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Fluid flow and heat transfer features in a twisting elliptical tube

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Abstract

Thermally walls heat tiny twisting elliptical tube (*TOTs*) for examine flow behavior and heat transmission within the pipes. Because as hydraulic fluid, freshwater is used to mimic various tubes designs, having Higher Reynolds number ranging from 500 to 1100 depending just on thermal thermostatically characteristics. Using known correlations and actual findings to validate the numerical approach, the effectiveness for TOTs was tested with that of a smoothly plain cylinder. Thermal efficiency for TOTs is investigated, as well as the findings are examined in terms of a field's cooperation concept and energy production. TOTs are rated for actual quality. TOTs' improved heat transfer capability comes at the cost of a greater pressure loss, as research shows. As a result, it's determined that the pipe twist's flow pattern enhances convective heat transfer, leading to improved heat transfer characteristics. Improved synergy among flow and thermal gradients, as well as reduced inevitability, lead to improved TOT energy transfer.

Introduction

There are many applications for heat exchangers with in automobile and aerospace sectors, refrigeration, power grids, and climate control. Enhanced heat transfer rate and reduced pressure drop may increase heat exchanger performance [1]. There were many studies shedding light both passive or active techniques for heat transfer augmentation [2-4]. Surface vibrations and acoustical or electrically fields are examples of active techniques, whereas different surface designs and water additives are examples of alternative techniques [3, 5, 6]. TOTs, as seen in Fig.1, are still a passively higher heat transfer method [7]. Twist elliptical pipe heat exchanger (TOTHE) increase heat transfer performance by eliminating shell pressure loss while increasing efficiency of heat transfer on the pipe wall. TOTHEs are more popular these days because to their impressive capabilities. Since the invention of TOTHEs with in 1980s, the large number of scientists have been researching that tube's heat transmission or pressure drops properties both on within and the outside. It was discovered by Bishara et al. [8, 9] that laminar flow of Prandtl number (Pr) 3.0 in such a twist-ratio Six oval tube as well as AR (about 1.43 at Reynolds numbers (Re) below 1200) has excellent heat transfer properties. The flow properties of a twisted elliptical tube, which are crucial, were not taken into account while reporting on the enhancement of heat transfer in convoluted oval pipes. Heat transfer and friction factor values Re ranging from 7000 to 200,000 were correlated by Asmantas et al. [10], who carried out experiments on turbulent air flow in flat, twisted tubes It's still unclear how the twisting elliptical tube increases heat transmission. Flow of water through twisted elliptic pipes was studied out from laminar to turbulent regimes by Yang et al. [7] while those results were then compared to those of pipe. With regard to better performance in twisting elliptic pipes, they found that low-Re flow had the highest heat transfer as well as lowest pressure drop. There was a shift from smooth edges tube flow to turbulent flow with smaller Re, as they presented correlations to forecast twisting elliptical tubes' heat transfer coefficients and friction factor. Tan et al. [11] have studied experimentally and analytically the heat transfer coefficient and pressure drop properties of TOTs in a turbulent environment. Expanding cross section area AR or reducing twist ratio led in such a rise within both heat transfer coefficient and friction factor when geometrical variables were considered. They also looked at their findings from the perspective of field synergy. twisted flat tubes and provided heat transmission and friction factor correlations for Re ranging from 7000 to 200,000. It's still unclear how the twisted elliptical tube increases heat transmission. Many research looked at the impact of utilizing twisted oval tubing just on tube side for TOTHEs to learn more about their overall quality. As for removal of heat from the shell's surface and pressure loss, Dzyubenko and colleagues [12-14] used experiments to determine turbulence intensity, thickness of the boundary layer, as well as heat diffusion coefficients. They also provided numerical models that calculated these results numerically. Experiments have been conducted as well as computational analyses on the heat transfer and friction loss upon that shell side to create relationships. On both of the shell and tube heat exchanger sides, Tan et al. [17] investigated the effects of a TOTHE on heat transmission and pressure losses in quite an experimental situation When the discharge rate is so low with in tube as well as high as in shell, all TOTHEs perform better.

