Convection and Radiation Heat Transfer in a Tube with Core Rod and Multi Duct Inserts at High Temperature

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Abstract: Heat transfer and friction factor characteristics in a circular tube fitted with core rod and multi duct inserts at high temperature have been investigated experimentally. In the experiments, ambient air with Reynolds numbers in a range of 6000-20,000 is passed through a circular tube with uniform wall temperature and convection and radiation heat transfer phenomena are studied. Experiments have been performed at four constant wall temperature tubes with core rod insert. For each wall temperature considered, convection and radiation heat transfer coefficients have been determined. The experimental results show that at uniform wall temperature of 373 K, 473 K, 553 K and 633 K the average share of the radiation heat transfer coefficient to the total heat transfer coefficient are 11.5, 13.1, 15.3 and 17.8% for core rod and 16.9, 20.0, 24.3 and 27.8% for multi duct insert respectively. In addition it was noted that for the mentioned temperatures, the heat transfer coefficient increased by 227, 299, 327 and 369% for core rod and 279, 396, 453 and 539% for multi duct insert respectively in comparison to the plain tube. It was also noted that increasing the wall temperature also resulted in increase of friction factor. Based on the experimental results for Nusselt number and friction factor empirical correlation have been derived. Plotting the experimental findings and the correlations, it was noted that the majority of the data are within ±12% and ±8% of the proposed correlations for heat transfer coefficient and friction factor, respectively. The results were also tested against available and well proven correlations with reasonable agreement.

[M.R. Alijani, A.A. Hamidi. **Convection and Radiation Heat Transfer in a Tube with Core Rod and Multi Duct Inserts at High Temperature.** *Life Sci J* 2013;10(3s):471-478] (ISSN:1097-8135). http://www.lifesciencesite.com. 73

Keywords: High temperature, Radiation, Convection, Heat transfer enhancement, Friction factor, Core rod, multi duct insert.

1. Introduction

Heat transfer enhancement technology is the process of improving the performance of a heat transfer system by increasing the convection and radiation heat transfer coefficients. Over the past forty years, this technology has been extensively used in heat exchangers and other heat transfer equipments in thermal power plants, chemical processing plants, air conditioning equipment, refrigerators and vehicle radiators. Generally the main objective is to reduce the size and costs of these equipments.[1,2,3]

Heat transfer enhancement techniques can be divided into two categories. First is the passive method without the use of any external power source and the other is the active method, which requires extra external power sources. Reverse/swirl flow devices form an important group of the passive augmentation techniques.

The reverse flow, sometimes called ‘‘recirculation flow’’, devices or turbulators are widely employed in heat transfer engineering applications. Promvonge and Eiamsa-ard[4] reported the effect of conical-nozzle and snail entrance on heat transfer and friction characteristics in a uniform heat flux tube and found that the heat transfer rate increases considerably for using both enhancement devices. They also studied the effect of combined V-nozzle turbulator insert and snail entry on the heat transfer and friction factor characteristics under uniform wall temperature conditions.[5] Eiamsa-ard and Promvonge[6] further reported an effect of the V-nozzle turbulators on heat transfer rate in a circular tube and suggested that the nozzles have a significant effect on heat transfer enhancement. This indicates that the crucial effect of the reverse/re-circulation flow can promote the heat transfer rate in tubes. Sivashanmugam and Suresh[7] examined the turbulent heat transfer and friction factor characteristics of circular tube fitted with full-length helical screw element of different twist ratio under uniform heat flux conditions.

They also reported the heat transfer and friction factor characteristics of laminar flow through a circular tube fitted with straight helical screw-tape insert.[8] Naphon[9] experimentally studied the heat transfer and pressure drop in horizontal double pipes with and without twisted tape insert. In most of the notable published works, the experiments had been carried out at low temperatures, usually less than 373 K at which the effect of radiation has been completely ignored. The aim of the present experimental study was to determine the effect of radiation and convection heat transfer as well as the pressure drop in a tube with core rod and multi duct insert at high temperature. Experiments have been performed with Reynolds numbers in the range of 6000-20,000 at four constant tube wall temperatures of 373 K, 473 K, 553 K and 633 K.

