EXPERIMENTAL INVESTIGATION OF INCLINED CIRCULAR RING TURBULATORS ON HEAT TRANSFER AND FLUID FLOW CHARACTERISTICS.

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Abstract

This paper demonstrates influence of various inclination of Circular Ring Turbulator (CRT) on thermal characteristics in circular type heat exchanger. “A wall linked circular inserts were analyzed having constant inner diameter to achieve uniform flow blockage area (FBA) of 40%”. These CRTs located within the tube to prevailed different angular patterns (-10º, -15º, -20º and -25º) w. r. t. horizontal of tube. Influence of these geometries on fluid friction and heat performance were studied. Parameter constant throughout the investigation was diameter ratio and Pitch Diameter Ratio (PDR). Modified parameters throughout study were the angular inclinations of CRT and Reynolds number. Inner side of tube is preserved at constant heat flux having air as working medium at atmospheric temperature with variation of Reynolds number from 6,000 to 24,000 respectively. Implementation of CRT as turbulator favorably improves rate of heat exchange within heated enclosure of tube to fluid domain throughout the test model. Out of all studied inserts, CRT having angular deflection of -25º gives peak value of heat transfer at extreme Reynolds number and at slightest Reynolds number CRT having angle inclination of -10º offers greater friction factor which is at 2.5 and 5.5 times than smooth tube. CRT with angular inclination of -25º at minimum Reynolds number offers extreme thermal performance factor is 1.7.

Keywords

Heat transfer enhancement; Circular ring turbulator (CRT); friction factor; Heat Exchanger; Angular deflection (-10º, -15º, -20º and -25º); Nusselt number; Reynolds number.

Introduction

Heat exchangers plays a vital role in different thermal power plant, process industries, condensers, evaporators, space vehicles and radiators in automobiles, electronic cooling units, nuclear reactors etc. The thermal characteristics of the heat transfer inventions depend on the impact of convective factor between heated surface and moving fluid. “The overall thermal resistance is proposed by the development of the boundary layer across the heated wall of the heat transferring device”. In modern decades, most of investigators dealing with optimization techniques and overall stability of exchangers due to maximum material cost, which enhancing the transfer rate with less pumping energy [1]. Size as well as cost of heat exchanging device and pumping power may be shorten by raising the convective factor and shrinking pressure loss over the exchanger. “Heightening the turbulence by obstructing and blocking the fluid flow as well as swirl generation by suitable external geometry or surface modification causes increment in the rate of convective heat transfer”. It advances not only severance of the boundary layer but also appropriate and swift domain mixing. Normally, improvement in heat exchange is carried by active and passive process. Among these, passive process is most preferred by researchers because it can’t demand any external source of power, while active techniques require an additional source. In passive process, these geometries known as a turbulators through which working medium passing and fixed inner tube of exchanger. “In the process, along with enhancement in heat exchange, the pressure drop over the tube increases as a result of produced higher turbulence and secondary fluid flow”. Hence efficient process of improvement in thermal stability of exchanger is the one improves the heat exchange expressively with less power [2]. Now a days, various process were designed and investigated solutions have been represented by Sakr [3] and Dewan [4]. Most of the investigators predicted that passive processes as emerging domain of research for thermal enhancement. Implicit of these processes of heat enhancements may be thoughtful in modified refrigeration methods by Jomde et al. 2018a; 2018b [5,6].This paper consists passive heat exchange improvement technique, most use passive methods and allied patterns from the survey data mentioned below.

