CHAPTER 5

RESULTS AND DISCUSSION

In this chapter, experimental results are validated using the standard correlations. The heat transfer and friction factor characteristics of a double pipe heat exchanger fitted with variant turbulators for turbulent flow are also presented.

5.1 PLAIN TUBE DATA

The present experimental results on heat transfer and friction characteristics in a plain tube are first validated in terms of Nusselt number and friction factor. It is important to compare the experimental results obtained for the fully developed turbulent flow with the correlations from the literature. The Nusselt number and friction factor obtained from experiment on the plain tube are compared with the correlations of Sieder and Tate (1936), Petukhov (1970) and Blassius correlations respectively found in the open literature for turbulent flow in circular tubes.

Sieder and Tate correlation,

\[ Nu = 0.027 Pr^{0.33} Re^{0.8} (\frac{\mu_b}{\mu_w})^{0.14} \]  

(5.1)

Petukhov correlation,

\[ Nu = \frac{(f/8) Re Pr}{1.07 + 12.7\sqrt{f/8 Pr^2 (Pr^2 / 3 - 1)}} (\frac{\mu_b}{\mu_w})^{0.11} \]  

(5.2)

where \( f \) is the friction factor and for plain tube it is given as (1970)
\[ f = (1.82 \log_{10} \text{Re} - 1.64)^{-2} \]  

Blassius correlation

\[ f = 0.3164(\text{Re})^{-0.25} \]  

Figure 5.1 shows the variation of Nusselt number obtained from experiment and Nusselt number estimated using Sieder and Tate with Reynolds number for the case of plain tube.

![Nusselt Number vs Reynolds Number](image)

**Figure 5.1 Data verification of Nusselt number for plain tube**

It is observed from Figure 5.1 that the Nusselt number estimated from experimental data lies within ±15% that of theoretical values calculated using Sieder and Tate and Petukhov correlation.

Plain tube experimental correlation for Nusselt number and friction factor are obtained as given in Equations (5.5) and (5.6).

\[ Nu = 0.228 \text{Re}^{0.641} Pr^{0.4} \]  

\[ f = 2.248 \text{Re}^{-0.45} \]  

(5.5)  

(5.6)
Figure 5.2 shows the variation of friction factor with Reynolds number. The experimental data matches with the Blassius and Petukhov correlation for plain tube with a discrepancy of less than ±7%.

![Figure 5.2 Data verification of Friction factor for plain tube](image)

### 5.2 TWISTED TAPE WITH PINS (TTP) AND TWISTED TAPE WITH PINS BONDED TO THE INNER SURFACE OF THE TEST SECTION (TTPB)

#### 5.2.1 Effect of y/w Ratios on Heat Transfer Enhancement for TTP

Figure 5.3 shows the variation of Nusselt number with Reynolds number for the tube fitted with TTP of three different y/w ratios (3.33, 4.29, and 5.71). From Figure 5.3 it can be concluded that the Nusselt number for the tube fitted with twisted tape with pins are higher than that of plain tube for a given Reynolds number. This is because the twisted tape with pins interrupts the development of the boundary layer of the fluid flow near the wall of the test section. Hence it increases the average temperature of the fluid in the radial direction. Due to the larger contact surface area, the heat transfer rate increases. Also it creates the turbulence and whirling motion to the water
which is flowing inside the test section. The whirling makes the flow to be highly turbulent, which leads to improved convection heat transfer. As the Reynolds number increases for a given y/w ratio of the twisted tape with pins, the Nusselt number also increases, indicating enhanced heat transfer coefficient. As the pitch of the tape decreases, the intensity of swirl flow increases leading to higher heat transfer rate and the maximum being for the twisted tape with pins of y/w=3.33. Throughout the experimental results it is seen that the smaller y/w (3.33) yields the higher values of heat transfer of about 23.86% than plain tube. Similarly for y/w=4.29 and 5.71 the enhancement are 19.9% and 14.4% respectively.

![Figure 5.3 Nusselt number vs. Reynolds number for TTP of different y/w ratios](image)

5.2.2 Effect of y/w Ratios on Friction Factor for TTP

Generally, the friction factor decreases conventionally with the increasing Reynolds number for different y/w ratios. From Figure 5.4 it can be seen that friction factor for the tube fitted with TTP is higher for a given Reynolds number. It indicates that friction factor for a given Reynolds
number increases with the decreasing y/w ratio due to swirl flow generated by TTP and reaches the maximum for y/w=3.33. From Figure 5.4, it can be seen that the friction factor for y/w=4.29 and 5.71 are less when compared with y/w=3.33. This is due to less contact surface area of the turbulator.

