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Numerical Analysis of a Plain and Finned Circular Tubes With Twisted Tape Inserts

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ABSTRACT

Computational fluid dynamic (CFD) studies were carried out by using the ANSYS FLUENT 13.0 to find the effects of twisted tape insert on heat transfer, friction loss and thermal performance factor characteristics in a circular tube at constant wall temperature. Simulation was performed with Reynolds number in a range from 800 to 10,000 using water as a working fluid. Four turbulent models are examined such as a standard k-ϵ, RNG k-ϵ, standard k-ω and SST k-ω and those compared with standard twisted tape correlations developed by Manglik and Bergles.

Plain tube with four different full width twisted tape inserts (FWTT) of twist ratios (y = 2, 3, 4 and 5) were examined, based on constant flow rate. The heat transfer coefficient were found to be 2.67 to 3.35, 2.43 to 2.19, 2.10 to 2.64,and 1.87 to 2.35 times respectively in laminar region, and 1.92 to 1.56, 1.74 to 1.41, 1.65 to 1.34, and 1.6 to 1.3 times of that in the plain tube in the turbulent region. For the same twist ratio (H/w) three different reduced width twisted tapes (RWTT) (of width 12, 14 and 16 mm), were examined in the finned tube. The simulation results revealed that both heat transfer rate and friction factor in the finned tube equipped with twisted tapes were significantly higher than those in the plain tube. Over the range of Reynolds number investigated, based on overall thermal performance factor (η) it is revealed that the plain tube with FWTT (η = 1.12-1.51 in laminar regime & 0.91 – 1.08 in turbulent regime) are suitable in laminar flow region and finned tube with RWTT (η = 0.58-0.91 in laminar regime & 0.83 – 1.31 in turbulent regime) are suitable for turbulent regime.

INTRODUCTION

Effective utilization, conservation and recovery of heat are critical engineering problems faced by the process industry. The economic design and operation of process plants are often governed by the effective usage of heat. A majority of heat exchangers used in thermal power plants, chemical processing plants, air conditioning equipment, and refrigerators, petrochemical, biomedical and food processing plants serve to heat and cool different types of fluids. Both the mass and overall dimensions of heat exchangers employed are continuously increasing with the unit power and the volume of production [1]. This involves huge investments annually for both operation and capital costs. Hence it is an urgent problem to reduce the overall dimension characteristics of heat exchangers. The need to optimize and conserve these expenditures has promoted the development of efficient heat exchangers. Different techniques are employed to enhance the heat transfer rates, which are generally referred to as heat transfer enhancement, augmentation or intensification technique [2-4].

Heat transfer enhancement

Heat transfer enhancement is one of the key issues of saving energies and compact designs for mechanical and chemical devices and plants. In the recent years, considerable emphasis has been placed on the development of various augmented heat transfer surfaces and devices. Energy and materials saving considerations,

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space considerations as well as economic incentives have led to the increased efforts aimed at producing more efficient heat exchange equipment through the augmentation of heat transfer.

The heat exchanger industry has been striving for enhanced heat transfer coefficient and reduced pumping power in order to improve the thermo hydraulic efficiency of heat exchangers. A good heat exchanger design should have an efficient thermodynamic performance, i.e. minimum generation of entropy or minimum destruction of energy in a system incorporating a heat exchanger. It is almost impossible to stop energy loss completely, but it can be minimized through an efficient design. The major challenge in designing a heat exchanger is to make the equipment compact and to achieve a high heat transfer rate using minimum pumping power [6].

Heat transfer augmentation techniques

Heat transfer augmentation techniques are generally classified into three categories namely: Active techniques, Passive techniques and Compound techniques [8].

Active techniques

As the name indicates, these techniques involve some external power input for enhancement of heat transfer. This has not shown much potential due to complexity in design.

They are classified as below:

- Mechanical aids
- Surface vibrations
- Fluid vibration
- Electrostatic fields (DC or AC)
- Jet impingement

Passive techniques

Passive techniques do not require any direct input of external power. They generally use geometrical or surface modifications to the flow channel by incorporating inserts or additional devices. Except for the case of extended surfaces, they promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour.

Passive techniques are classified as below:

- Treated surfaces
- Rough surfaces
- Extended surfaces
- Displaced enhancement devices.
- Swirl flow devices
- Coiled tubes
- Surface-tension devices
- Additives for gases
- Additives for liquids.