Literature Review
Most of passive patterns were made for exchange of heat enhancement in working medium of circular tube which consist perforated twisted pattern [7], coil wire, circular rings [8-9], V-nozzle [10], mesh inserts, conical nozzle ring [11], conical ring, ribbed pipes, V-ring, brush and strip turbulators, baffles etc. Distinctive twisted patterns insert mostly beneficial over laminar region also geometry such as conical ring, CRT, wire coil etc. are highly useful for turbulent model. Wire coil pattern doesn’t mix the fluid domain satisfactorily; also nonconductive for laminar zone which overcome efficiently with twisted type patterns. Because of this, twisted tapes are more useful in laminar region. Lot of researchers studied various geometries which generate the secondary flow and vortices by obstructing the flow effectively. “The inserts investigated were a V-nozzle, conical nozzles, conical ring, diamond-shaped turbulators [12], perforated ring [13], conical-nozzle with free-spacing entry, anchor-shaped inserts” [14], tube corrugation turbulators [15] baffle used geometry [16-17], wavy stripes [18], dimpled surface [19], converging-diverging ring, etc. An investigation by Eiamsaard was carried to predict thermal phenomenon throughout circular tube by twisted tape including conical-nozzles insert, shows better solution in overall performance coefficient. Heat flow improvement along with vibration produced in exchangers having conical-ring patterns, investigated by Yakut [20] with varying Reynolds number from 8000 to 18000. It concluded that, increment in Nusselt number possible by increasing Reynolds number at lowest pitch arrangement. Tandiroglu studied baffle inserts for convective heat exchange across turbulent model. Parameters varied of flow geometry were angular baffle orientation from 45°, 90°, 180° with PDR, empirical data was presented to calculate Nusselt number and fall in pressure [21]. Eiamsa-ard [22] concluded that, inserts created more frictional losses because of the flow blockage. An investigation by Rattanawong [23] stated that increment in the convective factor gained with propeller type swirl propagator effectively. An investigation carried on flow friction as well as heat exchange in round tube consisting porous type ring insert by Akansu [24]. The effect of inserts like inclined vortex rings (IVR) were studied with uniform heat flux in a round tube and its influence on heat transfer and fluid friction were investigated numerically by Promvonge [25]. Effect of the IVRs in moving flow offers significant influence on thermal parameters and pressure loss compared to smooth tube. Promvonge [26] studied allied horseshoes baffles along with various angular inclinations in a round heat exchanger. “This type of turbulator was inserted significantly into the circular tube exchanger at different flow blockage ratios; width and PDR factor (25%, 50%, 100%, 200%)”. The range of overall thermal performance coefficient was varied 1.34 to 1.92. The optimum scale observed at PDR of 25%, 50% having FBA of 25%, 100%. Chingtuaithong [27] investigated influence of V-type inserts on heat performance as well as friction factor under the uniform heat flux. In this work he presented the V-shaped ring inserts with four relative ring-pitches diameter ratio (0.5, 1.0, 1.5 and 2.0), three flow blockage ratios (0.1, 0.15 and 0.2), and angular orientation of 30° were fitted in tube. The overall factor was gained up to 1.36 to 1.63 across FBA 0.1 with relative diameter 1.0. V-ring achieves better thermal stability than other patterns such as twisted tapes, inclined rings and wire-coils. Singh [28] carried experiment with solid rings twisted tapes in a round heat exchanger to study heat transfer rate and fluid friction factor of moving flow along with uniform heat flux for the turbulent flow in which Reynolds number ranges from 6300 to 22,500. It shows at lowest pitch factor 1 with twist ratio 2 achieves excellent thermal performance. “Twisted ring (TRs) or Circular rings (CRs) inserts were comparatively tested for augmentation of friction factor and heat transfer in tube exchanger for Reynolds number 6000-20000”. Most of CRs achieve highest Nusselt numbers with flow friction compare to TRs excluded slightest PDR and highest width ratio, studied by Thianpong [29]. Gear-ring turbulators (GRTs) investigated to demonstrate effects on friction phenomenon with heat exchange rate in round tube exchanger. Friction characteristics and heat transfer rate increase with decreasing free-space length ratio and tooth number. Study was carried with CRT along with PDR and different diameter factor by Nanan [30]. It showed a CRT having lowest diameter factor produces the largest pressure drop along with the higher Nusselt number value. As per the investigation at the extreme diameter ratio 0.7 and lowest pitch 6, higher thermal performance is achieved. Promvonge investigated effect of conical type inserts having numerous alignments (converge geometry [CG] converge-diverge geometry [CDG] and diverge geometry [DG]) to heat transfer and overall thermal performance in circular tube. “By using these alignments, DR array of conical rings achieves the higher results as far as the thermal parameters are concern” [31]. Effect to overall thermal performance as well as moving flow friction of V-nozzle and conical ring inserts was studied by Promvonge [32]. Sane presented modern variable as flow blockage area applied for circular inserts terminology. “Kore conduct investigation on circular rings of different inner diameters for the tests which creates 30%, 40%, and 50% flow blockage area [33]. These ring type turbulator were mounted one after another to create insert of converging, diverging and converging-diverging model”. It showed that, CRT is efficient insert which improves transfer rate and fluid friction favorably with 40% blockage area. Also overall thermal performance decreased with higher Reynolds number.
Summary of study determine, turbulence magnitude may be predicted higher than smooth tube by obstructing flow beneficially by various arrangements in tube. Current manuscript comes with an idea of different angular inclinations of circular ring turbulators (CRTs). “Main focus of study is to demonstrate effect of different geometrical parameters (angular deviation w.r.t. horizontal) of CRT on rate of heat transfer and fluid friction”. Each insert has a Circular ring with the same diameter ratio; PDR is 2 along with 40% FBA. The CRT of various angular inclinations (-10°, -15°, -20° and -25°) has been selected to check influence on thermal performance. The inserts were tested for turbulent model with Reynolds number from 6,000 to 24,000.

