

Effect of Diameter Ratio on Heat Transfer Enhancement in a Circular Tube with Twisted–Tape

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ABSTRACT— *The influences of diameter ratios ($DR = W/D$), $DR = 0.2–0.8$, with single twisted–ratio (y/W) of 2.0 on heat transfer, pressure loss and thermal performance in a circular tube are investigated numerically in 3D. The finite volume method (FVM) and the SIMPLE algorithm are applied in this study. The fluid flow and heat transfer characteristics are presented for Reynolds number (Re) ranging from 100 to 2000. As the numerical results, the twisted–tape can generate the swirling flow that helps to increase heat transfer rate, but also increase in pressure loss. Additionally, the rising DR value results in the increase of Nusselt number and friction factor. The computational results reveal that the optimum thermal enhancement factor of using the twisted–tape insert is about 2.4 at $DR = 0.2$ and $Re = 2000$.*

Keywords— heat transfer, laminar flow, periodic flow, thermal performance, twisted–tape

[1] INTRODUCTION

Many types of twisted–tape such as single twisted–tape, double twisted–tapes, perforate twisted–tape, etc., have been used for enhancing the heat transfer rate and thermal performance in the heat exchanger system. The twisted–tape can change the flow structure in the heating or cooling tube and also increase the turbulence of the fluid flow. Due to the uncomplicated for the installing twisted–tape to the system, the twisted–tape is extensively used in many industries.

The parameters of the twisted–tape; twisted–ratio, twisted–length, configuration, etc., are investigated by many researchers. Ghadirijafarbeigloo et al. [1] studied the thermal performance enhancement in a solar heating system with the louvered twisted–tape by numerical method. They concluded that the use of the louvered twisted–tape leads to the significant increase on both heat transfer and pressure loss in comparison with general type twisted–tape and plain tube. Pal and Saha [2] experimental investigated the use of combined turbulators; spiral ribs and twisted–tapes, on fluid flow and heat transfer in a circular tube. They reported that the combined turbulators perform better thermal performance than the alone turbulators for the laminar flow regime. Azmi et al. [3] examined the heat transfer enhancement of SiO₂/water and TiO₂/water nanofluid in a tube with twisted–tape. They claimed that the rising twisted–ratio results in the decrease in heat transfer rate. Eiamsa–ard et al. [4] experimental investigated the heat transfer, pressure loss and thermal performance in a tube heat exchanger with regularly–spaced twisted tapes at turbulent regime. They showed that the normal twisted–tape performs higher heat transfer rate and thermal performance than the regularly–spaced twisted tapes due to a higher turbulence degree of the fluid flow. Naik et al. [5] compared the heat transfer performance between twisted–tape and wire–coil in a tube heat exchanger at turbulent regime. They concluded that the wire coil gives a higher heat transfer coefficient, pressure loss and thermal performance than the twisted–tape for all cases. They also reported that the thermal performance of the twisted–tape and wire–coil are around 1.24 and 1.36 compared against water value, respectively. Bhuiya et al. [6] presented the performance evaluations of double counter twisted–tape in a heat exchanger. They found that the heat transfer rate and friction loss are around 60 – 240% and 91 – 286%, respectively, in comparison with smooth tube, while the optimum thermal performance are about 1.34. Eiamsa–ard and Wongcharee [7] investigated the heat transfer characteristics in a micro–fin tube with double twisted–tapes. They claimed that the thermal performance enhancement is around 56.4%. Nanan et al. [8] experimental investigated the influences of perforated helical twisted–tapes on heat transfer, pressure loss and thermal performance in a tube. They stated that the maximum thermal performance factor of 1.28 is obtained by using the perforated helical twisted–tapes at the Reynolds number of 6000. Bhuiya et al. [9] presented the effects of triple twisted–tape in the heat exchanger on thermal enhancement factor. They

pointed out that the heat transfer and friction loss are higher than the smooth tube around 3.85 and 4.2 times, respectively.

As above literature reviews, the numerical investigation on the effect of the diameter ratio (W/D) for the twisted-tape has not been reported. Therefore, the three-dimensional numerical investigations for laminar forced convection, heat transfer and thermal enhancement in a circular tube with different diameter ratios for the twisted-tape are presented in the current work. The reductions of diameter ratio may help to decrease in the pressure loss of the heating system that leads to the increase in thermal performance.