Compound techniques

Two or more of the above techniques may be utilized simultaneously to produce an enhancement that is larger than the individual technique applied separately.

Some examples of compound techniques are given below:

- Rough tube wall with twisted tape
- Rough cylinder with acoustic vibrations
- Internally finned tube with twisted tape insert
- Finned tubes in fluidized beds
- Externally finned tubes subjected to vibrations
- Gas-solid suspension with an electrical field
- Fluidized bed with pulsations of air

Performance evaluation criteria

In most of the practical applications of enhancement techniques, the following performance objectives, along with a set of operating constraints [10] and conditions, are usually considered for evaluating the thermo hydraulic performance of a heat exchanger:

- Increase in the heat duty of an existing heat exchanger without altering the pumping power or flow rate requirements.
- Reduction in the approach temperature difference between the two heat exchanging fluid streams for a specified heat load and size of exchanger.
- Reduction in the size or heat transfer surface area requirements for a specified heat duty and pressure drop.
- Reduction in the process stream pumping power requirements for a given heat load and exchanger surface area.

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Table 1.1 Performance Evaluation Criteria

Case	Geometry	m	P	Q	ΔT_i	Objective
FG-1a	N, L, d	x			x	Q↑
FG-1b	N, L, d	x		x		ΔTi↓
FG-2a	N, L, d		x		x	Q↑
FG-2b	N, L, d		x	x		ΔTi↓
FG-3	N, L, d			x	x	P↓
FN-1	N, d		x	x	x	L↓
FN-2	N, d	x		x	x	L↓
FN-3	N, d	x		x	x	P↓
VG-1	-------	x	x	x	x	NL) ↓
VG-2a	N, L,	x	x		x	Q↑
VG-2b	N, L,	x	x	x		Δ Ti↓
VG-3	N, L,	x		x	x	P↓

Table 1.2 Performance Evaluation Criteria of Bergles et al.

		Criterion number							
		R1	R2	R3	R4	R5	R6	R7	R8
Fixed	Basic Geometry	x	x	x	x				
	Flow Rate	x						x	x
	Pressure Drop		x				x		x
	Pumping Power			x		x			
	Heat Duty				x	x	x	x	x
Objective	Increase Heat Transfer	x	x	x					
	Reduce pumping power				x				
	Reduce Exchange Size					x	x	x	x

Thermal Performance factor (η):

This is defined by equation 1.1 as follows and issimilar to enhancement of heat transfer at constant pumping power is criteria FG-2a

$$\eta = \frac{(Nu_a/Nu_0)}{(f_a/f_0)^{(1/3)}} \qquad (1.1)$$

Where Nu_a, f_a, Nu_0 and f_0 are the Nusselt numbers and friction factors for a duct configuration with and without inserts respectively.

It may be noted that FG-1a & FG-2a are similar to R1& R3 respectively. Evaluation criteria R_1 (i.e. Nu_a/Nu_0) & R_3 (i.e. η) have been used for present numerical simulation work to determine heat transfer enhancement for different types of twisted tapes [12].

Applications of heat transfer enhancement

The petrochemical and chemical industries are under economic pressure to increase the energy efficiency of their processing plants to compete in today's global market. Hence, these industries must invest in innovative thermal technologies that would significantly reduce unit energy consumption in order to reduce overall cost. In recent years, heat transfer enhancement technology has been widely applied to heat exchanger applications in boiling and refrigeration process industries. Most significantly, the uses of enhancement extend well beyond surface reduction i.e., they can also be used for capital cost reduction, the improvement of exchanger operability, the mitigation of fouling, the improvement of condenser design and the improvement of flow distribution within heat exchangers [3].

Important applications of heat transfer enhancement are listed below:

- Heating, Ventilating, Refrigeration and air conditioning
- Automotive Industries
- Power sector
- Process Industries
- Industrial Heat Recovery
- Aerospace and others.

Swirl flow devices

This is coming under category of passive heat transfer enhancement technique [5]. These include a number of geometric arrangements or tube inserts for forced flow that create rotating and/or secondary flow i.e. inlet vortex generators, twisted-tape inserts and axial-core inserts with a screw-type winding etc.