Model Description
Schematic configuration of simulation model is display in figure 1. Test section i.e. heat exchanger tube is open-type model having blower attachment of 2 HP capacity. To measure the flow rate, U-tube manometer is connected to orifice meter at the calming section. Calibration of orifice meter is done by using the anemometer and it has been fixed up according to ASME standard. According to the relation given [34] to smooth tube, distance of calming part is predicted at maximum Reynolds number. “It is taking 1.4 m which offers flow fully develops before entering the test section also uniform fluid entry inside the tube a bell mouth shape opening is given”.

![Schematic model](image)

Figure 1.Illustration structure.

Testing model is a hollow circular tube consist length of 1 m with inside diameter 49 mm and thickness is provided 3 mm framed by galvanized iron. Selection of test section length is so to predict measurable pressure as well as temperature variation at moderate Reynolds number. Bunch of 16 K-type temperature indicator are mounted on tube wall to predict surface temperature (T_s) of test section. To permits accuracy in the range bound of ±0.1° thermocouples are calibrated accordingly. “Pressure indicator is fixed at the end of test section which is attached to micro-manometer to predict significant pressure drop along the test section, outside surface is enclosed with Nichrome winding wire heater having 1.2 Ω/m resistance and single phase variac transformer is adapted to maintain a tube wall condition at constant heat flux of 1000 W/m² along with control input power”.

![Typical circular ring turbulators](image)

Figure 2.Typical circular ring turbulators.

| Variables data                  | Specification |
|---------------------------------|---------------|
| Dimension of Calm part (mm)     | 1400          |
| Dimension of test model (mm)    | 1000          |
| Inside dia. of tube (mm)        | 49            |
| Outside dia. of tube (mm)       | 55            |
| PDR                            | 2.0           |
| Re                             | 6000–24000    |
| Temperature of fluid (air)      | At ambient condition |
| Angular deflection of CRTs      | 0°, -10°, -15°, -20°, -25° |

Table 1. Details of parameters
To minimize losses to surrounding from exchanger, asbestos insulation with glass wool layers of thickness 5 cm is provided. Two set of thermocouples used to distinguish working fluid temperature at inlet as well as outlet. While testing, fluid i.e. air is pass throughout model by blower. Flow control valve is attached to tube for adjusting flow inside the round test section and obtain required Reynolds number. The CRT made by mild steel having 3 mm thickness was fitted in the test section display in figure 2. Specification of dimensions and variables modify within experiment display in table 1.

