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# Experimental Investigation of Thermal Performance in a Tube With Detached Circular Ring Turbulators

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*This article deals with the experimental investigation of thermal performance of detached circular ring turbulators in turbulent flow regime. The experiments were conducted by four different circular rings. Two different pitch ratios (1 and 2) and two different distances (1 mm and 2 mm) at which the circular rings were detached from the tube wall were considered in the experiments. Uniform heat flux was applied to the external surface of the tube and Reynolds number was varied from 4350 to 24,920. The tube with circular ring turbulators introduced both higher heat transfer and higher pressure drop than the smooth tube, depending on pitch ratio and distance. According to the experimental results, the Nusselt number and friction factor increased with decreasing of pitch ratio and distance. Therefore, the circular ring turbulator with a pitch ratio of 1 and distance of 1 mm offered the highest heat transfer rate accompanied by the highest pressure drop. The highest thermal performance efficiency of 26% was also achieved for this case at a Reynolds number of 4421. Evidently, the experimental results revealed that the best operating regime of all circular ring turbulators was found at low Reynolds number, leading to a more compact heat exchanger.*

## *INTRODUCTION*

High-performance thermal systems are of great importance in many engineering applications. Many efforts have been made on heat transfer enhancement according to the progress of thermal systems. The recent research studies in heat transfer enhancement lead to the development of currently used heat transfer techniques. These research studies focused on finding a technique not only increasing heat transfer, but also achieving higher efficiency. Achieving higher heat transfer rates through various enhancement techniques can result in substantial energy savings and in more compact and less expensive equipment with higher thermal efficiency.

Heat transfer enhancement technology has been improved and widely used in heat exchanger applications such as refrigeration, the automotive industry, process industry, chemical industry, solar water heaters, etc. Placing different-shaped turbulators with different arrangements is one of the heat transfer enhancement techniques extensively used in channel or tube flow due to the advantages of easy fabrication, operation, and maintenance. There are numerous studies on heat transfer enhancement by inserting various turbulators such as ribs [1–3], fins [4–8], baffles [9–12], rings [13, 14], twisted tapes [15–18], and coiled wire inserts [19–22].

In the last few years, several researchers conducted studies with conical or circular rings. The effects of the conical ring turbulators with different arrangements such as converging conical ring (*CR* array), diverging conical ring (*DR* array), and converging-diverging conical ring (*CDR* array) on the heat transfer, friction factor, and overall heat transfer enhancement were studied by Promvonge [23]. The results showed that the rings with *DR* array provided better thermal performance factor than the rings with the *CR* and *CDR* arrays. Another study with conical ring turbulators was performed by Durmus [24]. Four conical angles (5°, 10°, 15°, and 20°) were considered in the study, and it was reported that both heat transfer rates and friction coefficients increased with increasing turbulator angles.

Yakut and Sahin [25] experimentally investigated the flow-induced vibration characteristics of conical ring turbulators with different pitches used for improving the heat transfer in heat exchangers. Their findings showed that the conical ring

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turbulators increase the heat transfer and friction factor, and also produce vortices in the flow. The highest heat transfer was achieved for the smallest pitch arrangement. Yakut et al. [26] determined the thermal performance factor at constant pumping power and also evaluated the entropy generation in another study with conical ring turbulators having different pitches. The combined effects of the conical rings together with twisted tape to enhance the heat transfer in a circular tube were introduced by Promvonge and Eiamsa-ard [27]. Their experimental results revealed that heat transfer rate in the tube with the conical rings together with twisted tape was determined to be approximately 10% over that in the tube with the conical ring alone.

The heat transfer enhancement mechanism with ring turbulators is based on the flow separation and reattachment and provides an unfavorable effect in increasing the pressure drop for the enhanced tube. Both hydrodynamic and thermal boundary layers on the wall will become thinner in the case of the secondary flow occurrence, which causes the heat transfer in that region to be enhanced. However, the reverse stimulated heat transfer enhancement causes an unavoidable shear stress and pressure drop with ring turbulators inserted in the tube. Recently, Kongkaitpaiboon et al. [28] introduced an experimental study pertaining to the convective heat transfer and pressure loss in a tube fitted with circular ring turbulators in a range of Reynolds number varying from 4000 to 20,000. Three different pitch ratios ($P/D = 6$, 8 and 12) and three different diameter ratios ($d/D = 0.5$, 0.6 and 0.7) were considered in the study. As reported, the highest heat transfer rate and the largest pressure loss were provided by the circular ring turbulators smallest pitch and diameter ratios. In another study, Kongkaitpaiboon et al. [29] used perforated conical rings with three different pitch ratios ($P/D = 4$, 6, and 12) and three different numbers of perforated holes (4, 6, and 8 holes) for heat transfer enhancement in a tube under uniform wall heat flux condition. The obtained thermal performance analysis results at the same pumping power demonstrated that the perforated conical rings are more efficient than the typical circular rings in terms of overall heat transfer enhancement.

As mentioned in the literature review, most researchers are focused on investigating the effect of pitch ratio (*P/D*), diameter ratio (*d/D*), conical angles, or arrangement type on heat transfer and pressure drop in a tube with circular or conical ring turbulators. These ring turbulators are attached on the tube wall for enhancing the heat transfer by increasing the effective heat transfer area and disturbing the laminar sublayer. It is also known that there may be an additional resistance to heat transfer in the case of attaching the inserts onto the tube wall due to the formation of contamination with time. Accordingly, detaching the inserts from the tube wall may be appropriate in terms of preventing the occurrence of contamination.

The present experimental study differs from those available in the literature, in that the detached circular ring turbulators (*CR1*, *CR2*, *CR3*, and *CR4*) were used in order to improve the heat transfer by only disturbing the laminar sublayer. The experiments were carried out in the range of Reynolds number from 4350 to 24,920. Two different pitch ratios ($P/D = 1$ and $P/D = 2$) and two different distances ($s = 1$ mm and $s = 2$ mm) from the tube wall were considered. The effect of pitch ratio (*P/D*) and distance (*s*) from the tube wall on Nusselt number and friction factor were investigated and finally the thermal performance results were also presented for investigated cases (*CR1*, *CR2*, *CR3*, and *CR4*).

