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Simulation of Turbulent Flow and Heat Transfer in A double Tube Heat Exchanger with Twisted Tape

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Abstract

Enhancing the thermal performance of a heat exchanger is a target for designers and researchers. This work uses numerical analysis to predict the thermal performance of a double-tube heat exchanger with a twist tape insert. Various twisting ratio (y/w) values of (2, 4, 6) were examined in the Re range of 5000 - 30000. The aim of this work is to assess the thermal performance of a double-tube heat exchanger by inserting a twisted tape. To model this physical issue, in addition to the turbulence model, the Navier-Stokes and energy equations are taken into account. ANSYS Fluent was adopted to simulate the model equations. The findings show that the twisted tape has a significant effect on the thermal performance of a heat exchanger. An increased coefficient of overall heat transfer and efficiency was observed when the twisted tape was inserted into the tube. The effectiveness of the heat exchanger improved by 34% compared with the double tubes without tape.

Keywords—ANSYS Fluent, CFD, Double tube- heat exchanger, Twisted tape

1 Introduction

The development of efficient heat exchanger devices has received a lot of attention from scientists and researchers due to the rising cost of energy and materials, as well as the increasing demand for energy. These days, heat exchangers' thermal performance characteristics are improved by using heat transfer enhancement approaches. Many engineering applications, such as refrigeration systems, chemical industries, power generation, oil and gas, heating, ventilation, and air conditioning, food and beverage processing, etc., heavily utilise heat transfer improvement techniques (Bhuiya et al., 2020). The double tube heat exchanger is a fundamental heat exchanger utilised in numerous industries. This type has gained popularity recently due to its widespread usage and simplicity. To improve total heat performance, a variety of strategies have been used, such as compound, active, and passive approaches. Furthermore, one of the most important ways to improve heat transmission with nanofluids is to hybridize flow liquids. Additionally, various twisted heat exchanger tube designs, including oval, square, triangular, and elliptical, were investigated. These shapes produced secondary eddy flow, which intensifies turbulence and mixing and enhances heat transfer. Additionally, conical rings, ribs, and twisted tapes are additives found inside heat exchanger tubes (A. k. Abdul Razzaq & Mushatet, 2023).The double pipe heat exchanger is arguably the most basic type of heat exchanger. Benefits include ease of use even in extremely fouled circumstances, and cleaning and maintenance. High-pressure fluids can also be employed. An alternative method for enhancing heat transfer is by utilising a shell and tube heat exchanger with an identical configuration. One of the drawbacks is the challenge of cleaning the tubes due to fouling. This section has conducted multiple investigations on the thermo-hydraulic efficiency of the twin pipe-heat exchanger. (Milani

