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Effect of Twisted Tapes Insert on Heat Transfer and Pressure Drop Augmentation for High Viscous Fluids inside Rotating Tubes

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ABSTRACT

The present work shows the results obtained from the numerical simulation of the heat transfer enhancement for the high viscous flow inside horizontal axially rotating tubes, using twisted tapes with the different twist and width ratios. The simulation is performed with the twisted tapes of three twist ratios (TR = 5, 7.5 and 10) and four width ratios (WR = 0.9, 0.7, 0.5 and 0.3). Rotation Reynolds number and Reynolds number are ranged from 0.9 to 5927 and 5 to 849, respectively. A computational fluid dynamics (CFD) methodology using ANSYS FLUENT 14.0 is used to perform the numerical analysis by solving the Navier-Stokes and energy equations through the viscous model at all cases of rotation Reynolds numbers and Reynolds numbers. The results revealed that, thermal performance factor due to the insertion of the twisted tapes in rotating tubes is strongly depended on the rotation speed. The increase in rotation speed decreases the thermal performance factor for tubes with Engine oil and Oil (SN-500). Whereas, in the tubes with Ethylene glycol, the thermal performance factor increases as the rotation speed increases. The influence of the twist ratio variation on the friction factor and heat transfer is small, as compared with the tape width ratio. The best twisted tape geometry, to achieve best thermal performance is found at WR of 0.9 for tubes with Engine oil and Oil (SN-500), and 0.5 for tubes with Ethylene glycol.

KEYWORDS:

Rotating tubes, fluid flow, heat transfer enhancement, high viscous fluids, twisted tapes, CFD.

1. INTRODUCTION

Twisted tapes are one of the effective methods to improve performance/efficiency of heat transfer equipment. Twisted tapes have been used to create swirling flows that modify the velocity profile due to various vortices distributions in the core region and near the tube wall. Different configurations of twisted tapes, like full length twisted tape, short length twisted tape, regularly spaced twisted tape, full length twisted tape with varying pitch, reduced width twisted tape and tapes with different surface modifications, have been studied widely by many researchers, in order to achieve a good thermal performance and decrease the size and cost of the involving equipment; especially in heat exchangers. However, most of the available data in the literature had been gained from the investigation of the thermal and flow fields inside the stationary tubes.

Saha and Dutta [1] studied experimentally the effect of varying length and pitch twisted tape (H) on heat transfer rate and friction factor. Laminar flow of a large Prandtl number ($205<Pr<518$) was considered. They concluded that the friction factor and heat transfer rate are lower for short length twisted tape than those for full length twisted tape, and the twisted tape with gradually decreasing pitch performs worse than the uniform pitch twisted tape. Ray and Date [2] predicted numerically characteristics of laminar flow and heat transfer through square duct with twisted tape inserts under constant heat flux condition. In this study, four different fluids were considered. They were air (Pr = 0.7), water (Pr = 5), moderately high prandtl number fluid (Pr = 50), and fluid with very high Prandtl number (Pr = 500). For each fluid, four TR were considered (1.5, 2.5, 5 and 10) and the Re was varied from 10 to 1000. They found that, Nusselt number (Nu) for ducts with twisted tape inserts increases with the increase in Re and Pr and with the decrease in TR. They also found that the square duct containing twisted tape offers better thermal performance than plain duct for all Re, Pr and TR. Saha and Mallik [3] reported an experimental investigation of the heat transfer and pressure drop characteristics of laminar flow of viscous oil through horizontal rectangular and square ducts inserted with full length twisted tapes, short length twisted tapes, and regularly spaced twisted tape elements, under constant heat flux boundary conditions. They found that, as the tape length decreases, both friction factor and Nu decrease. An experimental study of heat transfer in an axially rotating tube fitted with twin twisted tapes was performed by Chang et al. [4] to unravel the effect of rotation on heat transfer performance. The rotating speed was in range of 0 - 1000 rpm and Re = 5000, 75000, 10000, 12500 and 15000. In this study air was used as the working fluid. The results showed that, due to the swirl flow structures generated by twin twisted tapes, the Coriolis force effect plays a dominant role to initiate the

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heat transfer reduction from the stationary tube and that is followed by a subsequent recovery which could lead to heat transfer improvement as the relative strength of Coriolis force increased.

Influences of the twisted tape insertion on enhancement of heat transfer efficiency, Nusselt number and flow behaviors in concentric double pipe heat exchanger have been studied experimentally by Noothong et al. [5]. The experiment was carried out at the Re = 2000 - 12000 and TR = 5.0 and 7.0. The experimental results revealed that, the efficiency, Nu and friction factor increase with decreasing the twist ratio. Eiamsa-ard et al. [6] investigated experimentally heat transfer and friction factor characteristics in a double pipe heat exchanger fitted with regularly spaced twisted tape elements. The twisted tapes were inserted in the test tube section in two different cases: (1) full length typical twisted tape at different TR (6.0 and 8.0), and (2) twisted tape with various free space ratios (s = 1.0, 2.0 and 3.0). The results obtained from the tube with twisted tape insert were compared with those without twisted tape. They showed that the heat transfer coefficient increases with decrease in TR. Whereas the increase in the free space ratio will improve both the heat transfer coefficient and friction factor. Laminar convective heat transfer in a circular tube fitted with twisted tape was investigated numerically by Lin and Wang [7]. The effects of conduction in the tape on the Nusselt number and the sensitivity of heat transfer enhancement to the thermal boundary conditions were discussed. The Re and TR were ranged between 200 - 800 and 4 - 6, respectively. The results revealed that: (1) for fully developed laminar heat convective transfer, different tube wall thermal boundaries lead to different effects of conduction in the tape on heat transfer characteristics; (2) the efficiency of heat transfer enhancement is dependent on both the tube wall thermal boundaries and the intensity of secondary flow, and the ratio of Nusselt number decreases with increasing TR, while it increases with increasing Re, for both uniform wall temperature and uniform heat flux conditions. Eiamsa-ard and Thianpong et al. [8] studied experimentally the mean Nu, friction factor, and enhancement efficiency characteristics in tube with short length twisted tape inserts under uniform wall heat flux boundary conditions. Reynolds numbers were taken in a turbulent region with air as the test fluid. The full length twisted tape is inserted into the tested tube at a TR of 4.0, while the short length the tapes were mounted at the entry test section. The experimental results indicated that the short length tapes perform lower heat transfer and friction factor values than the full length. In addition, the enhancement efficiency of the tube with the short length tape insert is found to be lower than that with the full length one. Heat transfer and flow behavior in a twisted tape inserted tube are investigated experimentally by Bas and Ozceyhan [9]. The effects of TR (2, 2.5, 3, 3.5 and 4) were discussed in the range of Re from 5132 to 24,989. Uniform heat flux was applied to the external surface of the tube wall. The air was selected as a working fluid. The experimental results showed that, using of twisted tapes supplies considerable increase in heat transfer and pressure drop when compared with those from the plain tube. The Nu increased with the decrease of TR, also with increase of Re. Heat transfer and friction factor characteristics of laminar flow in a circular tube with short width twisted tape have been investigated numerically by Guo and Fan et al [10]. In this study the water was used as working fluid with Re = 500, 750, 1000, 1250, 1500 and 1750. The computation results showed that the Nu decreases as the width ratio decreases, and the larger the width ratio, the better the heat transfer enhancement is. Liu and Sakr [11] reviewed experimental and numerical works taken by researchers on the passive techniques of heat transfer augmentation since 2004 such as twisted tape, wire coil, swirl flow generator, etc. The authors found that variously developed twisted tape inserts are popular researched and used to strengthen the heat transfer efficiency for heat exchangers. They also observed that twisted tape inserts perform better in laminar flow than turbulent flow, because in laminar flow the thermal resistant is not limited to a thin region. The twisted tape insert is more effective, if no pressure drop penalty is considered. Experimental investigation of heat transfer and friction factor of water in a circular tube fitted with twisted tape inserts was carried out by Sivakumar and Rajan et al. [12]. The Re varied from 7628 to 13720. The results showed that the enhancement of heat transfer with twisted tape inserts was more compared with plain tube with increase of Re due to the high swirl flow. Heat transfer enhancement for laminar flow in circular tubes with twisted tape inserts and a constant wall temperature has been studied experimentally by Morad [13]. The experiment was performed for two different tube diameters with Re ranging from 500 to 2200. Three different twist ratios and tape thicknesses were considered and water was used as working fluid. He found that, the Nu increases with a decreasing TR, increasing tape thickness and increasing Re; while the friction factor increases with an increasing TR and tape thickness. It was also found that, the thermal performance ratio decreases with an increasing Re and TR, whereas the tape thickness has a slight effect on the thermal performance ratio.

The aim of the current work is to study the thermal performance of the highly viscous fluids inside the rotating tubes with the twisted tapes insert; because the twisted tapes are one of the most effective techniques to improve the thermal performance of laminar flow in heat exchangers.

2. PHYSICAL MODEL

Steel tube with 1000 mm in length and 20 mm in diameter (D) is rotated about its axis at constant angular velocity, ω. The thickness of the tube is set to be 1 mm. Steel twisted tape of 1mm thickness, pitch H, and width W_T is inserted into the full length of the tube. Wall temperature of the tube is assumed constant at 300˚C. Engine oil, Oil (SN-500) and Ethylene glycol are used as working fluids with constant inlet temperature of 25˚C. The configuration of the model is shown in Fig.1.

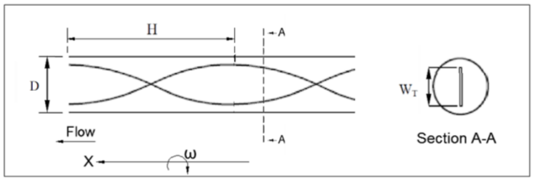

Fig.1 Configuration of the model.

3. MATHEMATICAL MODEL

The governing equations of the theoretical model consist of a set of fluid flow partial differential equations (Navier–Stokes equations, which represent continuity and momentum equations and the energy equation). The flow in the present study is considered to be three dimensional, laminar, steady and incompressible. No-slip boundary condition at the tube wall and tape surface is implemented.