Twisted tape inserts

To enhance the heat transfer rate, some kind of insert is placed in the flow passages and they also reduce the hydraulic diameter of the flow passages. Heat transfer enhancement in a tube flow is due to flow blockage, partitioning of the flow and secondary flow. Flow blockages increase the pressure drop and leads to viscous effects, because of a reduced free flow area. The

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selection of the twisted tape depends on performance and cost. The performance comparison for different tube inserts is a useful complement to the retrofit design of heat exchangers. The development of high performance thermal systems has stimulated interest in methods to improve heat transfer [7].

Geometry of the twisted tape

A schematic diagram of a twisted tape insert inside a tube is shown in Fig.1.2. The enhancement is defined geometrically in terms of thickness of the tape δ and its twist ratio. The twist ratio (y) is defined as the axial length (H) for a 1800 turn of the tape divided by the internal diameter (d) of the tube. This is the most common definition used in research literature and that used here.

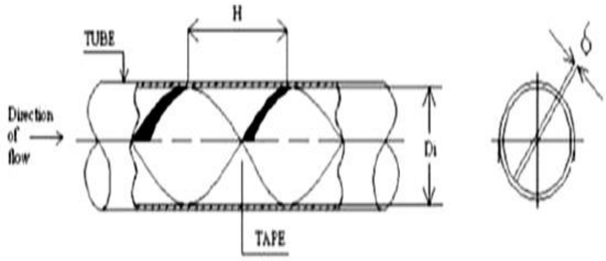

Fig.1.2.Diagram of a twisted tape insert inside a tube

COMPUTATIONAL FLUID DYNAMICS MODEL EQUATIONS

Computational Fluid Dynamics (CFD) [9] is the use of computer-based simulation to analyse systems involving fluid flow, heat transfer and associated phenomena such as chemical reaction. A numerical model is first constructed using a set of mathematical equations that describe the flow. These equations are then solved using a computer programme in order to obtain the flow variables throughout the flow domain. Since the arrival of the digital computer, CFD has received extensive attention and has been widely used to study various aspects of fluid dynamics. The development and application of CFD have undergone considerable growth, and as a result it has become a powerful tool in the design and analysis of engineering and other processes.

CFD analysis procedure

CFD analysis requires the following information:

- A grid of points at which to store the variables calculated by CFD
- Boundary conditions required for defining the conditions at the boundaries of the flow domain and which enable the boundary values of all variables to be calculated
- Fluid properties such as viscosity, thermal conductivity and density etc.
- Flow models which define the various aspects of the flow, such as turbulence, mass and heat transfer.
- Initial conditions used to provide an initial guess of the solution variables in a steady state simulation.
- Solver control parameters required to control the behaviour of the numerical solution process.

CFD methodology

The mathematical modelling of a flow problem is achieved through three steps:

- Developing the governing equations which describe the flow
- Discretisation of the governing equations
- Solving the resulting numerical equations.

Equations describing fluids in motion

The mathematical equations used to describe the flow of fluids are the continuity and momentum equations, which describe the conservation of mass and momentum. The momentum equations are also known as the Navier-Stokes equations [11]. For flows involving heat transfer, another set of equations is required to describe the conservation of energy. The continuity equation is derived by applying the principle of mass conservation to a small differential volume of the fluid. In Cartesian coordinates, three equations of the following form are obtained:

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho \vec{v}) = S_m \tag{3.1}$$

Equation 3.1 is the general form of the mass conservation and valid for incompressible as well as

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compressible flows. The source S_m is the mass added to the dispersed second phase.

The momentum equations are derived by applying Newton's second law of motion to differential volume of the fluid. According to Newton's second law, the rate of change of momentum over a differential volume of fluid is equal to the sum of all external forces acting on this volume of fluid. The resulting momentum equations in Cartesian coordinates take the general form

$$\frac{\partial}{\partial t}(\rho\vec{v}) + \nabla.(\rho\vec{u}\vec{v}) = -\nabla P + \nabla\bar{\bar{\tau}} + \rho\vec{g} + \vec{F} \qquad (3.2)$$

Where P is the static pressure , $=$ is the stress tensor (described below), and $\vec{}$and $\vec{}$are the gravitational body force and external body force.

$$\bar{\bar{\tau}} = \mu\left[(\nabla\vec{v} + \nabla\vec{v}^T) - \frac{2}{3}\nabla.\vec{v}I\right] \qquad (3.3)$$

Where μ is the molecular viscosity, I is the unit tensor, and the second term on the right hand side is the effect of volume dilation.