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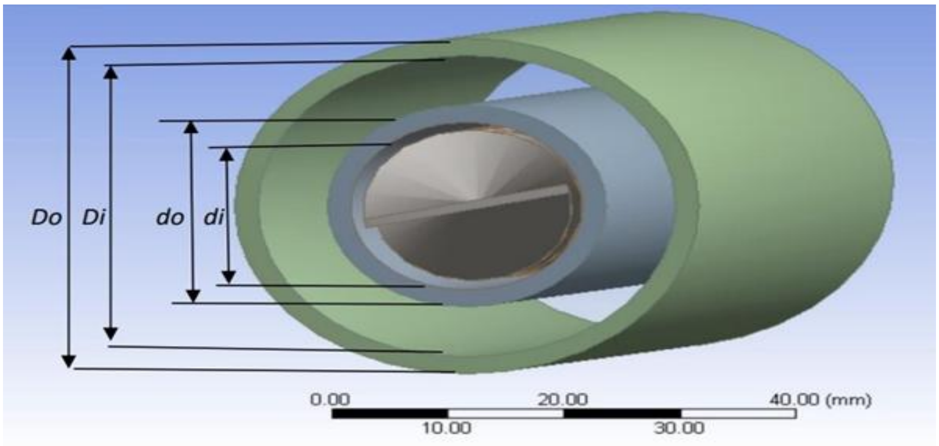
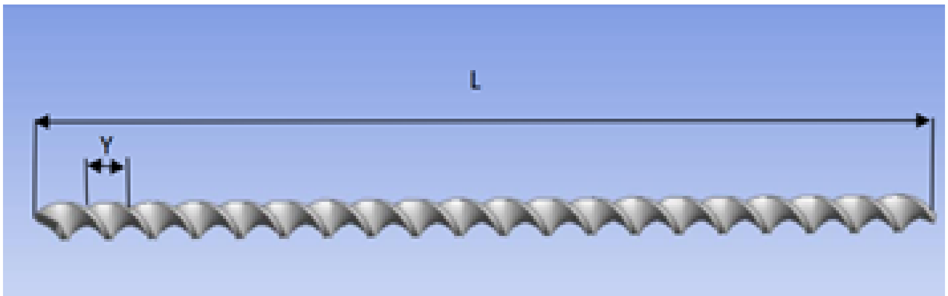
Shirvan et al., 2016)studied the enhancement of efficiency and heat transfer in a double pipe heat exchanger packed with porous material. It was discovered that as Re rises, Darcy number (Da) decreases, and the thickness of the porous substrate(δ) decreases, the mean Nu rises as well. Maximum effectiveness would occur at Re = 5000, Da = 10-3, and δ = 1. At Re = 5000, the efficacy was maximized simultaneously with Da = 10-3 and δ = 1 Nu. (Kotian et al., 2020) The analysis of thermos-hydraulic properties in a basic double-pipe heat exchanger arrangement greatly improves heat exchanger design. Due to the thermal characteristics of the hot and cold fluids, including their output temperatures, are greatly impacted by changes in the hot fluid's entrance temperature. It has been noted that the rate of heat transfer rises with increasing temperature differential. (El Maakoul et al., 2020) investigated the split longitudinal fins (SLF)to replicate an entrance region-like effect, in addition to the length's flow, which was discovered to be a benefit. The findings demonstrated that annuli with split longitudinal fins had higher heat transfer rates than annuli with traditional longitudinal fins (LF) by (31%–48%) with the same unit weight and pumping power. (Cavazzuti et al., 2015) examined the impact of conventional and the impact of perforated conical ring tabulators on the hydrothermal parameters of a twin pipe-heat exchanger, which facilitates heat transfer from air to water. The ideal setups resulted in a 10.8% improvement in their performance. There were two arrays taken into consideration: the Direct Conical Ring array and the Reverse Conical Ring. .it was discovered that for direct conical ring arrays, thermal performance increased as the conical angle increased. (Bezaatpour & Goharkhah, 2020) studied a novel strategy to lessen the penalty for pressure drop and enhance convective heat transfer. This is accomplished by creating a swirling flow in the magnetic working fluid by adding an external magnetic field. With this application, the pressure drop was reduced and heat transfer was increased by 320% due to the absence of any new obstacles in the flow path. (Bashtani & Esfahani, 2019) examined numerically, assuming three distinct wave amplitudes, a double pipe heat exchanger with a straightforward corrugated tube. Based on the findings, the process of corrugating increases the Nusselt number in a comparable Reynolds number range. At its peak performance, the average Nusselt number of the corrugated heat exchanger is approximately 1.75 times higher than that of the plain heat exchanger. The corrugated heat exchanger has an effectiveness ratio of 1.73 and a second law thermodynamic efficiency ratio of 1.17, when compared to the plain heat exchanger. (Zhang et al., 2019) The thermal properties of a double pipe-heat exchanger equipped with self-rotating twisted-tapes perforated at six different ratios—0%, 1.16%, 3.63%, 6.46%, 10.1%, and 14.49%—were experimentally studied. It was demonstrated in experiments that perforated self-rotating twisted tapes outperform perforated stationary twisted tapes in terms of thermal performance. (A. K. Abdul Razzaq & Mushatet, 2021) studied the employing twisted tubes rather than plane tubes, this study seeks to improve the heat exchanger's double tube performance. The optimal overall thermal-hydraulic performance is achieved with a development ratio of 5. The heat exchanger with a flow rate of 0.4962 and (Tr = 5) has an efficiency gain of 14.8%. (Sharifi et al., 2018) investigated the effects of coiled wire inserts on total efficiency, friction coefficient, and Nu. It was discovered that the Nu might increase by more than 1.77 times when coil inserts were used properly. Nu correlations and the friction coefficient were then suggested for numerical simulation. (Mushatet & Hmood, 2021) Enhanced turbulent heat transfer in heat exchangers using twisted triangular tubes. With significant friction losses, the twisted ratio (Tr=5) demonstrated the best heat transmission. For (Tr=5), the heat transfer enhancement for the twisted tube under consideration is increased by 150%. (Wijayanta et al., 2018) Improved the heat transfer by inserting double-sided delta wing tape. Research was conducted for 5300 ≤ Re ≤ 14,500. With a wing-width ratio of 0.63, the T-W tape insert produced the highest average Nusselt number, which is 177% greater than the plain tube's average. The greatest thermal performance factor (1.15) and Nu are likewise produced by the T-W tape insert. The thermal performance factor rises as the T-Ws' wing-width ratio does as well. (Mushatet et al., 2020) studied the effects on thermal and hydrodynamic properties of new tapered configurations of twisted tape. The largest increases for the Nusselt number and friction factor, respectively, were 75% and 100% in the experimental and numerical respectively, performance factors. A high Nusselt number and friction factor are produced by a tiny SW or a short length of tapered twisted tape. Several increased tapered twisted tap(ITTT) configurations yield a higher thermal performance factor (TPF) than the standard twisted tape (TT). (riyam ali & Khudheyer Salim, 2024) studied the twisted tape inserts and twisted double tubes. The twisted ratios of the tubes were evaluated at 5, 10, and 15, while the torsion ratios of the tape were assessed at 4, 6, and 8. The use of a twin twisted tube-heat exchanger with a twisted tape insert led to improved heat transfer. The double twisted tube heat exchanger (DCTTHE) achieves an efficiency of around 40% when the hot water flow rate is 0.024 and the temperature ratio (Tr) is 5. (Bazgir et al., 2018) suggested that cooling the vortex tube's main, or hot, the addition of a tube would improve the thermal efficiency and performance of a counter-flow vortex tube. It was determined that chilling the hot-tube contributes to raising the cold air temperature differential (ΔTc), and enhancing the system's isentropic efficiency. The vortex tube, when equipped with cooling water, exhibits increased efficiency and higher differentials in cold air temperature, ranging from 5.8% to 10.6% and 6.3% to 10.3%, respectively, than the one without cooling water. (Razzaq & Mushatet, 2022)studied the twin circular tube heat exchanger by using nanofluids and a hybrid twisting tube technique. Al2O3 nanoparticles are utilised with clean water as the base fluid. Due to the greater thermal conductivity of the nanofluids (Al2O3-water), the double twisted-circular tube exhibited superior performance while using a nanofluid (Al2O3-water) with a diameter of Ø = 4% compared to pure water, with an improvement of roughly 10.7%. (Heeraman et al., 2023) utilise a cylindrical conduit constructed from coiled adhesive tape with indentations. Impact TT inserts are specifically engineered with a slightly elevated surface on the front side, which incorporates a hole-forming characteristic. The pipe is adjacent to it at different depth ratios (D/H) along a 1500 mm length. The twisted tape insert with dimples offers a significant peak in heat values. There is a higher friction factor and gearbox rate. When the Reynolds number is increased, the twisted tape with dimples has a greater impact on the friction factor than the Nusselt number. Out of all the configurations, the twisted bar without a dimple has the least sensitive Performance evaluation criteria Dimensionless (PEC), suggesting that in this instance, the total friction factor may hold greater significance than the Nusselt number. (Mushate & Youssif, 2018) analyzed numerically the Al2O3 nano fluid flows through a horizontal pipe with a twin twist tape insert. The twisting ratio values of (2, 4, 6) are examined at volume concentrations (ϕ) of 0.5% and 4%, within the Re range of 5000 to 35000. The use of an alternating twin twisted tape with Al2O3/water can significantly increase heat transmission by roughly 214% compared to a plain tube. (Khaled & Mushatet, 2023) A numerical analysis has been conducted on double elliptical twisted tubes equipped with twisted tape. The utilisation of twisted tape (TT) significantly influenced heat transfer along the wall by enhancing fluid and centrifugal force mixing. The analysis uncovered a substantial improvement in both heat transmission and performance when compared to a traditional double-tube heat exchanger. (Mushatet & Youssif, 2018) studied numerically a double heat exchanger, where they combined twin twisted tape and nano fluid. A twin-twisting tape wound, both clockwise and anticlockwise, is utilised. The results show that the twist is more active than the single twist for improving heat transmission. This increase is amplified by the rise in volume concentration. Moreover, an anticlockwise twisting tape performed better thermally than a clockwise twisting one compared to the single-twisted tape. In comparison to a plan tube, the coupled counter twin is twisted together with an. Alumina-water nano fluid showed a 191%. increase in heat transmission. (Tang et al., 2024) studied a novel approach to enhance the flow and thermal efficiency by integrating elliptical tubes with twists that may change direction. This study elucidates that the variable-direction twisted oval tube enhances thermal performance by increasing the degree of mixing in both the main stream and the tail vortex, hence reducing pressure loss. As the angle of twist increases, there is a simultaneous increase in both the pressure drop and the heat transfer coefficient. Additionally, there is a negative association between the total performance and the twist angle. The twist time has minimal impact on the overall performance. Compared to a circular tube bundle, a variable-direction twisted oval tube with a 120° twist angle has a convective heat transfer coefficient that is 5.8%–10.2% higher. Additionally, the pressure drop is 5.0%–7.8% lower, resulting in an overall performance gain of 7.4%–10.7%. (Fagr et al., 2020) investigated the effects of twisted tapes with less tapered geometries within a tube has a significant effect on the thermal and hydrodynamic fields. The collected results indicate that there is no noticeable difference in the thermal performance factor when comparing the insertion of standard twisted tape with certain researched scenarios involving the insertion of tapered twisted tape with reduced dimensions. The friction factor and Nusselt number go up when the tapes are twisted. When a traditional TT is used, it results in higher values of TPF, friction factor, and Nusselt number. (Wu et al., 2018) studied the experimental investigation of heat transfer and flow resistance in a twisted elliptical tube, considering parameters such as Rey and twist pitch. Compared to oval tubes, it was shown that twisted elliptical tubes cause rotational motions in the fluid flow, strengthening the relationship between the temperature gradient and the velocity vector. In a 128-mm twisted elliptical tube, the best thermal-hydraulic performance was recorded.