Experimental Set-up and Procedure

The experiments were performed in an open loop experimental equipment as shown in Figure 1. The set up consisted of a blower (4 kW , 2400 Pa), orifice meter to measure the air flow rate, a 4 kW inverter to adjust the speed of motor in blower and the heat transfer test section. The orifice meter used to measure the flow rate was constructed and calibrated according to ASME standard.[10]

The test section consisted of a stainless steel tube, length, 2500 mm, inside diameter, 80 mm and outside diameter, 88.9 mm. Twenty three belt heaters each with maximum power of 800 W were belted on the outside surface of the tube. Suitable control system was installed in the set up to adjust for the necessary power for the heaters to provide uniform wall temperatures of 373 K , 473 K , 553 K and 633 K by receiving the feed back signal of thermocouples located on the surface of the tube. The outer surface of the test tube and heaters was well insulated with glass wool to minimize convective heat loss to the surroundings, and necessary precautions were taken to prevent air leakages from the system.

At the inlet and outlet of the tube, two K type thermocouples were placed in the center of pipe to measure the inlet and outlet air temperature of the test section. Thirteen K type thermocouples were lined up along the test tube wall surfaces and embedded in grooved tube surfaces. The average wall temperature was obtained using the temperature readings of all thirteen thermocouple. The thermocouples had been calibrated to within ±0.2°C.

Two pressure tapes, one just before and the other just after the test section were provided and attached to a digital on-line differential pressure transducer for pressure drop measurement. Differential pressure transmitter was calibrated within ± 0.25 Pa deviation before being used. The core rod and multi duct insert used in these experiments were made of carbon steel with a length of 2500 mm, and the outer diameter for core rod is 43.5 mm, and the diameter of square at multi duct was 25 mm. These are shown in Figure 2. Thirteen K type thermocouples were attached on the surfaces of core rod and multi duct to determine its average surface temperature.

In each experiment the flow rate of the ambient air from the blower was adjusted by the inverter to the required amount. An orifice meter with measuring range of 1.6- 250 m^3 hr^{-1} was used to measure the air flow rate. The required temperature of the tube wall was set on the controller. Once reaching steady state conditions, all temperatures and the pressure drop which were logged on the data logger and noted. Using the data obtained, radiation and convection heat transfer coefficients were calculated as discussed latter. Experimental uncertainty was calculated following Coleman and Steele method[11] and ANSI/ASME standard.[12] The calculations showed a maximum uncertainty of ±6%, ±5%, and ±8% for Reynolds number, heat transfer coefficient and friction factor, respectively.

Data Reduction

At the steady state conditions, the net heat transfer rate (Q_{net}) from the inner tube surface to the fluid flowing through the test tube can be calculated by subtracting the heat losses (Q_{loss}) from the total electrical power input (Q_{vol}). This is also equal to the rate of the heat transfer to the fluid passing through the test section, and is determined using inlet and outlet temperature difference and mass flow rate of air. The energy balance equations can be written as follows:

$$\dot{Q}_{net} = \dot{Q}_{vol} - \dot{Q}_{loss} = \dot{m}C_{p,a}(T_o - T_i) \tag{1}$$

$$\dot{Q}_{loss} = h_a \times A_s \times (T_s - T\infty) \tag{2}$$

The electrical power input ($\dot{Q}_{vol}$) to the heater can be measured by:

$$\dot{Q}_{vol} = \frac{V^2}{R} \tag{3}$$

The mass flow rate of air was calculated by:

$$\dot{m} = \rho_b \times u \times A_{cs} \tag{4}$$

The heat losses ($\dot{Q}_{loss}$) and the heat released by heaters ($\dot{Q}_{vol}$) were not measured or calculated in this work and heat absorbed by the flowing air was simply determined using equations (1) and (4). Subtracting the heat absorbed by the flowing air from the heat input from the heater, the heat losses ($\dot{Q}_{loss}$) were found to be 5 to 8% of the total electrical power input. The heat transfer from the test section can be written by:

$$\dot{Q}_{net} = h_{tot}\, A\, \Delta T_{\ln} \tag{5}$$

Where:

$$\Delta T_{\ln} = (T_i - T_o)/\ln\left(\frac{T_w - T_o}{T_w - T_i}\right) \tag{6}$$

