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STUDY OF ENHANCED FORCED CONVECTION HEAT TRANSFER FROM A FLAT PLATE BY SOLID AND DRILLED FINS UNDER DIFFERENT RELATIVE HUMIDITY CONDITION

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ABSTRACT
This study represents a synergistic integration of numerical and experimental study on enhanced forced heat transfer characteristics from a horizontal flat plate. The enhancement in heat transfer is provided through the use of solid and drilled fins of rectangular profile. This paper details the changes in the Nusselt number and heat transfer rates due to the use of number of equally spaced high conductivity drilled fins placed at the heater surface at different humid condition and compare these performances with that of solid fins under same condition. The results obtained are in good agreement with the available results in the literatures and its been presented graphically.

Keywords: Forced convection, Nusselt Number, Permeable fin, Fin efficiency.

1. INTRODUCTION
A number of techniques have been developed to improve the heat transfer performance of forced convection and various devices such as roughness elements, turbulence promoters, interrupted fins, fin arrays and so on, have been utilized to enhance forced convection Heat Transfer. Extended surfaces or fins are used frequently to enhance heat transfer. The term extended surface is commonly used in reference to solid that experiences energy transfer by conduction within its boundaries, as well as energy transfer by convection between its boundaries and the surroundings.

Laminar forced convection across a heated plate is an important problem in heat transfer. Accurate knowledge of the forced convection from heated plate is important in many fields such as electrical conductors, refrigeration, heat exchangers, automobile and so on. Fins have always been used to as a passive method of enhancing the conduction heat transfer from heated vertical and horizontal plates.

The presence of solid fins has an effect on both the aerodynamics as well as the thermal characteristics of the flow. The fins tend to obstruct the airflow near the plate surface, thus reducing heat transfer from the plate to the surrounding fluid. On the other hand, the fins increase the heat transfer area resulting in an increase heat transfer from the cylinder to the surrounding fluid. The net result of these two opposing effects depend on the combination of fin geometry, number of fins, fin height etc.

Extensive researches were related to this subject with single-phase heat transfer. However there are many applications for fins involved with two-phase flow. For instances, in the case of cooling coils, when warm and humid air encounters a cold surface that is below the dew point temperature, condensation will take place and mass transfer occurs simultaneously with heat transfer. The performance of the extended surfaces in those cases is different from that of dry surfaces.

2. LITERATURE REVIEW
A number of previous researchers were engaged in the study of heat transfer enhancement of forced convection from a heated plate using rectangular fins. Both active and passive heat transfer-enhancing techniques have been proposed and tested.

Chu and Simons [3] reports that forced air cooling is still and has been the most widely used cooling technology and the principal advantage of cooling with is its ready availability and easy application.

The heat sink with parallel plate fins was the original designed optimized by Tuckerman and Pease [6]. A key feature of the model is that the channels were long and narrow in the direction of flow, that the flow is fully developed hydro dynamically and thermally.

The results of the relative humidity on the fully wet fin efficiency of extended surfaces are quite confusing. The methods are based on one-dimensional analysis by the researchers. Some researchers also revealed that the influence of relative humidity on the wet fin efficiency is also very small. Conversely, analysis by Rosario and Rahman [4] shows a significant decrease in fin efficiency when relative humidity is increased.

Bassam and Hijleh [1] performed an experimental study concerning performance of a rectangular fin in both dry and wet conditions. They found that the effect of dry bulb temperature on the wet fin efficiency is very small. Depending on the frontal velocities, efficiency was about 15 to 25 percent higher than the corresponding wet fin efficiency.

For fully wet condition, the effect of relative humidity on the fully wet fin efficiency is also small. The results agree well with some of the previous numerical calculations. However, some of the previous investigations show significant influence of inlet relative humidity on the fully wet fin efficiency. The main causes of the controversy between these calculated results from various investigators may be attributed to the formulation of the relation between the humidity ratio and fin temperature, and mixed with partially wet fin efficiency. However, they showed that the effect of dry fin efficiency is relatively small. In their works, it was also seen that for a partially wet surface, a significant drop of fin efficiency is observed as the inlet relative humidity is decreased.