![Figure 3. CRTs having angular deflection of -10°, -15°, -20° and -25° fitted in testing model.](image)

Simulation models of various angular deflections -10°, -15°, -20° and -25° from the horizontal is shown in figure 3. McClintock and Kline adopted method for uncertainty inspection [36]. Uncertainty equation expressed as,

$$\mu_F = \left[ \frac{\partial F}{\partial a_1} \mu a_1 \right]^2 + \left[ \frac{\partial F}{\partial a_2} \mu a_2 \right]^2 + \ldots + \left[ \frac{\partial F}{\partial a_n} \mu a_n \right]^2 \right]^{\frac{1}{2}}$$

(1)

Such as F= F (a_1, a_2, ..., a_n) with a_n as parameter which influence on outcome of F.

“For parameter Reynolds number i.e. non-dimensional, friction factor and Nusselt number higher uncertainties were estimated to be 6.4%, 8.3%, and 7.2% respectively, for the average axial velocity predicted uncertainty was found to be less than ±7%; the uncertainty in pressure drop predicted has been found as ±6%, while the uncertainty in surface temperature was estimated about ±0.5%.”

**Data computation**

**Calculation of Parameters**

At steady state condition all the data were measured during conducting tests. To achieve steady state phenomena boundary temperatures offers fixed data. Current simulation values was additionally utilize for prediction of the pressure drop, fluid friction, coefficient of heat exchange, Nusselt number, and thermal performance criteria for dissimilar geometry and motion condition. Reynolds number is given by $R_e = \frac{UD}{\nu}$ where $D$, $U$ and $\nu$ are the diameter of tube, velocity of air and the working fluid kinematic viscosity respectively. Enhancement in heat transfer indicates by Nusselt number which is dimensionless number expressed as $N_u = hD/k$ in particular operating criteria, where $h$ is heat transfer coefficient and $k$ is conductivity of material used.

“At steady state condition, the convective heat transfer ($Q_{convection}$) of tube from inner surface is equal to the heat gained ($Q_a$) by air. Due to insulation provided radiation effect from wall of tube to air is considered negligible”.

$$Q_{convection} = Q_a$$

(2)

$$hA(T_w - T_p) = mC_p(T_2 - T_1)$$

(3)

Where,

$$T_w = \frac{(16 + 19 + \ldots + 16t_k)}{16}$$

And

$$T_p = \frac{(1 + 1 + \ldots + t_k)}{2}$$

Parameter of heat transfer, $h$ is predicted from enthalpy (3). Fluid friction, $f$ is given by,

$$f = \frac{\Delta p}{\rho \frac{v^2}{2}}$$

(4)

Prediction of the overall outcome of CRTs calculated within various conditions with modifies variables. Net overall ingredient ($\eta$) consists heat exchange along with moving flow factor synchronously having ratios of $(Nu_t/Nu_p)$ to $(f_t/f_p)$ expressed below.

$$\eta = \frac{(Nu_t/Nu_p)}{(f_t/f_p)^2}$$

(5)
Determination of results

Validation of testing model

Investigated data regarding friction factor as well as Nusselt number were predicted from the values obtained from round exchanger. Calculated data compared with outcome received from Blasius and Dittus-Boelter relation to validate experimental methodology adopted given in the literature. For fully developed turbulent fluid flow tube Dittus and Boelter correlation is expressed as

\[
Nu = 0.023Re^{0.8}Pr^{0.4}
\]  

(6)

![Figure 4. Validation of Nu for current model.](image)

Blasius relation to evaluate fluid friction for the smooth tube in turbulent region, expressed as

\[
f = 0.316 \left(\frac{1}{Re}\right)^{\frac{1}{4}}
\]  

(7)

![Figure 5. Validation of f for current model.](image)

Image 4 shows that Nusselt number obtained from correlation expressed by Equation (6) are comparable with Nu determined from simulation values of circular exchanger with a highest deviation of 14%.