![Friction factor vs. Reynolds number for TTP of different y/w ratios](image)

**Figure 5.4** Friction factor vs. Reynolds number for TTP of different y/w ratios

### 5.2.3 Effect of y/w Ratios on Heat Transfer Enhancement for TTPB

The main objective of providing the twisted tape with pins is to transfer the heat from the wall surface of the test tube to the centre core of the water and to disturb the boundary layer. When only TTP is used in the test section, there is air gap between the twisted tape with pins and the inner surface of the test section as shown in Figure 4.2. The water near the wall receives more heat because it is in direct contact with the wall surface. But the water at the centre core of the tube receives less heat. It is due to the fact the heat will be transferred to centre of the flow due to normal convective mode of heat transfer.
Due to this, there is decrease in heat transfer when only TTP is used. In order to increase the temperature of water which is flowing at the centre of the tube, bonding is done between the twisted tape with pins and the inner surface of the test section. Due to this process, the air gap is filled with bonding due to soldering. So there will be metal to metal contact between the twisted tape and the wall of the test section and the twisted tape with pins acts as a fin. The TTPB disturbs the boundary layer near the wall of the test section and hence there is increase in convective heat transfer. Also the TTPB picks the heat from the wall and transfers the heat to the centre of the tube. So the temperature of the water at the centre of the tube also increases. Due to the above mentioned reason, there is increase in heat transfer for TTPB as shown in Figure 5.5. TTPB with smaller y/w (3.33) yields the higher values of heat transfer of about 32.9% than plain tube. Similarly for y/w=4.29 and 5.7, the enhancement are 27.5% and 20.75% respectively.

![Figure 5.5 Nusselt number vs. Reynolds number for TTPB of different y/w ratios](image-url)
5.2.4 Effect of y/w Ratios on Friction Factor for TTPB

From Figure 5.6 it can be seen that friction factor for the tube fitted with TTPB inserts is higher for a given Reynolds number when compared with plain tube. It indicates that friction factor for a given Reynolds number increases with the decreasing y/w ratio and reaches the maximum for y/w=3.33. This can be attributed to dissipation of the dynamic pressure of the fluid due to higher surface area and flow blockage of the TTPB along the tube wall. From Figure 10 it can be seen that the friction factor for y/w=4.29 and 5.71 are less when compared with y/w=3.33. This is due to the less contact surface area of the TTPB with y/w ratio 4.29 and 5.71.

![Friction factor vs. Reynolds number for TTPB of different y/w ratios](image)

**Figure 5.6** Friction factor vs. Reynolds number for TTPB of different y/w ratios

5.2.5 Comparison of Experimental Nusselt Number and Friction Factor for all the TTP and TTPB Configurations

From Figure 5.7 it can be seen that for TTPB with y/w=3.33, the enhancement in heat transfer is very high when compared to other
configurations of TTP and TTPB. The percentage increase in heat transfer of TTPB (y/w=3.33) when compared to TTP of same y/w=3.33 is 13.14%. Similarly for y/w=4.29 and 5.71 of TTPB, the enhancement are 11.95% and 10.81% respectively when compared to TTP of same y/w=4.29 and 5.71.

From Figure 5.8 it can be seen that the friction factor for TTPB is high when compared to the TTP for the same y/w=3.33. The percentage increase in friction factor for TTPB of y/w=3.33 is only 4% when compared to TTP of y/w=3.33. But the augmentation of heat transfer is 13.14% more than the increase in pressure drop for TTPB (y/w=3.33) when compared to TTP of same y/w=3.33. This is due to the increase in heat transfer in radial direction due to the bonding of the turbulator. The percentage increase in heat transfer is lower for TTP of y/w=5.71 when compared to all the other TTP and TTPB configurations. The percentage increase in heat transfer for TTP (y/w=5.71) is about 3.8%. This is due to less turbulence intensity created by the twisted tape with pins.

![Figure 5.7: Nusselt number vs. Reynolds number for all TTP and TTPB configurations](image-url)
Figure 5.8 Friction factor vs. Reynolds number for all TTP and TTPB configurations

5.2.6 Empirical Correlation for Combined TTP and TTPB with Different y/w Ratios

The empirical correlation for combined TTP and TTPB with different y/w ratios is given by Equations (5.7) and (5.8)

\[ Nu = 0.528 \Re^{0.543} \Pr^{0.4} (y/w)^{-0.192} \]  \hspace{1cm} (5.7)

\[ f = 51.29 \Re^{-0.36} (y/w)^{-0.259} \]  \hspace{1cm} (5.8)

The fitted values of Nusselt number by Equation (5.7) and friction factor by Equation (5.8) are compared with the experimental values and are shown in Figures 5.9 and 5.10 respectively. The correlated Nusselt number and friction factor results in maximum discrepancies of ±7.28 and ±7.16% respectively, when compared with experimental results.
Figure 5.9  Comparisons of experimental and predicted Nusselt number for tube with TTP and TTPB configurations