The basic model equations in inertial (non-accelerating) reference frame can be written as follows [14]:

Continuity equation:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (1)$$

Momentum equation:

$$\frac{\partial}{\partial x_j}\left(\rho u_i u_j\right) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] \quad (2)$$

Energy equation:

$$\frac{\partial}{\partial x_j}\left(\rho u_j C_p T - k\,\frac{\partial T}{\partial x_j}\right) = 0 \quad (3)$$

The governing equations in moving reference frame (non-inertial reference frame) are augmented by additional acceleration terms that appear in the momentum equation due to the transformation from the stationary to the moving reference frame. However, the governing equations of fluid flow and heat transfer in steadily moving reference frame are as follows:

Continuity equation:

$$\nabla \cdot \rho\vec{v}_r = 0 \quad (4)$$

Momentum equation:

$$\nabla \cdot \left(\rho\vec{v}_r\vec{v}\right) + \rho\left[\vec{\omega} \times (\vec{v} - \vec{v}_t)\right] = -\nabla p + \left[\mu\,(\nabla\vec{v})\right] \quad (5)$$

Energy equation:

$$\nabla \cdot \left(\rho\vec{v}_r C_p T\right) = \nabla(k\,\nabla T) \quad (6)$$

Where

$$\vec{v}_r = \vec{v} - \vec{u}_r\,,\ \vec{u}_r = \vec{v}_t + \vec{\omega} \times \vec{r} \quad (7)$$

In the above equations, $\vec{v}_r$ is the relative velocity (the velocity viewed from the moving frame), $\vec{v}$ is the absolute velocity (the velocity viewed from the stationary frame), $\vec{u}_r$ is the velocity of the moving frame relative to the inertial reference frame, $\vec{v}_t$ is the translational frame velocity and $\vec{\omega}$ is the angular velocity. The term $(\vec{\omega} \times \vec{\omega} \times \vec{r})$ represents the Coriolis and centripetal accelerations.

The Re, Nu, friction factor (f) and thermal performance factor (TPF) proposed by Webb [15] are defined as follows:

$$Re = \frac{\rho u_i D}{\mu} = \frac{u_i D}{\nu} \quad (8)$$

$$Nu = \frac{hD}{k_f} \quad (9)$$

For circular tube of length L subjected to constant surface temperature, the average Nu for thermal entrance region can be determined from:

$$Nu = 3.66 + \frac{0.065(D/L)Re\,Pr}{1+0.04[(D/L)Re\,Pr]^{2/3}} \quad (10)$$

$$C_f = \frac{\tau_s}{\frac{\rho u_i^2}{2}} = \frac{f}{4} \quad (11)$$

$$TPF = \frac{(Nu_\omega/Nu_s)}{(f_\omega/f_s)} \quad (12)$$

Rotation Reynolds number, TR, and width ratio are computed by equations 13, 14 and 15, respectively

$$Re_\omega = \frac{\omega D^2}{\nu} \quad (13)$$

$$TR = \frac{H}{D} \quad (14)$$

$$WR = \frac{W_T}{D} \quad (15)$$

To calculate the average convective heat transfer coefficient inside the tube, the general governing heat balance equation is used as

$$hA_s\Delta T_m = \dot{m}C_p\,(T_e - T_i) \quad (16)$$

Where

$$\Delta T_m = \frac{T_i - T_e}{\ln\left[\frac{(T_s - T_e)}{(T_s - T_i)}\right]} \quad (17)$$

4. NUMERICAL METHOD

ANSYS FLUENT 14.0 is chosen as the CFD tool for this study. FLUENT works on the finite volume method to solve the above-mentioned governing equations accompanied with boundary conditions. The second order upwind discretization scheme for momentum and energy equations is employed in the numerical model. The pressure - velocity coupling is handled by the SIMPLE algorithm. In addition, convergence criteria of 10^{-3} for continuity and velocity components and 10^{-6} for energy are used, respectively.

Grid independent test has been performed for the physical model. The unstructured grid is used for meshing, as depicted in Fig. 2. The grid is highly concentrated near the tube wall and tape surface. Four grid systems (mesh 1, mesh 2, mesh 3 and mesh 4) with about 1,009,489, 1,009,765, 1,064,613 and 1,095,786 elements are adopted to calculate grid independence, at constant Re and at the same boundary conditions. The comparison had been hold between the predicated values of the estimated pressure and fluid velocity at a specified point in the flow. The solution accuracy is measured by the value of the percentage difference between the flow variable of certain mesh and same flow variable of mesh 4. The result of the mesh independence test is shown in Fig. 3, which demonstrates that, the difference between the calculated results of 1,064,613 elements (mesh 3) and 1,095,786 elements (mesh 4) are very small. Therefore, the gird system with 1,064,613 elements is adopted for the computational domain to give the acceptable accuracy of the solution, reasonable convergence of the solution and best executing time.

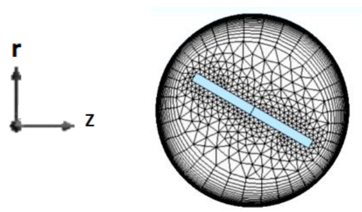

Fig.2 Mesh generated

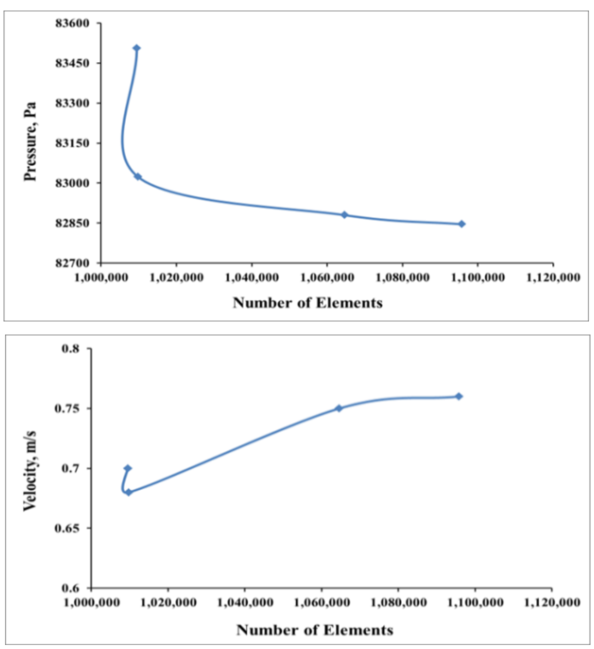

Fig.3 Grid independence test in rotating tubes with the twisted tape inserts.

5. CFD SIMULATION MODEL VALIDATION

To evaluate the validity of the numerical results, the numerical results are compared with the results obtained by Edward correlation (Equation 10), under similar operating conditions. Numerical analysis is conducted at Re ranging from 8 -101 and Engine oil is used as a working fluid. ANSYS Fluent 14.0 is used to predict the heat transfer characteristics inside the tube. The obtained numerical data are then compared with the Edward correlation values under similar conditions, in order to evaluate the validity of CFD results for the smooth tube. Figure 4 shows the comparison of predicted Nu results with correlation results. The figure reveal that, the numerical results are in better agreement compared to those from available correlation and the maximum percentage obtained between the numerical results and the correlation results is about 7%.

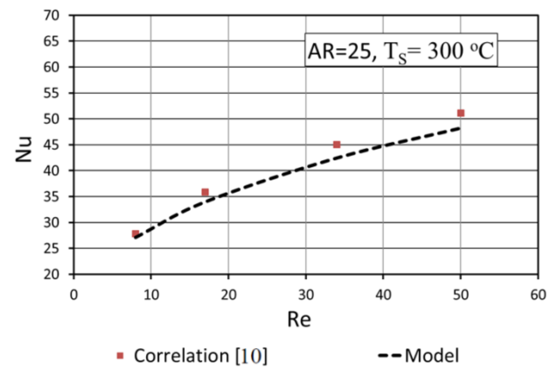

Fig.4 Comparison between numerical and correlation results of the average Nu in smooth passage at various Re.

6. RESULTS AND DISCUSSION

Different series of simulations under same operating condition are performed to study the effect of twisted tapes on the flow and thermal fields of the high viscous fluids inside rotating tubes. The objective of using twisted tapes here is to enhance the rate of heat transfer and the thermal performance through the high viscous fluids in the rotating tubes. Three values of the TR and four values of WR are used. Rotation speed of 25, 50 100, 500, 1000 and 2000 rpm are used within the inlet velocities of 0.3, 0.4, 0.5 and 0.6 m/s. Tables 1 and 2 list the range of operating dimensionless parameters considered in the this work. The average Nu and the total pressure drop in the rotating tubes with the twisted tapes are estimated and explained. The different results obtained from the simulations are compared with each other and with those for plain rotating tubes.

6.1 EFFECT OF THE TWIST RATIO (TR)

This investigation is conducted to know which of the TR gives best thermal performance through the high viscous fluids in the rotating tubes. The investigation is performed using twisted tapes with three different twist ratios, TR = 5, 7.5 and 10 (corresponding to the twist pitch, H = 100, 150 and 200 mm, respectively).Width of the twisted tapes remained constant during the investigation at 14 mm (WR = 0.7).

6.1.1 FLOW FIELD INVESTIGATION

The effect of using the twisted tapes with different twist ratios in the rotating tubes on flow of the fluids in axial direction at rotation speed, n = 1000 rpm, is shown in Fig.5. It can be seen that, use of the twisted tapes partitions the boundary layer without introducing any change in the flow pattern of the fluids. The lower value of twist ratio increases disturbance of the flow through the tube. Figures 6(A, B and C) show the predicted dimensionless axial velocity component (U*) profiles due to insertion of the twisted tapes. It is found that, presence of the twisted tapes in the rotating passages increases the axial velocity gradient near the tube wall, as compared to the plain tubes. In tubes with Ethylene glycol, the axial velocity gradient due to the twisted tapes was smaller than that for the tubes with Engine oil and Oil (SN-500). This is attributed to the drastic increment of pressure loss caused by rotation in the plain tubes. Figures below also show that, velocity profiles are slightly affected by the twist ratio variation. In general, velocity profile tends to decrease as the twist ratio decreases, as a result of the reduction in fluid flow through the tube.

