sustain the enhancements characteristics. Examples of passive enhancing methods are: (a) Treated surfaces (b) Rough surfaces (c) Extended surfaces (d) Displaced enhancement devices (e) Swirl flow devices (f) Coiled tubes (g) Surface tension devices (h) Additives for fluids and many others. Also compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger. When any two or more of these techniques are employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by either of them when used individually, is termed as compound enhancement [2].

II. LITERATURE REVIEW

Many researchers investigate the performance parameters of pin-fin and compared it with different profiles of the fins and suggested that, pin-fin elements are best suited for enhancement of heat transfer. Following are the various literatures suggested the best suitable geometry and parameters.

M. Tahat *et al*. [1] investigated experimentally, steady-state heat-transfers from pin-fin arrays for staggered and in-line arrangements of the pin fins, which were orthogonal to the mean air-flow. For the applied conditions, the optimal spacing's of the fins in the span-wise and stream-wise directions have been determined. The dependences of the Nusselt number upon the Reynolds number and pin-fin pitch (in both directions) have been deduced. These steady-state design data correlations facilitate predicting the performances of aligned and staggered pin fins, when used as arrays in heat exchangers and hence optimal designs to be chosen.

Kadir Bilen et al [2] investigated heat transfer and friction loss characteristics experimentally. Installing a finned surface kept at constant temperature 45 ℃ in rectangular channel through which air is used as working fluid. The fins are attached either inline or staggered arrangement. Parameters for study were chosen as Reynolds number (3700-30000), depending on hydraulic diameter, the distance between fins in flow direction (S_y/D= 1.69-4.41) and fin arrangement. For both fin arrangements, it was found that increasing Reynolds number increased Nusselt number, the heat transfer occurred at S_y/D=2.94. Also thermal performance of the system were also determined and compared with respect to heat transfer from the same surface without fins. With the staggered array, a heat transfer enhancement up to 33% at constant pumping power was achieved.

Giovanni Tanda [3] studied Heat transfer and pressure drop experiments were performed for a rectangular channel equipped with arrays of diamond-shaped elements. Both in-line and staggered fin arrays were considered, for values of the longitudinal and transverse spacing relative to the diamond side, from 4 to 8 and from 4 to 8.5, respectively. The height-to-side ratio of the diamonds was 4. Thermal performance comparisons with data for a rectangular channel without fins showed that the presence of the diamond-shaped elements enhanced heat transfer by a factor of up to 4.4 for equal mass flow rate and by a factor of up to 1.65 for equal pumping power.

O. N. Sara [4] presents the heat transfer and friction characteristics and performance analysis of convective heat transfer through a rectangular channel with square cross-section pin fins attached over a flat surface. The pin fins were arranged in a staggered manner. Various clearance ratios (C/H) and inter fin distance ratios S_x/D were used. The performance analysis was made under a constant pumping power constraint. The experimental results showed that the use of square cross-section pin fins may lead to an advantage on the basis of heat transfer enhancement. For higher thermal performance, lower inter fin distance ratio and clearance ratio and comparatively lower Reynolds numbers should be preferred for the staggered arrangement. The results of the staggered configurations were also compared with the results of the inline arrangement.

N. Sahiti *et al.* [5] suggested that, considerable enhancements by using small cylindrical pins on surfaces of heat exchangers. High performance of the wavy, strip or louvered fins has been investigated and optimized for a long time, resulting in very compact heat exchangers. A pin arrangement is considered optimal if the associated pressure drop does not exceed the benefits of enhanced heat transfer rate resulting from high population of the base plate with pins. Otherwise the pin to height diameter ratio influences the pin efficiency and the heat transfer rate of pins. However, it is known that for pin fins a good compromise between the pin efficiency and pin heat transfer rate is achieved if the ratio of pin height to pin diameter is of order of 15.

N. Sahiti *et al.* [6] suggested that, best suitable pin fin arrays with highest heat transfer rate. In order to check how the form of pin cross-section influences the pressure drop and heat transfer capabilities, six forms of pin cross-section in both arrangements staggered and inline were numerically investigated. The results of the simulation of six different pin cross-sections show that for both comparison criteria of the staggered arrangement the elliptic profile performs better than all other pin cross sections. One can conclude that for practical application of pin fins as heat transfer enhancement elements, the elliptic profile offers the highest heat transfer rate for a given base area and for the same energy input.