Permeable fins resulted in much larger aerodynamic and thermal wakes, which significantly reduces the effectiveness of the downstream fins. In their studies, a single long permeable fin tended to offer the best convection heat transfer from a cylinder.

So, because of its industrial importance, this class of heat transfer has been the subject of many experimental and analytical studies. Though a lot of work has been done in this area, it is still remained the subject of many investigations. Recent economic and environmental concerns have raised the interest in methods of increasing or reducing th**e** forced convection heat transfer depending on the application from a horizontal plate. Researchers continue to look for new methods of heat transfer control under these conditions.

3. GOVERNING EQUATIONS

3.1 Assumptions

Besides the assumptions used in the derivation of general conduction equation for fin, the following assumptions and simplifications apply-

i. The convection heat transfer coefficient ‘h’ is uniform and constant over the entire fin surface.
ii. Heat transfer through the fin is at steady state, and there is no physical thermal energy source in the fin.
iii. The temperature of the fluid between fins is uniform over the entire fin surface.
iv. The thermal conductivity of fin material ‘k’ is uniform and constant.
v. The base of the fin is at the temperature of the heater surface, which is uniform over the perimeter of the plate. The effect of holes on heat conduction of fins is taken into account by setting zero thermal conductivity for the holes, which does not violate the fourth assumption. In that idealization it is assumed that, the fin material, but not the fin including the holes, has a constant and uniform thermal conductivity.

3.2 The Empirical Method of Thermal Boundary Layer Equation for Laminar Flow

The major convection parameters may be obtained by solving The major convection parameters may be the appropriate form of the boundary layer equations. Assuming steady, incompressible, laminar flow with constant fluid properties and negligible viscous dissipation and recognizing that $\frac{dp}{dx} = 0$, the boundary layer equations reduce to

Continuity

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad (1)$$

Momentum

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = \upsilon\frac{\partial^2 u}{\partial y^2} \qquad (2)$$

Energy

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\frac{\partial^2 T}{\partial y^2} \qquad (3)$$

Species

$$u\frac{\partial \rho_A}{\partial x} + v\frac{\partial \rho_A}{\partial y} = D_{AB}\frac{\partial^2 \rho_A}{\partial y^2} \qquad (4)$$

Solution of these above equations is simplified by the fact that for constant properties, conditions in the velocity (hydrodynamic) boundary layer are independent of temperature and species concentration.
The wall shear stress may be expressed as

$$\tau_s = 0.332u_\infty\sqrt{\frac{\rho\mu u_\infty}{x}} \qquad (5)$$

The local friction coefficient is then

$$C_{f,x} = \frac{\tau_{s,x}}{\frac{\rho u_\infty}{2}} = 0.664Re_x^{-1/2} \qquad (6)$$

The appropriate boundary conditions are

$T^*(0)\ 0$ and $T^*(\infty) = 1$

Applying this boundary conditions

$$\frac{dT^*}{d\eta}(\text{at } \eta = 0) = 0.332Pr^{1/3} \qquad (7)$$

The local Nusselt number is of the form

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.332$$

Applying appropriate boundary conditions and necessary manipulation, we get the final form as-

$$\overline{Nu}_x = \frac{\bar{h}_x x}{k} = 0.664\mathrm{Re}_x^{1/2}\,\mathrm{Pr}^{1/3} \qquad (\mathrm{Pr} \geq 0.6) \qquad (8)$$

3.3 General Conduction Analysis for Rectangular Straight Fin of Uniform Cross Sectional Area

The analysis is simplified if certain assumptions are made. We choose to assume one-dimensional conditions in the longitudinal (x) direction, even though conduction within the fin is actually two-dimensional. The rate at which energy is convected to the fluid from any point on the fin surface must be balanced by the rate at which energy reaches that point due to conduction in the transverse (y, z) direction. However, in practice the fin is thin and temperature changes in the longitudinal direction are much larger than those in the transverse direction. Hence we may assume one-dimensional conduction in the x direction**.** We will consider steady-state conditions and also assume that the thermal conductivity is constant, that radiation from the surface is negligible, that heat generation effects are absent, and that the convection heat transfer coefficient h is uniform over the surface.