Additionally, Tan et al. [18] carried out a computational analysis of TOTHE's casing friction and heat thermophysical properties with in turbulence domain. ץ The ability to save energy and produce better effective TOTHEs depends heavily on our ability to better comprehend the heat transfer characteristics in TOTs. Exergy assessment may assist in understanding the process of heat transfer increase even if the first rule of thermodynamic has always been applied. Entropy production within conduits or pipes has also been the focus of several studies for optimizing heat transmission and fluid flows at the same time [19-22]. To date, many analyses upon that heat transfer enhancement for TOTs have relied on Average nusselt number, that do not adequately explain the enhanced heat transfer. As addition, despite the lot of interest in heat transfer studies, little attention has been paid to the streamlines and how it affects heat transmission. As a result of the excellent potential of twisting elliptical tubing and the absence of researches on the processes that improve laminar flow behavior as well as heat transfer, more research into the convective heat transfer properties of twisted elliptical tubes is required. Using a finite element model, this research examines the laminar forced convection heat transport in twisted oblong tubing with various ARs. Modelling are being run to determine the impact of the twisting oval tubes on flow characteristics at various Reynolds numbers (Re). Ultimately, we want to assess the total heat transfer efficiency of twisty elliptical tubes and discover, to a best of knowledge, what drives the fields synergy concept and heat production analysis that we've used in this project.

Design modeling

Four distinct twisted elliptical tube topologies with varying cross-sectional ARs (a/b) were simulated in three dimensions. Fig.1 depicts the mathematical model for twisting tubes utilized in this research as well as the essential geometrical factors. The main and secondary axes of a tube's cross-sectional area have lengths of 2a and 2b. The hydraulically diameter (Dh) was 5 mm as well as the twisting section lengths (S) equals 60 Dh throughout all instances examined in this research. Mathematical model is divided into three areas: the flowing development region with isothermal walls with in intake domain, the outlets sector with an exit length about 10Dh and thermal walls that avoid any possible back flowing flows, as well as the warmed zone with both the warmed wall in the heated area. For the twisting region, it's important to choose the shortest possible length of an entry boundary I to have a nearly completely formed flow. On average, there are around 30 degrees of rotation in the twisting phase, as well as the flow path starts off to be a straight line.

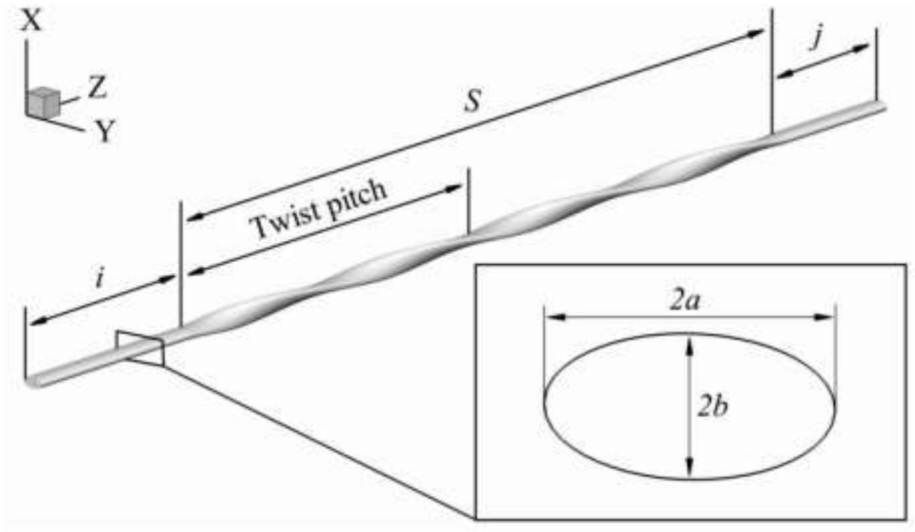


Fig. 1 Physical model and relevant geometrical parameters of twisted tubes.