[2] PRINCIPLES AND METHODS

2.1 Computational design

Referred from Promvonge et al. [10], the configurations of the twisted-tape in a circular tube and the computational domain are shown in Figure 1. The periodic condition (Patankar et al. [11]) on both flow structure and heat transfer characteristics apply to the current computational domain. The air enters the tube at an inlet temperature, T_{in} where D is the tube diameter set to 0.05 m, W is the twisted-tape width/diameter and y is the twisted-length. The effects of the diameter ratios ($DR = W/D$), $DR = 0.2 - 0.8$, at a single twisted-ratio, $y/W = 2.0$ are investigated numerically.

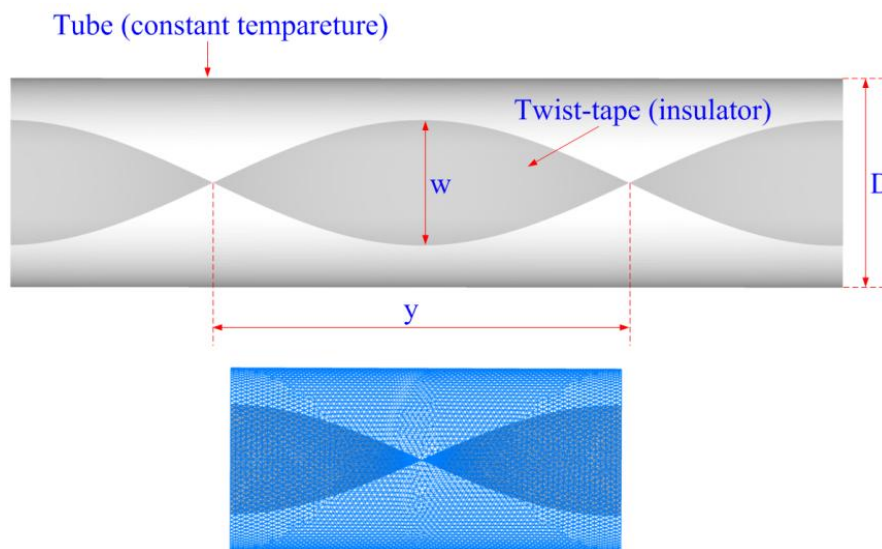


Figure 1: The circular tube equipped with the twisted-tape and the computational domain.

2.2 Computation methods

The computations are based on a finite volume method with the SIMPLE algorithm for handling the pressure-velocity coupling and using the QUICK scheme for the convection terms. The governing equations are discretized by the power law scheme, coupling of velocity and pressure with the SIMPLE algorithm and solved using the finite volume approach (Patankar [12]). The solutions are considered to be converged when the normalized residual values are less than 10^{-5} for all variables, but less than 10^{-9} only for the energy equation.

Periodic boundaries are used for the inlet and outlet of the flow domain. The constant mass flow rate of air with 300K ($Pr = 0.7$) is assumed in the flow direction rather than constant pressure drop due to periodic flow conditions. The inlet and outlet profiles for the velocities must be identical. The physical properties of the air are assumed to remain constant at average bulk temperature. Impermeable boundary and no-slip wall conditions are implemented over the tube wall as well as the twisted-tapes. The constant temperature of the tube wall is maintained at 310K while the twisted-tape is assumed at adiabatic wall condition.

[3] MATHEMATICAL FOUNDATION

The mathematical foundation is referred from Promvonge et al. [10]. The numerical model for fluid flow and heat transfer in a tube is developed under the following assumptions:

- Steady three-dimensional fluid flow and heat transfer.
- The flow is laminar and incompressible.

- Constant fluid properties.
- Body forces and viscous dissipation are ignored.
- Negligible radiation heat transfer.

