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EXPERIMENTAL INVESTIGATION ON THE PERFORMANCE OF LONGITUDINAL FINS WITH DIFFERENT NOTCHES USING MIXED CONVECTION HEAT TRANSFER

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ABSTRACT

An a experimental investigation of mixed convection heat transfer on rectangular fins with different notches in a horizontal rectangular channel for a wide range of Rayleigh numbers and different flow rate has been investigated with various dimensionless fins S/H=0.15 and fin height $H_f/H=0.5$ with fin spacing S=11 mm has been kept constant .The heat transfer coefficient of notched fin array is more than the un-notched fin array.

Keywords: Mixed Convection Heat Transfer, Fins With Different Notches, Rayleigh Number, Different Flow Rates, Heat Transfer Coefficient.

INTRODUCTION

Fins

The fins are commonly used for increasing the heat transfer rates from the surfaces whenever it is not possible to increase the rate of rate of heat transfer either by increasing the heat transfer coefficient on the surface or by increasing the temperature difference between the surface and surrounding fluids.

The fins are commonly used on small power developing machines as engines used for scooters and motorcycles as well as small capacity compressors. They are also used in refrigerating systems for increasing heat transfer rates.

An extended surface configuration is generally classed as a

- Straight fin
- An annular fin
- A Spine

HEAT TRANSFER

Heat transfer is thermal energy in transit due to spatial temperature difference. Whenever there exists a temperature difference in a medium or between media, heat transfer must occur.

The heat transfer may occur in three

modes-

- Conduction heat transfer
- Convection heat transfer
- Radiation heat transfer

In the above three modes of heat transfer our project comes under convection heat transfer.

Convection heat transfer

The convection heat transfer mode is comprised of two mechanisms. In addition to energy transfer due random molecular motion, energy is also transferred by the bulk, or macroscopic, motion of the fluid. This fluid motion is associated with the fact at any instant; large numbers of molecules are moving collectively or as aggregates. Such motion, in the presence of a temperature gradient, contributes of heat transfer. Because of the molecules in the aggregate retain their random motion, the total heat transfer is then due to a superposition of energy transport by the random motion of the molecules and by the bulk motion of the fluid.

the rate of convection heat transfer is observed to be proportional to the temperature difference, and is conveniently expressed by Newton's law of cooling as

$Q_{conv} = hA_s(T_s - T_\infty)$

Where As is the surface area through which convection heat transfer takes place

h is the convection heat transfer coefficient in W/m2K

Ts and T∞ is the surface and fluid temperatures respectively.

The above expression is known as Newton's law of cooling.

It depends on the condition of boundary layer, which are influenced by surface geometry, the nature of fluid motion, and as assortment of fluid thermodynamic and transport properties.

Classification

Convective heat transfer may be classified according to the nature of the flow. They are

i. Natural convection

The natural convection is induced by the buoyancy forces, which are due to density differences caused by the temperature variations in the fluid.

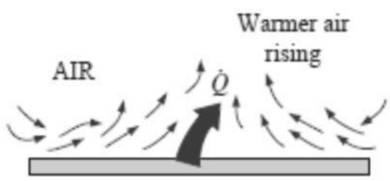

Fig 1.6 Natural convection

1 Forced convection

When the flow is caused by the external means such as by a fan, a pump, or atmospheric winds it is called forced convection.

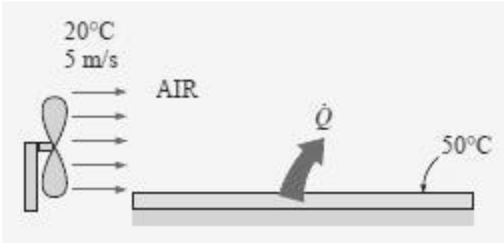

Fig 1.7 Forced convection

iii. Mixed convection

It is the combination of both free and forced convection. A process occurs when the effect of the buoyancy force in forced convection or the effect of forced flow in free convection becomes significant.

MIXED CONVECTION HEAT TRANSFER

In a mixed convection, both natural and forced convection participate in the heat transfer process

Mixed convection flows arise in many transport processes in natural and engineering devices. Atmospheric boundary layer flow, heat exchangers, solar collectors, nuclear reactors, and electronic equipment are some examples.

Such a process occurs when the effect of the buoyancy force in forced convection or the effect of forced flow in free convection becomes significant. For horizontal ducts, the buoyancy force is normal to the main flow direction and induces secondary flows in the cross plane.