Figure 5.10  Comparisons of experimental and predicted friction factor for tube with TTP and TTPB configurations

CONCLUDING REMARKS

i) The Nusselt number of bonded twisted tape with pins is noticeably higher than the tube fitted with twisted tape with pins for the same twist ratios $y/w = 3.33, 4.29$ and $5.71$ on
account of the secondary flow development in the twist with main swirl flow created by the twisted tape with pins for same Reynolds number range, and the second reason is due to heat transfer in radial direction in the test tube fitted with bonded twisted tape with pins.

ii) The empirical correlations for Nusselt number and friction factor are developed for the tube fitted with twisted tape with pins and for maximum deviation of ±7.28% and ±7.16% respectively.

5.3 WIRE COILED COIL MATRIX TURBULATOR (WCCMT)

5.3.1 Effect of Pitch on Heat Transfer Enhancement without Bonding

Figure 5.11 shows the variation of Nusselt number with Reynolds number for the tube fitted with WCCMT inserts of three different pitch to diameter ratios of 0.23, 0.45 and 0.68. From Figure 5.11 it can be concluded that the Nusselt number for the tube fitted with WCCMT inserts are higher than that of plain tube for a given Reynolds number. This is because the WCCMT interrupts the development of the boundary layer of the fluid flow near the wall of the test section. Hence it increases the average temperature of the fluid in the radial direction. Due to the larger contact surface area the heat transfer rate increases. Also it creates the turbulence and whirling motion to the water which is flowing inside the test section. The whirling makes the flow to be highly turbulent, which leads to improved convection heat transfer.

As the Reynolds number increases for a given pitch, the Nusselt number also increases, indicating enhanced heat transfer coefficient. It is also observed from Figure 5.11 that Nusselt number for a given Reynolds number increases with decreasing pitch of the coil. As the pitch of the coil decreases,
the intensity of swirl flow increases leading to higher heat transfer rate and the maximum being for the WCCMT of P/D=0.23. Throughout the experimental results, it is seen that the smaller pitch to diameter (P/D=0.23) yields the higher values of heat transfer of about 25.4% than plain tube. Similarly for P/D of 0.45 and 0.68 the enhancement in heat transfer are 20.7% and 16.8% respectively.

![Figure 5.11](image)

**Figure 5.11** Nusselt number vs. Reynolds number for wire coiled coil matrix turbulator of different pitches without bonding.

### 5.3.2 Effect of Pitch on Friction Factor without Bonding

Generally, the friction factor decreases conventionally with the increasing Reynolds number for different pitches. Figure 5.12 shows that friction factor for a given Reynolds number increases with the decreasing pitch due to swirl flow generated by WCCMT and reaches the maximum for the P/D=0.23. From Figure 5.12 it can be seen that the friction factor is less when compared with turbulator of P/D=0.23. This is due to the less contact surface area of the turbulator, because the number of turns of the coil is only 148, so more area is available for the water to flow in the test section.
Figure 5.12 Friction factor vs. Reynolds number for wire coiled coil matrix turbulator of different pitches without bonding

Hence the pressure drop for the turbulator of P/D=0.45 is less when compared to turbulator of P/D=0.23. Similarly there is more area for the water to flow in the test section when the turbulator of P/D=0.68 is used in the heat exchanger. This is because only 95 turns are provided for this turbulator. Hence the friction factor for the turbulator of P/D=0.68 is less when compared with P/D=0.23 and P/D=0.45.

5.3.3 Effect of Pitch on Heat Transfer Enhancement with Bonding

From Figure 5.13, it can be seen that the Nusselt number increases with increase in Reynolds number. From, Figure 5.14 it can be seen that friction factor for the tube fitted with WCCMT inserts with bonding is higher for a given Reynolds number when compared with plain tube.

The increase in heat transfer is because of two reasons, first due to the turbulence and swirl motion created by the turbulator to the water which is flowing inside the tube. Second is due to the bonding of the turbulator with the inner surface of the tube. When the steam is allowed in the annulus region of the concentric tube heat exchanger, the outer surface of the inner tube of
the heat exchanger picks the heat and the heat is conducted through the wall of the inner tube. The turbulator coil which is bonded with the inner surface picks heat and the heat will conduct through the coil of the turbulator to the centre rod of the turbulator. Hence the water core which is flowing in the center receives heat due to the bonding effect.

**Figure 5.13** Nusselt number vs. Reynolds number for wire coiled coil matrix turbulator of different pitches with bonding

**Figure 5.14** Friction factor vs. Reynolds number for wire coiled coil matrix turbulator of different pitches with bonding
Due to the above mentioned reasons the heat transfer rate is high when compared with the turbulator without bonding which can be seen from the Figure 5.15.

