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Experimental Investigation of Tube Side Heat Transfer Enhancement Using Segmented Cross-Strip Insert

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Abstract- *Enhancement of heat transfer by inserting different type turbulators into the tube is one of the conventional passive augmentation methods. This paper represents the experimental investigation of heat transfer in a circular tube with segmented cross strip inser using air as the test fluid in both laminar and turbulent flow rigime. A 900 mm long copper tube of 24.45 mm inner diameter and 28.3 mm outer diameter was used as the test section. The experiments were performed with a 938 mm long and 2.25 mm thick stainless steel cross strip insert, segmented at 7 parts, where the distance between each segment were 1.5 cm. Uniform heat flux condition was maintained at the external surface of the tube and K-type thermocouples were used at four points of the test section to measure the outer surface temperature of the tube. Reynolds numbers were varied from 1445 to 5678 in the experiment with heat flux variation 68 to 178 W/m^2 for smooth tube and 106 to 296 W/m^2 for tube with insert. The experimental Nusselt number obtained from a smooth tube were compared with Gnielinski [9] correlation and error were found to be in the range of -44% and +10% with r.m.s value of 33%. At equal Reynolds number, the investigation reveals that using of these segmented cross strip inserts are thermodynamically advantageous.*

Keywords: Heat transfer enhancement, segmented cross strip, heat flux, Nusselt number.

1. INTRODUCTION

Heat transfer enhancement provides a simple and economical method of improving the thermal performance of heat exchangers. The use of such a technique results in a reduced surface area requirement and consequently a smaller heat exchanger and reduced equipment cost. The augmentation of the heat transfer by inserting different type turbulators [1-2] into the channels is the conventional passive enhancement method. The cross strip insert is one of the common heat transfer devices. These tube inserts enhanced the convective heat transfer by interrupting the boundary layer development and raising the degree of turbulence or by increasing the heat transfer area or by generating the swirl flow. Hsieh et al. [3] experimentally investigated the heat transfer and flow characteristics of turbulent flow in a horizontal tube with strip type insert. They explained the underlying physical phenomena that were responsible for heat transfer enhancement. An experimental and computational fluid dynamics modeling study was performed on heat transfer in an air cooled heat exchanger with different tube inserts by Shabanian et al. [4]. Salam et al. [5] studied the enhancement of heat transfer in turbulent flow through tube with a twisted tape insert. Gunes et al. [6] conducted an experimental investigation of heat transfer augmentation in a tube with equilateral cross-sectioned coiled wire inserts. A numerical study was undertaken by Ozceyhan et al. [7] for the investigation of the heat transfer enhancement in a tube with circular cross sectional rings. Hsieh et al. [8] presented the results of an extensive study of turbulent heat transfer in a horizontal tube with strip type inserts. They found that for Re ~ 6500 to 19500 the heat transfer enhancement of the tube with insert was about four to two times that of bare tubes. Therefore, the main aim of the present work is to extend the experimental data available on the strip type insert and also improve the heat transfer by increasing the disturbance of flow due to the segment of the cross strip.

2. EXPERIMENTAL SETUP

The experimental apparatus, shown in Fig. 1, consists of a heated test section, air and power supply system, mixing box, instrumentation to measure flow rate and temperature. The test section was made of copper having 24.45 mm ID, 28.3 mm OD and 900 mm in length. A stainless steel segmented cross strip of 938 mm length and 20 mm height was inserted into the smooth tube. The cross strip insert had seven segments only at one side of the strip, where the distance between each segment was 15 mm with a thickness of 2.25 mm. The length of six segments was 120 mm each and one segment was of length of 128 mm. The nichrome resistance wire was spirally wound uniformly on the outer surface of the test section to supply the heating power. Mica sheet was used between the tube and heating wire for electrical