Here (T_w) is the constant inner surface temperature of the tube test, whose average is being measured by the thermocouples. The total average heat transfer coefficient is assumed to be:

$$h_{tot} = h_{conv} + h_{rad} \quad (7)$$

Where (h_{conv}) is the average convection heat transfer coefficient and (h_{rad}) is the average radiation heat transfer coefficient.

$$h_{rad} = 5.67 \times 10^{-8} \times \varepsilon \times F_{jk} \times \left(T_w^2 + \widetilde{T}_{pac}^2\right)\left(T_w + \widetilde{T}_{pac}\right) \quad (8)$$

Where $\left(\widetilde{T}_{pac}\right)$ is the average surface temperature of each insert (the core rod or the multi duct), which is measured.

$$\widetilde{T}_{pac} = \sum T_{pac}/13 \quad (9)$$

And $\left(F_{jk}\right)$ is the shape factor from the inner surface of the tube test to the outer surface of each insert. The shape factor equations can be written as follows:

$$F_{jk} = \frac{1}{A_j}\int_{A_j}\int_{A_k}\frac{\cos\theta_j \times \cos\theta_k}{\pi \times X^2}\,dA_j dA_k \quad (10)$$

$$A_j \times F_{jk} = A_k \times F_{kj} \quad (11)$$

Where θ_j and θ_k are the angles between the unit normals to the areas A_j and A_k and X is the distance between the two areas. There is conduction heat transfer in the body mesh of the multi duct insert. Therefore at steady state condition the temperature of the inner surfaces of the insert are approximately equal and uniform. It can therefore be assumed that radiation heat transfer in the inner surface of the multi duct insert is negligible.

The Reynolds number and average Nusselt number are given by:

$$\mathrm{Re} = \frac{uD_h}{\upsilon} \quad (12)$$

$$Nu = \frac{hD_h}{k} \quad (13)$$

In fully developed tube flow, the friction factor **(f)** can be determined by measuring the pressure drop across the test tube length as follows:

$$f = \frac{\Delta P}{\left(\frac{L}{D_h}\right)\left(\rho\frac{u^2}{2}\right)} \quad (14)$$

Where ΔP is the pressure drop across the test tube measured by the differential pressure transducer with a ±2% Pa accuracy, L is the test tube length and u is the mean air velocity at the entrance of the test section which was calculated from volumetric flow rate divided by the cross-section area of the tube.

All of thermo physical properties of the air are determined at the overall bulk air temperature.

$$T_b = \left(T_o + T_i\right)/2 \quad (15)$$

For a constant pumping power:

$$\left(V'\Delta P\right)_p = \left(V'\Delta P\right)_t \quad (16)$$

The heat transfer enhancement efficiency (η) is defined as the ratio of the heat transfer coefficient for the tube fitted with the turbulator (h_t) to that of the plain tube (h_p) at similar pumping power. It can be written that[13] :

$$\left(f\,\mathrm{Re}^3\right)_p = \left(f\,\mathrm{Re}^3\right)_t \quad (17)$$

$$\eta = \left.\frac{h_t}{h_p}\right|_{pp} \quad (18)$$

Results and Discussion

In this section, results of the effect of installation of core rod and multi duct tube inserts at various wall temperatures on heat transfer rate and flow friction are presented. Investigation of the radiation and convection heat transfer at uniform wall temperature of 373 K, 473 K, 553 K and 633 K are performed. The total heat transfer (h_{tot}) and radiation heat transfer (h_{rad}) are calculated using equations (5) and (8) respectively. The total average heat transfer (with core rod or multi duct) in test section consists of radiation and convection heat transfer. Figure 3 shows the variation of total heat transfer coefficient and the radiation heat transfer coefficient as a function of Reynolds numbers and the tube wall temperature for each insert. It can be seen that the share of the radiation heat transfer coefficient is increased by increasing the surface temperature. The radiation share of heat transfer appears to be 11.5, 13.1, 15.3 and 17.8% for core rod and 16.9, 20.0, 24.3 and 27.8% for multi duct depending on the wall temperature.