Mathematical model

Newtonian modeling utilizes water as that of the heat transfer fluid. Laminar as well as deformable flow was examined via twisting elliptical pipes with an uniform heat flux, with thermally thermophysical characteristics Neither radiation nor gravitational influences are taken into consideration in this investigation. Mass, momentum, and energy conservation equations in steady - state condition may be expressed as follows:

$\nabla \cdot \rho \mathbf{U} = 0$ 1

$\nabla \cdot (\rho \mathbf{U}\mathbf{U}) = -\nabla p + \nabla \cdot [\mu(\nabla \mathbf{U} + (\nabla \mathbf{U})^T)]$ 2

$\nabla \cdot \left(\rho c_p T \mathbf{U}\right) = \nabla \cdot (k \nabla T)$ 3

in which U would be the fluid's motion direction where, ρ, μ, k, T, cp, with p were density, kinematic viscosity, thermal expansion, temperature, and heat capacity, correspondingly. To describe the thermally thermo-physical characteristics of water, which can be shown as follows [24-30].

$\rho_{\text{Water}} = 753.2 + 1.88T - 3.570 \times 10^{-3} T^2$ 4

$k_{\text{Water}} = -0.5981 + 6.53 \times 10^{-3} T - 8.354 \times 10^{-6} T^2$ 5

$\mu_{\text{Water}} = 2.591 \times 10^{-5} + (238.3/(T - 143.2))$ 6

$c_{p,\text{ Water}} = 4200$

Boundary Conditions

In which the water flow is determined by: somewhere at input, exit, and boundaries

Inlet: $u = \mathrm{U_i}, V = W = 0, T = \mathrm{T_i} = 300.15(\mathrm{K})$ 7

Outlet: $\frac{\partial u}{\partial z} = \frac{\partial v}{\partial z} = \frac{\partial w}{\partial z} = 0, \frac{\partial T}{\partial z} = 0$ 8

Adiabatic wall: $u = V = W = 0, k\frac{\partial T}{\partial y} = 0$ 9

Heated wall: $u = V = W = 0, T = \mathrm{T_{wall}} = 348.15(\mathrm{K})$ 10

where u, v and w are velocity components in X, Y and Z direction, respectively. The subscripts of "i" and "wall" stand for inlet and wall conditions, respectively.

Numerical analysis

ANSYS generates an organized quasi meshes through discretizing the mathematical model that used a 2^{nd} upwind technique and refining it walls to improve reliability while also minimizing computational expense. The numerical method code Ansys Software 14.0 is used for all computations. Whenever the normalized error terms of the energy, continuum, and governing equations were much less below $1.0*10^{-10}$, $1.0*10^{-15}$, overall resolution requirement is fulfilled. These dependent variables were added only for the sake of demonstration. The formulae below determine the Reynolds number (Re) depending on pipe diameter (D_h) with actual friction coefficient (f) [31-37].

$Re = \frac{\rho \mathrm{U_i} D_h}{\mu}$ 11

$f = \frac{2\Delta p}{\rho \mathrm{U_i^2}} \frac{D_h}{L}$ 12

$\Delta p = \left(\bar{p}_o - \bar{p}_i\right)$ 13

$\bar{p} = \frac{\int p dA}{\int dA}$ 14

$Nu = \frac{h D_h}{k}$ 15

$Q = \dot{m} c_p \left(\bar{T}_o - \bar{T}_i\right)$ 16

$\bar{T} = \frac{\int T\rho |\mathbf{U} \cdot d\mathbf{A}|}{\int \rho |\mathbf{U} \cdot d\mathbf{A}|}$ 17

$Nu_m = -\frac{D_h}{k_m} \ln\left(\frac{\mathrm{T_{wall}} - \bar{T}_i}{\mathrm{T_{wall}} - \bar{T}_o}\right) \frac{\dot{m} c_{p,m}}{A_{ht}}$ 18

$$\eta = \left(Nu_m / Nu_{m,0}\right) / (f / f_0)^{1/3} \qquad 19$$

The proportional entropy energy content may be calculated using the flowing field's velocity profiles variation.