## *EXPERIMENTAL RIG*

The details of the experimental rig used in the experiments are schematically shown in Figure 1a. The rig consists of a blower, a nozzle for air entering the tube uniformly, a venturimeter to measure the flow rate, a calming tube (6000 mm), and the heat transfer test tube (3000 mm) with circular ring turbulators. A solid model of the ring turbulators inserted in the tube is shown in Figure 1b. The SS304 seamless steel test tube has 56 mm inner diameter ($D$), 60 mm outer diameter ($D_o$), and 2 mm thickness ($t$). Two different pitch ratios ($P/D = 1$ and $P/D = 2$) and two different distances ($s = 1$ mm and $s = 2$ mm) from the tube wall were considered in the experimental study. The physical dimensions of the circular rings used in the experiments are given in Table 1.

The circular ring turbulators used in experiments were made of soft iron with 2.8 mm diameter. The rings were connected by welding along a line with a connection wire with 3 mm diameter, as depicted in Figure 1c. The smooth connection surfaces were obtained by sensitive grinding of the welding slag at welding points. The separators were used in order to detach the circular ring turbulators from the tube wall. The separators were manufactured from Teflon rods by precision turning. The circular ring turbulators pass through the separators by 120° arrangement, and thus the contact of the rings with tube inner wall was prevented. The circular ring turbulators used in the experiments are illustrated in Figure 2.

The test tube was heated by an electrical cable in order to achieve uniform heat flux along the entire length of the test section. The electrical output was controlled by a Variac transformer. The outer surface of the test tube was well insulated with glass wool to reduce convective heat loss to the surroundings. The surface temperatures of the tube wall were measured by 20 K-type thermocouples, which were located on the outer wall of

**Table 1** The physical dimensions of the circular rings used in the experiments

| Case | $S_i$ (mm) | $S_o$ (mm) | $s$ (mm) | *P/D* | Number of circular rings |
|---|---|---|---|---|---|
| *CR1* | 25.6 | 27 | 1 | 1 | 54 |
| *CR2* | | | | 2 | 27 |
| *CR3* | 24.6 | 26 | 2 | 1 | 54 |
| *CR4* | | | | 2 | 27 |

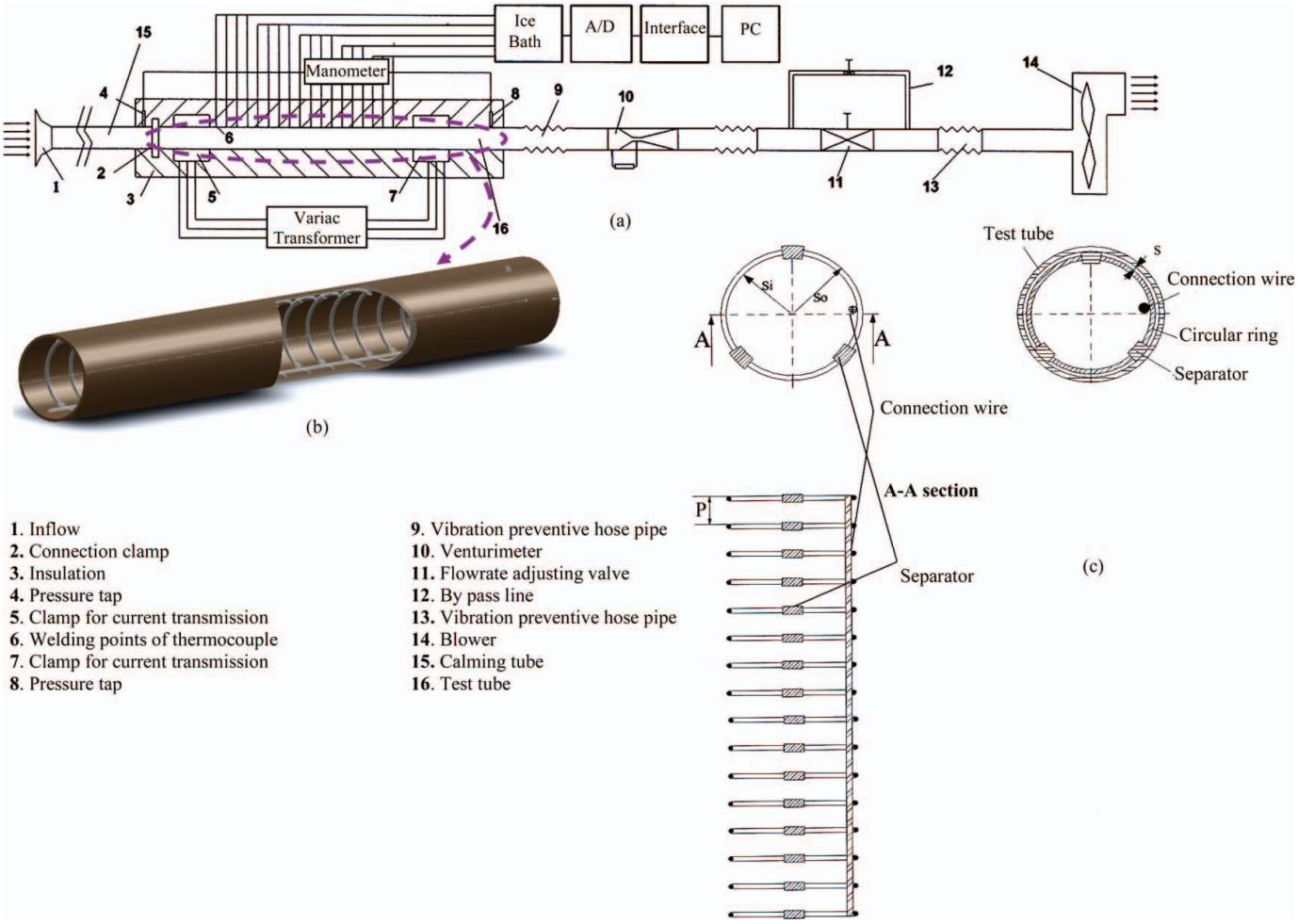



**Figure 1** (a) Schematic view of experimental rig. (b) Solid model of the test tube with circular ring turbulators. (c) The details of circular ring turbulators. (Color figure available online.)