This is due to the air gap between the turbulator and the inner surface of the tube in the case of turbulator without bonding. But in the case of turbulator with bonding the air gap is filled with soldering, so there is metal to metal contact, hence the heat is transferred both by conduction and convection. From Figure 5.16 it can be seen that there is only marginal increase in friction factor for turbulator with bonding when compared with turbulator without bonding.

![Graph showing comparison of experimental Nusselt number for the entire turbulator configuration](image)

**Figure 5.15** Comparison of experimental Nusselt number for the entire turbulator configuration
Figure 5.16 Comparison of experimental friction factor for the entire turbulator configuration

5.3.4 Empirical Correlation for WCCMT with Different P/D Ratios without Bonding with Centre Core Rod

The data were fitted by the following empirical correlations

\[ Nu = 0.491 \text{Re}^{0.569} (P / D)^{-0.072} \quad (5.8) \]

\[ f = 12.07 \text{Re}^{-0.245} (P / D)^{-0.156} \quad (5.9) \]

The fitted values of Nusselt number by Equation (5.8) and friction factor by Equation (5.9) are compared with the experimental values and are shown in Figures 5.17 and 5.18 respectively. The fitted values coincide with experimental data within ±3, and ±4%, respectively, for Nusselt number and friction factor.
Figure 5.17  Comparison of Experimental Nusselt number with the fitted value without bonding

Figure 5.18  Comparison of Experimental friction factor with the fitted value without bonding
5.3.5 Empirical Correlation for WCCMT with Different Pitches with Bonding with Centre Core Rod

The data were fitted by the following empirical correlations

\[ Nu = 0.459 \text{Re}^{0.606} P^{-0.073} \]  \hspace{1cm} (5.9)

\[ f = 26.43 \text{Re}^{-0.268} P^{-0.189} \]  \hspace{1cm} (5.10)

The fitted values of Nusselt number by Equation (5.9) and friction factor by Equation (5.10) are compared with the experimental values and are shown in Figure 5.19 and 5.20 respectively. The fitted values coincide with experimental data within ±6, and ±3%, respectively, for Nusselt number and friction factor.

![Graph showing comparison of experimental Nusselt number with fitted value with bonding](image)

**Figure 5.19** Comparison of Experimental Nusselt number with the fitted value with bonding
CONCLUDING REMARKS

i) The enhancement of heat transfer is high for tube fitted with bonded wire coiled coil matrix turbulator when compared to wire coiled coil matrix turbulator. This is due to the whirling motion created by WCCMT to the water which is flowing inside the tube; second reason is due to the presence of centre core rod of the turbulator the water is deflected to the side wall of the test section, which disturbs the development of thermal boundary layer near the wall. Third reason due to the increased heat transfer in the radial direction due to the effect of bonding the WCCMT to the inner wall of the test section. Due to the above mentioned reasons there is increase in heat transfer for the tube fitted with WCCMT when compared with plain tube.
ii) The correlations are developed for the Nusselt number and friction factor for the tube with WCCMT and bonded WCCMT by fitting turbulent experimental results with the maximum deviation of ±6% and ±4% respectively.

5.4 WCCMT WITHOUT CENTRE CORE ROD

5.4.1 Effect of Pitch on Heat Transfer Enhancement and Friction Factor for WCCMT without Center Core Rod with Bonding

From Figure 5.21 it can be seen that the Nusselt number increases with increase in Reynolds number. The increase in heat transfer is because of whirling motion created by the turbulator. Second the turbulator disturbs the thermal boundary layer near the wall of the tube. Due to the bonding the turbulator conducts heat through the coil to the centre of the tube. Due to this reason there is increase in heat transfer when compared to the plain tube. The percentage increase in heat transfer when compared to plain tube is 23.5%.

![Figure 5.21 Nusselt number vs. Reynolds number for wire coiled coil matrix turbulator of different P/D ratios without centre core rod with bonding](image)
From Figure 5.22 it can be seen that friction factor for the tube fitted with WCCMT inserts with bonding without centre core rod is higher for a given Reynolds number when compared with plain tube. Due to the absence of the centre core rod, there is more area for the water to flow in the tube. Hence the friction factor is less when compared with the turbulator with centre core rod. The percentage decrease in friction factor for WCCMT without center core rod when compared with the WCCMT with centre core rod is 16.5% for $P/D=0.23$, 18% for $P/D=0.45$ and 21.5% for $P/D=0.68$.

Due to the bonding effect there is increase in heat transfer. From Figure 5.23 it can be seen that there is increase in Nusselt number when there is increase in Reynolds number. The percentage increase in heat transfer for bonded turbulators with rod is about 14% when compared with bonded turbulator without rod.