It can be seen from the experimental results for the core rod and multi duct insert that the heat transfer coefficients increase with increasing tube wall temperature. This increase is mostly due to the radiation share of the heat transfer. The radiation heat transfer is proportional to the fourth power of temperature. In this investigation the shape factor of the multi duct and core rod are calculated by equations (10) and (11) and are 0.9 and 0.54 respectively. At the same wall temperature this results in higher radiation heat transfer coefficients for the multi duct as compared to the core rod.

Figure 4 shows the variation of heat transfer coefficient with Reynolds number. In this diagram the results for the heat transfer coefficient for plain tube and tube equipped with core rod or multi duct inserts at various wall temperatures are compared. For the plain tube the results for various wall temperatures are very similar to each other since the effect of radiation heat transfer between surfaces is not present.

On the other hand, the boundary layer disruption causes a better mixing between the insert and the wall regions, thus enhancing the convective heat transfer process. In addition, the use of the multi duct provides better heat transfer than that of the core rod at a similar wall temperature. This is due to the higher circulation and higher contact surface area between the fluid and the heating wall surface when fluid flows inside the multi duct insert. It is worth noting that for the considered wall temperatures, the heat transfer coefficient increased by 227, 299, 327 and 369% for core rod and 279, 396, 453 and 539% for multi duct insert respectively as compared to the plain tube.

In Figures 5 and 6, the results of Nusselt number and friction factor at considered wall temperatures for the plain tube are found to be in good agreement with previous correlation of Dittus-Boelter and Blasius.[14]

Using the experimental data, the following empirical correlations for Nusselt number and friction factor are derived for the plain tube and are presented in equations (19) and (20) respectively. They are found to represent the experimental data to within ±1 to 5 % error limits.

$$Nu = 0.2 \times \mathrm{Re}^{0.613} \times \mathrm{Pr}^{0.4} \tag{19}$$

$$f = 0.515 \times \mathrm{Re}^{-0.311} \tag{20}$$

Figure 7 shows the variation of friction factor with the Reynolds number at the various wall temperatures. For plain tube and tube equipped with inserts, it is clear that friction factor for the core rod and multi duct configuration are higher than that for plain tube. Also as expected for these cases friction factor decreases with increasing Reynolds number.

The predicted Nusselt number and friction factor for tube with core rod and multi duct insert were correlated and are shown in equations (21) to (24) respectively. The predicted Nusselt number and friction factor were compared with the experimental data in Figures 8 to 11. The majority of the data falls within ±12%, ±8% of the proposed correlations for heat transfer coefficient and friction factor, respectively.

$$Nu_{cor} = 0.723 \times \mathrm{Re}^{0.518} \times \mathrm{Pr}^{0.4} \times \left(\frac{T_w}{\tilde{T}_{rod}} \right)^{1.736} \tag{21}$$

$$f_{cor} = 0.331 \times \mathrm{Re}^{-0.262} \times \left(\frac{T_w}{\tilde{T}_{rod}} \right)^{0.88} \tag{22}$$

$$Nu_{mul} = 10.19 \times \mathrm{Re}^{0.27} \times \mathrm{Pr}^{0.4} \times \left(\frac{T_w}{\tilde{T}_{mul}} \right)^{2.246} \tag{23}$$

$$f_{mul} = 0.528 \times \mathrm{Re}^{-0.297} \times \left(\frac{T_w}{\tilde{T}_{mul}} \right)^{2.179} \tag{24}$$

Using equations (17), (20), (22) and (24), the Reynolds number for the plain tube (Re_p) can be written as a function of the Reynolds number for the core rod and multi duct insert turbulator (Re_t):

$$(\mathrm{Re}_p)_{cor} = 0.848 \times \mathrm{Re}_t^{1.018} \times \left(\frac{T_w}{\tilde{T}_{rod}} \right)^{0.327} \tag{25}$$

$$(\mathrm{Re}_p)_{mul} = 1.009 \times \mathrm{Re}_t^{1.005} \times \left(\frac{T_w}{\tilde{T}_{mul}} \right)^{0.810} \tag{26}$$