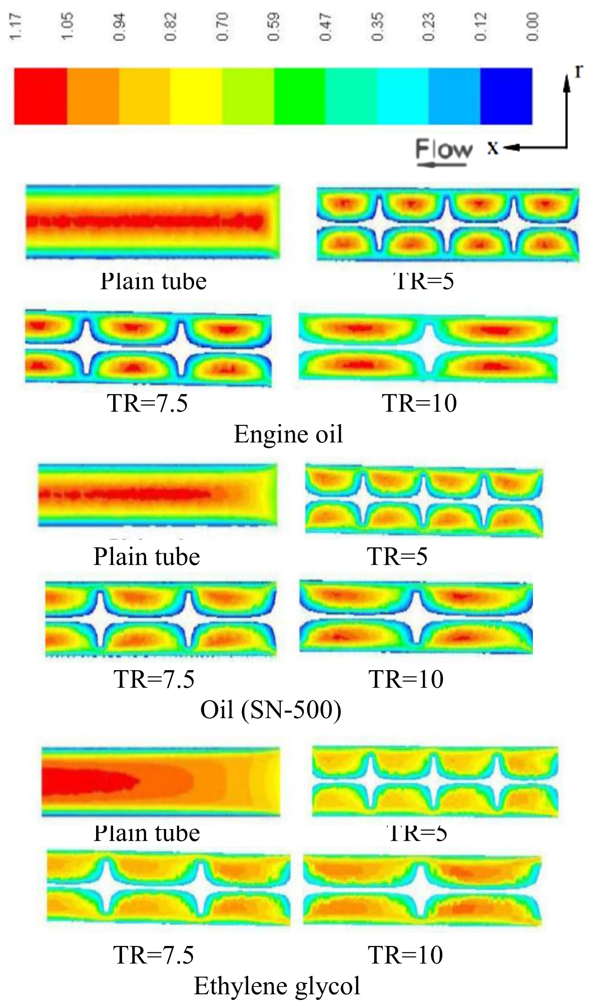

Fig.5 Axial velocity component (u, m/s) contours on longitudinal planes, from tube inlet to X* = 0.2, for all fluids, inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

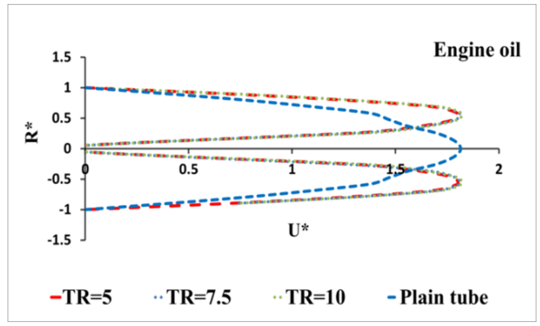

Fig.6(A) The predicted dimensionless axial velocity (U*) profiles, for Engine oil inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

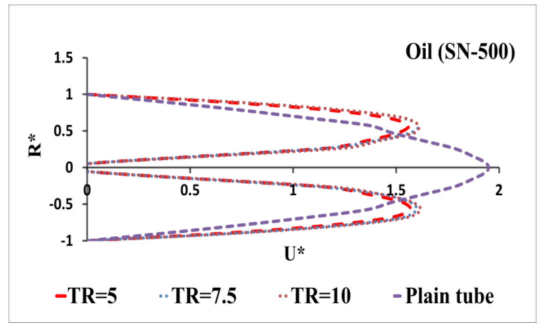

Fig.6(B) The predicted dimensionless axial velocity (U*) profiles, for Oil (SN-500) inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

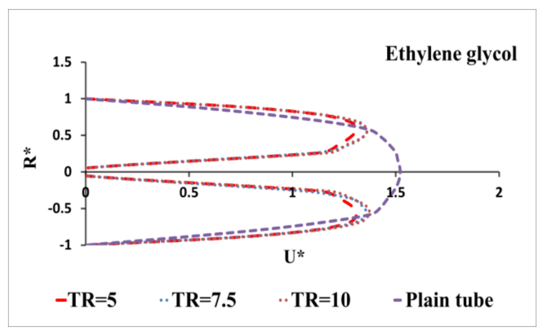

Fig.6(C) The predicted dimensionless axial velocity (U*) profiles, for Ethylene glycol inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

Figure 7 shows the tangential velocity component contours (w) for all fluids inside rotating plain tubes and tubes fitted with twisted tapes, at n = 1000 rpm. Figures 8(A, B and C) and 9(A, B and C) illustrate the predicted dimensionless tangential velocity component (W*) profiles, at n = 100 and 1000 rpm, respectively. It is observed that, use of the twisted tapes in the rotating tubes increases the tangential velocity component in the flow, as compared with the plain tubes. Effect of the twisted tapes on the tangential velocity component is strongly influenced by rotation, where the increment of rotation speed decreases the effect of the twisted tapes on the tangential velocity component. However, use of small value of the TR yields increase in tangential velocity component than that of large value of TR. Use of twisted tapes in the tubes with Ethylene glycol prevents the occurrence of the large viscous effects that appeared in plain tubes at high rotation speed.

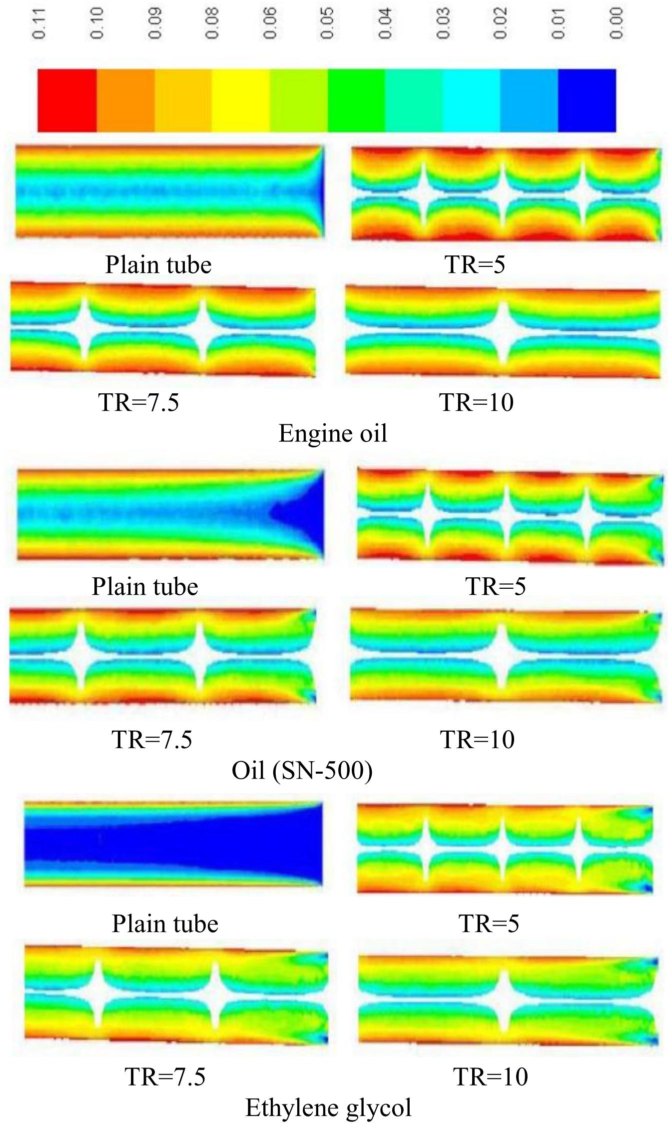

Fig.7 Tangential velocity component (w, m/s) contours on longitudinal planes, from tube inlet to X* = 0.2, for all fluids inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

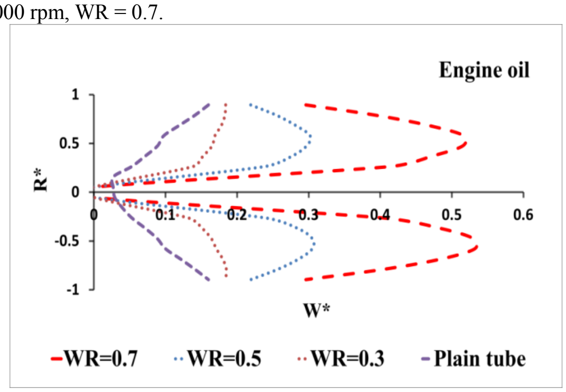

Fig.8(A) The predicted dimensionless tangential velocity (W*) profiles for Engine oil, inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 100 rpm, WR = 0.7.

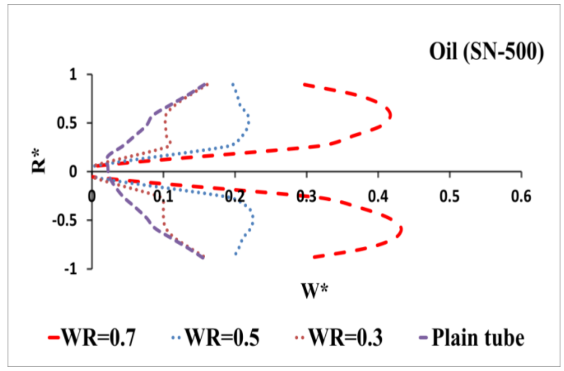

Fig.8(B) The predicted dimensionless tangential velocity (W*) profiles for Oil (SN-500), inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 100 rpm, WR = 0.7.

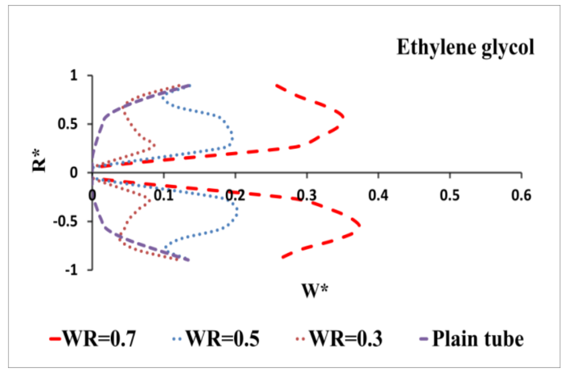

Fig.8(C) The predicted dimensionless tangential velocity (W*) profiles for Ethylene glycol, inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 100 rpm, WR = 0.7.