Based on the above assumptions, the tube flow is governed by the continuity, the Navier–Stokes equations and the energy equation. In the Cartesian tensor system these equations can be written as follows:

Continuity equation:

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

Momentum equation:

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

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j} \left(\Gamma \frac{\partial T}{\partial x_j} \right) \quad (3)$$

where Γ is the thermal diffusivity and is given by

$$\Gamma = \frac{\mu}{Pr} \quad (4)$$

Four parameters of interest in the present work are the Reynolds number, friction factor, Nusselt number and thermal enhancement factor.

The Reynolds number, friction factor, Nusselt number and thermal enhancement factor are calculated as follows:

$$Re = \rho U D / \mu \quad (5)$$

The friction factor, f , is computed by pressure drop, Δp , across the length of the periodic tube, L , as

$$f = \frac{2(\Delta p/L)D}{\rho U^2} \quad (6)$$

The heat transfer is measured by local Nusselt number which can be written as

$$Nu = \frac{hD}{k} \quad (7)$$

The average Nusselt number can be obtained by

$$Nu = \frac{1}{A} \int Nu_x \partial A \quad (8)$$

The thermal enhancement factor (TEF) is defined as the ratio of the heat transfer coefficient of an augmented surface, h to that of a smooth surface, h_0 , at an equal pumping power and given by

$$TEF = \frac{h}{h_0} \bigg|_{pp} = \frac{Nu}{Nu_0} \bigg|_{pp} = (Nu/Nu_0)/(f/f_0)^{1/3} \quad (9)$$

where Nu_0 and f_0 stand for Nusselt number and friction factor for the smooth tube, respectively.

[4] RESULTS

4.1 Verifications of the smooth circular tube

The verifications of the heat transfer and friction factor for the smooth circular tube are done by comparing with the analytical values under similar operating conditions as shown in Figure 2a and 2b, respectively. The current results are found to be in excellent agreement with exact solution values (Incropera and Dewitt [13]) for both the Nusselt number and friction factor, less than $\pm 0.5\%$ deviation. The exact solutions of the Nusselt number and the friction factor for laminar flows over the smooth tube with constant wall temperature are as follows:

$$Nu = 3.66 \quad (10)$$

$$f = 64 / Re \quad (11)$$

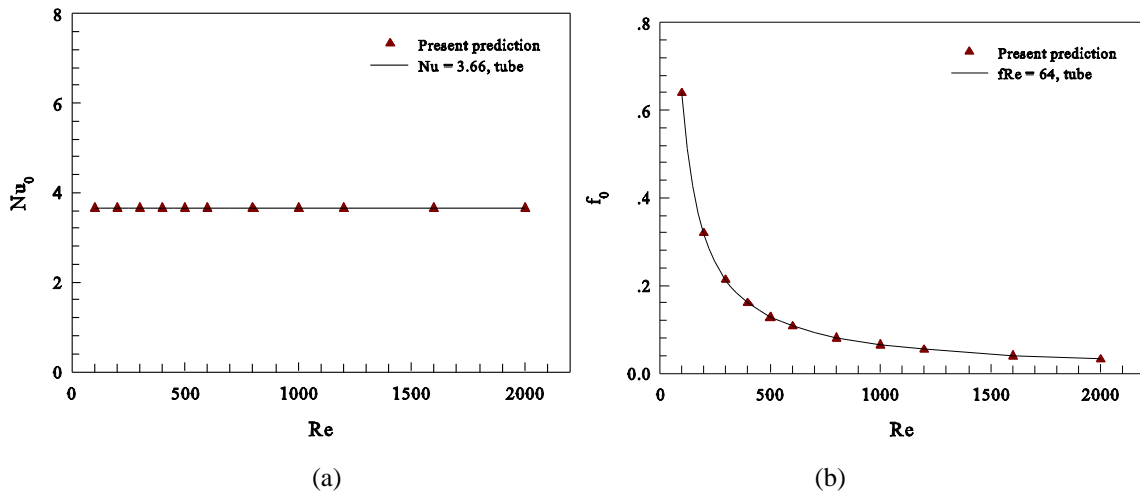


Figure2: Verification of (a) Nusselt number and (b) friction factor for smooth tube.

4.2 Grid independent test

The variations of the Nusselt number and friction factor values for the twisted-tape at $DR = 0.6$, $y/W = 2.0$ and $Re = 800$ are less than 0.2% when rising the number of grids from 350,000 to 450,000. Considering both convergent time and solution precision, the grid system of 350,000 cells is used for the current computational model.