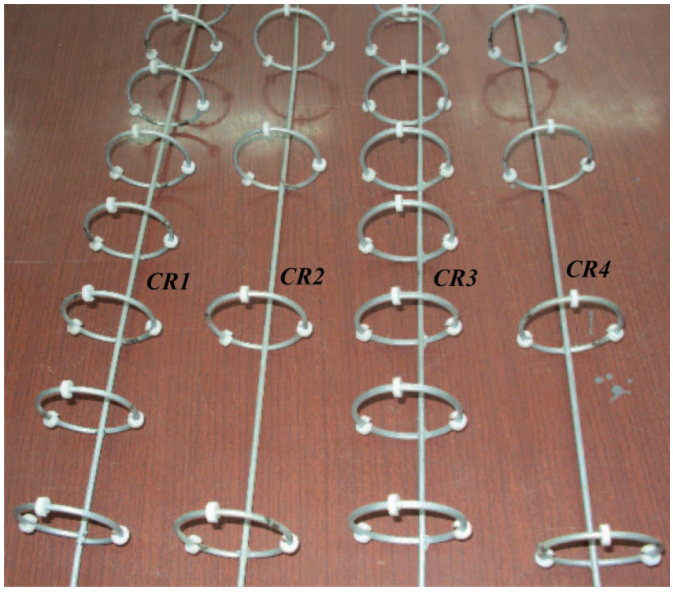



**Figure 2** The circular ring turbulators used in the experiments. (Color figure available online.)

the test tube, and the outputs of thermocouples were obtained by passing through anice bath used as a reference temperature, thereafter acquired via a data logger. Four thermocouples (1, 2, 3, 4) that were in the same axial direction were fixed at the end of the heated test tube in order to determine whether the applied heat flux was uniform in radial direction or not. Consequently, in preliminary tests, it was seen that the readings were same for these four thermocouples and the heat flux was uniform in the radial direction. The locations of the thermocouples through the test tube and the connection of the thermocouples (1, 2, 3, and 4) are given in Figure 3. The inner and outer temperatures of the bulk air were also measured by K-type thermocouples at certain points. A piping system was used to remove the heated air in the test tube from the experimental field to the atmosphere. Therefore, the temperature of the experimental field ($T_\infty$) was not changed considerably during the experiments, so it was negligible.

The inlet bulk air was first drawn by vacuum through the calming tube and later the air was directed through the test tube by a 4-kW blower. After passing through the test tube, the air entered the venturimeter, which was used for flow

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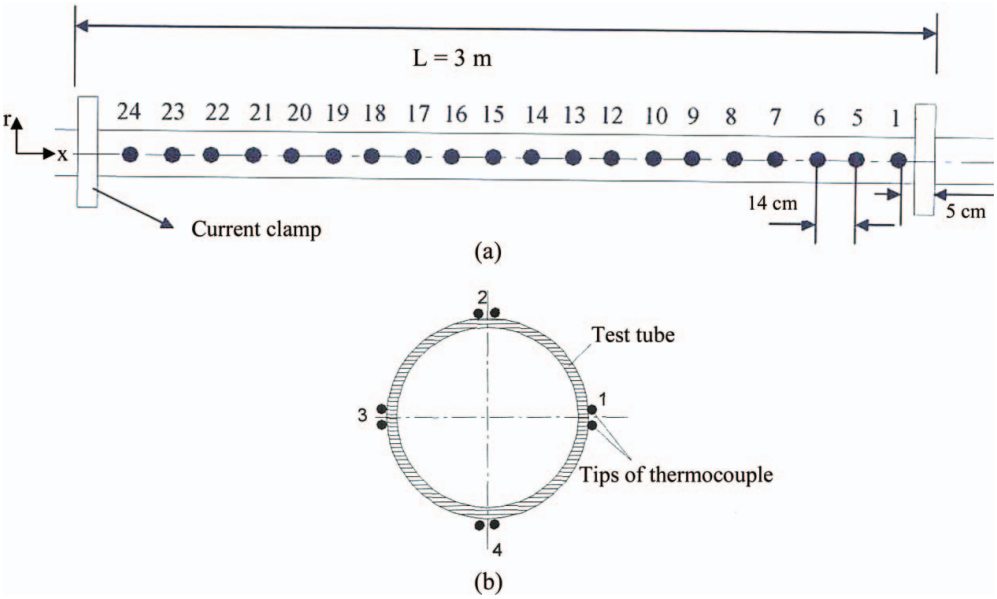



**Figure 3** (a) The location of thermocouples through the test tube. (b) The connection of the thermocouples (1, 2, 3 and 4) at the end of the test tube. (Color figure available online.)

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rate measurements. The venturimeter was calibrated before the experiments and the calibration procedure is given in detail in reference [30]. The volumetric air flow rates from the blower were adjusted manually by varying motor speed through a valve. The low pressure drop and high pressure drop across the test tube were measured by using an inclined U-tube manometer and a U-tube manometer, respectively. The density ($\rho$) of the manometer fluid was 780 kg/m$^3$. Experiments were conducted by varying the flow rate and electrical power. The test tube was heated during the experiments and of the temperature, volumetric flow rate, and pressure drop of the bulk air and electrical output were recorded as data after the system was allowed to approach the steady-state condition. The data except for the volumetric flow rate and differential pressure drop were recorded by a multi-channel data logger with high resolution. The Reynolds number of the bulk air varied from 4350 to 24,920. The Nusselt number, the Reynolds number, and the various characteristics of the flow were calculated based on the average tube wall temperatures and the inlet and outlet air temperatures by a computer program written in FORTRAN. The local wall temperature, inlet and outlet air temperatures, pressure drop across the test section, and volumetric flow rate were measured for investigating the heat transfer of the heated test tube. The method introduced by Kline and McClintock [31] was used in order to determine the uncertainties of experimental measurements. The uncertainty calculation method used involves calculating derivatives of the desired variable with respect to the individual experimental quantities and applying known uncertainties. As a result, the maximum uncertainties of the nondimensional parameters were found to be $\pm 4.3\%$ for Reynolds number, $\pm 9.8\%$ for Nusselt number, and $\pm 8\%$ for friction factor.