Friction factor value for smooth tube obtained from experiment was compared with those calculated from the Blasius correlation with highest deviation 8.4% which shown in Figure 5.

Numerical Study

Adoption of turbulence model

In practical engineering applications most commonly used turbulence models, consist the realizable k-ε, standard k- ω and SST k- ω model were helpful to predict three-dimensional turbulent flow, the steady-state and Thermal characteristics of model [37]. The simulation data obtained from each models was compared with testing result for smooth tube; observation made that calculation of Nu by realizable k-ε model is more correspondents to investigated values than other models.

Comparatively, the highest variation for the average Nusselt number for numerical results and experimental data by using realizable k-ε model, the standard k-ω model and the SST k- ω model are 8.9%, 7.8% and 6.5%,
respectively, concluding that the $k$-$\varepsilon$ Model is more reliable than other models”. Therefore, for further numerical investigations $k$-$\varepsilon$ Model was applied in the present work shown in figure 6.

**Grid independence study**

“The fluid domain was discretized by unstructured mesh generated with the commercial software ANSYS CFD 18.0 as shown in figure 7, to obtain more accurate simulation results it is ensure that $y^+$ remained less than 1, prism grids are orthogonal to the wall surfaces and with faces perpendicular to the boundary layer fluid flow direction, were extruded from the surface of the smooth tube”.

“A hex core mesh, with hexahedron grids covering the majority of the volume, was created to obtain the characteristics of the fluid flow in the major region, to cover the areas inside the surface of the prisms and the hex core, tetrahedron grids used”. Compute highest accuracy of simulations, six bunches of grid, with 286868, 377728, 506131, 842525, 996171 and 999845 count size, were offered in computation to check appropriate grid study at $Re$ 24000. Comparative Nu as well as $f$ data determined from grid is display in figure 8.
The results indicate that the deviation between the calculated values for grids comprising 996171 and 999845 elements is about 0.9% for Nusselt number and 1.2% for friction factor, indicating that the grid system with 996171 elements is adequately dense for the simulations shown in fig 9, hence grid system with 996171 elements was approved in the subsequent simulation reflecting a compromise of computational time and precision in solution.

In the present simulation work, a uniform heat flux of 1000 $\text{W/m}^2$ is specified as boundary condition on the tube wall. Before simulation inlet tube is extruded by 10 times diameter to achieve fully enlarge velocity outline. Pressure, boundary conditions concerned at outlet region. No-slip condition is offered on the surface of tube wall and turbulator surface.

Governing factors such as energy, momentum and continuity equations are computed within steady state phenomena. For momentum and convective diffusion evaluation, QUICK method (Quadratic upstream...
interpolation for convective kinetics differencing) scheme and SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) method is used for velocity-pressure relation and for pressure variation”. To capture accurate results near-wall, enhanced wall treatment model is used.

“A solution is supposed to be converged when residuals were less than $10^{-9}$ for the energy equation and less than $10^{-5}$ for other of the governing relation”. Velocity variation contour at Reynolds number 9000 and 24000 displayed on three planes for various angular deflections of -10°, -15°, -20° and -25° shown in figure 10a and figure 10b. “In case of a CRTs, behind the ring the phenomenon of secondary reverse direction flow is higher which gives more effective separation of flow and tiny boundary thickness in that region, which promotes the strong wall shear stress on a tube”.

**Accomplishment of heat transmit**

Figure 11[a, b] present influence of CRTs having angular deflections of 0°, -10°, -15°, -20° and -25° and Re from 6000 to 24000 on heat exchange ($Nu$ and $Nu_t/Nu_p$) resp. Figure 11(a) gives diversity of Nu along with Re for tested turbulators.