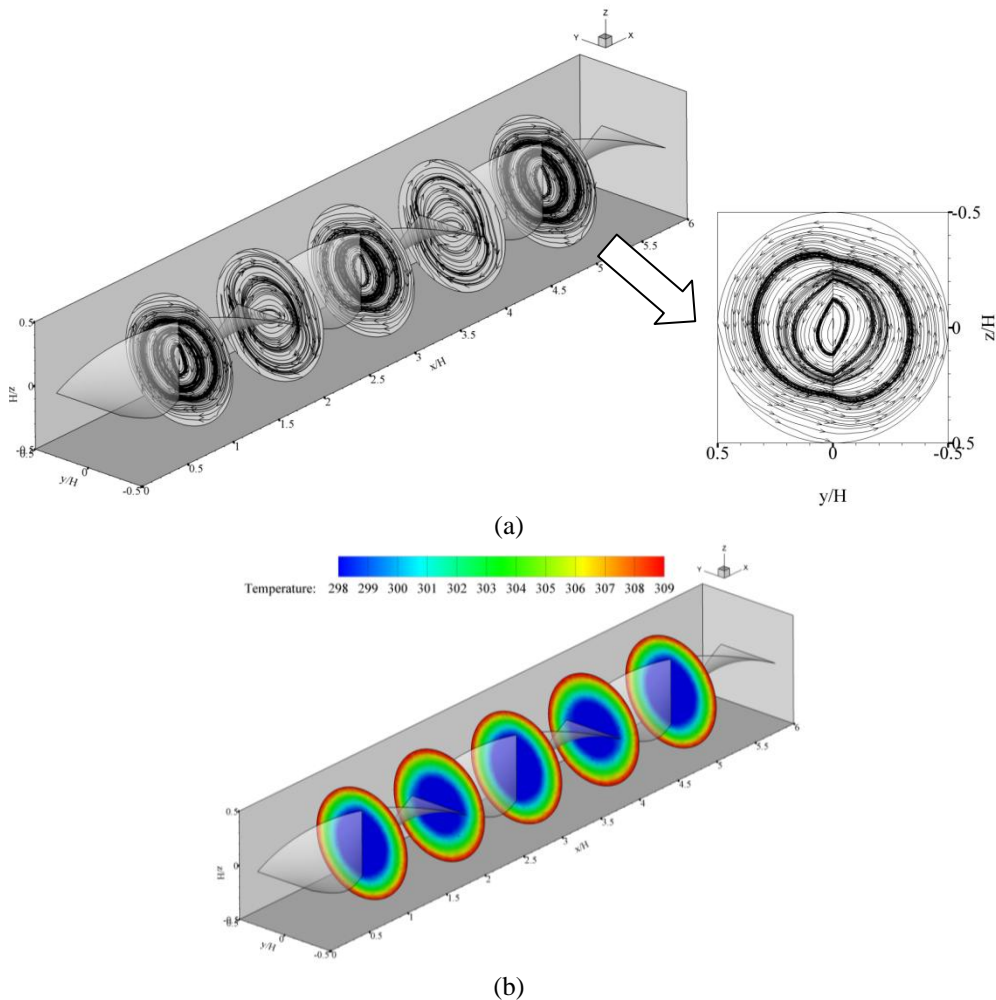


Figure 3: (a) streamlines in transverse planes and (b) temperature contours in transverse planes for twisted-tape at $Re = 800$ and $DR = 0.6$.