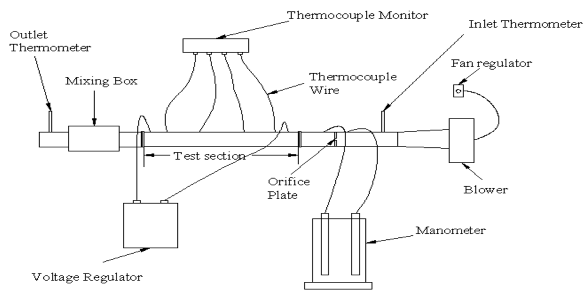

Fig. 1: Schematic diagram of experimental apparatus

insulation. The heating wire was covered with mica sheet and fiber glass. The heating wire was connected to 220 volt main. To measure the outer surface temperatures of the tube four K-type thermocouples were placed on equally spaced points of the test section. Two thermometers were placed at the inlet and outlet of the tube to measure the inlet and outlet water temperatures respectively. To measure the outlet temperature, the thermometer was placed in a mixing chamber, which was thermally insulated to minimize the heat loss. A mercury manometer was provided to measure the air flow rate, where the pressure difference was taken across an orifice. The taping positions for the orifice plate were 12 cm before the orifice and 6 cm after the orifice. A blower was used for the purpose of air

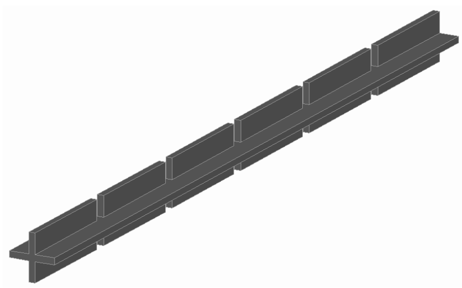

Fig.2: Diagram of portion of segmented cross strip insert

supply. The maximum speed of the blower was 13000 rpm and the blower speed was controlled by a fan regulator. After switching on the heating power, the sufficient time was given to attain the steady state condition. In each run, data were taken for air flow rate, air inlet, air outlet and tube outer surface temperatures.

3. DATA REDUCTION

The measured data were reduced using the following procedure:

The heat transfer rate by the heater to air was calculated by measuring heat added to the air. Heat added to air was calculated by,

$$Q = mC_p (T_o - T_i) \quad (1)$$

Heat transfer coefficient was calculated from,

$$h = Q / A (T_{wi} - T_b) = q / (T_{wi} - T_b) \quad (2)$$

where, $A = \pi d_i L$ (3)

The bulk temperature was obtained from the average of air inlet and outlet temperatures,

$$T_b = (T_i + T_o) / 2 \quad (4)$$

Tube inner surface temperature was calculated from one dimensional radial conduction equation,

$$T_{wi} = T_{wo} - Q.\frac{\ln(d_o/d_i)}{2\pi k_w L} \quad (5)$$

Tube outer surface temperature was calculated from the average of four local tube outer surface temperatures,

$$T_{wo} = \sum_{i=1}^{4} T_{wo,i} / 4 \quad (6)$$

Theoretical Nusselt number was calculated from Gnielinski [9] correlation and f from Petukhov [10] relation.

$$Nu_{th} = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \quad (7)$$

where,

$$f = (0.79 \ln Re - 1.64)^{-2} \quad (8)$$

$$3000 \le Re \le 5\times10^6$$

$$Re = \rho V d_i/\mu \quad (9)$$

$$Pr = \mu C_p/k \quad (10)$$

$$Nu = hd_i/k \quad (11)$$

Volume flow rate,

$$\dot{Q} = C_d A_o \sqrt{2gh}_{air} \quad (12)$$

For a circular orifice,

$$\dot{Q} = C_d(1/4\pi d^2)\sqrt{2gh}_{air}$$

Here, the value of C_d was taken as 0.75

Mass flow rate, $m = \dot{Q}\rho = VA_t\rho$ (13)

4. RESULTS AND DISCUSSIONS

Before undertaking the experiments using the tube equipped with tube inserts, the heat transfer data were taken in a plain tube. In order to evaluate the validity of the set up, the plain tube experimental data were compared with Gnielinski [9] correlation. Figure 3 shows