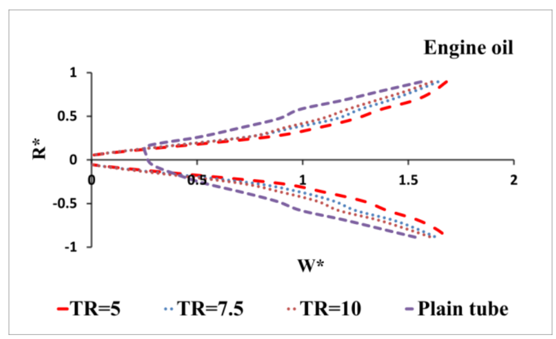

Fig.9(A) The predicted dimensionless tangential velocity (W*) profiles for Engine oil, inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

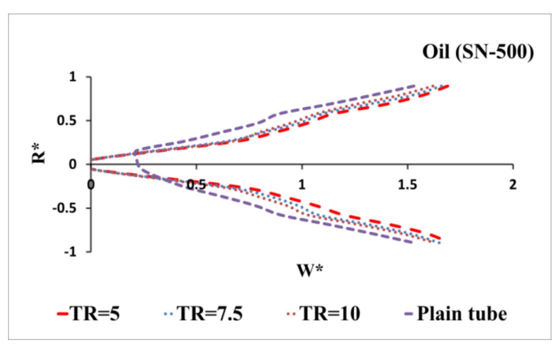

Fig.9(B) The predicted dimensionless tangential velocity (W*) profiles for Oil (SN-500), inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

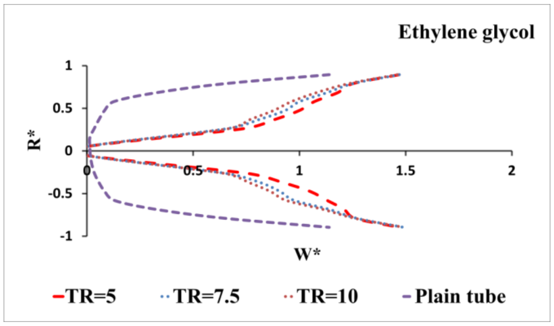

Fig.9(C) The predicted dimensionless tangential velocity (W*) profiles for Ethylene glycol, inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

Figures 10(A, B and C) describe the predicted dimensionless radial velocity component (V*) profiles for all fluids in the rotating tubes with and without twisted tapes. It can be inferred from these figures that the radial velocity component in the rotating tubes influenced by insertion of the twisted tapes, which provide better distribution for the radial velocity near the tube wall than that in case of the plain tubes. The radial velocity component tends to increase toward the tube wall with the decrease in the twisted ratio. In general, effect of the radial velocity component on the heat transfer enhancement remained weak, as compared with the tangential velocity component effect, because of its small values.

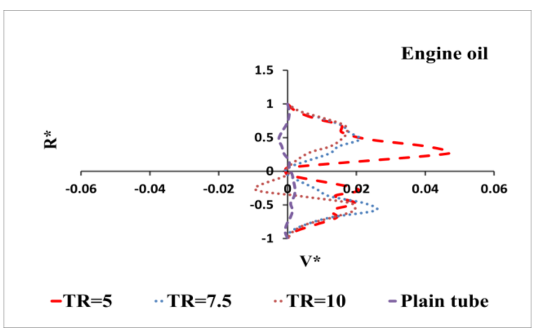

Fig.10(A) The predicted dimensionless radial velocity (V*) profiles for Engine oil inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

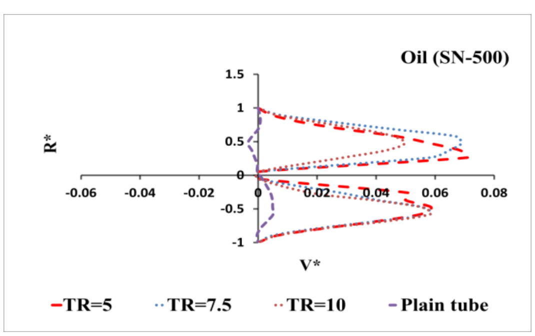

Fig.10(B) The predicted dimensionless radial velocity (V*) profiles for Oil (SN-500) inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

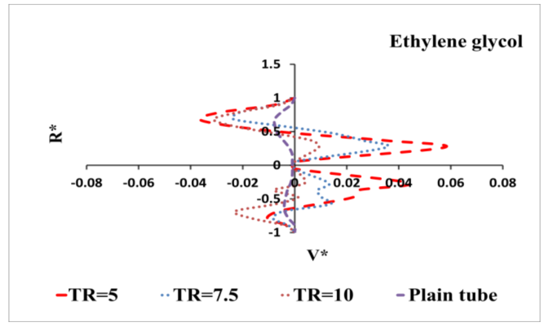

Fig.10(C) The predicted dimensionless radial velocity (V*) profiles for Ethylene glycol inside rotating tubes with and without twisted tapes insert, at different TR and u_i = 0.6 m/s, n = 1000 rpm, WR = 0.7.

FRICTION FACTOR

Figures 11, 12 and 13 discuss effect of the twisted tapes of different twist ratios on the average friction factor in the rotating tubes, for all Re considered. The figures show that, use of the twisted tapes in the rotating tubes results in increase the average friction factor considerably. This is due to reduction of the flow area which leads to the substantial pressure loss action of the fluid between the tape surface and the tube wall. From results, no significant difference in the average value of friction factor are seen when the twist ratio is varied. Generally, average friction factor slightly increases with decrease in the twist ratio, at all Reynolds numbers considered. This is attributed to the increase of fluid disturbance when the twist ratio decreases, as illustrated in Fig.5. It is also seen that, at higher rotation Reynolds number, the average friction factor for the tubes with Engine oil and Oil (SN-500) increases significantly due to the use of the twisted tapes than that for tubes with Ethylene glycol. This is because of the large pressure drop generated by rotation in the plain tubes with Ethylene glycol (see Fig.13).

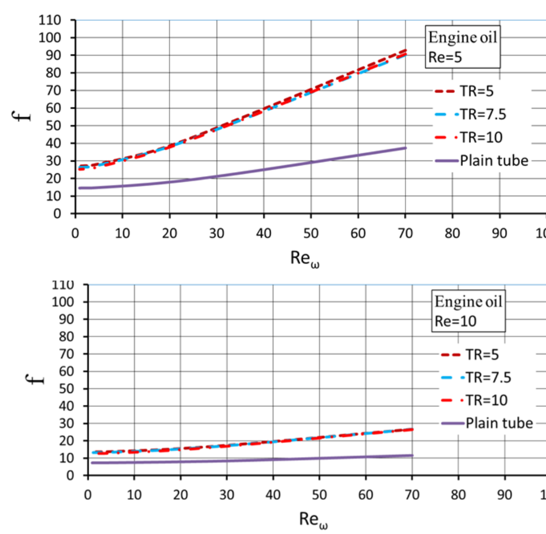

Fig.11 The variation in the predicted average value of the friction factor with Re$_\omega$, due to presence of twisted tapes in the tubes with Engine oil, at Re = 5 and 10, WR = 0.7.

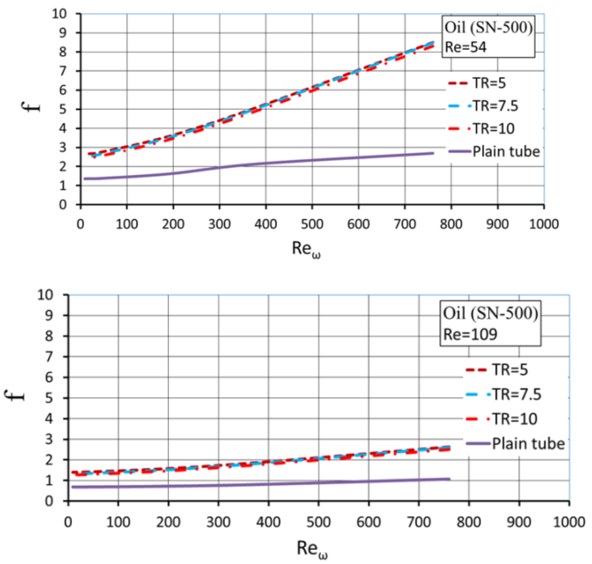

Fig.12 The variation in the predicted average value of the friction factor with Re_ω, due to presence of twisted tapes in the tubes with Oil (SN-500), at Re = 54 and 109, WR = 0.7.

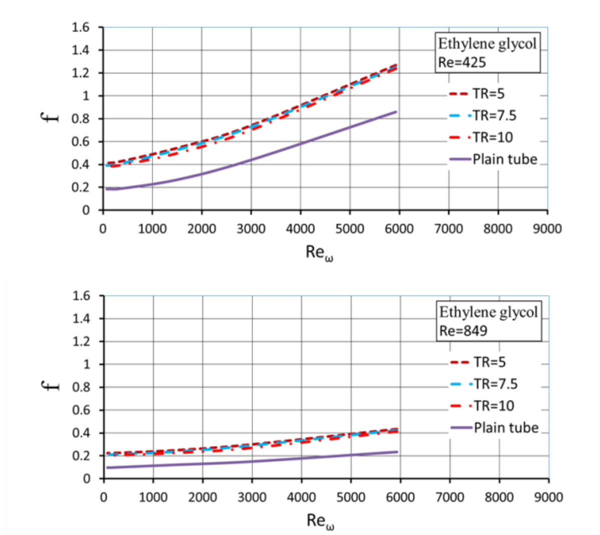

Fig.13 The variation in the predicted average value of the friction factor with Re_ω, due to presence of twisted tapes in the tubes with Ethylene glycol, at Re = 425 and 849, WR = 0.7.

6.1.2 THERMAL FIELD INVESTIGATION

Figure 14 illustrates the predicted dimensionless temperature profiles, for all fluids inside rotating plain tubes and tubes fitted with twisted tapes of different TR, at n = 1000 rpm. It's seen that, insertion of the twisted tapes in the rotating tubes improves uniformly the temperature distribution across fluid layers than in case of the plain tubes.