The Energy equation is derived from the first law of thermodynamics which states that the rate of change of energy of fluid particle is equal to the rate of heat addition to the fluid particle plus the rate of work done on the particle. The resulting energy equation in general form

$$\frac{\partial}{\partial t}(\rho E) + \nabla.(\vec{v}(\rho E + P)) = \nabla.(k_{eff}\nabla T - \Sigma_j h_j\vec{J_j} + (\bar{\bar{\tau}}_{eff}.\vec{v})) + S_h \qquad (3.4)$$

Where is the effective conductivity (k + ki, where ki is the turbulent thermal conductivity, defined according to the turbulence model being used), and $\vec{}\vec{}\vec{}$ is the diffusion flux of species j. the first three terms on the right- hand side of equation (3.4) represent energy transfer due to conduction, species diffusion, and viscous dissipation.Includes the heat of chemical reaction, and any other volumetric heat sources

In equation (3.4), Energy E per unit mass is defined as

$$E = h - \frac{P}{\rho} + \frac{u^2}{2} \qquad (3.5)$$

Turbulence modelling

Turbulent flows are characterized by fluctuating velocity fields. These fluctuations mix transported quantities such as momentum, energy, and species concentration, and cause the transported quantities to fluctuate as well. It is an unfortunate fact that no single turbulence model is universally accepted as being superior for all classes of problems [8]. The choice of turbulence model will depend on considerations such as the physics encompassed in the flow, the established practice for a specific class of problem, the level of accuracy required, the available computational resources, and the amount of time available for the simulation.

Boussinesq Approach vs. Reynolds Stress Transport Models

$$\frac{\partial\rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0 \qquad (3.6)$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_l}{\partial x_l}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho\overline{u_i'u_j'}\right) \qquad (3.7)$$

Equations 3.6 and 3.7 is called Reynolds – averaged Navier -stokes (RANS) equations. They have the same general form as the instantaneous Navier-Stokes equation, with velocities and other solution variables representing time averaged values. Additional terms now appear that represent the effects of turbulence. These Reynolds stresses, must be modelled in order to close equation 3.7.

The Reynolds-averaged approach to turbulence modelling requires that the Reynolds stresses in Equation (3.7) is appropriately modelled. A common method employs the Boussinesq hypothesis to relate the Reynolds stresses to the mean velocity gradients:

$$\left(-\rho\overline{u_i'u_j'}\right) = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\left(\rho k + \mu_t\frac{\partial u_k}{\partial x_k}\right)\delta_{ij} \qquad (3.8)$$

The Boussinesq hypothesis is used in the Spalart-Allmaras model, the k-ϵ models, and the k-ω models. The advantage of this approach is the relatively low computational cost associated with the computation of the turbulent viscosity, μ_t. In the case of the Spalart-Allmaras model, only one additional transport equation (representing turbulent viscosity) is solved. In the case of the k-ϵ andk-ω models, two additional transport equations (for the turbulence kinetic energy, k, and either the turbulence dissipation rate, ϵ, or the specific dissipation rate, ω are solved, and μ_t is computed as a function of k and ϵ or k and ω.