## *DATA REDUCTION*

The uniform heat flux applied to the test tube causes an increase in the outer surface temperature ($T_{ow}$) of the test tube in the axial direction. Therefore, the heat loss was calculated for each part of the test tube in which the thermocouples exist. The total heat loss was taken as the sum of these 20 parts. The heat loss $Q_{loss}$ is the heat transfers from the outer tube wall to the surroundings and is calculated as follows:

$$Q_{loss} = F(T_{ow} - T_{\infty}) \tag{1}$$

$$\frac{1}{F} = \frac{1}{h_o A_{ins}} + \frac{c}{k_{ins} A_{12}} \tag{2}$$

In Eq. (2), $h_o$ indicates the heat transfer coefficient of the natural convection that occurs between the outer surface of the insulated test tube and the surroundings and is determined by iterative computation via FORTRAN computer code. $A_{ins}$ and $k_{ins}$ represent the surface area and conductive heat transfer coefficient of the insulated test tube, respectively. $A_{12}$ is the log mean area of the insulated test tube surface and the test tube outer surface.

The steady-state heat transfer rate and heat flux applied to the test tube can be written as:

$$Q_{air} = Q_{conv} \tag{3}$$

$$Q_{air} = \dot{m} C_{p,\,air} (T_0 - T_i) = \Delta VI - Q_{loss} \tag{4}$$

The heat provided by the electrical cable in the test tube was about 4–5% higher than the heat absorbed by the air for the thermal equilibrium test because of the convection and

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radiation heat losses ($Q_{loss}$) from the test section to the surroundings. Therefore, only the heat transfer rate absorbed by the air was taken into consideration for the convective heat transfer coefficient calculation.

$$q = \frac{Q_{air}}{\pi D_o L} \tag{5}$$

The local convective heat transfer coefficient through the heated test tube in any axial $x$ direction was defined as:

$$h(x) = \frac{q}{T_{iw}(x) - T_b(x)} \tag{6}$$

Here, $T_{iw}(x)$ and $T_b(x)$ represent the local inner wall temperature of the heated test tube and local bulk temperature of the fluid, respectively. The determination of these temperatures is given in detail in reference [30]. All of the thermophysical properties of air were determined at the overall bulk mean temperature.

Consequently, the local Nusselt number can be calculated from:

$$Nu(x) = \frac{h(x)D}{k} \tag{7}$$

where $k$ is the thermal conductivity of fluid.

There was almost no difference between the local and mean Nusselt numbers for fully developed flow. Therefore, the average of last three local Nusselt numbers at the end of the test tube was taken into account for presenting the heat transfer results.

The Reynolds number is defined by:

$$Re = \frac{UD}{\nu} \tag{8}$$

The friction factor ($f$) was calculated as follows:

$$f = \frac{\Delta P}{\frac{1}{2}\rho U^2 \frac{L}{D}} \tag{9}$$

where U indicates the mean fluid velocity in the tube.

According to constant pumping evaluation criteria [21],

$$(\dot{V}\,\Delta P)_s = (\dot{V}\,\Delta P)_c \tag{10}$$

and the relationship between the friction factor and Reynolds number can be given as follows:

$$(fRe^3)_s = (fRe^3)_c \tag{11}$$

$$Re_s = Re_c(f_c/f_s)^{1/3} \tag{12}$$

The thermal performance factor η was expressed as the ratio of the $h_c$ of an enhanced tube with circular ring turbulators to that of a smooth tube, $h_s$, at a constant pumping power, as introduced by Webb [32]:

$$\eta = \frac{h_c}{h_s}\Big|_{pp} = \frac{Nu_c}{Nu_s}\Big|_{pp} = \left(\frac{Nu_c}{Nu_s}\right)\left(\frac{f_s}{f_c}\right)^{1/3} \tag{13}$$

## *RESULTS AND DISCUSSION*

### *Validation of Smooth Tube Tests*

The present heat transfer and pressure drop results for a smooth tube were first validated in terms of Nusselt number and friction factor before the experiments with circular ring turbulators. The obtained experimental results of Nusselt number and friction factor for smooth tube were compared with the results obtained from the well-known steady-state flow correlations of Gnielinski [33] and Petukhov [34], for the fully developed turbulent flow in circular tubes.

The Gnielinski [33] correlation as given in Eq. (14) is used to find the heat transfer in smooth tubes:

$$Nu = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \quad 3000 \le Re \le 5\,x\,10^6 \tag{14}$$

The correlation of Petukhov [34] is given as follows:

$$f = (0.790 ln Re - 1.64)^{-2} \qquad 3000 \le Re \le 5\,x\,10^6 \tag{15}$$

Figure 4 reveals that the heat transfer results of the present study agree well with those obtained from Eq. (14), with the deviations within ±9.8% for the Reynolds number above 6000. The maximum deviation is ±21% at Re = 3115. Consequently, the experimental results of the present smooth tube in terms of the Nusselt number can be correlated as follows:

$$Nu = 0.094 Re^{0.653} Pr^{0.39} \quad 3115 \le Re \le 25000 \tag{16}$$

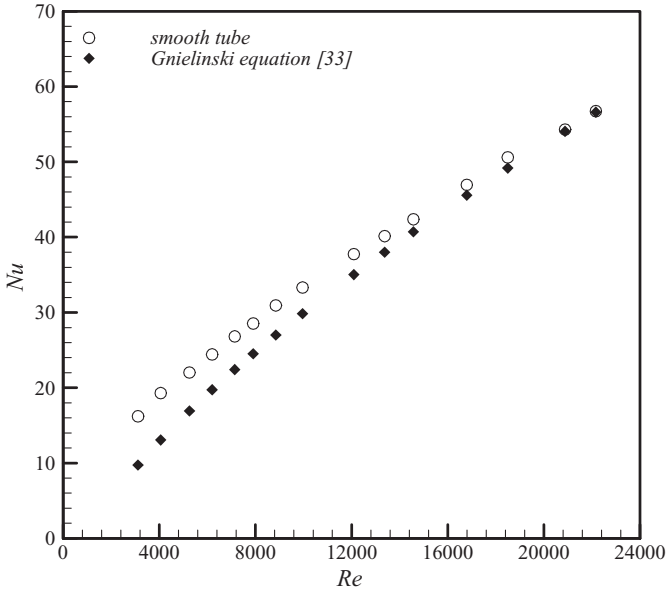



**Figure 4** Validation of smooth tube tests for Nusselt number.

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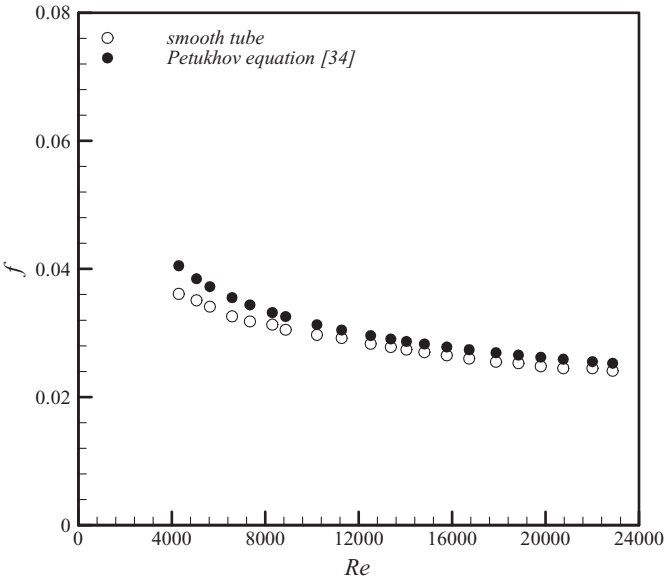



**Figure 5** Validation of smooth tube tests for friction factor.

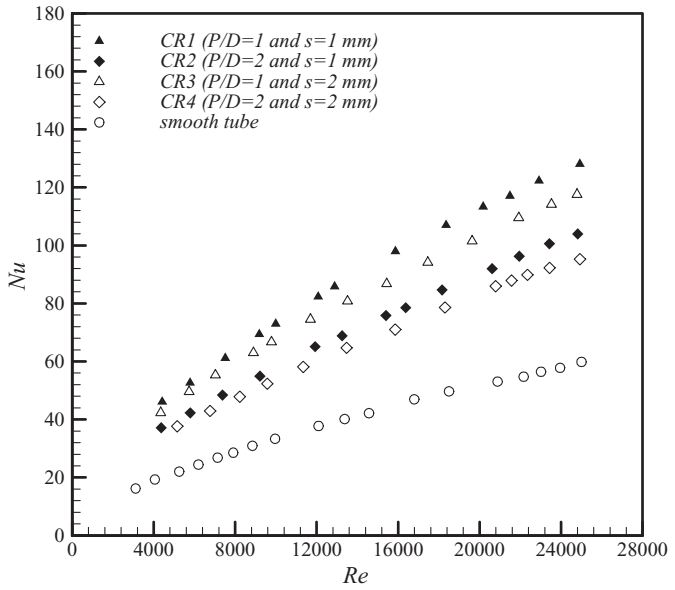



**Figure 6** Variation of Nusselt number with Reynolds number for different circular ring turbulators.

Figure 5 presents the friction factor results for smooth tube. As seen, the friction factor values of the present smooth tube are in good agreement with the correlation of Petukhov [34], with the deviations within ±8%. The correlation of friction factor for the present smooth tube can be expressed as:

$$f = 0.282Re^{-0.245} \qquad 4250 \leq Re \leq 27000 \quad (17)$$

The present smooth tube data fall within ±3.5% and ±2.5% for the predicted correlations of the Nusselt number and the friction factor, respectively.

### *Heat Transfer Results*

This study was performed pertaining to the investigation of thermal performance of detached circular ring turbulators in a tube under a uniform heat flux boundary condition. The experiments were conducted with four different circular ring turbulators (*CR1*, *CR2*, *CR3*, and *CR4*) having two different pitch ratios (*P/D* = 1 and 2) and two different distances ($s = 1$ mm and 2 mm) at which the circular ring turbulators were detached from the tube wall. The effects of pitch ratio (*P/D*) and the distance ($s$) on heat transfer and friction factor under turbulent flow conditions are shown in Figures 6 and 7, respectively. It is evident from these figures that the tube with circular rings introduces both higher heat transfer rate and higher flow friction than the smooth tube.

The variation of Nusselt number with Reynolds number for circular ring turbulators (*CR1*, *CR2*, *CR3*, and *CR4*) is presented in Figure 6. It is obvious from this figure that the heat transfer increases significantly with decreasing pitch ratio, because the circular rings with smaller pitch ratios increase the turbulence intensity and yield a better mixing of the flow in the tube. Heat transfer also increases with decreasing distance at which the circular ring turbulators were detached from the tube wall. The clearance between the wall and circular ring turbulators causse higher velocities occur in the clearance gap, thus increasing the heat transfer rate. The circular rings (*CR1* and *CR3*) that were placed at a distance of 1 mm from the tube wall provided higher heat transfer than the ones (*CR2* and *CR4*) at a distance of 2 mm from the tube wall, because the circular rings that were close to the near wall region interrupted the development of the boundary layer of the fluid flow better than the ones farther from the tube wall. Consequently, the highest and lowest heat transfer were achieved for the cases of *CR1* (*P/D* = 1, $s = 1$ mm) and *CR4* (*P/D* = 2, $s = 2$ mm), respectively. The increase

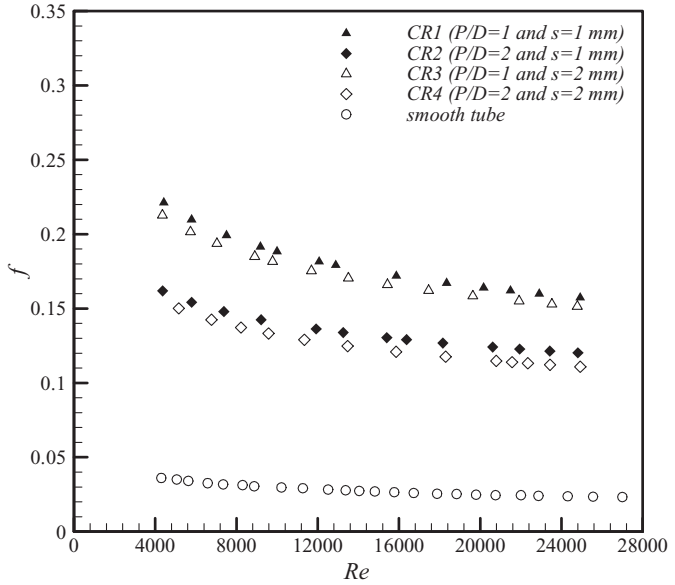



**Figure 7** Variation of friction factor with Reynolds number for different circular ring turbulators.

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in Nusselt number for the cases of *CR1* (*P/D* = 1, *s* = 1 mm) and *CR4* (*P/D* = 2, *s* = 2 mm) were in the range 115–132% and 60–72% compared to that of the smooth tube, depending on Reynolds numbers, respectively.