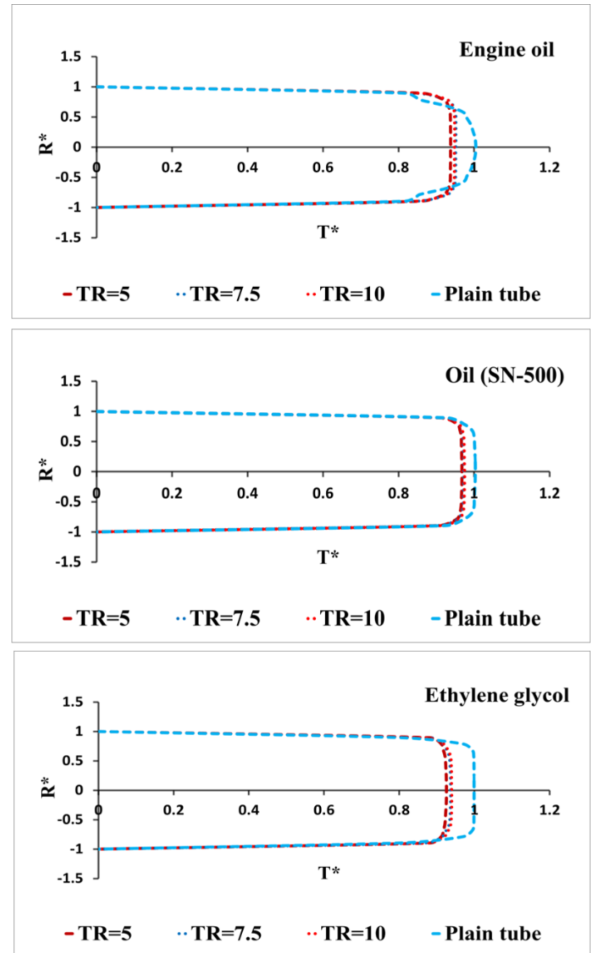

Fig.14 The predicted dimensionless temperature (T*) profiles on cross-sectional planes, at positions, X* = 0.9 (plain tubes), 0.925 (TR = 5), 0.8875 (TR = 7.5) and 0.85 (TR = 10), for all fluids inside rotating tubes with and without twisted tapes, at u_i = 0.6 m/s, n = 1000 rpm, and WR = 0.7.

HEAT TRANSFER

Figures 15, 16 and 17 show the effect of the twisted tapes of different TR on the average value of Nu in rotating tubes, for all Re considered. It is evident from these figures that the Nu for the rotating tubes with the twisted tapes is higher than that for the plain rotating tubes, at all Re_ω considered. This can be attributed to effect of tangential velocity component on the heat transfer, which increases when the twisted tapes are used in the rotating tubes. Also it is noticed that, using the smaller twist ratio gives increase in average value of Nu than the greater TR. This is because the tangential velocity component increases with decrease in the TR (as seen in Figs. 8(A, B and C) and 9(A, B and C)).Figures below also illustrated that, enhancement of heat transfer by the twisted tapes decreases with increase of the Re_ω. This is because the impact of the twisted tapes on the tangential velocity component decreases as rotation speed increases. In the tubes with Oil (SN-500), it is seen that the observed enhancement in heat transfer increases again slightly when Re_ω reaches 760

(2000 rpm), due to the decrease of the heat transfer enhancement in the plain tubes. In the tubes with Ethylene glycol, the absence of the large viscous effects due to the use of twisted tapes substantially enhanced the tangential velocity component in the flow (Fig. 9(C)). This caused a high rate of heat transfer at higher Re_ω than that for the tubes with Engine oil and Oil (SN-500) (Fig.17).Results also show that, the variation of the heat transfer enhancement ratio with the twist ratio is more obvious in the case with smaller Re_ω. This is because when the rotation speed is large, the twisted tapes have little impact on the tangential velocity component (Figs. 8 (A, B and C) and 9(A, B and C)).

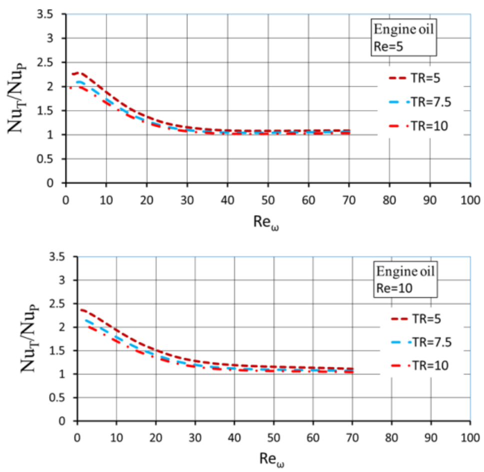

Fig.15 The variation in the predicted average value of the Nu ratio with rotation Reynolds number, due to presence of twisted tapes in the tubes with Engine oil, at Re = 5 and 10, WR = 0.7.

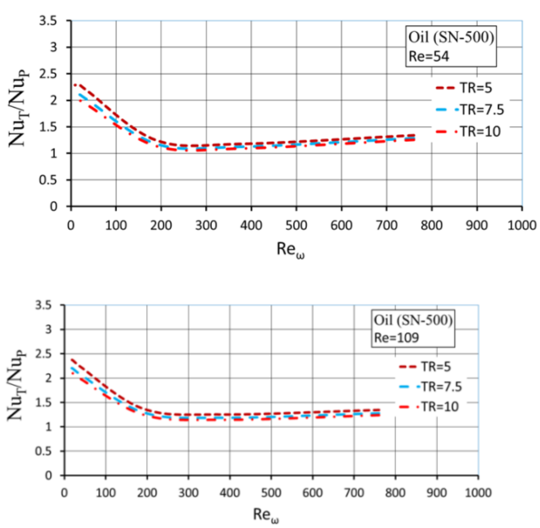

Fig.16 The variation in the predicted average value of the Nu ratio with rotation Reynolds number, due to presence of twisted tapes in the tubes with Oil (SN-500), at Re= 54and109, WR=0.7.

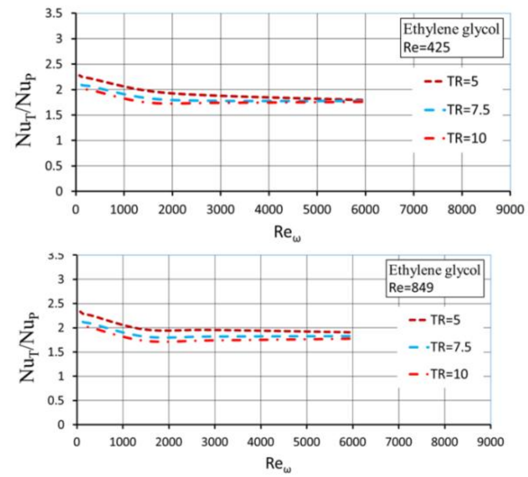

Fig.17 The variation in the predicted average value of the Nu ratio with rotation Reynolds number, due to presence of twisted tapes in the tubes with Ethylene glycol, at Re = 425 and 849, WR = 0.7.

6.1.3 THERMAL PERFORMANCE FACTOR (TPF)

Figures 18, 19 and 20 show the variation in the TPF due to the insertion of the twisted tapes in the rotating tubes, with the Re_ω. It's found that, the thermal performance factor is strongly influences by Re_ω variation. In the rotation speed ranging between 25-100 rpm and when TR = 5, the thermal performance in the tubes with Engine oil and Oil (SN-500) increases due to presence of twisted tapes to about 20-26% (at Re = 10, Re_ω = 0.9-4) and 14-19 % (at Re = 109, Re_ω= 9-38), respectively, then decreases with increase in Re_ω than that for plain tubes. Generally, the increase in the TR decreases the predicted average thermal performance.

In the tubes with Ethylene glycol, the enhancement in thermal performance factor by twisted tapes is found at higher rotation Reynolds numbers, Re_ω = 2964 and 5927 (1000 and 2000 rpm),which reaches about 11-22% (at Re = 425). Improvement of the thermal performance at these values of rotation Reynolds numbers is less than unity, as a result of increase in pressure drop.

The improvement in the thermal performance factor for the tubes with Engine oil and Oil (SN-500) due to the use of the twisted tapes increases with increase in Re, as compared with that for plain tubes. Whereas, in the tubes with Ethylene glycol, the enhancement in the thermal performance factor decreases with the increase in Re, as compared with that for plain tubes. This is attributed to the large pressure drop that occurred at the higher Re.

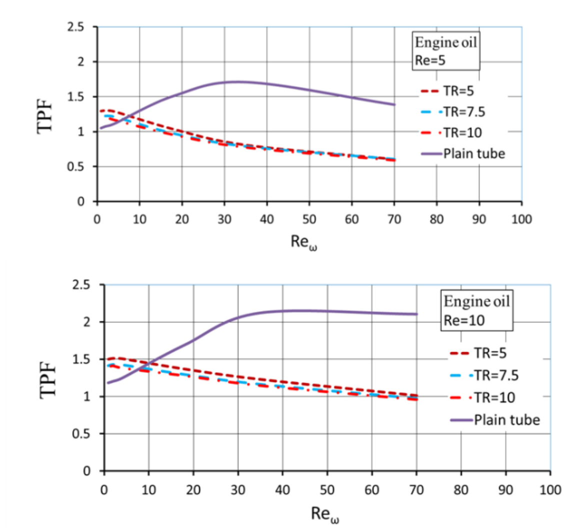

Fig.18 The variation in the thermal performance factor with Re$_\omega$, due to presence of twisted tapes in the tubes with Engine oil, at Re = 5 and 10, WR = 0.7.