Employing equations (18), (19), (21) and (23), the enhancement efficiency at constant wall temperature for the core rod and multi duct insert turbulator can be written as:

$$\eta_{cor} = \left. \frac{h_t}{h_p} \right|_{pp} = 3.998 \times \mathrm{Re}_t^{-0.106} \times \left(\frac{T_w}{\tilde{T}_{rod}} \right)^{1.536} \tag{27}$$

$$\eta_{mul} = \left. \frac{h_t}{h_p} \right|_{pp} = 50.671 \times \mathrm{Re}_t^{-0.346} \times \left(\frac{T_w}{\tilde{T}_{mul}} \right)^{1.749} \tag{28}$$

Figure 12 shows Variation of enhancement efficiency with Reynolds number for both core rod and multi duct inserts. It indicates that the enhancement efficiency of inserts increases with increasing temperature, since the ratio $T_w \cdot \tilde{T}_{pac}^{-1}$ is increased at high temperature. In the multi duct the fluid had higher turbulency, higher contact surface area and bigger radiation shape factor than the core rod. Therefore the total heat transfer and the enhancement efficiency for this insert are higher. The average enhancement efficiency for the considered uniform wall temperatures of 373, 473, 553 and 633 K were 1.63, 1.83, 1.97 and 2.28 for the core rod and 2.32, 2.83, 3.14 and 3.36 for the multi duct insert respectively. The average ratio of enhancement efficiency of multi duct insert to that of the core rod for these temperatures and at various Reynolds numbers were 1.42, 1.55, 1.59 and 1.47, respectively.

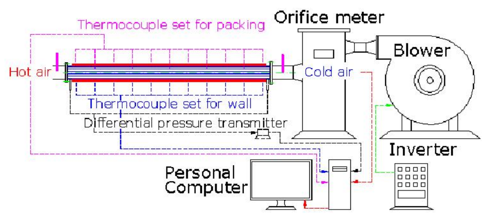

Figure 1: Schematic diagram of the experimental apparatus

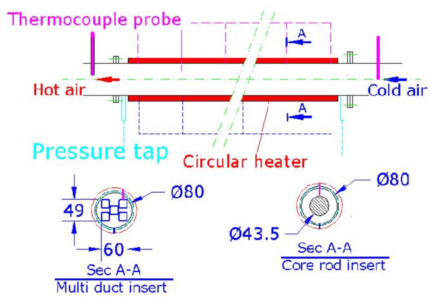

Figure 2: The tube fitted with core rod and multi duct network insert

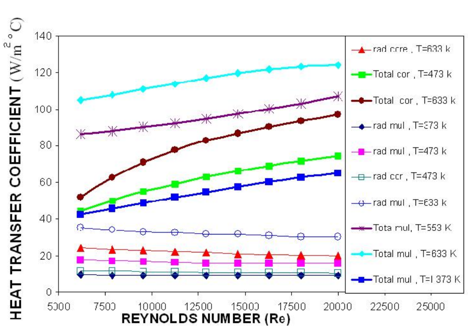

Figure 3: Verification of radiation and total heat transfer coefficient at various temperature of core rode and multi duct insert

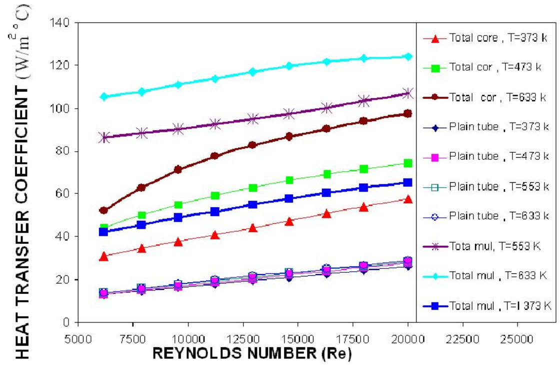

Figure 4: Verification of heat transfer coefficient at various temperature between plain tube , multi duct and core rode insert