2 Physical Model

Figure 1 illustrates the composition of the real device, which consists of a dual-tube heat exchanger incorporating a twisted tape. Displays a rudimentary image of a counter-flow heat exchanger employed in this study. The building is made up of cylinder-shaped tubes that are all (1 m) long. Fluid that is hot can move through the inner tube, and fluid that is cold can move through the outer tube. The inside tube has a semi-major diameter of (di)=0.02 m, and the outside tube has a semi-major diameter of (do)=0.024 m. This tube's semi-minor diameter is (Di)=0.036 m, and this tube's semi-minor diameter is (Do)= 0.04 m. Two tubes that are 0.004 m thick each other. The twisted tape was tried with twisted ratios of 2, 4, and 6 all simulations were done Re from 5000 to 30000.



(a) Cross-section of for double pipe heat exchanger with the twisted tape



(b) Longitudinal section view for the twisted tape

Figure 1: A schematic illustration of the physical problem

3 Mathematical Model and Numerical Analysis

Assumed to be incompressible with consistent thermal and physical characteristics. The governing equations of continuity, momentum, energy, and turbulence model (Razzaq & Mushatet, 2022):

$$\frac{\partial \rho u}{\partial x}+\frac{\partial \rho v}{\partial y}+\frac{\partial \rho w}{\partial z}=0 \tag{1}$$

$$\left(\frac{\partial u^2}{\partial x}+\frac{\partial uv}{\partial y}+\frac{\partial uw}{\partial z}\right) = -\frac{1}{\rho}\frac{\partial P}{\partial x}+\frac{\partial}{\partial x}\left(2\mu_{eff}\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial u}{\partial z}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial x}\right) \tag{2}$$

$$\left(\frac{\partial \boldsymbol{v}u}{\partial x}+\frac{\partial \boldsymbol{v}^2}{\partial y}+\frac{\partial vw}{\partial z}\right) = -\frac{1}{\rho}\frac{\partial P}{\partial x}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial \boldsymbol{v}}{\partial x}\right)+\frac{\partial}{\partial y}\left(2\mu_{eff}\frac{\partial \boldsymbol{v}}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial \boldsymbol{v}}{\partial z}\right)+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial y}\right) \tag{3}$$

$$\left(\frac{\partial wu}{\partial x}+\frac{\partial w\boldsymbol{v}}{\partial y}+\frac{\partial w^2}{\partial z}\right) = -\frac{1}{\rho}\frac{\partial P}{\partial x}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial w}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial w}{\partial y}\right)+\frac{\partial}{\partial z}\left(2\mu_{eff}\frac{\partial w}{\partial z}\right)+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial z}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial \boldsymbol{v}}{\partial z}\right) \tag{4}$$