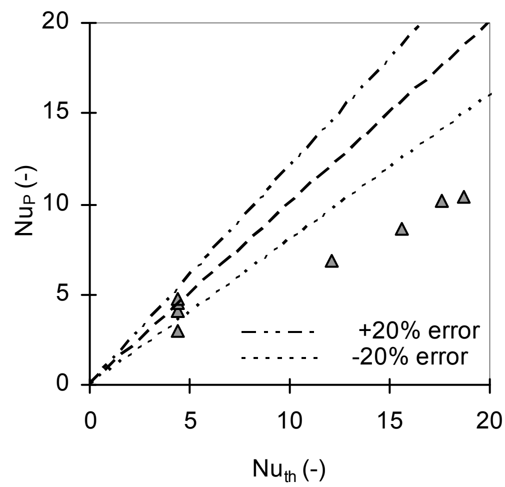

Fig. 3: Comparison between experimental and theoretical Nusselt number

the comparison of experimental Nusselt number for plain tube, Nu_p, with those calculated from Gnielinski [9]correlation. Data fall between -44% and +10% of the Gnielinski [9] value with r.m.s value of error 33%. Salam et al. [5] used the same setup for different insert and errors were found within -13% and +18% comparing with Dittus Boelter [11] correlation. Nusselt numbers for the plain tube and the tube with cross strip insert are shown in fig 4. It is seen that, Nusselt number increased with the increase of Reynolds number and segmented-cross strip insert gave higher values of Nusselt number than those for plain tube. Reynolds number was calculated based on inner diameter of the tube. For plain tube Nu_p increased from 3 to 10.44 with the increase of Reynolds number from 1445 to 5678 respectively. For tube with insert Nusselt number, Nu_i increased from 5.68 to 24.37 with the increase of Reynolds number from 1445 to 5678. Therefore, at equal Reynolds numbers, Nusselt number in tube with segmented-cross strip insert were enhanced by 89.37% to 133.35%. Hsieh et al. [8] found that the tube with insert provided more heat transport opportunity. For these reason they found that, the enhancement factor ranged from a minimum to 2.5 at Re=19500 to maximum 3.5 at Re= 6500. Figure 5 shows the heat flux variation for smooth tube and tube with segmented cross strip insert. It is seen that, heat fluxes increased with the increase of Reynolds number and insert gave higher heat flux than those for plain tube. The insert contribute to continuous

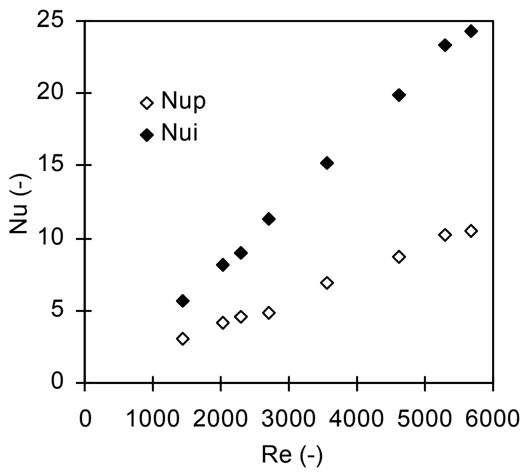

Fig. 4: The variation of Nusselt number with Reynolds number

interruption of the development of boundary layer as well as to generate the secondary flow and increased mixing resulting in higher heat transfer coefficients as compared to the tubes without insert. The temperature difference between wall and bulk fluid significantly decreased for tube with insert, figure 6. Average of 63% enhancement of heat flux was observed for tube with insert than that of plain tube.