$$\frac{d^2T}{dx^2} + \left(\frac{1}{Ac}\frac{dAc}{dx}\right)\frac{dT}{dx} - \left(\frac{1}{Ac}\frac{h}{k}\frac{dAs}{dx}\right)(T-T_\infty)$$

This result provides a general form of the energy equation for one-dimensional conditions in an extended surface. Its solution for appropriate boundary conditions would provide the temperature distribution, which could then be used to calculate the conduction rate at any x.

$$\frac{\theta}{\theta_b} = \frac{\cosh m(L-x) + (h/mk)\sinh m(L-x)}{\cosh mL + (h/mk)\sinh ml} \qquad (9)$$

We are also interested in the total heat transferred by the fin. It is evident that the fin heat transfer rate q_f may be evaluated in two alternative ways, both of which involve use of the temperature distribution. The simpler procedure, and the one that we will use, involves applying Fourier's law at the fin base. That is,

$$q_f = q_b = -kA_c\frac{dT}{dx} = -kA_c\frac{d\theta}{dx} \quad (\text{at } x=0) \qquad (10)$$

Hence, knowing the temperature distribution, θ(x), q_r may be evaluated, giving

$$q_f = \sqrt{(hPkA_c}\theta_b\frac{\sinh mL + (h/mk)\cosh ml}{\cosh mL + (h/mk)\sinh ml} \qquad (11)$$

4. EXPERIMENTAL SETUP

The experiments were performed in a controlled atmospheric condition in the laboratory. The temperature and humidity of air were controlled by using air-conditioning devices. The test apparatus and arrangement was based on the air-enthalpy method proposed by ASHRAE/ANSI standard.

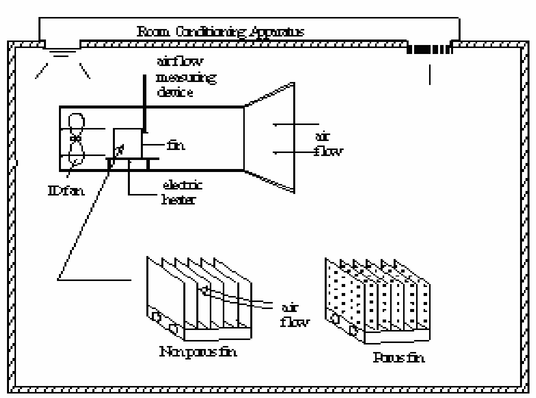

Fig 1. Schematic diagram of experimental setup

4.1 Fin Section

For the experimental investigation, we choose a model of extended surface of straight rectangular shape having a dimension of 60 mm in length, 60 mm in width and with a thickness of 3 mm. The material chosen for base wall and fin was 'Brass' because of its rigidity, relatively high conductivity and ease of availability. The fin was provided permeability by drilling very small holes (with a diameter of $\frac{3}{16}$ inch, 6×5 holes per fin) in it. There was an array 6 fins integral to the base and spacing between arrays of fins was 7.5 mm.

4.2 Wind Tunnel

The experiment was performed inside a wind tunnel made of plywood having the specifications as 130 cm long, 30 cm wide and also 30 cm high. The wind tunnel was provided with a 46cm opening or inlet section at the bell mouth inlet to minimize breaking of streamlines. Provisions were made inside the wind tunnel for accommodating fins, heaters and thermocouple wires with a 30cm removable plastic cover, at a distance 15cm from the fan side (opposite of bell-shaped inlet) of the wind tunnel. The wind tunnel was placed on a wood table and was fixed with it at a convenient height to work with

4.3 Fan Section

An induced draft, forward curved blade fan was placed at the opposite end the bell-shaped inlet of wind tunnel to suck the air through the tunnel to provide air-velocity range. Induced draft fan was chosen because it is capable of handling hot gases (often from 260-480°C) and gases containing soot, ash and other dust particles. The speed of the fan was regulated by a voltage regulating circuit.