### *Friction Factor Results*

Effect of pitch ratio (*P/D*) and distance (*s*) on friction factor was introduced in Figure 7. Apparently, the friction factor tends to decrease with increasing Reynolds number. The circular ring turbulators caused a remarkable fluid friction as well, in comparison with the smooth tube, because of the dissipation of the dynamic pressure of the fluid due to higher surface area. The fluid friction increased with the decrease in pitch ratio (*P/D*) and distance (*s*). The number of rings increases by decreasing the pitch ratio. Therefore, the highest pressure drop was observed for the case of *CR*1 (*P/D* = 1, *s* = 1 mm) due to the fact that increasing the number of rings disturbed the entire flow field and caused more friction, while the lowest pressure drop was obtained for the case of *CR4* (*P/D* = 2, *s* = 2 mm). Flow passes through the clearance between the wall and circular rings, and fluid friction results with an increase because of the flow disturbance near the wall region due to the circular ring turbulators. The friction factors for the cases of *CR1* (*P/D* = 1, *s* = 1 mm) and *CR4* (*P/D* = 2, *s* = 2 mm) were on average 6.45 and 4.53 times more than for the smooth tube, respectively.

### *Thermal Performance Factor Results*

Figure 8 presents the dimensionless heat transfer ratio ($Nu_c/Nu_s$) versus Reynolds number for all investigated cases (*CR1*, *CR2*, *CR3*, and *CR4*). The dimensionless heat transfer ratio must be greater than unity for a net energy gain. The increase in heat transfer obtained by circular ring turbulators is remarkable and tends to decrease with the increase of Reynolds number, as indicated in Figure 8. The reverse flow provided by circular ring turbulators improves the convection of the tube wall by raising the mean velocity and temperature gradient and reducing the cross section of the flow field. The highest dimensionless heat transfer ratio of 2.3 was achieved in the case of *CR1* (*P/D* = 1, *s* = 1 mm), which interrupted the development of the boundary layer of the fluid flow and increased the turbulence intensity in the flow field better than the others. The case of *CR1* (*P/D* = 1, *s* = 1 mm) produced higher heat fluxes and momentum transfers depending on not only the large effective driving potential force but also the higher pressure drop in comparison with the other cases.

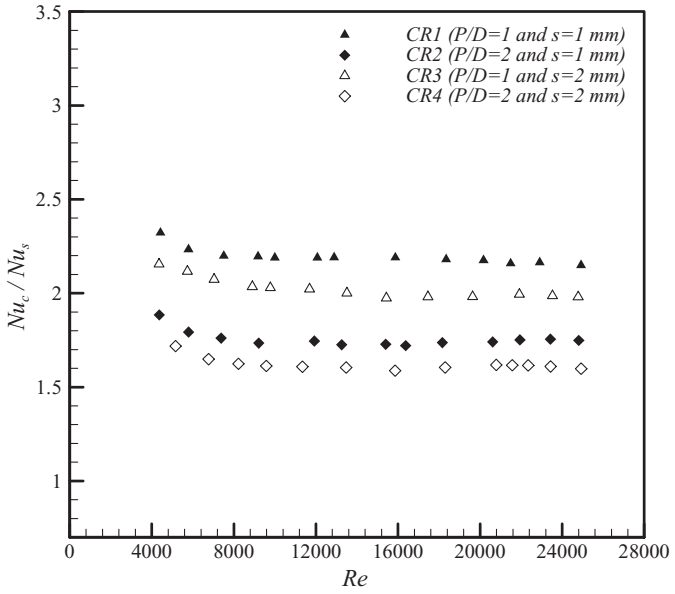



**Figure 8** Variation of dimensionless heat transfer ratio ($Nu_c/Nu_s$) with Reynolds number.

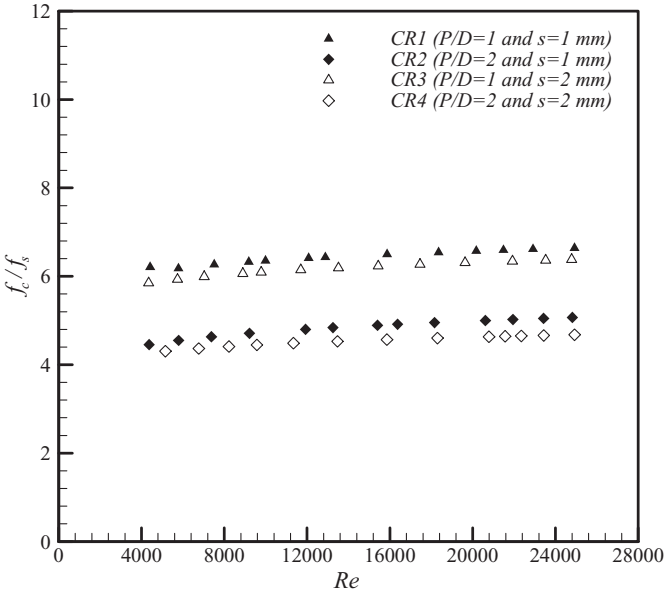



**Figure 9** Variation of friction factor ratio ($f_c/f_s$) with Reynolds number.

The variation of friction factor ratio ($f_c/f_s$) with Reynolds number was illustrated in Figure 9. The friction factor ratio increases with the decrease of pitch ratio (*P/D*) and distance (*s*), similar to the heat transfer rate. Obviously, the circular ring turbulators (*CR1* and *CR3*) that are close to the tube wall exhibit more flow resistance than the farther ones (*CR2* and *CR4*). The highest friction factor ratio of 6.3 was observed for the case of *CR1* (*P/D* = 1, *s* = 1 mm), in which there was higher surface area and flow blockage of the circular ring turbulators through the flow field.