Tzer-Ming Jeng et al. [7] studied the pressure drop and heat transfer of a square pin-fin in a rectangular channel. The variable parameters are the relative longitudinal pitch (X_L = 1.5, 2, 2.8), the relative transverse pitch (X_T = 1.5, 2, 2.8) and the arrangement (inline or staggered). The result indicates that the in-line square pin-fin array has smaller pressure drop than the in-line circular pin-fin array at high X_T (X_T = 2.0 or 2.8) but an equivalent (or even slightly higher) pressure drop at low X_T (such as X_T = 1.5). Additionally, the staggered square pin fin array has the largest pressure drop of the three pin fin arrays (in-line circular pin-fins, in-line square pin-fins and staggered square pin-fins). Most in-line square pin-fin arrays have poorer heat transfer than an in-line circular pin-fin array, but a few, as when X_L = 2.8, exhibit excellent heat transfer at high Reynolds number. For instance, when X_L = 2.8, X_T = 1.5. and Re_d=12500, the fin Nusselt number of square pin fins is around 40% higher than that of circular pin fins. Moreover, when Re_d exceeds 6000, the Nusselt number of the staggered square pin-fin array generally exceeds that of the in-line circular pin-fin array. When X_T = 1.5, X_L = 2.8 and Re_d =12,500, the heat transfer of the staggered square pin-fin array is around 20% higher than that of the in-line circular pin-fin array. The optimal inter-fin pitches are determined by the largest Nusselt number at a given pumping power. The optimal inter-fin pitches of in-line square pin fin arrays are X_T = 2 and X_L = 1.5, its Nu_d is around 20% higher than that of the in-line circular pin-fin array. However, the staggered square pin-fin array performs best at X_T = 1.5 and X_L = 1.5. Furthermore, at X_T = 1.5 and X_L = 1.5 and when $f_d Re_d^3$ exceeds 2.5 x10^{10}, the Nu_d of the staggered square pin-fin array exceeds that of the in-line circular pin fin array; and when $f_d Re_d^3$ =2.0 x 10^{11}, the Nu_d of the staggered square pin fin array is 25% higher than that of the in-line circular pin-fin array.

Bayram Sahin *et al.* [8] studied the heat transfer enhancement and the corresponding pressure drop over a flat surface equipped with square cross-sectional perforated pin fins in a rectangular channel. In this paper experimental range of some parameter like inter fin distance ratio (Sy/D) 1.208,1.524, 1.944,and 3.417, clearance ratio (C/H) 0,0.33, and 1 , Reynolds number 13500-42000. Enhancement efficiencies depending on the clearance ratio and inter fin spacing ratio and its varied between 1.1 to 1.9. The effects of flow geometrical parameters on the heat transfer and friction factors were determined, and the enhancement efficiency correlations have been obtained. Bayram Sahin et al. analyse that the maximum heat transfer rate was observed at 42000 Reynolds number, 3.417 Sy/D and 50 mm fin height.

Isak Kotcioglu *et al*. [9] investigated that, convective heat transfer and pressure drop in a cross flow heat exchanger with hexagonal, square and circular (HSC) pin-fin arrays experimentally. The variable parameters are the relative longitudinal pitch (S_L/D=2, 2.8, 3.5), and the relative transverse pitch was kept constant at S_T/D=2. The performances of all pin-fins were compared with each other. The experimental results showed that the use of hexagonal pin-fins, compared to the square and circular pin-fins, can lead to an advantage in terms of heat transfer enhancement. The optimal inter-fin pitches are provided based on the largest Nusselt number under the same pumping power, while the optimal inter-fin pitches of hexagonal pin-fins are S_T/D=2 and S_L/D=2.8.

M. G. Mousa [10] investigated that, forced convective heat transfer and friction factor for air flowing inside rectangular horizontal duct over a set of pin–fins experimentally studied under uniform heat flux. Experiments are conducted with air subjected to a magnetic body force through a magnetic gradient field. Experiments are carried out for two different fin geometries. The obtained experimental results show that, both Nusselt number and pressure drop increases in case of using pin–fin in a staggered arrangement. Higher rates of heat transfer and pressure drop are obtained from the duct provided by a pin–fin set compared to flow in a plain duct under similar conditions. The performance factor of the pin–fin inserts for both inline and staggered is evaluated.