Results and discussion

Grid generation

Using non-uniform patterning polyhedral volumetric pieces and comparing the average Nusselt number with seeming friction coefficient at every refinement stage, it is determined whether the computationally findings are independent of the mathematical formulation. There's many 5 different cell size taken into account: 95,400 (very coarse), 200,800 (coarse), 337,000 (intermediate), 566,800 (fine), and 772,850 (very fine). Table 2 shows the grid study findings for an A_R = 2.0 twisted tube at Re=800. In the end, the "fine" grid of 566,800 computationally nodes were selected as a compromise between duration, expense, and needed precision. Four alternative inlet region distances (5, 10, 15 and 20 time that tube hydraulically diameter) are explored to approach a fully developed flow state so at twisting region's intake. Fig.2 shows the influence of the intake obtaining on the average Nusselt number and apparent skin friction coefficient for a helical tube of various aspect ratios for Re=800, as well as an intake sector long of *15D_h* is used for the remainder of the computations.

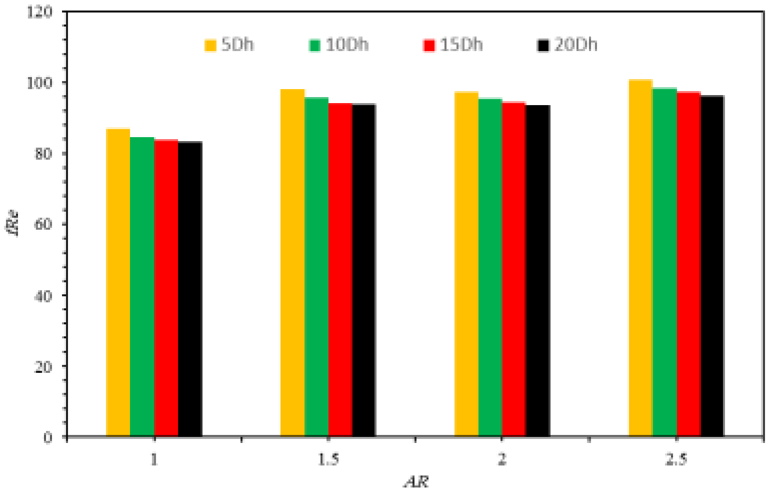


Fig. 2 TOTs with various aspect ratios and Re=800 have their apparent friction factor affected by the inlet zone length.

Code verification

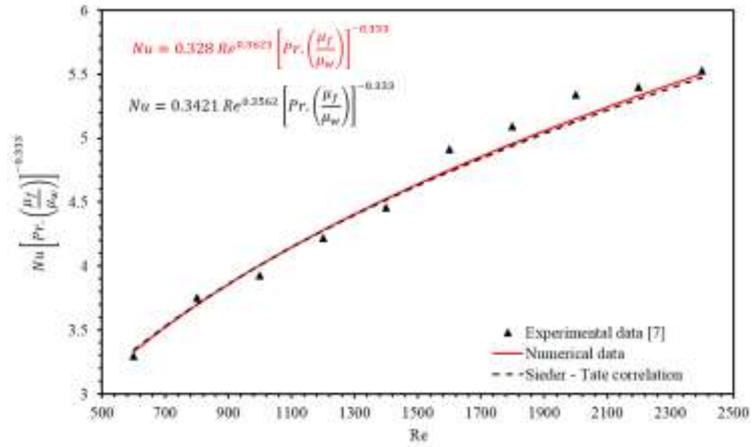
In a straight circular pipe, numerical simulations of laminar flow and heat transfer demonstrate the solver's precision and dependability. [27, 38] has extensive information regarding system testing. Nusselt number findings are seen in Fig.3a alongside predictions from Sieder-Tate relation (Eq. 24) and Yang et al. (Eq. 7) experimental data.

$$Nu = 1.86 Pr_f^{1/3} Re^{1/3} \left(\frac{D_h}{L}\right)^{1/3} \left(\frac{\mu_f}{\mu_w}\right)^{0.14} \qquad 20$$