4.4 Thermocouple

To measure the fin temperature at different points, in both longitudinal and transverse direction, thermocouples were used in this experiment. Total of six thermocouples were mounted on each fin and two on the base wall. The thermocouples used were of T-type thermocouples (copper + Constantine) with an applicable range of 0-400^{0}C.

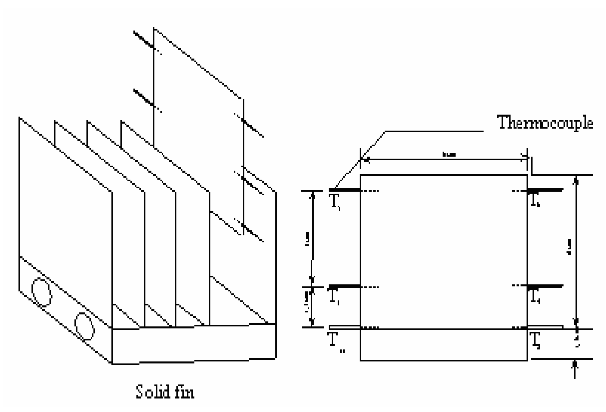

Fig 2. Connection of thermocouples in the fin

To connect the thermocouple leads to the fin and base, highly adhesive epoxy-resins were used. Extension wire (Cu was used here) was to connect the thermocouple leads to the selector switch provided with a digital calibrated voltmeter.

5. RESULT AND DISCUSSION

The effect of rectangular plate fin on the forced convection heat transfer over a flat plate was experimentally studied for several combinations of Reynolds number, heat energy input and relative humidity conditions. All the calculations and computations were done for four values of Reynolds numbers corresponding to four different relative humidity conditions. The nondimensional temperature distribution was plotted against dimensionless distance (x/L) for different values of Reynolds number to compare the experimental results with the theoretical ones. The variation of average Nusselt number was plotted against Reynolds number for both solid and permeable fins. The variation of average Nusselt number as a function of relative humidity for various values of selected Reynolds number was also plotted on the graph.. The following figure shows the change in the average Nusselt number as a function of Reynolds number at different values of relative humidity and base plate temperature for both solid and permeable fins

The lower line corresponds to solid fins while the upper line corresponds to permeable or drilled fins. It can be seen that in all the cases, the use of drilled fins results in a significant heat transfer enhancement over the solid fins under the same environmental and flow conditions. The advantage of using drilled fins was optimum at higher values of Reynolds number. In almost all the conditions except for one, the use of drilled fins resulted in higher Nusselt number than solid fins. In the figure, we can also see that the average Nusselt number increases about linearly with increasing Reynolds number which is quite expected.

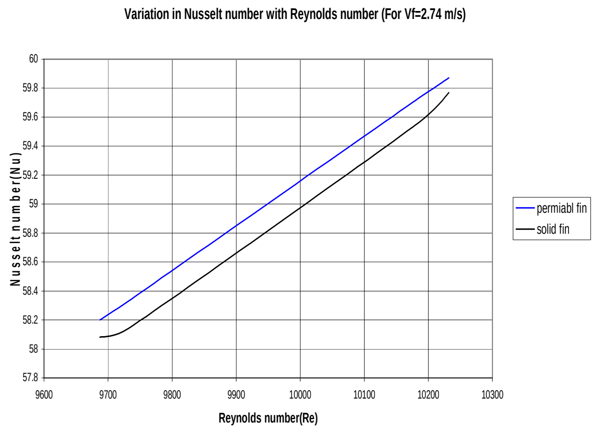

Fig 3. Variation of Nusselt number as a function of Reynolds number

The above observations seen can be discussed from both thermal and aerodynamic point of view. The reduction in Nusselt number or lower Nusselt number is caused by the reduction of air speed ahead of and in the wake of the solid fins. Reduced heat transfer effect results when the array of fins is sufficiently dense to trap a blanket of air in the small space between them. As the number of fins in a fin array increases, the spacing between the fins becomes smaller. This causes part of the hot air to get trapped between the fins resulting in a recirculation region, which further increases the ineffective portion of the fin.