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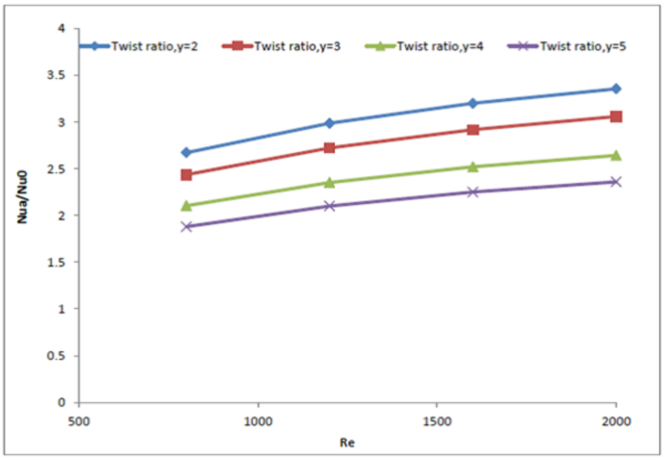
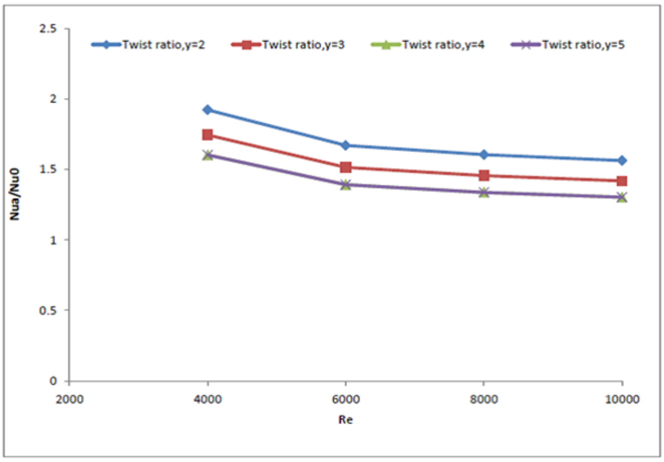
Heat transfer results

Effect of the twist ratios on the heat transfer rate is numerically studied; the values are given in Table.A.5. The results for the tube fitted with all twisted tapes are also compared with those for a plain tube under similar operating conditions. The heat transfer rate is considered in terms of Nusselt numbers Fig, 5.12 and 5.13 shows the Nusselt number ratio (Nua/Nu0) with Reynolds number of the tube equipped with four different twist ratios (y = 2, 3, 4 and 5). The Nusselt number in the tube with the twist ratios y= 2, 3, 4 and 5, are around 2.67 to 3.35, 2.43 to 2.19, 2.10 to 2.64, and 1.87 to 2.35 times of that in the plain tube in laminar region. Similarly in turbulent region 1.92 to 1.56, 1.74 to 1.41, 1.65 to 1.34, and 1.6 to 1.3 times of that in the plain tube [13].



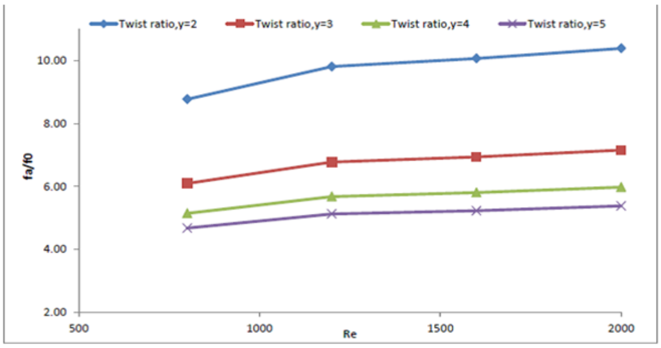
Variation of Nua/Nu0 with Reynolds number in laminar regime for FWTT



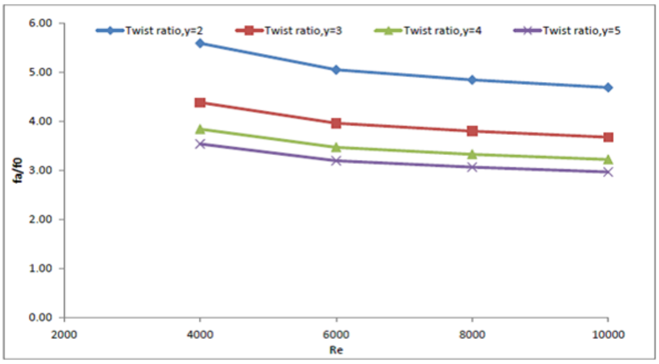
Variation of Nua/Nu0 with Reynolds number in turbulent regime for FWTT

Friction factor results

Effect of the twist ratios on the friction factor is numerically studied, the values are given in Table.A.6.The friction factor characteristic in a plain tube fitted with twisted tape at various twist ratios is displayed in Fig. 5.14 and Fig. 5.15. Friction factor decreases with increasing twisted ratio. The friction factors in the tube with the twist ratios y= 2, 3, 4 and 5, are around 8.77 to 10.39, 6.09 to 7.15, 5.14 to 5.97, and 4.67 to 5.37 times of that in the plain tube in laminar region. Similarly in turbulent region 5.59 to 4.69, 4.39 to 3.68, 3.84 to 3.22, and 3.54 to 2.97 times of that in the plain tube.