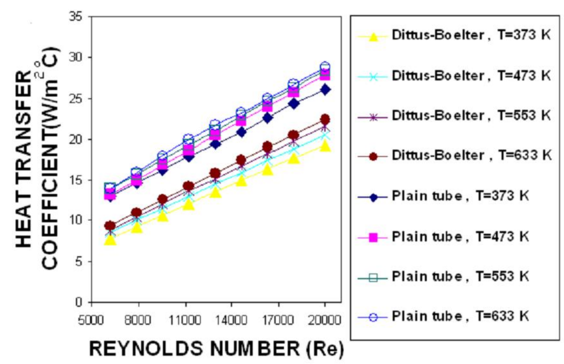

Figure 5: Verification of heat transfer coefficient at various temperature in plain tube

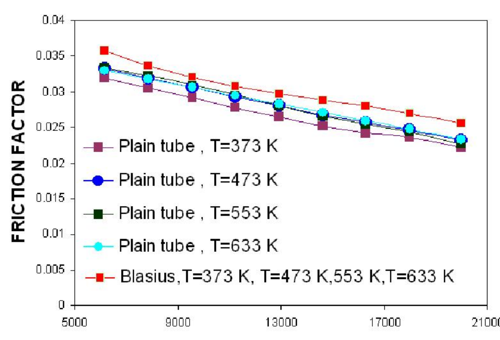

Figure 6: Verification of friction factor at various temperature in plain tube

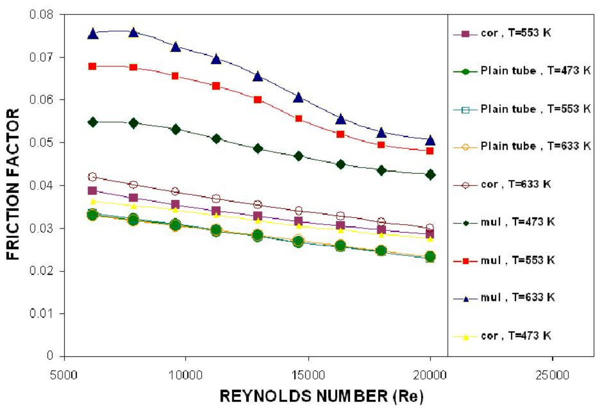

Figure 7: Verification of friction factor at various temperature between plain tube, core rod and multi duct insert

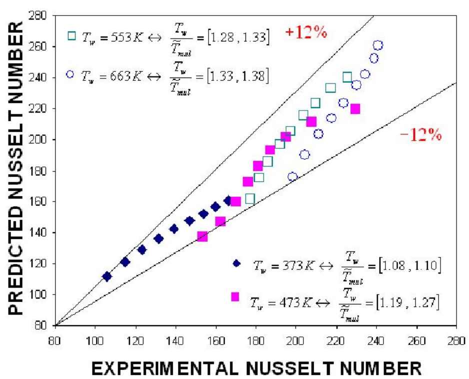

Figure 8: Nusselt numbers obtained from the present correlation and experimental data for multi duct insert

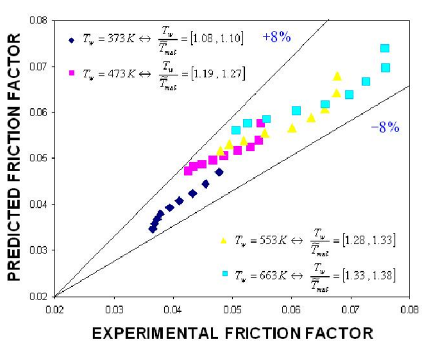

Figure 9: Friction factors obtained from the present correlation and experimental data for multi duct insert

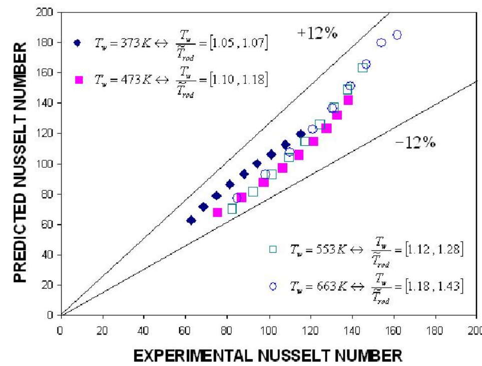

Figure 10: Nusselt numbers obtained from the present correlation and experimental data for core rod insert