$$\frac{\partial \boldsymbol{u}T}{\partial x}+\frac{\partial \boldsymbol{v}T}{\partial y}+\frac{\partial \boldsymbol{w}T}{\partial z}=\frac{\partial}{\partial x}(\Gamma_{eff}\frac{\partial T}{\partial x})+\frac{\partial}{\partial y}(\Gamma_{eff}\frac{\partial T}{\partial y})+\frac{\partial}{\partial z}(\Gamma_{eff}\frac{\partial T}{\partial z}) \tag{5}$$

$$\mu_{eff}=\mu+\mu_t \tag{6}$$

$$\Gamma_{eff}=\Gamma+\Gamma_t \tag{7}$$

3.1 Turbulence model

The standard k-ε model, along with the enhanced wall treatment method, is used to accurately simulate turbulent flow, particularly in the near-wall region. The standard k-ε model has been extensively utilised in the field of heat transfer flow due to its practical accuracy in a variety of turbulent flows. The values of k and ε are provided in reference (Razzaq & Mushatet, 2022).

$$\rho\left(\frac{\partial}{\partial x}(ku)+\frac{\partial}{\partial y}(kv)+\frac{\partial}{\partial z}(kw)\right)=\frac{\partial}{\partial x}\left(\frac{\mu_t}{\sigma_k}\frac{\partial k}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\mu_t}{\sigma_k}\frac{\partial k}{\partial y}\right)+\frac{\partial}{\partial z}\left(\frac{\mu_t}{\sigma_k}\frac{\partial k}{\partial z}\right)+G-\rho\varepsilon \quad (8)$$

$$\rho\left(\frac{\partial}{\partial x}(\varepsilon u)+\frac{\partial}{\partial y}(\varepsilon v)+\frac{\partial}{\partial z}(\varepsilon w)\right)=\frac{\partial}{\partial x}\left(\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial y}\right)+\frac{\partial}{\partial z}\left(\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial z}\right)+C_{1\varepsilon}\rho\frac{\varepsilon}{k}G-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \quad (9)$$

Here, μ_t represents the turbulent dynamic viscosity. The value of μ_t is zero for laminar flow. The term "G" is employed in this context to refer to different generations. Its meaning is further upon in reference.

$$G=\mu_t\left[2\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2+2\left(\frac{\partial w}{\partial z}\right)^2+\left(\frac{\partial v}{\partial y}\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}\frac{\partial w}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}\frac{\partial w}{\partial y}\right)^2\right] \quad (10)$$

Also,

$$k=\frac{1}{2}\left(\overline{u'^2}+\overline{v'^2}+\overline{w'^2}\right) \quad (11)$$

$$\varepsilon=\overline{e'_{ij}.e'_{ij}} \quad (12)$$

$$\mu_t=\rho c_\mu\frac{\mathbf{k}^2}{\varepsilon} \quad (13)$$

3.2 Data reduction

The log mean temperature difference (LMTD) for counter flow is calculated using the following formula (Khaled & Mushatet, 2023):

$$\text{LMTD}=\Delta T_1-\Delta T_2/Ln\left(\frac{\Delta T_1}{\Delta T_2}\right) \quad (14)$$

$$\Delta T_1=(T_{hi}-T_{co}) \quad (15)$$

$$\Delta T_2=(T_{ho}-T_{ci}) \quad (16)$$

The heat exchanger effectiveness is:

$$\varepsilon=q_{act}/q_{max} \quad (17)$$

$$C_h=m_h cp_h \quad (18)$$

$$C_c=m_c cp_c \quad (19)$$

The overall heat transfer coefficient of double pipe-heat exchanger (Khaled & Mushatet, 2023).

$$U=Q_{avg}/A_{s,o}LMTD \quad (20)$$

Where , $A_{s,o}=\pi d_o L, Q_{avg}=\frac{Q_h+Q_c}{2}$

3.3 Boundary condition

The model under examination is subject to the following boundary conditions:

1. Constant temperature (Tci=303, Thi=343) K and Re (5000 -30000) are used at the two-tube inlet.
2. No slip condition, adiabatic wall.
3. At the outlets, zero relative gauge pressure is presumed.

4 Numerical computation

4.1 Convergence and coding

The entire performance of the double pipe heat exchanger with the twisted tape was investigated in order to forecast and depict the impact with relation to fluid dynamics. The fluid flow and heat equations that were previously discussed can be numerically solved using the mass, momentum, and energy equations as computational parameters [1, 2, and 3].

4.2 Grid study

Structured meshes offer several advantages over unstructured meshes, including the ability to use fewer cells and produce higher quality outputs. Additionally, structured meshes facilitate faster and easier convergence. A structured mesh was constructed in this study. The ANSYS meshing software was utilised to construct the present mesh. Figure 2 provides a high-level overview of the computing grid, specifically designed for analysing the solutions' independence from the created mesh. The collection of tetrahedral elements forms the computational domain. The mesh system consists of 1,326,567 components.