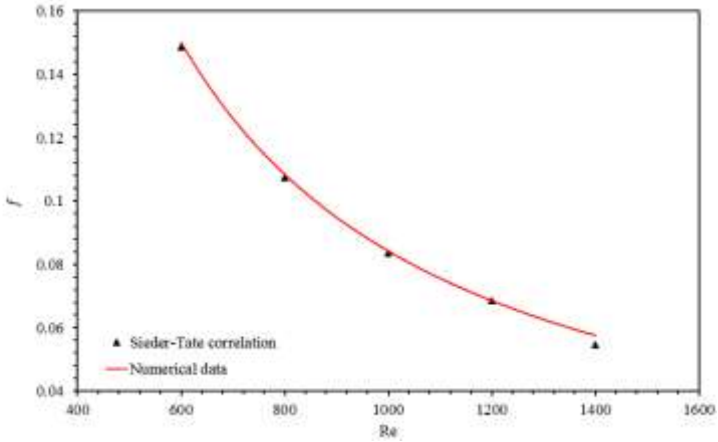
Fig.3b depicts a comparison between the frictional factor's mathematical findings and the Darcy friction formulas [23, 39]. (Eq. 25).

$$f = 64 / Re \qquad 21$$

With an average variation of less than 1 percent, the numerical findings accord well with the Sieder-Tate correlation, the Darcy friction coefficient, as well as the Yang et al. [7] experimental measurements.



(a) Numerical data of Nu compared with experimental data [7] and Sieder-Tate correlation



(b) Friction factor data compared with that for Darcy.

Fig.3 comparison between the present study and others in literature.

Characteristics of fluid flow

In this study, a range of Re from 500 to 1100 is simulated. Fig.4 shows non-dimensional velocity (U/Ui) and secondary flow vectors patterns with Re of 500, 800, and 1100 at z/Dh=30 of the pipe. Round pipes do not even have flow pattern, but twisting elliptical pipes have because the spiral surface provides normal force. According to the findings of Yang et alexperiments .'s [7], a larger AR leads to improved eddy currents hence more intense disturbance and successful combining. The auxiliary stream of an instance with just an AR of 1.5 differs significantly from secondary flows of instances had ARs of 2.0 to 2.5. Unlike with the earlier, whose secondary flow seems to be a helical along tube direction, the former's secondary flow was bidirectional and parallel to the main cross sectional axial direction. It's mostly due to differences in the pipes' geometrical features as compared to other instances. Axially spiral curving enforces an increased angular momentum in situations of greater AR, which results in a spiraling flow with a greater secondary flow even in the lower AR scenario without helical flowing. As a consequence, the located close fluid has a propensity to swirl, which increases a secondary flow. This behavior is seen when an induced vortices is present inside the tubes cross - sectional area. The closeness of the turning point generates a flow pattern with a greater velocity for oval tubes. Additionally, the cross-section circulating in twisting elliptical pipes be approximately to five times greater than that in tubing, and this is clearly correlated to an improvement in pipe A_R.