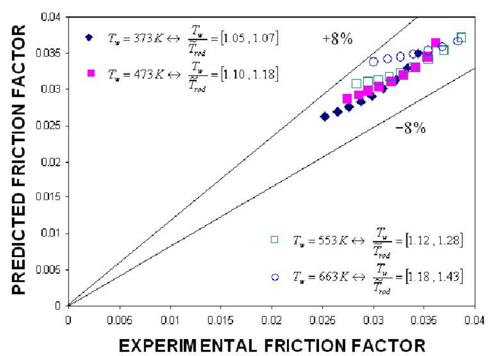

Figure 11: Friction factors obtained from the present correlation and experimental data for core rod insert

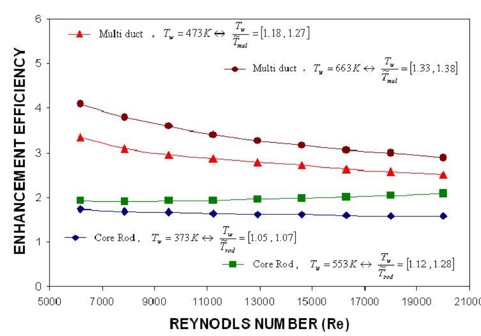

Figure 12: Variation of enhancement efficiency with Reynolds number for core rod and multi duct insert

Conclusions

Experimental investigations have been conducted to study the heat transfer in a circular tube equiped with core rod and multi duct inserts. For Reynolds numbers in the range of 6000 to 20000 for any given uniform wall temperature, the share of radiation heat transfer to the total heat transfer are 11.5, 13.1, 15.3 and 17.8% for core rod and 16.9, 20.0, 24.3 and 27.8% for multi duct insert respectively. For all cases studied the total heat transfer coefficient for both the core rod and multi duct insert increased due to increased turbulency, enhanced surface area and increased radiation effects. As mentioned the ratio of radiation heat transfer was separated from the total average heat transfer, and by increasing the shape factor in multi duct to core rod, It was seen that this parameter for multi duct was higher than that for the core rod. It was noted that the heat transfer coefficients increase with increasing tube wall temperature. This increase is mostly due to the radiation share of the heat transfer and it can be seen that the radiation heat transfer is proportional to the fourth power of temperature. Moreover the increased enhancement efficiency for the multi duct as compared to that of the core rod at any given uniform wall temperature is attributed to the increasing of the shape factor, the circulation, and the increased surface area for the multi duct.

Nomenclature

A : Heat transfer surface area, m^2
Cp,a : Specific heat capacity of air, J kg^{-1} K^{-1}
D_h : Hydraulic diameter, m
f : Friction factor
F_{jk} : Shape factor of surface j with respect to surface k
h : Average heat transfer coefficient, W m^{-2} K^{-1}
k : Thermal conductivity, W m^{-1} K^{-1}
L : Length of test section, m
$\dot{m}$: Mass flow rate, kg s^{-1}
Nu : Average Nusselt number
Pr : Prandtl number
Q : Heat transfer rate, W
R : Electrical resistance of the heater element, Ω
Re : Reynolds number
T :Temperature, K
$\tilde{T}$: Average temperature, K
u : Mean axial velocity, m s^{-1}
V : Voltage output from the Auto-transformer,V
V' : Volumetric flow rate, m^3 s^{-1}
X : Distance between the two areas

Greek symbols

η : Enhancement efficiency
υ : Kinematic viscosity, m^2 s^{-1}
ρ : Density of the fluid, kg m^{-3}
ε : Wall emissivity
θ : Angle between the unit normals and the area
$\Delta T_{\ln}$: Logarithmic mean temperature difference, K
ΔP : Pressure drop, Pa

Subscripts

a : Air
b : Bulk
cor : Core rod
Conv : Convection
cs : Cross section area
mul : Multi duct
i : Inlet
j : Inner
k : Outer
loss : losses
o : Outlet
p : Plain tube
pac : Packing
pp : Pumping power
rad : Radiation
rod : Rod
s : Side long area of heater
t : Turbulator
tot : Total
vol : Voltage
w : Wall
∞ : Atmospheric air

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1/15/2013