![Figure 11a.Computing Re vs Nu](image)

The main aim to predict; enhancement in heat transfer achieved, while testing with CRTs comparably higher 55% to 75% than the round plain tube. This is achieved because; it advances not only rupture of the thermal boundary layer but also appropriate and swift fluid mixing along with enhancement in heat exchange, produced higher turbulence and secondary fluid flow with eddy’s as well as vortex structure across CRT shown in figure 9.

![Figure 11b.Computing Re vs $Nu_t/Nu_p$](image)
Figure 11(b) gives the ratio of the theoretical to plain tube $\frac{Nu_t}{Nu_p}$ which reduces with raising Re in the range of 6000 to 24000 for all tested inserts. As results show, CRT having angular deflection of -10°, -15°, -20° and -25° improves rate of enhancement in heat transfer respectively around 1.43-1.95, 1.58-2.06, 1.65–2.304 and 1.747–2.51 times than that of smooth tube exchanger. It is observed that, obstruction across moving flow is higher in case of larger angular deflection i.e. -25°. “Because of additional blockage, extreme secondary flow is generated across the heated wall which leads to tiny thermal boundary thickness”. Position at which CRT is in contact with a heated medium, which shows desolation of boundary thickness offers suitable circumstance to heat exchange. At, maximum turbulence magnitude with maximum Re shows convection transfer more favorably.

**Characteristics of fluid friction**

Effect of CRTs having angular deflection of 0°, -10°, -15°, -20° and -25° on fluid friction factor with Reynolds number vary from 6000 to 24000 shown in Figure 12(a, b).

![Figure 12](image)

Figure 12[a]. Computing Reynolds number vs $f$

Figure 12[a] indicates, increment in Reynolds number causes slightly decrease in the friction factor for all tested inserts. Among the tested CRT with maximum angular deflection i.e. -25° indicates higher pressure drop for highest Reynolds number due to stronger secondary flow and higher turbulence intensity.

![Figure 12](image)

Figure 12(b). Computing Reynolds number vs $\frac{f_t}{f_p}$

Figure 12(b) gives the ratio of the theoretical to plain tube friction factor ($\frac{f_t}{f_p}$) which increases with increases Re all tested inserts. This show at lower angular deflection value of friction factor is higher than other inserts. Also, the friction factor decreases as Reynolds number increase with higher CRT. The insert with angular geometry of -25° has the lowest friction factor ranging from 125%–210% which is greater than that of the smooth
round tube. In the range bound of 6000 to 24000 Reynolds number, CRT with angular deflection of 0° and -10° gives the maximum value of friction factor in the range of 165%-250% and 149%-230% respectively. The having angular deflection i.e. (-25°) and Re from 6000 to 24000 gives maximum pressure reduction 18.5 to 31.7 times additional than smooth tube.

**Effect on overall achievement factor**

To demonstrate overall achievement of CRTs having angular deviation of -10°, -15°, -20° and -25° for Reynolds number ranging 6000 to 24000, it’s mandatory to measure overall performance factor (η).

![Figure 13 Reynolds number vs η](image)

“Improvement in heat exchange as a result of flow separation is attendant by an increase in pressure drop. Overall performance factor includes heat transfer characteristics \( \frac{N_u}{\nu} \) and fluid friction performance \( \frac{f_t}{\nu} \) favorably”. The significant of passive method adapted effectively if offers the η greater than 1. Influence of Re as well as CRT with inclination of -10°, -15°, -20° and -25° on η display in figure 13. It shows overall thermal characteristics for all CRTs under test decrease with increase in Reynolds number. This happens due to friction across fluid offers dominant effect on the heat transfer rate at higher Re. “CRT including maximum angular deviation of -25° gives larger thermal performance factor. Also, decrease in Reynolds number tends to increases in η”. Maximum η predicted is 1.77 for a CRT with angular deviation (-25°) at Reynolds number 6000.