Variation of fa/f0 with Reynolds number in laminar regime for FWTT



Variation of fa/f0 with Reynolds number in turbulent regime for FWTT

Thermal Performance factor (η):

The Thermal Performance factor for different case are compared and shown in Fig.6.20 in laminar region Fig.6.21 is turbulent region and the related values are given in Table. A.10. In laminar region it is shows that the maximum value is 0.91 it means that there is no

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effect in this region. In turbulent region the thermal performance factor increase with Reynolds number increases corresponding tape widths of the, w = 12 mm, 14 mm, 16 mm.

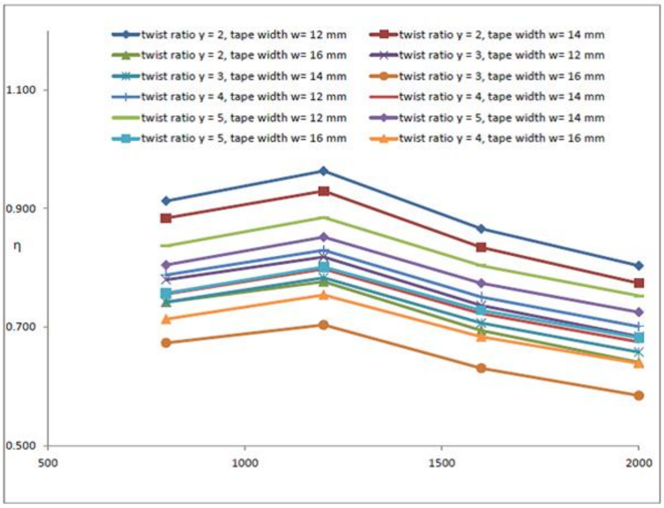

Fig. Thermal performance factor for finned tube with twisted tape inserts in laminar regime, (RWTT)

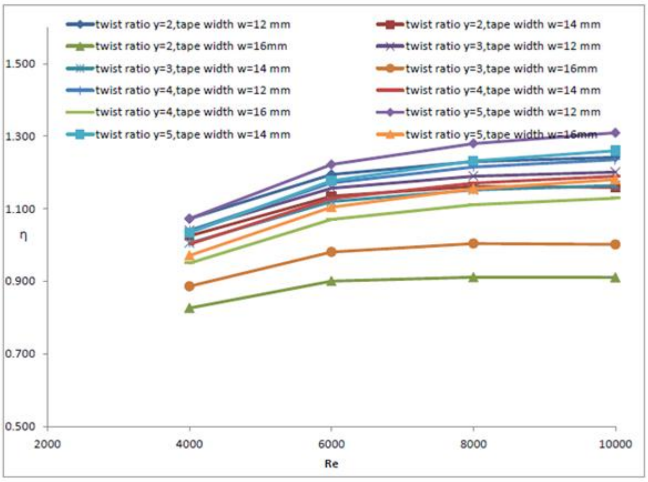

Fig. Thermal performance factor for finned tube with twisted tape inserts in turbulent regime, (RWTT)

CONCLUSION

The numerical analysis of heat and fluid-flows through a constant wall temperature circular plain tube and finned tube fitted with twisted tape is carried out, with the aim to investigate the effect of tape twist ratio (y=H/d) and tape width on heat transfer (Nu), friction factor (f) and thermal performance (η) behaviours in the laminar and turbulent flow regime.

The main findings can be drawn as follows.

1. Turbulent modelling was done with standard k–ε turbulence model, the Renormalized Group (RNG) k–ε turbulence model, the standard k–ω turbulence model, and the Shear Stress Transport (SST) k–ω turbulence model.

2. The CFD results of Nusselt number and Friction factor are compared with those obtained from Manglik and Bergles equations. It is clearly seen that the predicted Nusselt numbers obtained from the SST k–ω turbulence models is in better agreement compared to those from other models. The SST k–ω turbulence model is valid within ±20.2% error limit with measurements for Nusselt number and ±26.4% for friction factor. Therefore this model used for further studies.