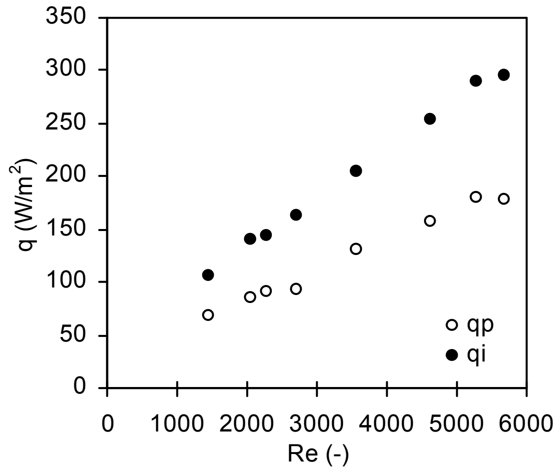

Fig. 5: The variation of heat flux with Reynolds number

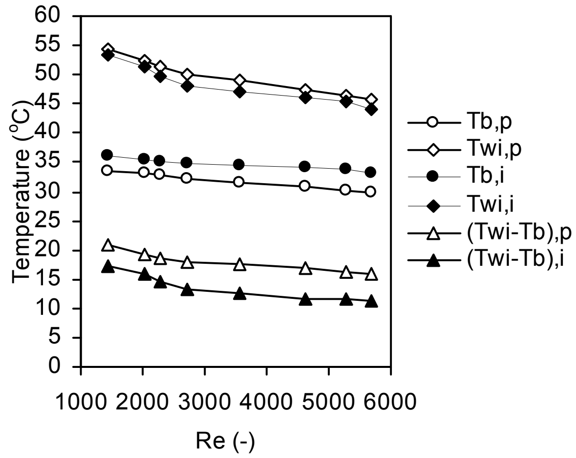

Fig. 6: The variation of temperatures with Reynolds number

5. CONCLUSIONS

An experimental study was carried out to investigate the tube side heat transfer coefficient of air for both laminar and turbulent flow in a circular tube fitted with segmented cross strip insert. The results can be summarized as follows:

- The Nusselt number and heat flux increased with increasing Reynolds number.
- The experimental Nusselt numbers for smooth tube fall with in -44% and 10% of Gnielinski [9] value.
- The experimental Nusselt numbers for tube with segmented cross-strip insert were enhanced by 1.8 to 2.4 times compared to those of smooth tube with average value of 2.16 times.

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8. NOMENCLATURE

Symbol	Meaning	Unit
A	Area of the heated region of tube	(m^2)
A_o	Orifice area	(m^2)
A_t	Tube x-sectional area	(m^2)
C_p	Specific heat of air at constant pressure	(J/kg.K)
C_d	Coefficient of discharge	(-)
d	Orifice diameter	(m)
d_i	Tube inner diameter	(m)
d_o	Tube outer diameter	(m)
f	Friction factor [10]	(-)
h	Heat transfer coefficient	($W/m^2.K$)
h_{air}	Orifice pressure drop	(m air)
k	Thermal conductivity of air	(W/m.K)
k_w	Thermal conductivity of tube material	(W/m.K)
L	Effective tube length	(m)
m	Mass flow rate of air	(kg/s)
Nu_i	Experimental Nusselt number with insert	(-)
Nu_p	Experimental Nusselt number for smooth tube	(-)
Nu_{th}	Nusselt number from Gnielinski [9] correlation	(-)
Pr	Prandtl number	(-)
Q	Heat transfer rate	(W)
q	Heat flux	(W/m^2)
q_i	Heat flux for tube with insert	(W/m^2)
q_p	Heat flux for smooth tube	(W/m^2)
$\dot{Q}$	Volume flow rate	(m^3/s)
Re	Reynolds number	(-)
T_b	Bulk temperature	(^0C)
$T_{b,i}$	Bulk temperature with insert	(^0C)
$T_{b,p}$	Bulk temperature for smooth tube	(^0C)
T_i	Air inlet temperature	(^0C)
T_o	Air outlet temperature	(^0C)
T_{wi}	Tube inner surface temperature	(^0C)
$T_{wi,i}$	Tube inner surface temperature with insert	(^0C)
$T_{wi,p}$	Tube inner surface temperature for smooth tube	(^0C)
T_{wo}	Tube outer surface temperature	(^0C)
$T_{wo,i}$	Local tube outer surface temperature	(^0C)
V	Mean velocity	(m/s)
ρ	Density of air	(kg/m^3)
μ	Dynamic viscosity of air	(kg/m.s)