Figure 5.22 Friction factor vs. Reynolds number for wire coiled coil matrix turbulator of different P/D ratios with bonding without centre core rod.
From Figure 5.24 it can be seen that there is decrease in pressure drop for turbulator without centre core rod when compared with turbulator with centre core rod. This is due to the less obstruction created by the turbulator to the water which is flowing inside the tube in the case of turbulator without centre core rod.

**Figure 5.23** Comparison of experimental Nusselt number for the entire turbulator configuration

**Figure 5.24** Comparison of experimental friction factor for the entire turbulator configuration
5.4.2 Empirical Correlation for WCCMT with Different Pitches with Bonding without Centre Core Rod

The data were fitted by the following empirical correlations

\[ Nu = 0.433 \text{Re}^{0.578} (P / D)^{-0.076} \]  

(5.11)

\[ f = 24.88 \text{Re}^{-0.34} (P / D)^{-0.21} \]  

(5.12)

The fitted values of Nusselt number by Equation (5.11) and friction factor by Equation (5.12) are compared with the experimental values and are shown in Figures 5.25 and 5.26 respectively.

The fitted values coincide with experimental data within ±4.6 and ±6.7%, respectively, for Nusselt number and friction factor.

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**Figure 5.25** Comparison of experimental Nusselt number with the fitted value for bonding with centre core rod
CONCLUDING REMARKS

i) The enhancement of heat transfer is high for tube fitted with bonded wire coiled coil matrix turbulator when compared to bonded wire coiled coil matrix turbulator without centre core rod. This is due to the whirling motion created by WCCMT to the water which is flowing inside the tube; second reason is due to the presence of centre core rod of the turbulator the water is deflected to the side wall of the test section, which disturbs the development of thermal boundary layer near the wall. Third reason due to the increased heat transfer in the radial direction due to the effect of bonding the wire coiled coil matrix turbulator to the inner wall of the test section. Fourth reason is due to more contact surface area of the WCCMT with centre core rod when compared with WCCMT without centre core rod and also due to increased fluid velocities wall owing to the presence of the tape. Due to the
above mentioned reasons there is increase in heat transfer for the tube fitted with WCCMT when compared with plain tube. But the pressure drop for WCCMT without center core rod is less when compared to the WCCMT with centre core rod.

ii) The correlations are developed for the Nusselt number and friction factor for the tube with WCCMT without centre core rod by fitting turbulent experimental results with the maximum deviation of 4.6% and ±6.7% respectively.

5.5 WIRE COIL TURBULATOR

5.5.1 Effect of Pitch to Diameter Ratios of the WC on Heat Transfer Enhancement

Figure 5.27 shows the variation of Nusselt number with Reynolds number for the tube fitted with WC inserts of four different P/D ratios (0.697, 0.93, 1.162 and 1.86). From Figure 5.27 it can be concluded that the Nusselt number for the tube fitted with WC inserts are higher than that of plain tube for a given Reynolds number.

![Figure 5.27 Nusselt number vs. Reynolds number for WC turbulator of different P/D ratios](image_url)
This is because the WC interrupts the development of the boundary layer of the fluid flow near the wall of the test section. Hence it increases the average temperature of the fluid in the radial direction. Also it creates the turbulence and whirling motion to the water which flows inside the test section. The whirling makes the flow to be highly turbulent and leads to improved convective heat transfer. The WC also drifts the warm fluid particles at the tube wall towards the cooler inner core and vice versa, owing to a centrifugal force field generated by the wire coil, and thus creating mixing of the colder and warmer particles, which results in increased heat transfer.

As the Reynolds number increases for a given pitch, the Nusselt number also increases, indicating enhanced heat transfer coefficient. It is also observed from Figure 5.27 that Nusselt number (for a given Reynolds number) increases with decreasing pitch of the coil. As the pitch of the coil decreases, the intensity of swirl flow increases leading to higher heat transfer rate and the maximum being for the WC of P/D=0.697. Throughout the experimental results, it is seen that the smaller P/D (0.697) yields the higher values of heat transfer of about 16.7% more than plain tube. Similarly for P/D=0.93,1.162 and 1.86 the enhancements are 14.4%, 10.7% and 4.5% respectively.

5.5.2 **Effect of Pitch to Diameter Ratios of WC on Friction Factor**

Generally, the friction factor decreases conventionally with the increasing Reynolds number for different pitches. From Figure 5.28 it can be seen that friction factor for the tube fitted with wire coil inserts is higher for a given Reynolds number. It indicates that friction factor for a given Reynolds number increases with the decreasing pitch due to swirl flow generated by WC and reaches the maximum for the P/D=0.697.
From Figure 5.28 it can be seen that the friction factor for WC turbulator of P/D=0.93, 1.162, and 1.86 is less when compared with WC turbulator of P/D=0.697. This is due to less blockage of flow and less contact surface area of the turbulator. Blockage is less because the number of turns of the coil is less and therefore more area is available for the water to flow in the test section. The main reason for the increase in pressure drop is due to the increase in velocity in radial direction.