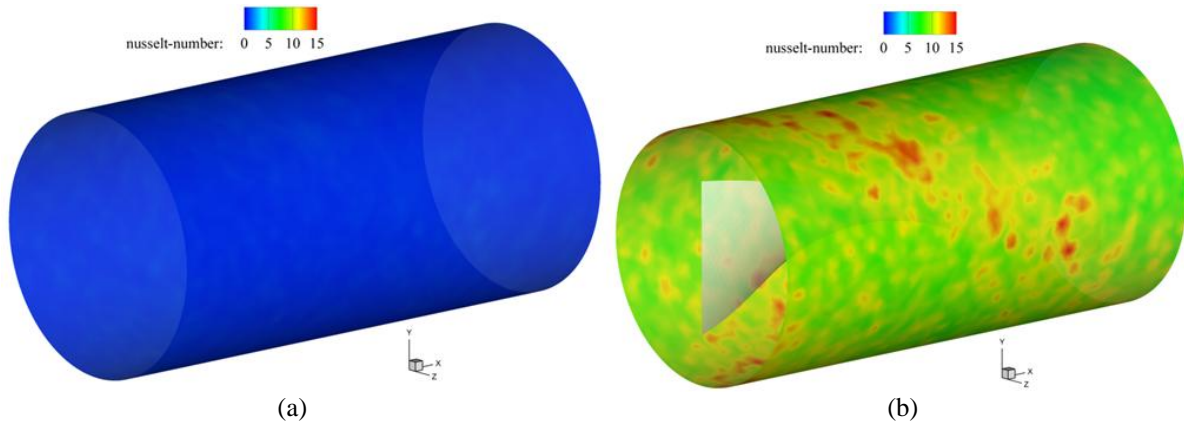


Figure 4: Nu_x Contours for (a) smooth tube and (b) $DR = 0.60$ and $Re = 800$.

4.3 Flow configuration and heat transfer behavior

Figure 3a presents the flow configuration of the twisted-tape in a circular tube in term of streamlines in transverse planes at $DR = 0.6$ and $Re = 800$. In general, the twisted-tape can create the clockwise swirling flow though the test tube for all cases. The turbulent level of the fluid flow is depended on the DR and Re values. The rise of the diameter ratio and Reynolds number leads to the increase in the turbulence level of the fluid flow.

Figure 3b shows the temperature contours in transverse planes for the twisted-tape in case of $DR = 0.6$ and $Re = 800$. It can be seen that the use of twisted-tape results in a better mixing of the fluid flow, especially, at the center of the circular tube, but the heat transfer behavior is terribly found at neat the tube wall.

Local Nu_x contours over the tube surface for smooth tube and circular tube inserted with twisted-tape at $DR = 0.60$ and $Re = 800$ are presented in Figure 4a and 4b, respectively. As the figures, it is found that the insertion of the twisted-tape can enhance heat transfer rate over the tube wall.

4.4 Performance evaluation

The performance evaluations can divided into three parts; heat transfer, pressure loss and performance in terms of Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and the thermal enhancement factor (TEF), respectively.

The variation of the average Nu/Nu_0 with diameter ratio at different Reynolds number values is depicted in Figure 5. In general, the Nu/Nu_0 tends to increase with the rise of diameter ratio and Reynolds number for all cases. The $DR = 0.2$ performs the lowest heat transfer rate while the $DR = 0.8$ gives the highest value. In range studied, the heat transfer rate is around 1.31 – 5.44 times higher than the smooth tube with no twisted-tape.

Figure 6 presents the variations of the f/f_0 with DR at various Reynolds numbers. As seen in the figure, the f/f_0 tends to increase with the rise of DR and Reynolds number. The $DR = 0.8$ provides the highest value of friction factor around 13.68 times above the smooth tube. The f/f_0 value for using the twisted-tape is found to be about 1.95 – 13.68 times over the smooth tube depended on the DR and Re values.

The variation of the thermal enhancement factor, TEF with diameter ratios is displayed in figure 7. The TEF performs slightly decrease when increasing diameter ratio, but increase with increasing Reynolds number. The TEF in range studies is found to be around 1 – 2.42 depended on diameter ratio and Reynolds number. The optimum thermal enhancement factor is found in the case of $DR = 0.2$ at the highest Reynolds number.