The bulk fluid flow direction can be any of the three possible directions in horizontal channel, forward, backward or upward. The forced flow can be in the same direction as the flow created by the natural convection, and this flow condition is called assisting mixed convection. Whereas, for the other case, forced flow is in an opposing direction to the flow that is created by buoyancy, and this flow condition is referred to as opposing mixed convection.

MATERIAL SELECTION AND EXPERIMENTAL SET UP

3.1 MATERIAL SELECTION

Generally there are two types of materials used for fins aluminum and copper. The thermal conductivity of aluminum is 225 W/mK and that of copper is 385 W/mK. The melting and boiling point of copper are 1084 ºC and 2595 ºC and that of aluminum are 658ºC and 2057 ºC.

EXPERIMENTAL SETUP

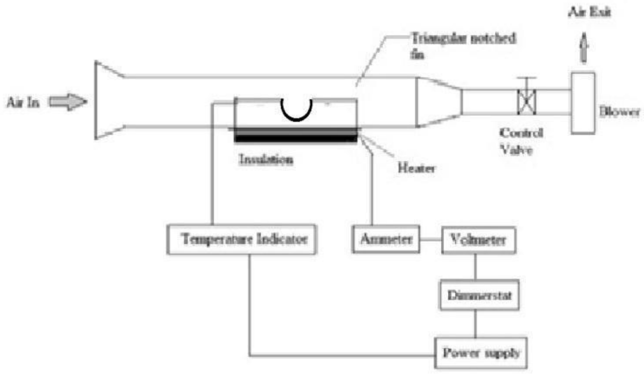

Fig. 3.1 Sketch of the experimental setup

The experimental set-up consists mainly of a nozzle, the test section, a diffuser, a flow control valve and a blower. The multipoint temperature indicator is used to measure the fin temperature, base plate temperature, and inlet and exit temperature of the air. A dimmer stat with digital voltmeter and digital ammeter is used to control the electric current supplied to the heater plate.

The test section is a rectangular duct heated at the bottom with a cross section 140 mm in width and 60 mm in height. The bottom of the heater is wounded with asbestos thread.

The unheated entrance region of the duct is 300 mm, the heated test section involving a copper plate and rectangular longitudinal fins is 127 mm and unheated exit region is 150 mm. At the channel entrance a nozzle is made of 1 mm thick galvanized iron sheet.

The test section consists of a 190 x 110 x 1 mm copper plate, copper longitudinal fins of 1 mm thickness and 127 mm in width and 30 mm height with notch at the center of the fin with 20% notch area.

Fins are attached to the plate using copper welding rod through gas welding. The thermocouples are attached to fins to measure the fin temperature and to the base plate to measure the base plate temperature. All surfaces are carefully cleaned and polished.
A plate heater was placed under the copper plate, with a size equal to the size of the copper plate having dimensions of 190 x 110 mm. Electric current was provided to the heater plate via a variac, providing a heat flux boundary condition specified for a decide experimental case.
The air velocity is measured with a hot-wire anemometer. All the thermocouples were attached to the fins by drilling a hole in the fins and connecting them with rigidity.

Using these different values collecting from the experiment, the following parameter has to be determined. The Nusselt number for the bottom plate and the heat transfer coefficient h has to be calculated. The experimental setup used in the evaluation of heat transfer coefficient for mixed convection is shown in figure 3.2 The fin array with and without notch used in the convection environment is shown in figure 3.3 and 3.4 respectively.