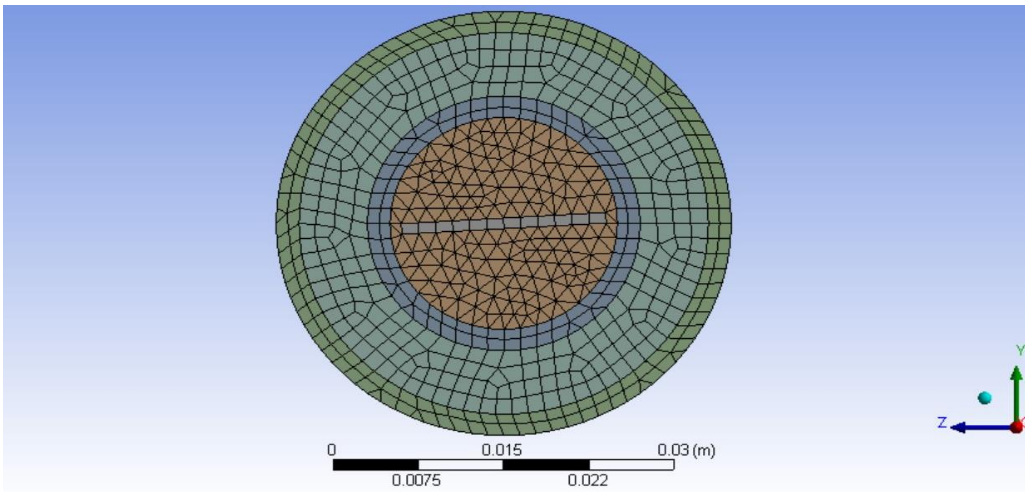


Figure 2: Mesh generation

The average total heat transfer coefficient is calculated for several mesh sizes to reach grid independence. Table 1 shows in a double tube heat exchanger (DTTHE) running at a Reynolds number of 5000 the total heat transfer coefficient for various mesh sizes. The last column shows the variations between the two scenarios that follow. The computation consists in subtracting the larger mesh size from the total heat coefficient for the smaller mesh size, then dividing the resulting number by the smaller value. The 1333455 increasing number of mesh elements has resulted in an error rate of 0.24% as seen in the table1. This implies that the total coefficient is not much changed by further changing the mesh. Consequently, in the prior arrangement the mesh's scale exactly corresponds to the number of mesh components used.

Table 1: Overall heat transfer coefficient for different mesh size

Cases	Mesh Size	Overall Heat Coefficient (U)	(Percentage Error)%
1	1065954	388.7891	0.00091
2	1227675	389.1442	0.00048
3	1314435	389.3342	0.000094
4	1325156	389.3711	0.00047
5	1326567	389.5567	0.00023
6	1333455	389.6502	-

5 Model Validation

Figure 3 illustrates the hot and cold water exit temperatures for the present computational model and experimental data by (Kotian et al., 2020) in terms of hot flow rate. To validate the numerical model, the identical experimental conditions outlined by (Kotian et al., 2020) are incorporated, including the inner and outer tube lengths of 1200 mm, inner tube diameters of 26 mm and 68 mm, and inner and outer tube thicknesses of 4 mm. It can be seen that the results from the current model and the experimental data given by (Kotian et al., 2020) accord well with one another. The largest discrepancy between the outcomes is 2.5%. As a result, this technique has been established as robust and accurate.

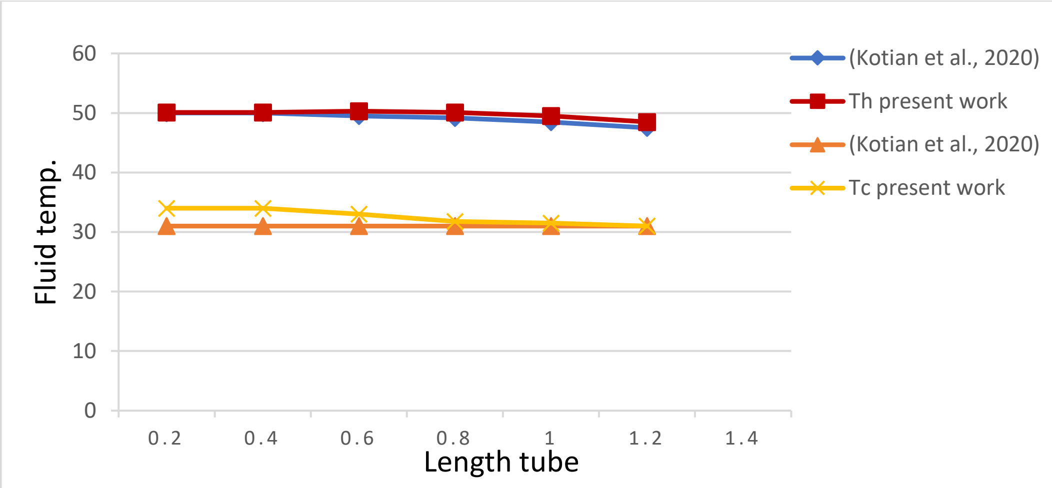


Figure 3: Validation of hot and cold liquid temperature between the present results and the results of (Kotian et al., 2020) published

Figure 4 shows the effectiveness of the present results compared to those published by (A. K. Abdul Razzaq & Mushatet, 2021). The numerical model is validated under the published as reported. These specifications include inner and outer tube diameters of 25 mm and 50 mm, inner and outer tube thicknesses of 4 mm, and inner and outer tube lengths of 1000 mm. It is clear that the outcomes from the current model and the published data supplied coincide quite well. The largest discrepancy between the findings is 2.4%. Thus, the accuracy and dependability of this approach have been demonstrated.