![Friction factor vs. Reynolds number for WC turbulator of different P/D ratios](image)

**Figure 5.28** Friction factor vs. Reynolds number for WC turbulator of different P/D ratios

### 5.5.3 Empirical Correlation for WC with Different P/D Ratios

The data are fitted using the following empirical correlations

\[
Nu = 0.372\Re^{0.545}(P/D)^{-0.131}Pr^{0.4} \tag{5.13}
\]

\[
f = 14.94\Re^{-0.286}(P/D)^{-0.170}\tag{5.14}
\]

The fitted values of Nusselt number by Equation (5.13) and friction factor by Equation (5.14) are compared with the experimental values and are
shown in Figures 5.29 and 5.30 respectively. The fitted values coincide with experimental data within ±3.15 and ±5%, respectively, for Nusselt number and friction factor.

**Figure 5.29** Comparisons of experimental and predicted Nusselt number for tube with WC turbulators

**Figure 5.30** Comparisons of experimental and predicted friction factor for tube with WC turbulators
CONCLUDING REMARKS

i) The enhancement of heat transfer is high for tube fitted with wire coil turbulator. This is due to the secondary motion created by wire coil to the water which is flowing inside the tube; second reason is due to the increased flow path to the water because of the wire coil, third reason is the wire coil is very nearer to the wall surface which disturbs the development of thermal boundary layer and hydrodynamic boundary layer near the wall, hence there is increased convective heat transfer. Fourth reason is due to more contact surface area of the wire coil when compared to plain tube. Due to the above mentioned reasons there is increase in heat transfer for the tube fitted with wire coil when compared with plain tube.

ii) The correlations are developed for the Nusselt number and friction factor for the tube with wire coil by fitting turbulent experimental results with the maximum deviation of ±3.15% and ±5% respectively.

5.6 WIRE COIL WITH TUBE AND BONDED WIRE COIL WITH TUBE

5.6.1 Effect of P/D Ratios on Heat Transfer Enhancement for WCT and BWCT

The main objective of providing the turbulator is to transfer the heat from the wall surface of the test tube to the centre core of the water and to disturb the boundary layer. When the wire coil is used, the contact surface area between the water and WC is less. Hence the heat transfer from the wall to the centre core of the water is reduced.
Figure 5.31  Nusselt number vs. Reynolds number for WCT turbulator of different P/D ratios

In order to increase the contact surface area between the turbulator and water, the coil is wound over the copper tube of 9.5mm outer diameter and 0.7mm thickness. Due to the insert of WCT, the flow of water inside the test tube will be separated into two regimes. First the water flows over the tube of the turbulator where coil is present.

Due to the presence of coil, swirling motion will be created and also the coil disturbs the boundary layer of the water nearer to the wall of the test tube. Hence there will be increased convective heat transfer. The wire coil picks the heat from the water which is very nearer to the wall and transfers the heat to the tube of the turbulator. From the tube the heat will be transferred to the second regime where the water flows inside the tube of the turbulator. From Figure5.31 it can be seen that there is increase in heat transfer as the pitch of the coil is decreased. Practically, there will be air gap between the wire coil and the tube in some areas of the turbulator as shown in Figure4.9. So the heat transfer from the coil to the turbulator tube decreases when compared with the BWCT.
Figure 5.32 Nusselt number vs. Reynolds number for BWCT turbulator of different P/D ratios

In the case of BWCT, there is increase in heat transfer because the air gap between the coil and turbulator tube is removed with the help of bonding between wire coil and the tube. From, Figure 5.32 it can be seen that the percentage increase in heat transfer for BWCT of P/D=0.697 when compared with the plain tube is about 27.7% and when compared with WCT of same P/D (0.697) is 21.49%. From these experimental values there is considerable increase in heat transfer for BWCT when compared to WCT. This is due to the effect of bonding the coil with the tube.

5.6.2 Effect of Pitch to Diameter Ratios on Friction Factor for WCT and BWCT

The presence of tube in the WCT results in reduction of hydraulic diameter for the flow of water inside the tube in tube heat exchanger.
Figure 5.33 Friction factor vs. Reynolds number for WCT turbulator of different P/D ratios

This leads to the more contact surface area between the turbulator and water, which in turn results in increase of velocity in the radial direction. Due to these reasons there is increase in pressure drop.

From Figure 5.33 it can be seen that friction factor for the tube fitted with WCT inserts is higher for a given Reynolds number. It indicates that friction factor for a given Reynolds number increases with the decreasing pitch due to swirl flow generated by WCT and reaches the maximum for the P/D=0.697. From Figure 5.33 it can be seen that the friction factor for WCT turbulator of P/D=0.93, 1.162, and 1.86 is less when compared with WCT turbulator of P/D=0.697.