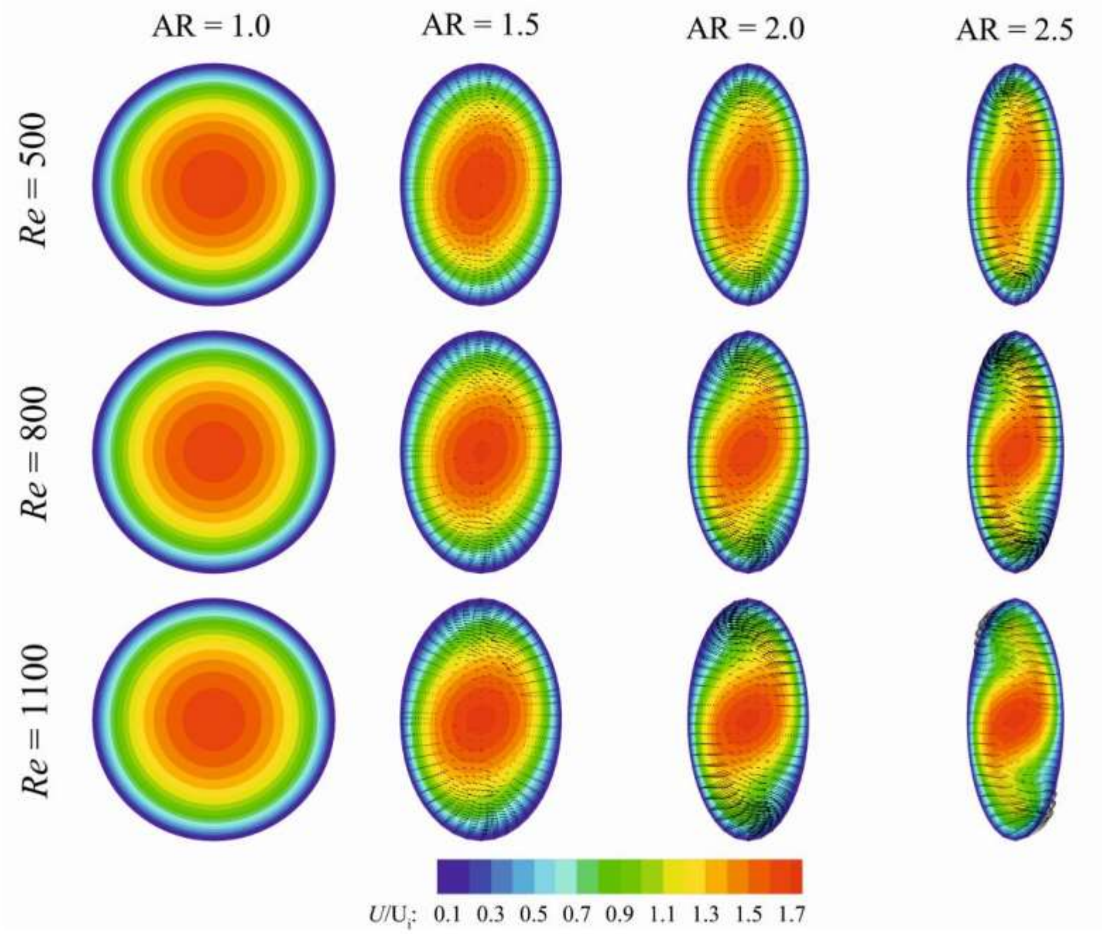


Fig. 4 Dimensionless velocity as well as secondary flow variables at various shapes of pipe contour.

Present study:

$$Nu = 0.328\, Re^{0.3623} \left[Pr.\left(\frac{\mu_f}{\mu_w}\right)\right]^{-0.333} \quad 22$$

Sieder -Tate correlation

$$Nu = 0.3421\, Re^{0.3562} \left[Pr.\left(\frac{\mu_f}{\mu_w}\right)\right]^{-0.333} \quad 23$$

Fig. 5 illustrate the effect of *Re vs.* that *fRe* combination of skin friction coefficient and *Re*. Spiral pipes have a greater pressure loss due to the additional contact area of a pipe wall and the accelerated secondary flow. Higher pressure difference is seen for pipes had twisting walls relative to a circular pipe throughout the full range for *Re* examined herein. Twisted oval tubes with varied A_R s have a higher *fRe* than a smooth circular tube with Re equal to 500 because of this. The *fRe* product's price also rises somewhat as compared to the 1100 *Re* smooth circular tube (about 17, 18 and 26 percent for A_R of 1.5, 2.0 and 2.5, respectively). The example with an A_R of 1.5 has a different slope, and this is thought to be due to the case's unique secondary flow pattern already being present (See Fig.4). Viscous effects dominate the centrifugal force at low Re and may be reduced by reducing the tube's cross-sectional A_R, while increasing the tube's cross sectional AR increases the core swirl because of the greater centrifugal force. As a result, reduced fluid resistance may be seen at same Reynolds number when greater fluid temperatures are observed inside pipes of larger A_R.