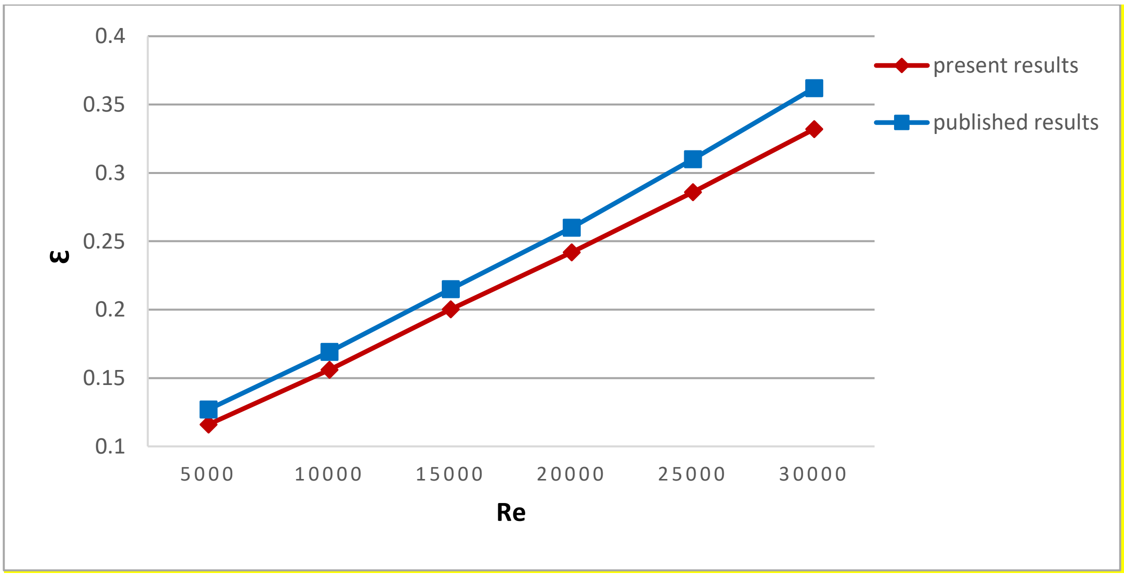


Figure 4: Validation effectiveness between the present results and the published results of (A. K. Abdul Razzaq & Mushatet, 2021)

6 Results and Discussion

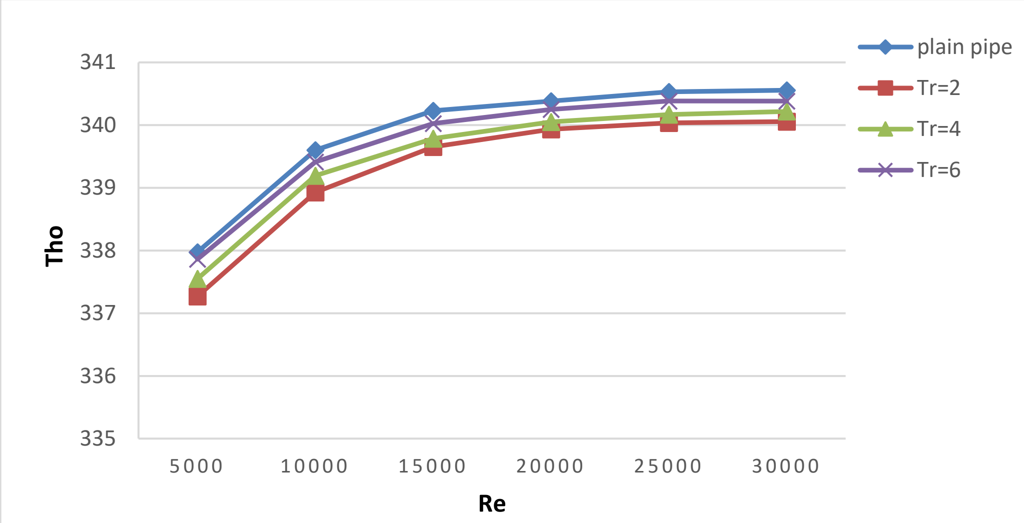


Figure 5: Difference of the hot fluid outflow temperature with Re for DTTHE

An essential component of comprehending the variables affecting the heat transfer rate is flow analysis. Figure 5 it shows the hot fluid outlet's temperature change ،Heat exchanger tube with twisted tape and a twisted ratio (Tr = 2,4 and 6) with Reynolds number. The figure illustrates how Tho increases when Re increases as a result of the difference in the flow of hot fluid, which decreases heat transfer time exchange. In comparison to a standard tube, the inner tube performs better thermally when the twisted tape is inserted. Figure 6 shows the relationship between the temperature of the cold water coming out of the heat exchanger and the Reynolds number for the tube with used difference the twisting ratio for the tape. When DTTEH is used, the cold water exit temperature is higher than when tape is not used. In all cases, it goes up when the Re goes up. The heat energy dissipate from the inner tube will grow as the Reynolds number rises. Heat exchanger tubes equipped with twisted tape demonstrate a noticeable increase in the average cold water output temperature compared to those without tape.