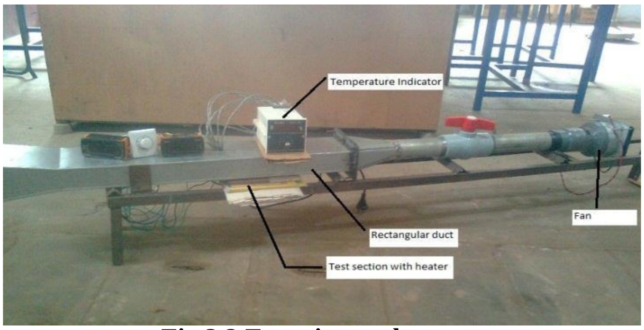

Fig 3.2 Experimental setup

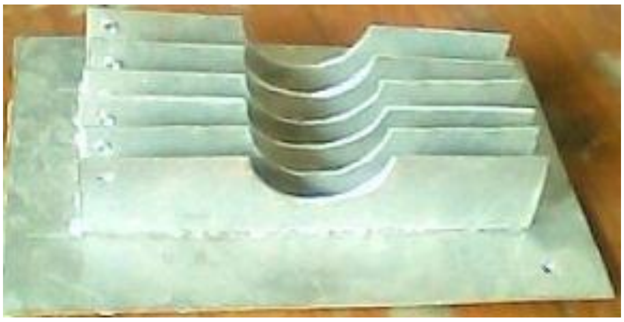

Fig 3.3 fin with circular notch

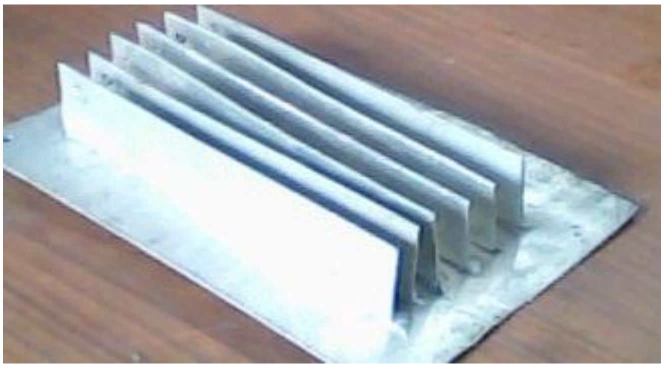

Fig 3.4 fin without notch

FORMULAE USED

- $Q_{conv} = hA_s(T_s - T_\infty)$

h- Heat transfer coefficient, W/m²K

A_S - Surface area of fin, m²

temperature difference between fin

Surface and ambient air

• = $L_f [(W_{bp} – N_f t_f) + N_f (2H_f + t_f)]$ L_f = length of the fin, m

$W_{bp=}$ width of the base plate,m
$N_{f\,=}$ number of fins

t_f = thickness of fin, m $H_{f\,=}$ height of the fin, m

- Nusselt number Nu = hd/K
- Hydraulic diameter= 4A/P
- Free flow area = $A_f = (W_d – N_f t_f) H_f + W_dC - W_{bp}t_{bp}$

W_d = width of the duct

C = Clearance between fin array and top margin of the duct
P = wetted perimeter of whole fin array assembly = $2[(N_f+1) H_f + C + W_d]$

- Reynolds number Re = VD/υ
- V = velocity of air in the duct, m/s
- υ = kinematic viscosity of air, m^2/s
- Grashof number $Gr = gl^3\varepsilon\Delta T/\upsilon^2$

- g = acceleration due to gravity, m s^{-2}
- l = representative dimension, m
- ξ = coefficient of expansion of the fluid, K^{-1}
- ΔT = temperature difference between the surface and the bulk of the fluid, K
- ν = kinematic viscosity of the fluid, m^2s^{-1}.

RESULTS & DISCUSSION

The mixed convection heat transfer with longitudinal fins with and without notch in a horizontal channel under bottom wall constant heat flux conditions has been investigated experimentally. By adjusting the flow control valve, the fluid velocity at the inlet of the duct was obtained as 0.5 m/s during the experiment. Experiments were conducted under various heat flux conditions. As a consequence of the above mentioned experimental conditions the Heat transfer coefficient was obtained between 7.79 to 8.4 for circular notched fin and 7.39 to 8 for without notch fins.

TABLE 1

Heat Transfer coefficient for various heat

Input

Heat inputs in watts	Fin without notch (h in W/m^{2}k)	Fin with notch (h in W/m^{2}k)
90	7.39	7.79
115	7.56	8.08
140	7.86	8.23
165	8	8.4

Heat input Vs Heat transfer coefficient

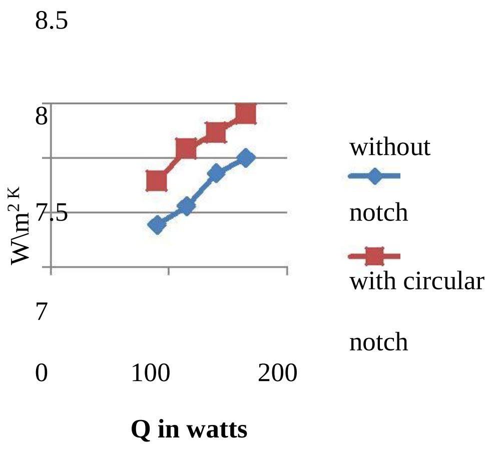

Fig 4.1 Comparison of HTC for with and Without notched fin array

CONCLUSION

Mixed convection heat transfer of longitudinal fins with circular notch in a horizontal rectangular channel with uniform heat flux boundary condition at the bottom surface has been studied experimentally. Experimental results for bottom heated fin arrays have been presented for different heat input and the effect of heat transfer have been investigated.

It has been determined that the average heat transfer coefficient for circular notch fin with fin spacing of 11 mm is more than for without notch fin array.

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