In the case of BWCT there is marginal increase in pressure drop because of bonding when compared to WCT. From Figure 5.34 it can be seen that the friction factor for P/D=0.697 is higher when compared to other BWCT turbulator. For the same Reynolds number, the increase in friction
factor for BWCT (P/D=0.697) when compared to WCT (P/D=0.697) is about 5.44%.

Figure 5.34 Friction factor vs. Reynolds number for BWCT turbulator of different P/D ratios

5.6.3 Empirical Correlation for WCT with Different P/D Ratios

The empirical correlation for WCT with different P/D ratios is given by Equations (5.15) and (5.16).

\[ Nu = 0.293 \text{Re}^{0.575} (P \div D)^{-0.134} Pr^{0.4} \quad (5.15) \]

\[ f = 14.2 \text{Re}^{-0.28} (P \div D)^{-0.066} \quad (5.16) \]

The fitted values of Nusselt number by Equation (5.15) and friction factor by Equation (5.16) are compared with the experimental values and are shown in Figures 5.35 and 5.36 respectively. The correlated Nusselt number and friction factor results in maximum discrepancies of ±2.5 and ±6.13%, respectively when compared with experimental results.
Figure 5.35  Comparisons of experimental and predicted Nusselt number for tube with WCT turbulators

Figure 5.36  Comparisons of experimental and predicted friction factor for tube with WCT turbulators
5.6.4 Empirical Correlation for BWCT with Different P/D Ratios

The empirical correlations developed relating P/D ratios and Reynolds number is given in Equations (5.17) and (5.18) for Nusselt number and friction factor.

\[
Nu = 0.267 \text{Re}^{0.586} (P / D)^{-0.163} Pr^{0.4} \quad (5.17)
\]

\[
f = 12.49 \text{Re}^{-0.257} (P / D)^{-0.062} \quad (5.18)
\]

The fitted values of Nusselt number by Equation (5.17) and friction factor by Equation (5.18) are compared with the experimental values and are shown in Figures 5.37 and 5.38 respectively. The fitted values coincide with experimental data within ±5 and ±3%, respectively for Nusselt number and friction factor.

![Figure 5.37 Comparisons of experimental and predicted Nusselt number for tube with BWCT turbulators](image)

Figure 5.37 Comparisons of experimental and predicted Nusselt number for tube with BWCT turbulators
Figure 5.38  Comparisons of experimental and predicted friction factor for tube with BWCT Turbulators

CONCLUDING REMARKS

i) The enhancement of heat transfer is high for tube fitted with bonded wire coil with tube when compared to wire coil with tube. This is due to the separation of water into two regimes which is flowing inside the test tube. First the water which is flowing outside the turbulator tube and the second regime flows inside the turbulator tube. The water which is flowing outside the tube of turbulator undergoes whirling motion and also it disturbs the thermal boundary layer to the water which is nearer to the wall; Second reason due to the increased heat transfer in the radial direction due to the effect of bonding the wire coil to the tube of the turbulator, because of this bonding there is no air gap between the turbulator tube and wire coil. Third reason is due to more contact surface area for tube fitted with bonded wire coil with tube. Due to the above mentioned reasons there is increase in heat transfer for the tube fitted with BWCT when compared with WCT. But the pressure drop for WCT is less when compared to the BWCT.
ii) The correlations are developed for the Nusselt number and friction factor for the tube with BWCT and WCT by fitting turbulent experimental results with the maximum deviation of ±5% and ±6.13% respectively.

5.7 BONDED WIRE COIL WITH TUBE BONDED TO THE INNER SURFACE OF THE TEST SECTION

5.7.1 Effect of Pitch on Heat Transfer Enhancement for the BWCTB

When only BWCT is used in the test section there is air gap between the turbulator and the inner surface of the test section as shown in Figure 5.39. Due to the presence of air gap between the wall surface of the test section and the turbulator there is decrease in heat transfer when compared to BWCTB. The main objective of providing the bond between the turbulator and the inner surface of the test section is to increase the water temperature at the centre of the flow. The water near the wall receives more heat because it is in direct contact with the wall surface. But the water at the centre of the tube receives less heat due to the fact that the heat will be transferred to centre of the flow due to normal convective mode of heat transfer.