**Empirical correlations**

Using present experimental data empirical correlations was obtained for Nu, f and η expressed by equation 8, 9 and 10 respectively.
These are,
\[ N_u = 0.1574R_e^{0.662}P_r^{0.3} \]  \( (8) \)
\[ f = 0.5632R_e^{-0.209} \]  \( (9) \)

Figure 16 shows comparative investigation of predicted versus experimental sheet. “For Nusselt number, friction factor and Overall performance factor maximum fluctuation occurs in the predictive data is 11%, 7.8%, and 5% respectively”.

\[ \eta = \left( \frac{Nu_t}{Nu_p}(f_t/f_p) \right)^{-0.3602} \]  \( (10) \)

Installation and manufacturing of the CRT inserts used in the current work are very simple as well as cost-effective. Also comparison of wall shear stress for all tested inserts with smooth tube, indicates wall shear stress is high for higher Reynolds number and at maximum angular deviation \(-25^\circ\). At higher rate of change of shear wall stress we get higher heat enhancement at higher Reynolds.

**Illustration with literature**

“Some of the numerically published data from the literature has been given in table 2 to compare the present investigation with equivalent methods”. Installation and manufacturing of the CRT inserts used in the current work are very simple as well as cost-effective. Also comparison of wall shear stress for all tested inserts with smooth tube, indicates wall shear stress is high for higher Reynolds number and at maximum angular deviation \(-25^\circ\). At higher rate of change of shear wall stress we get higher heat enhancement at higher Reynolds.

| Sr. No | Geometry                               | Nu\(_t\)/Nu\(_p\) | \( \eta \) |
|--------|----------------------------------------|--------------------|------------|
| 1      | Conical Ring [38]                      | 3.3                | 1.8        |
| 2      | Circular Ring with twisted tape [39]   | 4.5                | 1.42       |
| 3      | Conical ring with twisted tape [40]    | 3.67               | 1.95       |
| 4      | Conical nozzle with snail entry [41]   | 4.1                | 1.10       |
| 5      | Perforated conical ring [42]           | 3.8                | 54         |
| 6      | Hexagonal ring turbulator [43]         | 3                  | 1.34       |
| 7      | **Present Work**                       | **2.5**            | **1.72**   |

Table 2. Current work compare with other study

**Conclusions**

In current investigation work, a new modified geometry of turbulator i.e. Circular ring turbulator consisting negative angle deflection from the horizontal of tube was examined in rounded heat exchanger tube with working fluid air at ambient condition and influence on the rate of pressure drop, heat transfer and fluid friction with CRT including angular deviation of \(-10^\circ\), \(-15^\circ\), \(-20^\circ\) and \(-25^\circ\) were analyzed by changing the Re in the span of 6000 to 24000. The observations defined on different parameters are summarized below.

1. “CRT Insert improves the rate of heat transfer favorably as well as increases a pressure loss over a smooth tube within the tested range of all the parameters”.

(2) “Heat transfer rate increases with increasing the angular deflection towards the inlet and increasing Reynolds number. Also, by increment in Reynolds number having higher angular deflection shows increment in pressure drop”.

(3) “For all tested inserts with increase in angular deviation tends to slightly decreases friction factor with increasing Reynolds number. Friction factor (f_r/f_p) which increases with increases in Reynolds number in the range of 6000 to 24000 for all tested inserts. This demonstrates at lowest angular deflection geometry friction factor is higher than other tested inserts”.

(4) “Overall performance factor increases with decrease in Reynolds number for all tested inserts. Also, for significant Reynolds number, overall performance factor decreases with increase in angular deflection across the flow”.

(5) “Nusselt number (Nu_r/Nu_p) which decreases with increases in Reynolds number in the range of 6000 to 24000 for all tested inserts. Greater heat transfer improvement (Nu_r/Nu_p) 1.747–2.51 is obtained for insert angular deviation of (~25°). Also maximum thermal overall performance factor of 1.77 was achieved for a CRT with angular deviation (~25°) at Reynolds number 6000”.

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