Figure 10 introduces the variation of thermal performance factor (η) with Reynolds number for all cases. Heat transfer enhancement is provided with the expense of increasing pressure drop caused by circular ring turbulators. Therefore, a performance analysis is essential for the evaluation of the net energy gain to make a decision on whether the technique applied to improve the heat transfer is efficient from an energy point of view. The comparison was made based on the same pumping power in order to determine the net final gain.

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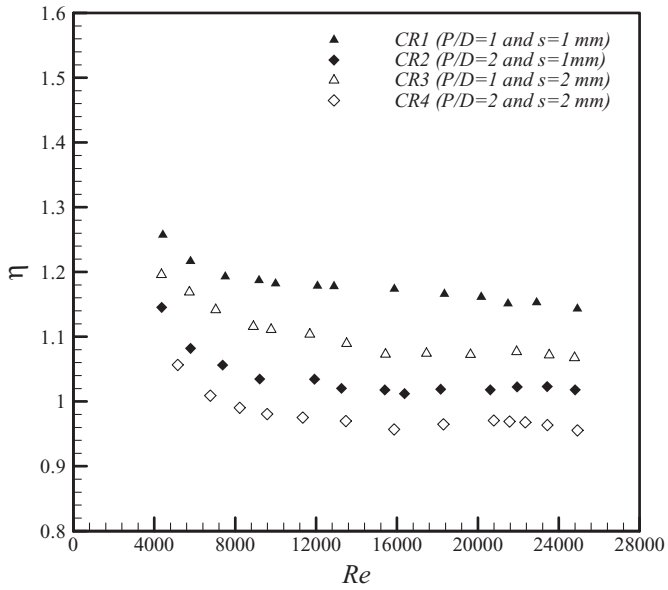



**Figure 10** Variation of thermal performance factor (η) with Reynolds number.

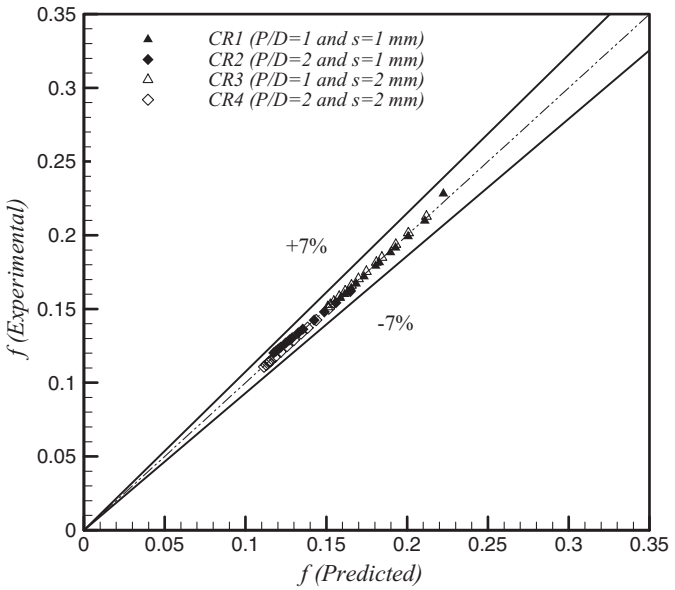



**Figure 12** Comparison of experimental data with correlation values of friction factor.

There will be a net energy gain only if η is greater than unity. The thermal performance factor increases with the decrease of pitch ratio (*P/D*) and distance (s) for all investigated cases (CR1, CR2, CR3, and CR4). It is clear from Figure 10 that using these rings at low Reynolds numbers offers higher thermal efficiency. The thermal performance factor tends to decrease with the increase in Reynolds number. Eventually, *CR1*, *CR2*, and *CR3* are thermodynamically advantageous ($\eta > 1$) at all Reynolds numbers; however, $\eta > 1$ for *CR4* after the Reynolds number is about 8000, because the friction factor increase dominates the increase in heat transfer. The highest thermal performance factor of 1.26 was achieved for the case of *CR1* ($P/D = 1$, $s = 1$ mm).

As a result, the experimental results of Nusselt number, friction factor, and thermal performance factor for circular ring turbulators were correlated as given here:

$$Nu = 0.188 Re^{0.611} (P/D)^{-0.314} (s/D)^{-0.119} Pr^{0.4} \quad (18)$$

$$f = 0.849 Re^{-0.195} (P/D)^{-0.436} (s/D)^{-0.074} \quad (19)$$

$$\eta = 1.351 Re^{-0.0527} (P/D)^{-0.174} (s/D)^{-0.089} \quad (20)$$

The comparison of the present experimental data and those calculated from the proposed correlations, Eqs. (18)–(20), are shown in Figures 11–13. The deviations found for