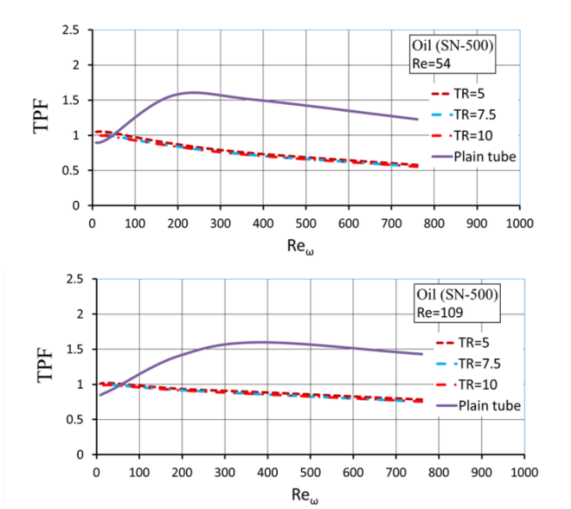

Fig.19 The variation in the predicted average thermal performance factor with Re$_\omega$, due to presence of twisted tapes in the tubes with Oil (SN-500), at Re = 54 and 109, WR = 0.7.

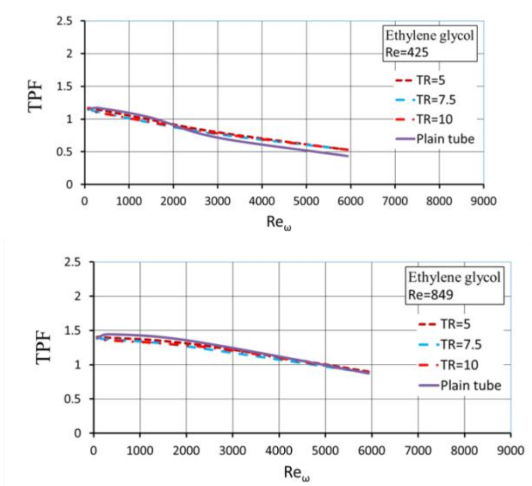

Fig.20 The variation in the predicted average thermal performance factor with Re$_\omega$, due to presence of twisted tapes in the tubes with Ethylene glycol, at Re = 425 and 849, WR = 0.7.

6.2 EFFECT OF THE TAPE WIDTH RATIO (WR)

The second geometrical parameter that can be investigated here is effect of the tape width ratio on the heat transfer augmentation and the pressure drop increase relative to that of plain tubes. The tape width ratio varies according to its values that are given in Table 2 (which are corresponding to the tape width of 6, 10, 14 and 18 mm, respectively).

6.2.1 FLOW FIELD INVESTIGATION

In this investigation twist ratio remained constant at TR = 5 (H = 100 mm).

Figure 21 illustrates the effect 0f tape width ratio on the axial velocity component (U*) profiles for rotating tubes fitted with twisted tapes, as compared to the plain rotating tubes. It is evident from this figure that the reduction of the tape width ratio decreases the axial velocity gradient near the tube wall. In the tubes with Engine oil and Oil (SN-500), velocity profiles tend to decrease at the center as the tape width ratio decreases, due to the increase in the fluid flow through the tube, whereas, in the tubes with Ethylene glycol, the velocity profiles will tend to increase at the center as a result of appearing the viscous effects in the boundary layer when the tape width ratio decreases.

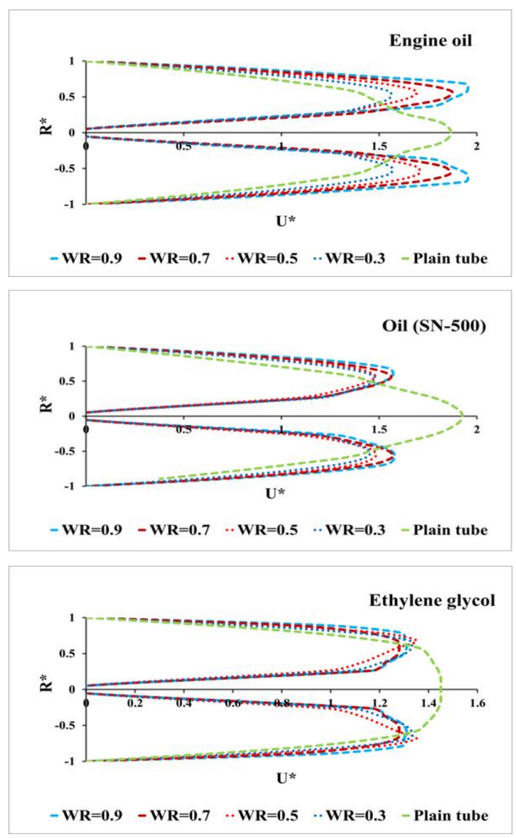

Fig.21 The predicted dimensionless axial velocity (U*) profiles for all fluids, inside the rotating tubes with and without twisted tapes, at different width ratios and u_i = 0.6 m/s, n = 1000 rpm, TR = 5.

The predicted dimensionless tangential velocity component (W*) profiles, for the rotating tubes with and without twisted tapes at different width ratios and different rotation speeds are depicted in Figs. 22 and 23. As shown in these figures, at higher rotation speed, the magnitude of the tangential velocity component due to the decrease in the tape width ratio approaches to that observed in the plain tubes. This effect decreases with decrease of the rotation speed.

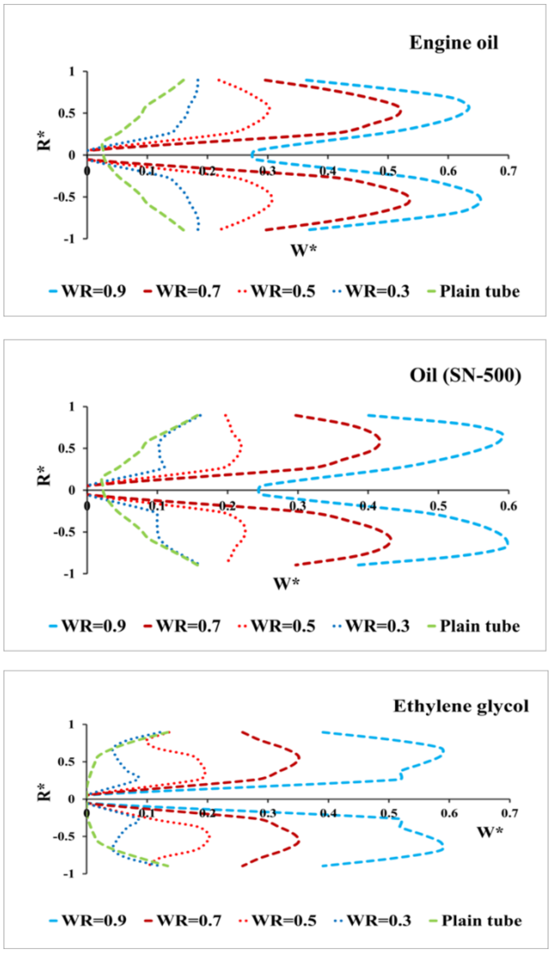

Fig.22 The predicted dimensionless tangential velocity (W*) profiles for all fluids, inside rotating tubes with and without twisted tapes, at different width ratios and $u_i = 0.6$ m/s, n = 100 rpm, TR = 5.

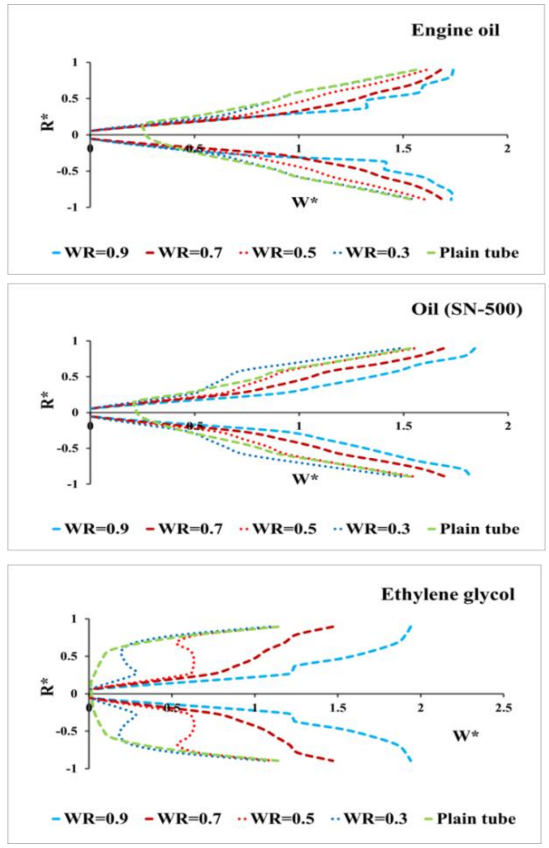

Fig.23 The predicted dimensionless tangential velocity (W*) profiles for all fluids, inside rotating tubes with and without twisted tapes, at different width ratios and $u_i = 0.6$ m/s, n = 1000 rpm, TR = 5.

FRICTION FACTOR

From Figures, it can be seen that the average friction factor decreases as the tape width ratio decreases. This is because the tapes with small width ratio have less blocking area. The results also indicate that the variation effect of the tape width ratio on the average friction factor is larger than the variation effect of the twist ratio (see Figs. 11, 12 and 13).

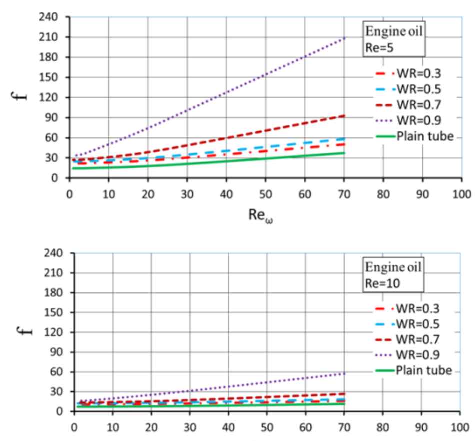

Fig.24 The variation in the predicted average value of the friction factor with Re_ω, due to presence of twisted tapes in the tubes with Engine oil, at Re = 5 and 10, TR = 5.