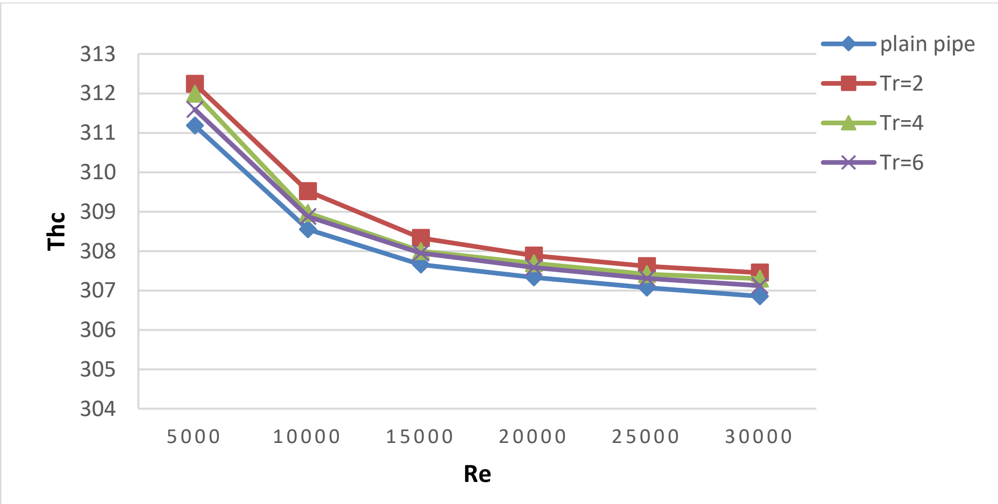


Figure 6: Variation of cold fluid outlet temperature with Re for DTTHE

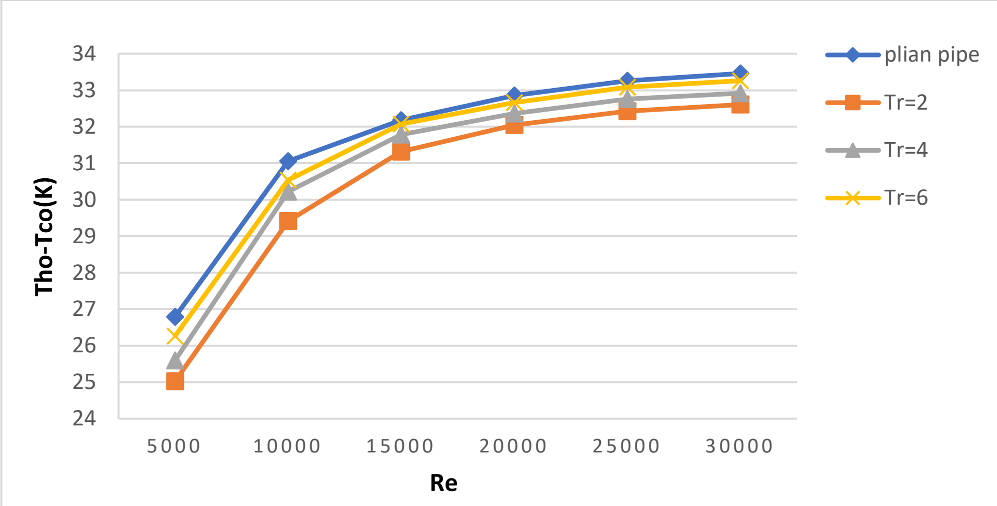
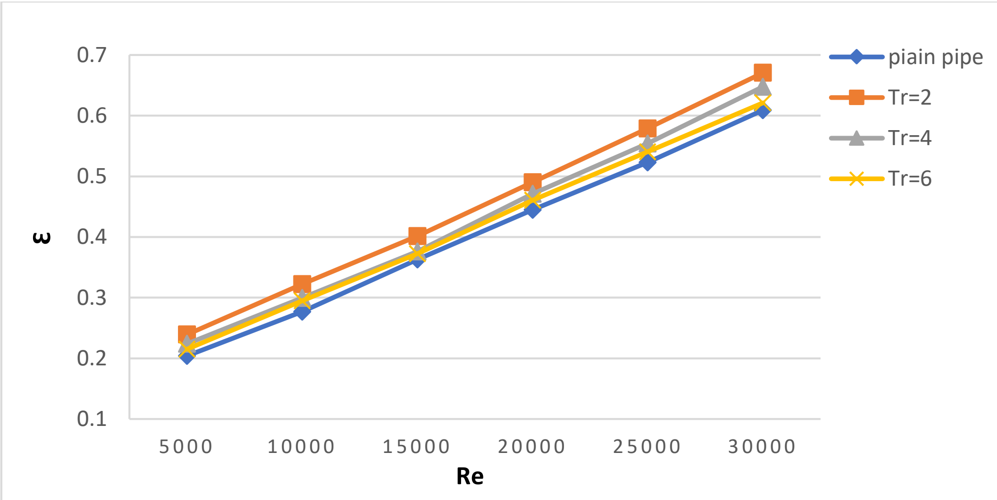


Figure 7: The difference between hot and cold fluid outlet Temperature with Re for DTTHE

Figure 7 displays the disparities in hot and cold fluid output temperatures between a standard double tube Heat Exchanger and one with a twisted tape insert, along with the corresponding Reynolds number. The figure shows how (Th-Tc) K increases with increasing Re, resulting in higher temperatures when producing hot and cold fluid.



Figuer 8: Effectiveness variation against Reynolds number for DTTHE

Figure 8 shows the variation of heat exchanger efficiency with the Re for fluid flow. It is noted that the effectiveness of the heat exchanger gradually increases when a twisted tape is added, reaching the highest value when a twisted tape is included for (Tr=2). The effectiveness of the heat exchanger improved by 34% compared with the double tubes without tape. This is due to an increase in the amount of heat exchanged due to increased turbulence of the fluid, in other words, an elevation in the thermal conductivity of the fluid.