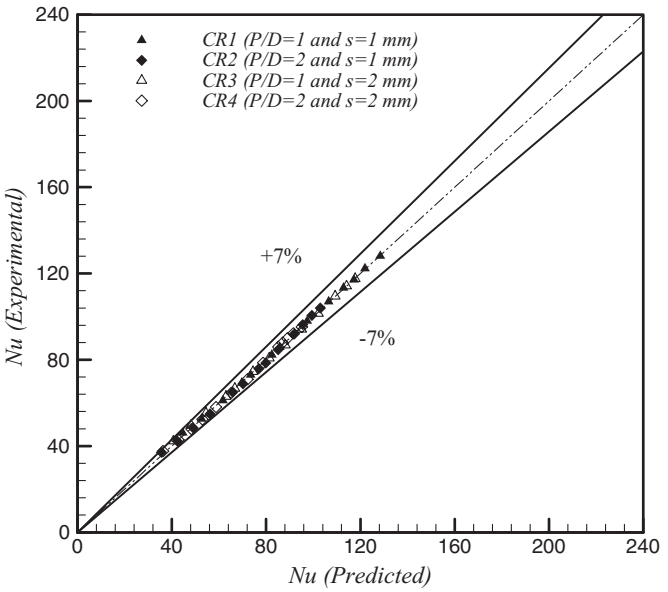



**Figure 11** Comparison of experimental data with correlation values of Nusselt number.

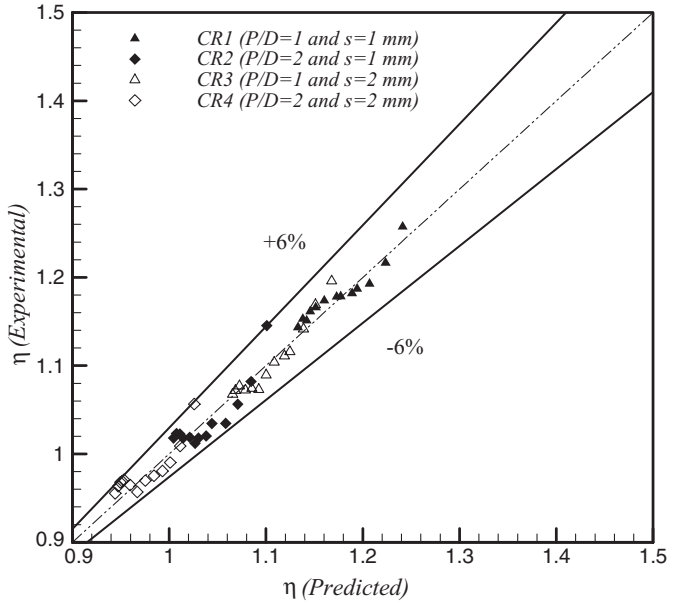



**Figure 13** Comparison of experimental data with correlation values of thermal performance factor.

Nusselt number, friction factor, and thermal performance factor were within ±7%, ±7%, and ±6%, respectively.

## CONCLUSIONS

The present experimental study was performed in order to investigate the thermal performance of circular ring turbulators detached from the tube wall for Reynolds numbers ranging from 4350 to 24,920 under a uniform wall heat flux condition. The effects of the pitch ratio (*P/D*) and distance (*s*) on the heat transfer enhancement in a tube were also considered. The interruption of flow by rings causes the destruction of thermal boundary layer near the tube wall and more blockage against the flow in the tube; therefore, the use of circular ring turbulators leads to a considerable increase in heat transfer and pressure drop over the smooth tube, depending on pitch ratio (*P/D*) and distance (*s*). It was observed from the results that the parameter of pitch ratio (*P/D*) is more dominant than the distance (*s*) on heat transfer and pressure drop.

The concluding remarks can be described as follows:

- The Nusselt number increases with decreasing pitch ratio (*P/D*) and distance (*s*). Therefore, the highest heat transfer was achieved for the case of *CR1* ($P/D = 1$, $s = 1$ mm). The increase in Nusselt number for the case *CR1* ($P/D = 1$, $s = 1$ mm) was in the range 115–132% over that of the smooth tube, depending on Reynolds number.
- The friction factor also increases with decreasing pitch ratio (*P/D*) and distance (*s*). The highest pressure drop was observed for the case of *CR*1 ($P/D = 1$, $s = 1$ mm). The friction factor for the case of *CR1* ($P/D = 1$, $s = 1$ mm) was on average 6.45 times more than the smooth tube.
- *CR1*, *CR2*, and *CR3* are thermodynamically advantageous ($\eta > 1$) at all Reynolds numbers; however, $\eta > 1$ for *CR4* after Reynolds number is about 8000, because the friction factor increase dominates the increase in heat transfer. The highest thermal performance factor of 1.26 was achieved for the case of *CR1* ($P/D = 1$, $s = 1$ mm) at a Reynolds number of 4421. The experimental results reveal that using these detached circular ring turbulators is advantageous in order to provide higher thermal performance, especially at low Reynolds numbers.
- Finally, the correlations for the Nusselt number, friction factor, and thermal performance factor obtained from the present experimental data were given for practical uses. The deviations found for Nusselt number, friction factor, and thermal performance factor were within ±7%, ±7%, and ±6%, respectively.

## NOMENCLATURE

| | |
|---|---|
| $A_{12}$ | log mean area, $m^2$ |
| $A_{ins}$ | surface area of the insulated test tube, $m^2$ |
| $c$ | insulation thickness, m |
| $C_{p,air}$ | specific heat capacity of air, J/kg-K |
| $CR$ | circular ring turbulator |
| $d$ | inner diameter of circular ring turbulator, m |
| $D$ | inner diameter of the tube, m |
| $D_o$ | outer diameter of the tube, m |
| $F$ | total heat transfer coefficient, $W/m^2$-K |
| $f$ | friction factor |
| $h$ | convective heat transfer coefficient, $W/m^2$-K |
| $I$ | current, A |
| $k$ | thermal conductivity, W/m-K |
| $L$ | length of the test tube, m |
| $\dot{m}$ | air mass flow rate, kg/s |
| $Nu$ | Nusselt number, $hD/k$ |
| $\Delta P$ | pressure drop, $N/m^2$ |
| $P$ | pitch of circular ring turbulator, m |
| $Pr$ | Prandtl number, $\mu C_{p,air}/k$ |
| $Re$ | Reynolds number, $UD/\nu$ |
| $S_i$ | inner radius of circular ring turbulator, m |
| $S_o$ | outer radius of circular ring turbulator, m |
| $s$ | distance between the circular ring turbulator and test tube inner wall, m |
| $Q$ | heat transfer, W |
| $q$ | heat flux, $W/m^2$ |
| $T$ | steady state temperature, K |
| $t$ | thickness of the test tube, m |
| $U$ | mean fluid velocity, m/s |
| $V$ | voltage, V |
| $\dot{V}$ | volumetric flow rate, $m^3/s$ |

### Greek Symbols

| | |
|---|---|
| $\rho$ | fluid density, $kg/m^3$ |
| $\mu$ | fluid dynamic viscosity, kg/ms |
| $\eta$ | thermal performance factor |
| $\nu$ | fluid kinematic viscosity, $m^2/s$ |

### Subscripts

| | |
|---|---|
| $b$ | bulk |
| $c$ | circular ring turbulator inserted in tube |
| $conv$ | convection |
| $i$ | inner |
| $ins$ | insulated test tube |
| $iw$ | inner wall of test tube |
| $o$ | outer |
| $ow$ | outer wall of test tube |
| $pp$ | pumping power |
| $s$ | smooth tube |
| $w$ | tube wall media |
| $\infty$ | ambient |

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