The reason for higher Nusselt number in case of drilled fins which is responsible for higher heat transfer rates is due to the fact that air can flow through the pore spaces or holes created in them without restriction and thus reduces the ineffective portion of the fin. Although a slight reduction in heat transfer area is caused by creating holes in the fins, it increases the flow of air through them compensating the effect of reduced area.

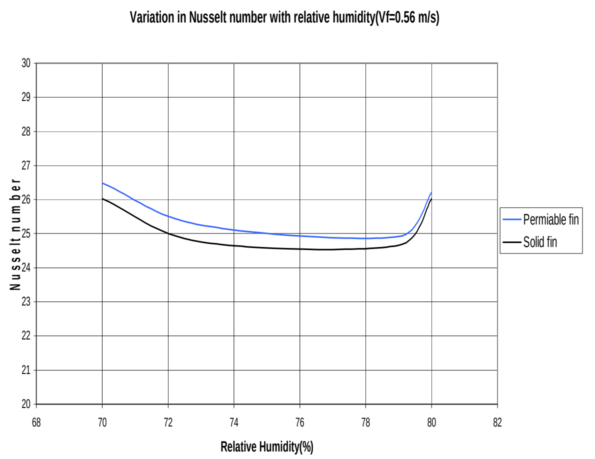

Fig 4. Variation of Nusselt number with relative humidity for both solid and drilled fin

The figure 4 shows the variation of Nusselt number for two different flow conditions (v =0.56 m/s and v =2.74 m/s) against four values of relative humidity from which it can be seen that the average Nusselt number does not vary much with relative humidity. The average Nusselt number remains nearly constant for relative humidity ranging from RH=70% to RH=80%. The results are similar to the findings obtained by Bassam and Hiljeh [2] in their experiment on rectangular fin efficiency. The slight increase in Nusselt no between RH = 79 % to 80% is due to the higher fin base temperature which resulted in a higher heat transfer rate. At high frontal velocities the difference in Nusselt number between solid and permeable fin is small with relative humidity and also shows some from of overlapping.

The results indicate that permeable or drilled fins can be very useful in situations which require higher heat transfer rates than what can be achieved by the use of regular solid fins. Drilled or permeable fins have great practical implementation in terms of weight and cost of fins needed to achieve a certain level of heat transfer. The use of drilled fins can be an excellent passive method for providing high heat transfer rates for electronic components in a small, light weight and low maintenance package

6. CONCLUSIONS

The phenomenon of laminar forced convection heat transfer from a horizontal flat plate with equally spaced rectangular solid and drilled fins was studied experimentally and numerically. Changes in the average Nusselt number at different combinations of Reynolds number and relative humidity were reported and compared to the results of those of solid fins. Permeable fins offered a higher Nusselt number than the solid fins, under same operating conditions. Both the solid and permeable fin showed nearly insensitive characteristics to changes in the relative humidity. Therefore this study suggests that permeable or drilled fins can be very useful in situations which require higher heat transfer rates than what can be achieved by the use of regular solid fins.

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8. NOMENCLATURE

Symbol	Meaning	Unit
A_f	Area of fin	m^2
C_p	Specific heat at const. pr	KJ/Kg.K
C_{fx}	Coefficient of friction	
D	Diameter of drilled hole	m
h	Convection heat transfer	
K	Temperature	K
k_f	Conductivity of fin mat.	W/m.K
k	Conductivity of air	W/mK
L_c	Length of fin	m
Nu	Nusselt number	
P	Perimeter of fin	m
Pr	Prandtl number	
Re	Reynolds umber	
T_b	Fin base temperature	K
T_f	Fluid film temperature	K
T_∞	Surrounding temperature	K
T	Surface temperature	K
t	Thickness	m
V_f	Flow velocity	m/s
μ	Viscosity of air	Pa/s
ρ	Density of air	kg/m^3