Swee-Boon Chin *et al.* [11] investigates experimentally and numerically the use of staggered perforated pin fins to enhance the rate of heat transfer in these devices. In particular, the effects of the number of perforations and the diameter of perforation on each pin are studied. The results show that the Nusselt number for the perforated pins is 45 % higher than that for the conventional solid pins and it increases with the number of perforation. Pressure drop with perforated pins is also reduced by 18 % when compared with that for solid pins. Perforations produce re-circulations in the x–y as well as the x–z planes downstream of the pins which effectively increase convective heat transfer. However, thermal dissipation decreases significantly when the ratio of pin diameter to perforation diameter exceeds 0.375. This is due to both a reduction in the number of perforation per pin and the decrease in the axial heat conduction along the pin.

Jaideep Pandit *et al.* [12] investigated heat transfer enhancement methods in order to improve the performance of thermoelectric power generators in automotive applications. Specifically, pin fin geometries were examined in a fixed configuration array at pin heights that ranged between 15% and 50% of the channel height. Data from these experiments show that maximum heat transfer is achieved with diamond shaped pin fin as compared to other shapes, when the pin fin to channel height ratio is 0.5. The pressure drop caused by the pin fins is small since they do not extend fully into the flow path and hence could not be measured. Thus these pin fins can be used without an adverse effect on the vehicle in terms of back pressure in the exhaust tail pipe.

III. METHODOLOGY

The convective heat transfer rate Q_{conv} from electrically heated test surface is evaluated by using [1, 8-10]:

$$Q_{conv} = Q_{elect} - Q_{cond} - Q_{rad} \tag{1}$$

Where, Q indicates the heat transfer rate in which subscripts conv, elect, cond and rad denote convection, electrical, conduction and radiation, respectively. The electrical heat input is obtained from the electrical voltage and current supplied to the surface. In similar studies, investigators [8-11] reported that total radiative heat loss from a similar test surface would be about 0.5% of the total electrical heat input. The conductive heat losses through the sidewalls can be neglected in comparison to those through the bottom surface of the test section. Using these findings, together with the fact that the two sidewalls and the top wall of the test section is well insulated therefore one could assume with some confidence that the last two terms of Eq. (1) may be ignored.

The heat transfer from test section by convection can be expressed as [1, 8-10]:

$$Q_{conv} = h_{av} A_s [T_s - (\frac{Tout + Tin}{2})] \tag{2}$$

Hence, the average convective heat transfer coefficient can be expressed as [1, 8-10]:

$$h_{av} = \frac{Q_{conv}}{A_s [T_s - (\frac{T_{out} + T_{in}}{2})} \tag{3}$$

Either the projected or the total area of the test section can be treated as the heat transfer area in the calculations. The total area is equal to the sum of the projected area and surface area contribution from the pin fins [4, 8-9]. These two areas can be related to each other by:

Total area = Projected area + Total surface area contribution from the blocks (4)

The dimensionless groups are calculated as follows:

Nusselt Number: This parameter is equal to the dimensionless temperature gradient at the surface, and it provides a measure of the convection heat transfer occurring at surface. It is computed by using equation [8-9]:

$$Nu = \frac{h_{av} D_h}{k_{air}} \tag{5}$$

Reynolds Number: It is ratio of inertia force to viscous force. It is computed by using the equation [8-9]:

$$\mathrm{Re} = \frac{D_h V}{v} \tag{6}$$

Friction Factor: It defined as dimensionless pressure drop for internal flow. The friction factor can be calculated as follows [8-9]:

$$f = \frac{\Delta P}{(\frac{Lt}{Dh})(\rho \frac{V^2}{2})} \tag{7}$$

In all calculations, the values of thermo physical properties of air were obtained at the bulk mean temperature [8-9], which is

$$T_m = \frac{T_{in} + T_{out}}{2} \tag{8}$$

The system of pin fin will be tested under forced convection for different combinations of hallow cylindrical pin fins in inline arrangement with different clearance ratio (C/H) and different flow velocities of air. The number of sets of experiment obtained by using factorial design of experiment, and evaluation of output parameter will be done by using above equations. Further validation of experimental result and optimization will be done by using ANOVA techniques.

IV. CONCLUSION

Intensification of heat transfer is very crucial in many industrial purposes. Recently a great challenge for design of heat exchanger equipment is very active heat removal from heated surface reducing material weight as well as cost. Recent techniques fabrication of heat exchangers is required to modernize of heat exchanger to exchange maximum amount of heat between extended surfaces and ambient liquid. In case of perforated fins there is higher contact area with fluid in comparison with solid fins result in average friction drag force for perforated fins compare to solid fins and also increase by adding perforations. There is drop in temperature from fin base to fin top surface increases with number of perforations. Increasing perforations, weight reduction is remarkable and this economical benefit is along with more enhancement heat transfer rate.

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