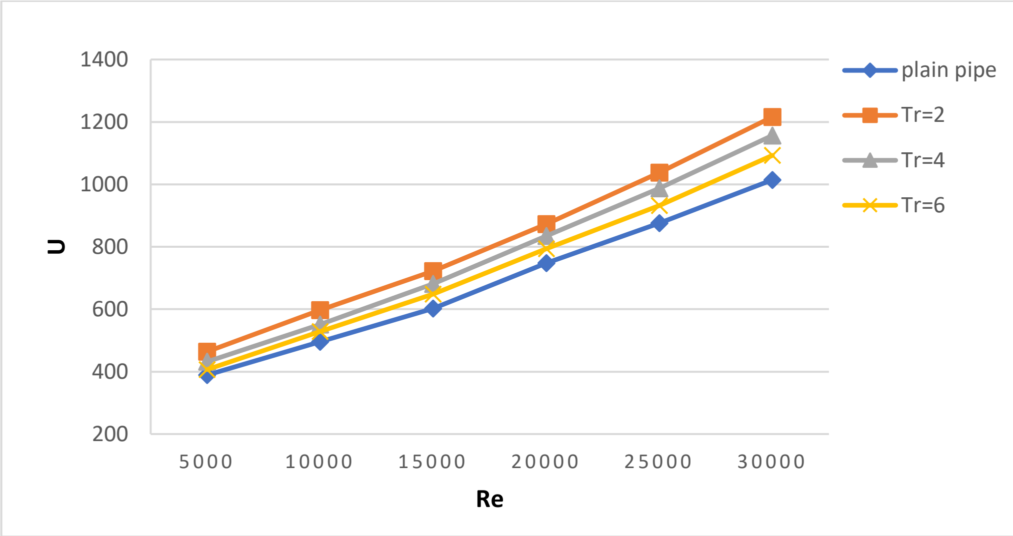
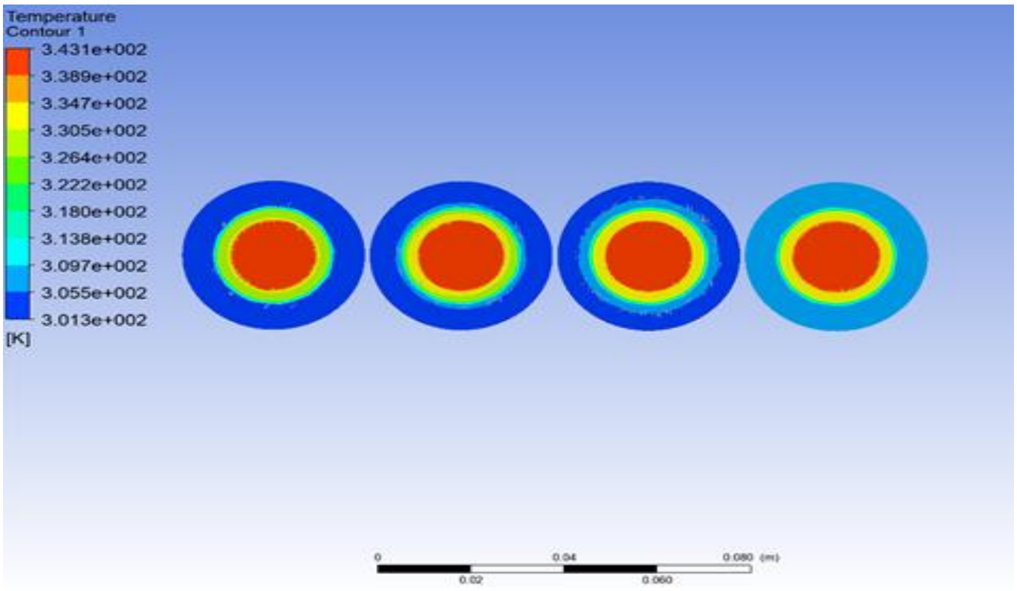
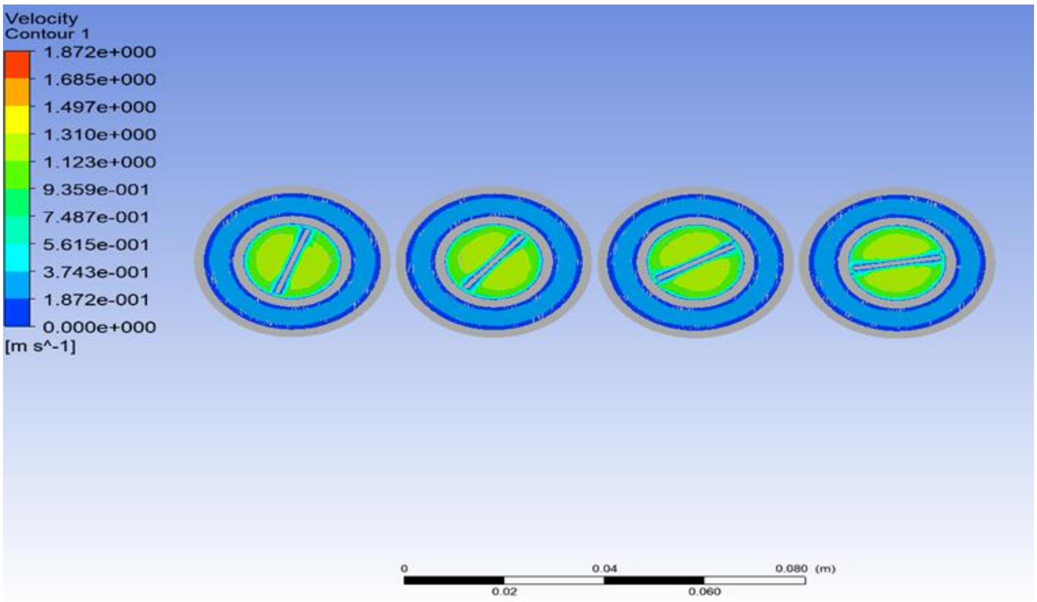


Figure 9: Variation of the coefficient of overall heat transfer versus Reynolds number

Describes the connection between the Re for a double tube-heat exchanger and the coefficient of total heat transfer. Figure 9 shows how a large amount of heat is transferred when a twisted tape is added with twisted ratios (Tr=2,4, and 6) compared to a conventional double tube-heat exchanger. Therefore, the higher the Reynolds number, the higher the overall heat transfer coefficient. Twisted tape is added to the double tubes to promote heat transfer because it creates a secondary flow, as shown in Figure 9.



Figuer 10: Cross-section temperature contours of the hot and cold fluid at
(x=0.2, x=0.4, x=0.6, x=0.8)m for DTTHE at Re (30000)



Figuer 11: Cross-section velocity contour at (x=0.2, x=0.4, x=0.6, x=0.8) m for DTTHE

7 Conclusions

This study demonstrates the practicality of improving convection heat transmission by incorporating a heat exchanger and utilizing a twisted tape. The primary discoveries of this investigation can be succinctly summarized. The insertion of a twisted tape into the dual heat exchanger leads to an increase in the rate of heat exchange between hot and cold water temperatures. Introducing twisted tape in different quantities has been proven to greatly improve the efficiency of the heat exchanger. When the twisted tape was inserted, an increase in pressure was observed with an increase in the Reynolds number as a result of increased fluid turbulence inside the tube. The heat exchanger with twisted tape has a higher overall heat transfer coefficient and is capable of transferring a greater amount of heat compared to a standard heat exchanger. The overall heat transfer coefficient rises as the Reynolds number increases. The use of twisted tape inserts results in a 25% increase in U compared to not employing any inserts.

8 References

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