![Figure 5.39 Nusselt number vs. Reynolds number for BWCTB turbulator of different P/D ratios](image)

Figure 5.39 Nusselt number vs. Reynolds number for BWCTB turbulator of different P/D ratios
In order to increase the heat transfer in the radial direction, bonding is done between the turbulator and the inner wall of the test section which results in enhanced convective mode of heat transfer to the water which is flowing at the centre. When bonding is done the turbulator acts as a fin. Due to the bonding there is metal to metal contact between the turbulator and the inner surface of the test section. There is no air gap between the test tube surface and turbulator because of bonding. The coil disturbs the boundary layer near the wall of the test section and hence there is increase in convective heat transfer. Also the coil picks the heat from the wall and transfers the heat to the tube of the turbulator. The tube transfers the heat to the water which is flowing through it. BWCTB also increases the flow length and also it causes centrifugal force superimposed over the longitudinal flow producing the secondary motion. Due to the above mentioned reasons there is increase in heat transfer.

From Figure 5.39 it can be seen that due to the bonding there is increase in heat transfer when compared to the plain tube and BWCTB. As the pitch of the coil decreases there is increase in heat transfer and also increase in pressure drop due to more contact area of the turbulator. The percentage increase in heat transfer for BWCTB with pitch to diameter ratio (0.697) is about 30.1% when compared with plain tube. Similarly for P/D=0.93, 1.162 and 1.86 the heat transfer enhancements are 26%, 23.9% and 16.5% respectively.

5.7.2 Effect of Pitch to Diameter Ratios on Friction Factor for BWCTB

In the case of BWCTB from Figure 5.40 it can be seen that the friction factor for P/D=0.697 is higher when compared to other BWCTB turbulator. For the same Reynolds number the increase in friction factor for BWCTB (P/D=0.697) when compared to BWCT (P/D=0.697) is about 2.1%.
Figure 5.40  Friction factor vs. Reynolds number for BWCTB turbulator of different P/D ratios

5.7.3  Empirical Correlation for BWCTB with Different P/D Ratios

The Equations (5.19) and (5.20) gives the empirical correlations developed for Nusselt number and friction factor by relating P/D ratios and Reynolds number ranging from 10,000 to 23,000.

Figure 5.41  Comparisons of experimental and predicted Nusselt number for tube with BWCT Turbulators
\[ Nu = 0.226 \text{Re}^{0.608} (P/D)^{-0.145} Pr^{0.4} \]  \hspace{1cm} (5.19)

\[ f = 12.21 \text{Re}^{-0.252} (P/D)^{-0.061} \]  \hspace{1cm} (5.20)

The predicted values from correlations (5.19) and (5.20) are within ±2.5 and ±3.5% when compared with experimental Nusselt number and friction factor. The fitted values of Nusselt number by Equation (5.19) and friction factor by Equation (5.20) are compared with the experimental values and are shown in Figures 5.41 and 5.42 respectively.

Figure 5.42  Comparisons of experimental and predicted friction factor for tube with BWCT turbulators

5.7.4  Comparison of Experimental Nusselt Number and Friction Factor for all the Turbulator Configurations

From Figure 5.43 it can be seen that for BWCTB with P/D=0.697, the enhancement in heat transfer is very high when compared to other wire coil turbulators. The percentage increase in heat transfer of BWCTB (P/D=0.697) when compared to BWCT of same P/D=0.697 is 8.66%. From Figure 5.44 it can be seen that the friction factor for BWCTB is high when compared to the BWCT for the same P/D=0.697. The percentage increase in
friction factor for BWCTB of P/D=0.697 is only 1.74% when compared to BWCT of P/D=0.697. The augmentation of heat transfer is very high than the increase in pressure drop for BWCTB (P/D=0.697) when compared to BWCT of P/D=0.697. This difference due to the increase in heat transfer in radial direction due to the bonding of the turbulator.

Figure 5.43 Nusselt number vs. Reynolds number for all the turbulator configurations

Figure 5.44 Friction factor vs. Reynolds number for all the turbulator configurations
The percentage increase in heat transfer is lower for WC of P/D=1.86 when compared to all the other wire coil turbulators. The percentage increase in heat transfer for WC (P/D=1.86) is about 4.5%. This is due to less turbulence intensity created by the turbulator.

**CONCLUDING REMARKS**

i) The enhancement of heat transfer is high for tube fitted with bonded wire coil with tube bonded to the inner surface of the test section when compared to BWCT, WCT and WC. This is because of the increased convective heat transfer inside the test section due to turbulence and secondary flow generated by the BWCTB, increased flow path of the water in the test section, disturbance created to the development of the boundary layer nearer to the wall of the test section and due to increased heat transfer in radial direction because of bonding the turbulator to the inner wall of the test section, because of this bonding there is no air gap between the turbulator tube and inner wall of the test section. Due to the above mentioned reasons there is increase in heat transfer for the tube fitted with BWCTB when compared with BWCT, WCT and WC.

ii) The correlations are developed for the Nusselt number and friction factor for the tube with BWCT and WCT by fitting turbulent experimental results with the maximum deviation of ±2.5% and ±3.5% respectively.