3. Plain tube with full width twisted tape inserts, (FWTT) of twist ratio, (y=2, 3, 4, 5) results show that Nusselt number and friction factor values were found to decrease with increasing in twist ratio. Twisted tape inserts for twist ratio (y=2) can enhance heat transfer rates up to 3.5 times at Reynolds number 2000 and increase in friction factors nearly 9 times in comparison with those of the plain tube. Thermal performance factor (η) was found to increase with increase in Reynolds number in the laminar region and decrease in the turbulent region. The maximum value of the thermal performance factor was found to be 1.6 for Twisted tape (y=3) in plain tube at a Reynolds number of 2000

4. Finned tube with reduced width twisted tape,(RWTT) simulation results are shown that the twisted tape inserts of twist ratio (y=2) can enhance heat transfer rates up to 3.76 times at Reynolds number 2000 with tape width of 16 mm and the corresponding increase in friction factors nearly 14 times in comparison with those of the plain tube. Thermal performance factor(η) was found to initially increase with increase in Reynolds number then decrease in the laminar region and increase with increase Reynolds number in the turbulent region. The maximum value of the thermal performance factor was found to be 1.31 for Twisted tape (y=5) and tape width of 12 mm in a Reynolds number of 10000.

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REFERENCES

[1]. Bergles.A.E. (2001), the implications and challenges of enhanced heat transfer for the chemical process industries‖ Trans IChemE Vol.79 pp.437-444

[2]. Syam sundar, L. sharma, K.V. (2011), Laminar convective heat transfer and friction factor of Al2O3 nanofluid in circular tube fitted with twisted tape inserts, International Journal of Automotive and Mechanical Engineering, Volume 3, pp. 265-278

[3]. Adrian, B. Allan, D. K. (2003), Heat transfer enhancement. In Heat Transfer Handbook, Chapter 14, pp.1033, -1101, Wiley-interscience.

[4]. Bergles, A.E. Blumenkrantz, A.R. Taborek, J. (1974), Performance evaluation criteria for enhanced heat transfer surfaces. Proc. of 5th Int. Heat Conf., Tokyo, Vol. 2, pp. 239-243

[5]. Lokanath, M. S. (1997), Performance evaluation of full length and half-length twisted tape inserts on laminar flow heat transfer in tubes. In Proceedings of 3rd ISHMT–ASME Heat and Mass Transfer Conference, India, pp. 319–324

[6]. Ray, S. Date, A.W. (2001), Laminar flow and heat transfer through square duct with twisted tape insert, International journal of Heat and Fluid Flow, Vol. 22, pp. 460-472

[7]. Manglik, R. K. and Bergles, A. E. (1993), Heat transfer and pressure drop correlations for twisted-tape inserts in isothermal tubes: Part I: laminar flows. Trans. ASME, J. Heat Transfer, Vol.115, pp.881–889.

[8]. Saha, S. K. and Dutta, A. (2001), Thermo-hydraulic study of laminar swirl flow through a circular tube fitted with twisted tapes. Trans. ASME, J. Heat Transfer, Vol.123, pp. 417–421.

[9]. Tariq, A. Kant, K. and Panigrahi, P. K. (2000), Heat transfer enhancement using an internally threaded tube. In Proceedings of 4th ISHMT–ASME Heat and Mass Transfer Conference, India, pp. 277–281.

[10]. Saha, S. K. and Bhunia, K. (2000), Heat transfer and pressure drop characteristics of varying pitch twisted-tape-generated laminar smooth swirl flow. In Proceedings of 4th ISHMT– ASME Heat and Mass Transfer Conference, India, pp. 423–428.

[11]. Ray, S. and Date, A. W.(2003), Friction and heat transfer characteristics of flow through square duct with twisted tape insert, Int. J. Heat and Mass Transfer, Vol. 46, pp.889–902.

[12]. Lokanath, M. S. and Misal, R. D. (2002), An experimental study on the performance of plate heat exchanger and an augmented shell and tube heat exchanger for different types of fluids for marine applications. In Proceedings of 5th ISHMT– ASME Heat and Mass Transfer Conference, India, pp. 863–868

[13]. Saha, S. K. Dutta, A. and Dhal, S. K. (2001), Friction and heat transfer characteristics of laminar swirl flow through a circular tube fitted with regularly spaced twisted-tape elements. Int. J. Heat and Mass Transfer, Vol. 44, pp. 4211–4223.