In contrast, raising higher Reynolds number lowers the bulk flow temperature, resulting in greater fluid viscosity. When Reynolds numbers are low, the combined effects of rising viscosity as well as secondary flow lead to a higher pressure loss for A_R =1.5 pipes; however, as Reynolds numbers increase as well as the consequences of friction force to viscous dissipation are overcome, a larger pressure penalty has been observed due to the greater eddy currents in pipes with relatively high A_R.

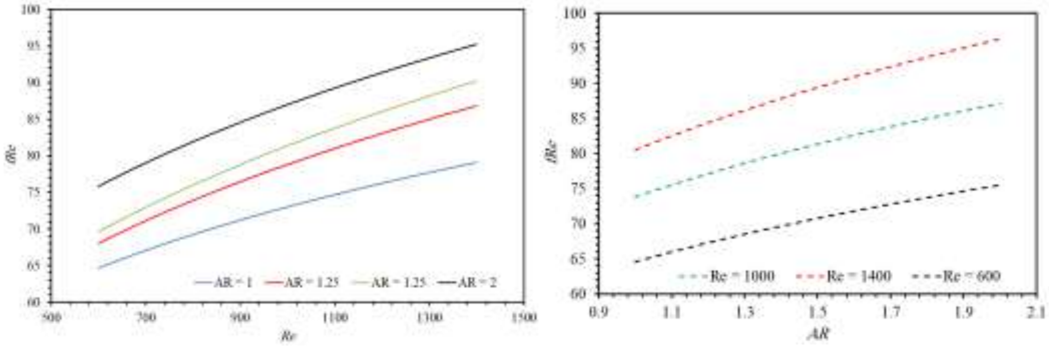


Fig. 5 effect of the A_R on the fRe with respect to plain tube.

Thermal performance

This study explores the total heat properties of twisting elliptical tubing using Nusselt number data (Nu). The tangled produced swirls create flow pattern, which alters the thickness of the thermal shear layer, increases turbulence inside the tubes, and also improves mixing among hot and cold streams near the tube wall and the center of the tube Because of the increased combining, flow field is much more powerful at higher A_R s, resulting in more intense heat transfer for twisting ellipse tubes, as previously stated (Fig.4).

Fig.6 depicts the relationship between Nu_m and Re, with distorted elliptical tubes having a much higher Nu_m as opposed to circular ones as AR increases. However, when Re and AR rise, the heat transfer rate becomes more apparent. The increase of Nu_m or, like a result, better heat transfer effectiveness may be achieved by raising Re. It also enhances mixing because it keeps that high thermal gradients along the tube - side constant, which in turn improves heat transmission.

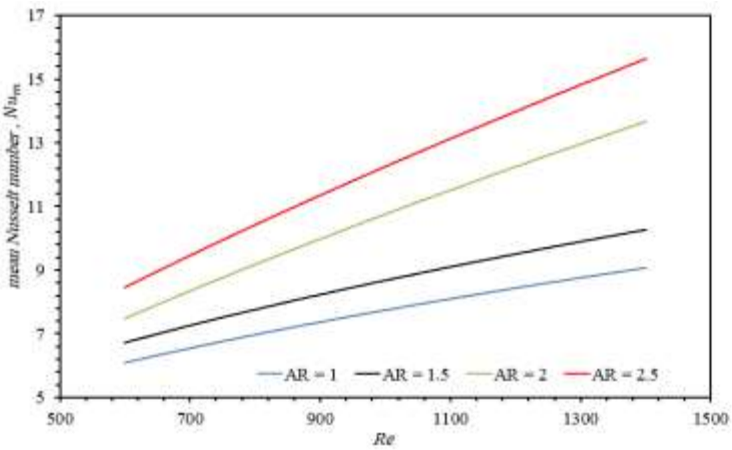


Fig. 6 variation of Nu_m according to aspect ratio, A_R.