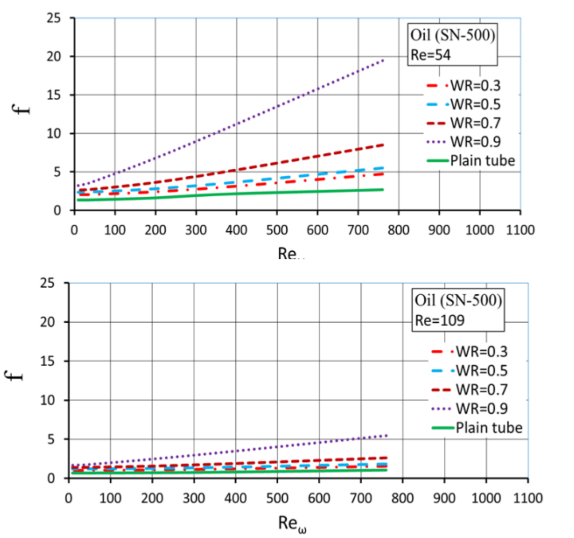

Fig.25 The variation in the predicted average value of the friction factor with Re_ω, due to presence of twisted tapes in the tubes with Oil (SN-500), at Re = 54 and 109, TR = 5.

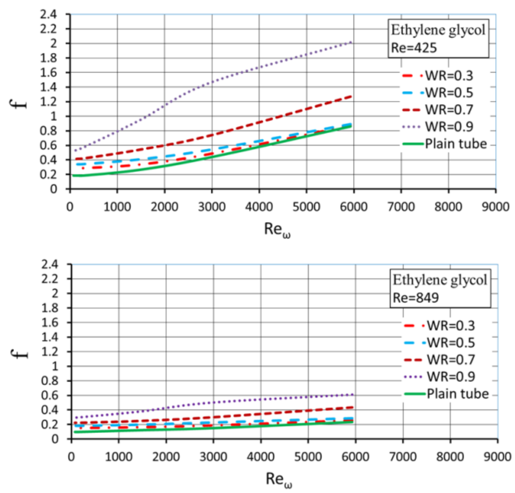

Fig.26 The variation in the predicted average value of the friction factor with Re_ω, due to presence of twisted tapes in the tubes with Ethylene glycol, at Re = 425 and 849, TR = 5.

6.2.2 THERMAL FIELD INVESTIGATION

Figure 27 shows the predicted dimensionless temperature profiles, for all fluids inside the rotating tubes with and without twisted tapes at different width ratios and n = 1000 rpm. It's seen that the fluid temperature due to the reduction of the tape WR approaches to that for the plain tubes. This is attributed to the weak enhancement of the tangential velocity component (see Fig. 23). However, this implies that the reduction of the tape WR has slight effect on the heat transfer.

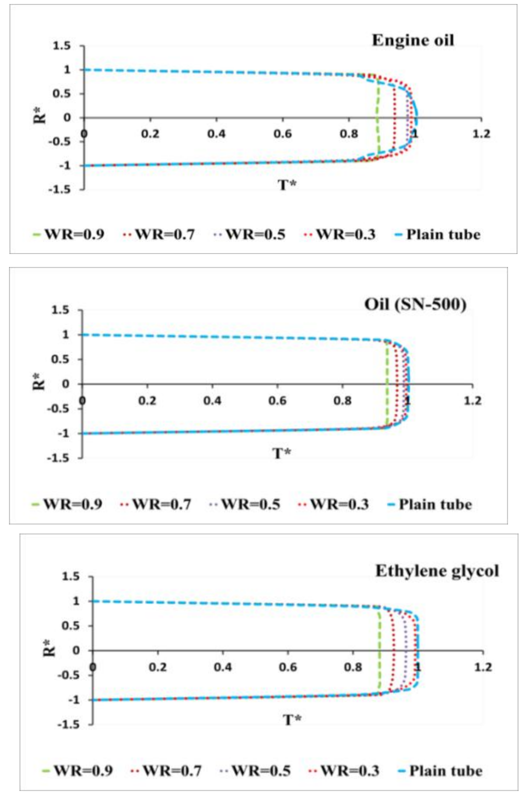

Fig.27 The predicted dimensionless temperature (T*) profiles for all fluids inside rotating tubes with and without twisted tapes, at different WR and u_i = 0.6 m/s, n = 1000 rpm, TR = 5.

HEAT TRANSFER

The computational results for the variation in the tape width ratio on the heat transfer rate are presented in Figures 28, 29 and 30. It can be observed that, reduction of the tape width ratio decreases the heat transfer enhancement in the rotating tubes. In the tubes with Engine oil, the predicted average Nu due to the reduction of the tape width ratio to 0.5 and 0.3 becomes lower than that for the plain tubes when Re_ω increases, as seen in Fig. 28. This is because the tangential velocity component was very small and not enough to induce higher rate of heat transfer than that for the plain tubes (as illustrated in Fig. 23). Also, in the tubes with Oil (SN-500), the Nu due to the reduction of the tape width ratio to 0.5 and 0.3 decreases than that for plain tubes when Re_ω increases. Then Nu increases slightly than that for the plain tubes when Re_ω reaches 760 (2000 rpm), due to the decrease of the heat transfer enhancement in the plain tubes. In the tubes with Ethylene glycol, the reducing in the viscous effects due to the use of twisted tapes led to enhance the rate of heat transfer than that for plain tubes at all tape width ratios and all cases of rotation speeds.

The results also illustrate that, the effect of the WR variation on the average Nu is larger than the effect of the TR variation (see Figs. 15, 16 and 17).

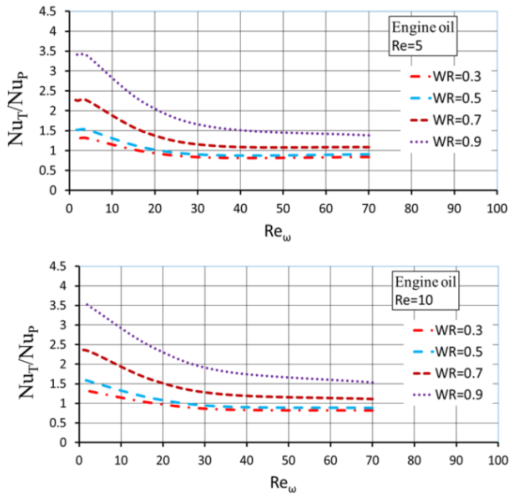

Fig.28 The variation in the predicted average value of the Nu ratio with Re_ω,, due to presence of twisted tapes in the tubes with Engine oil at Re = 5 and 10, TR = 5.

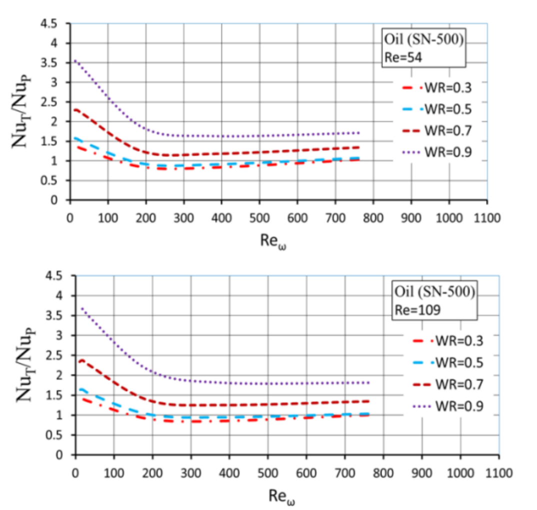

Fig.29 The variation in the predicted average value of the Nu ratio with Re_ω,, due to presence of twisted tapes in the tubes with Oil (SN-500), at Re = 54 and 109, TR = 5.

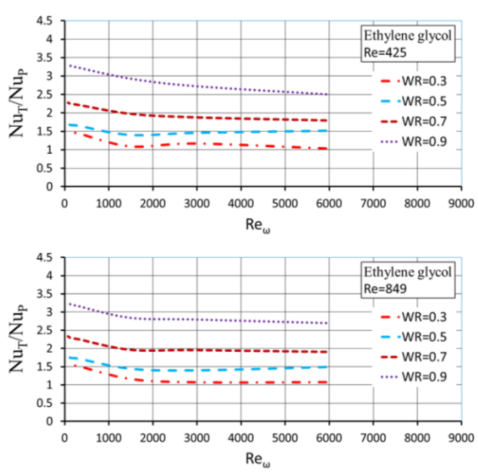

Fig.30 The variation in the predicted average value of the Nu ratio with Re_ω,, due to presence of twisted tapes in the tubes with Ethylene glycol, at Re = 425 and 849, TR = 5.

6.2.3 THERMAL PERFORMANCE FACTOR (TPF)

Figures 31, 32 and 33 present the variation of the TPF with the Re_ω, at different WR and different Re. It's shown that, in the tubes with Engine oil and Oil (SN-500), reduction of WR = 0.5 and 0.3 decreases the TPF than that for the plain tubes, for all Re_ω and Re considered. It's also seen that, at rotation speed ranged between 1000 -2000 rpm (at Re_ω = 35 and 70 for Engine oil, Re_ω = 380 and 760 for Oil (SN-500)), using the smaller tape width ratio yields higher values of TPF than the greater tape width ratio.

In the tubes with Ethylene glycol, using the tape width ratio, WR = 0.9 and 0.5 gives higher value of the TPF than that for the plai n tubes, at Re_ω = 74 and 5927, respectively.

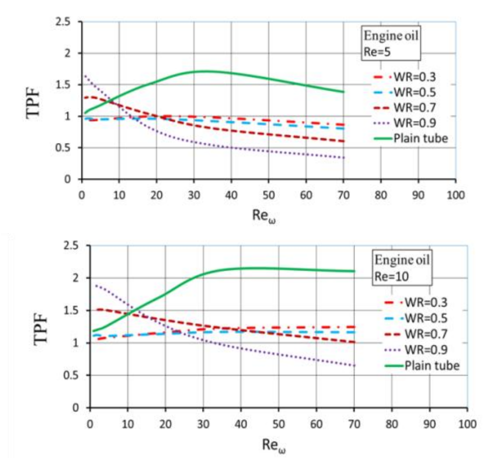

Fig.31 The variation in the thermal performance factor with Re_ω,, due to presence of twisted tapes in the tubes with Engine oil, at Re = 5 and 10, TR = 5.