All designs for Re = 800 have Nu fluctuations along the primary flow direction. Higher Nu values are found for twisted elliptical tubes than for circular ones, with cross-section A_R values peaking at the greatest point due to stronger eddy currents and greater temperature gradients. In addition, Nu declines independently of a cross-section A_R in the primary flow direction, while it is linear and variable for circular tubes as well as twisting oval tube. While gravitational force could cause that, secondary flow imposed harsh is along tube's primary flow direction, as well as Vortices interacting with its wall, may also cause this phenomena. An effectiveness assessment criteria developed by Webb [25] is used to evaluate the improvement of heat transfer so at cost of an improvement in operating pressure in order to examine the thermal properties performance for tube had various designs. A higher number means the tube performs better as a whole. As seen in Fig. 7, the round tube has a varying vs Re, whereas the twisted elliptical tube has three distinct configurations. Over the full range of Re, it is found that the performance factor is greater than the circular tube. Twisted tubes' overall performance improves substantially when Re rises, as well.

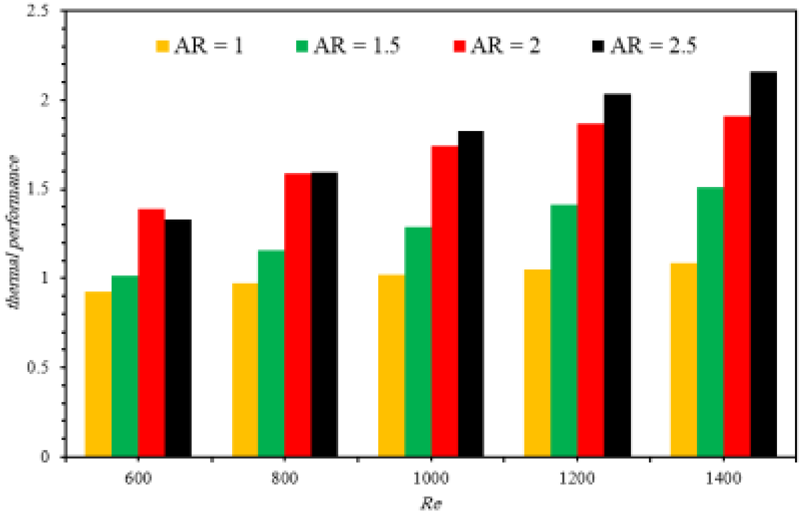


Fig. 7 effect the aspect ratio on thermal performance

Conclusions

Spiral elliptical pipes are subjected to something like a three-dimensional mathematical solution of any flow configuration as well as heat transfer effectiveness. Wrapped elliptical pipes' total performance in comparison with that of a circular pipe to see how geometric features and Re variation affected their performance. Entropy production and field's synergistic principles were also used to investigate higher heat transfer mechanisms, as well as the following major results and conclusions have been reached. The increased fluid mixing produced by the produced vortices within twisted elliptical tube results in a greater rate of heat transfer. Helical auxiliary flow occurs with high A_R tubes, whereas bidirectional and orthogonal to a pass main axis prevail for low A_R tubes. Spiral tubes with improved A_R -related recirculation zone transport hotter fluid from the walls to the core area and conversely, increasing temperature disparities at the walls and improving heat transfer within the tubes. A better synergy across different velocities as well as temperature difference leads to enhanced heat transfer, as shown by the findings. Warped elliptical pipes have a greater drop in pressure than tubing because of the increased surface area as well as intensified flow pattern, which are both linked to that same A_R of the tubular cross - sectional area.

When comparing twisted elliptical tubes versus circular cylinder, the former have superior thermal performance, but the latter have higher heat transfer effectiveness when Re or A_R are increased. Fluids having greater thermophysical properties are anticipated to perform better during heat transfer. A change in temperature is being described. The increase in flow rate is dominated by the increase in volume concentration when Re is increased. Wrapped elliptical tube could be a viable alternative to applications area when pumping energy is not a critical design requirement. According the fields synergistic concept as well as exergy assessment, the TOT designs examined in this research outperform the cylindrical tube in terms of effectiveness. The much more successful design seems to be a twisting pipe with an A_R around 2.5, following by convoluted tubes with A_R s of 2.0 and finally 1.5.

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