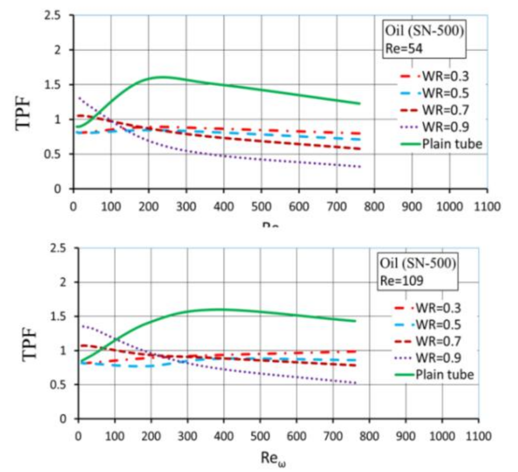

Fig.32(A) The variation in the thermal performance factor with Re_ω, due to presence of twisted tapes in the tubes with Oil (SN-500), at Re = 54 and 109, TR = 5.

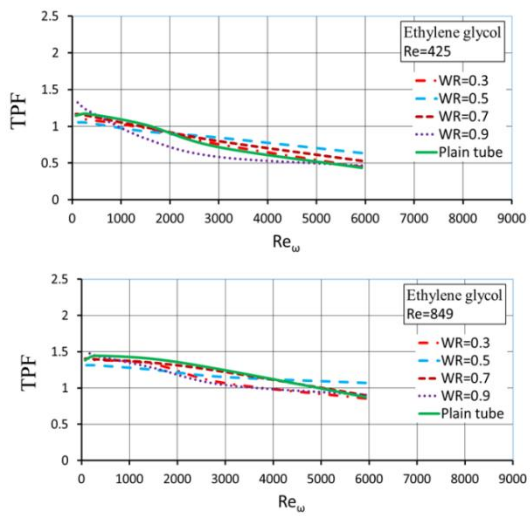

Fig.33 The variation in the thermal performance factor with Re_ω, due to presence of twisted tapes in the tubes with Ethylene glycol, at Re = 425 and 849, TR = 5.

CONCLUSIONS

Fluid flow and heat transfer characteristics for three different high viscous fluids (Engine oil, Oil (SN-500) and Ethylene glycol) inside axially rotating tubes with twisted tapes insert have been numerically investigated using ANSYS FLUENT 14.0.

The computational results of the present study can be summarized as follows:

1. Heat transfer enhancement for the high viscous fluids in the rotating tubes due to insertion of twisted tapes is strongly depending on the rotation speed. Increase of the rotation speed decreases effect of the twisted tapes on the tangential velocity component, leading to reduce the enhancement of the heat transfer at the wall.
2. The effect of inserting twisted tapes in rotating tubes is more observed with fluids of lower viscosity.
3. Tape width ratio (WR) has larger influence on the average friction factor and heat transfer than the twist ratio (TR). Therefore, tape width ratio is a strong variable for pressure drop and heat transfer of the high viscous fluids inside rotating tubes fitted with twisted tapes
4. Increase of the tape width ratio is a beneficial design for enhancement of the thermal performance for the tubes with Engine oil and Oil (SN-500). Whereas, in the tubes with Ethylene glycol, using the tape width ratio, WR = 0.9 and 0.5 gives higher value of the TPF than that for the plain tubes, at Re_ω = 74 and 5927, respectively.
5. In the rotating tubes with Engine oil and Oil (SN-500), improvement of the thermal performance factor by twisted tapes is found at lower rotation speed (n = 25, 50 and 100 rpm), which reaches about 60% for tubes with Engine oil and 61% for tube with Oil (SN-500), at Reynolds number, Re = 10 and 109, respectively. The best twisted tape geometry, which can be selected to enhance the thermal performance factor, is that having width ratio (WR) of 0.9.
6. In the rotating tubes with Ethylene glycol, improvement of the thermal performance factor by twisted tapes is occurred at rotation speed, n = 25 and 2000 rpm, which reaches about 18% at Re = 425 (for WR = 0.9) and 22 % at Re = 849 (for WR = 0.5). The best twisted tape geometry, which can be selected to enhance the thermal performance factor, is that having width ratio (WR) of 0.5.

NOMENCLATURE

A_s	Surface area, m^2
AR	Aspect ratio = L/D
C_f	Dimensionless skin friction coefficient
C_p	Specific heat, J/kg K
D	Tube diameter, m
f	Average friction factor= $4C_f$
$\frac{f_\omega}{f_s}$	Friction factor ratio
H	Twist pitch, m
h	Average convective heat transfer coefficient, W/m^2 K
k	Thermal conductivity, W/m.K
L	Tube length, m
$\dot{m}$	Mass flow rate of fluid, kg/s
N	Rotation rate = $\frac{W_w}{u_i}$
Nu	Average Nusselt number
$\frac{Nu_\omega}{Nu_s}$	Nusselt number ratio
n	Rotation speed, rpm
Pr	Prandtl number $= \frac{v}{\alpha} = \frac{C_p\mu}{k}$
p	Pressure, Pa
Q	Total rate of heat transfer, W
Re	Reynolds number $= \frac{u_i D}{v}$
Re_ω	Rotation Reynolds number $= \frac{\omega D^2}{\nu}$
s	Space ratio = $\frac{\text{Free spacing length}}{\text{Twisted tape length}}$

T	Temperature, K
TR	Twist ratio
TPF	Thermal performance factor
u_i	Inlet velocity, m/s
u	Axial velocity component, m/s
v	Radial velocity component, m/s
w	Tangential velocity component, m/s
WR	Tape width ratio
W_T	Width of the twisted tape, m
W_w	Tangential speed of the tube wall, m/s
x	Axial distance, m
x	Axial (flow) direction
X*	$= x/L$
U*	$= u/u_i$
V*	$= v/u_i$
W*	$= w/u_i$
T*	$= \frac{T_s - T}{T_s - T_i}$

GREEK SYMBOLS

α	Thermal diffusivity, m^2/s
Δ	Difference of variable
ΔT_m	logarithmic mean temperature difference, K
μ	Dynamic viscosity, kg/m.s
ν	Kinematic viscosity $= \frac{\mu}{\rho}$, m^2/s
ρ	Fluid density , kg/m^3
τ	stress tensor, N/m^2
τ_s	Wall shear stress, N/m^2
ω	Angular velocit, rad/s

SUBSCRIPTS

e	Exit
f	Fluid
i	Inlet
o	Outlet
P	Plain tube
s	Stationary
T	Twisted tape
w	Wall
ω	Rotation

SUPERSCRIPTS

*	Dimensionless

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Table 1: The dimensionless operating variables

Fluid	Re	$Re_{,\omega}$
Engine oil	5, 7, 9 and 10	0.9, 2, 4, 18, 35 and 70
Oil (SN-500)	54, 73, 91 and 109	9, 19, 38, 181, 380 and 760
Ethylene glycol	425, 566, 708 and 849	74, 148, 296, 1482, 2964 and 5927

Table 2: The dimensionless geometrical variables

TR	WR
5	0.3
7.5	0.5
10	0.7
-	0.9

تاثير ادراج الاشرطة الملتوية على انتقال الحرارة و زيادة الضغط للموائع عالية اللزوجة المتدفقة داخل الانابيب الدوارة

الخلاصة

يعرض البحث الحالي النتائج التي تم الحصول عليها من المحاكاة العددية التي أجريت لدراسة التحسن في انتقال الحرارة خلال الموائع عالية اللزوجة المتدفقة داخل الأنابيب المدارة باستخدام الأشرطة الملتوية ذات نسب لي و نسب عرض مختلفة

أنجز البحث باستخدام ANSYS FLUENT 14.0 لحل المعادلات الحاكمة للتنبوء بكل من مجالات التدفق و المجالات الحرارية ومن ثم حساب معدل انتقال الحرارة ومعامل الاحتكاك. و كانت النتائج كالآتي

- كفاءة الأشرطة الملتوية في زيادة مكون السرعة التماسية اعتمد بشكل كبير على سرعة الدوران فكلما زادت سرعة الدوران كلما قلت كفاءة الأشرطة الملتوية في زيادة مركبة السرعة التماسية في التدفق , مما أدى إلى انخفاض التحسن في انتقال الحرارة عند السرعات العالية.
- إدراج الأشرطة الملتوية في الأنابيب المزودة بالإثيلين جليكول منع حدوث التاثيرات اللزجة الكبيرة التي لوحظت في الأنابيب المدارة البسيطة عند سرعات الدوران العالية و هذا أسهم بشكل كبير في رفع معدل انتقال الحرارة عند السرعات العالية.
- زاد متوسط عدد نسلت بسبب إدراج الأشرطة الملتوية في الأنابيب المدارة (عند نسبة العرض 0.9) إلى حوالي 240% في الأنابيب المزودة بزيت المحرك ,245% في الأنابيب المزودة بزيت (SN-500) و 233% في الأنابيب المزودة بالإثيلين جليكول.
- زاد متوسط معامل الاحتكاك بسبب إدراج الأشرطة الملتوية في الأنابيب المدارة (عند نسبة العرض 0.9) إلى حوالي 457% في الأنابيب المزودة بزيت المحرك ,621% في الأنابيب المزودة بزيت (SN-500) و 160% في الانابيب المزودة بالإثيلين جليكول.
- معدل انتقال الحرارة و معامل الاحتكاك تاثر بالتفاوت في عرض الشريط من التفاوت في نسبة لي الشريط.
- يقل معامل الأداء الحراري نتيجة لإدراج الأشرطة الملتوية في الأنابيب المزودة بزيت المحرك وزيت (SN-500) بزيادة سرعة الدوران. بينما في الأنابيب المزودة بالإثيلين جليكول معامل الأداء الحراري يزداد بزيادة سرعة الدوران.
- زاد متوسط معامل الأداء الحراري بسبب ادراج الاشرطة الملتوية الى حوالي 60% في الانابيب المزودة بزيت المحرك, 61% في الانابيب المزودة بزيت (SN-500) و 22% في الانابيب المزودة بالاثيلين جليكول.
- في الانابيب المزودة بزيت المحرك وزيت (SN-500) افضل شكل هندسي للاشرطة الملتوية و الذي يعطي اعلى اداء حراري هو الذي له نسبة عرض تساوي 0.9. بينما في الانابيب المزودة بالاثيلين جليكول افضل شكل هندسي للاشرطة الملتوية هو الذي له نسبة عرض تساوي 0.5.