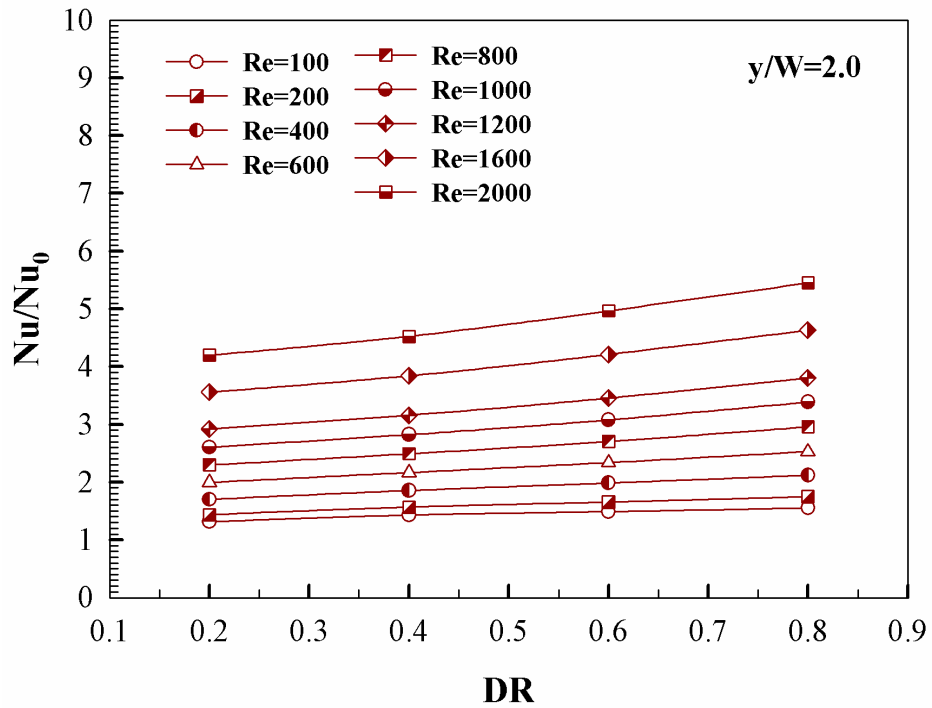


Figure 5: Variation of the Nu/Nu_0 with DRs for various Reynolds numbers.

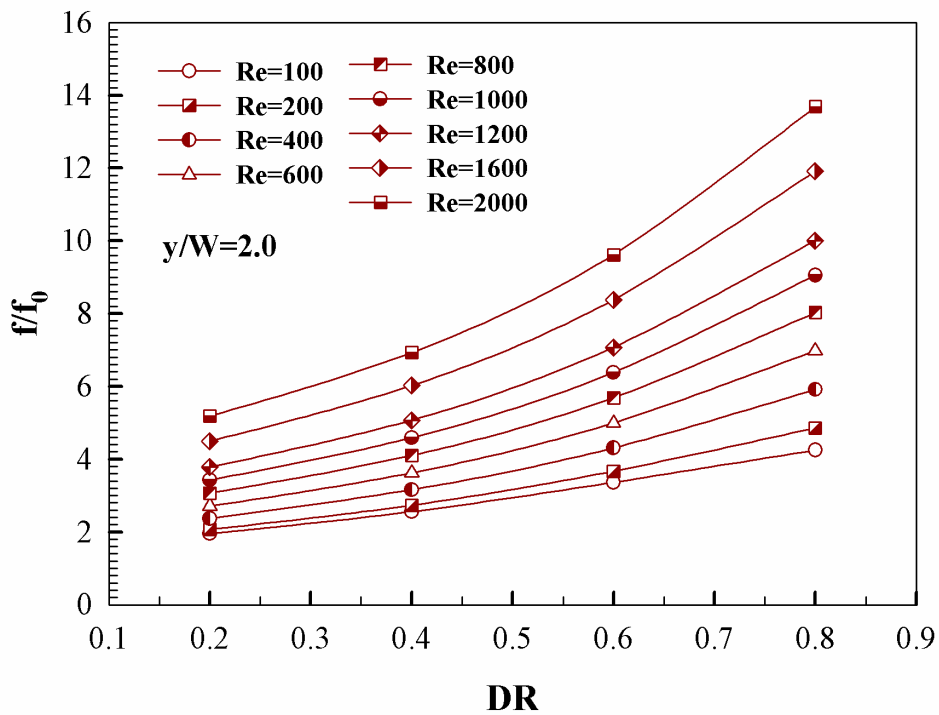


Figure 6: Variation of the f/f_0 with DRs for various Reynolds numbers.

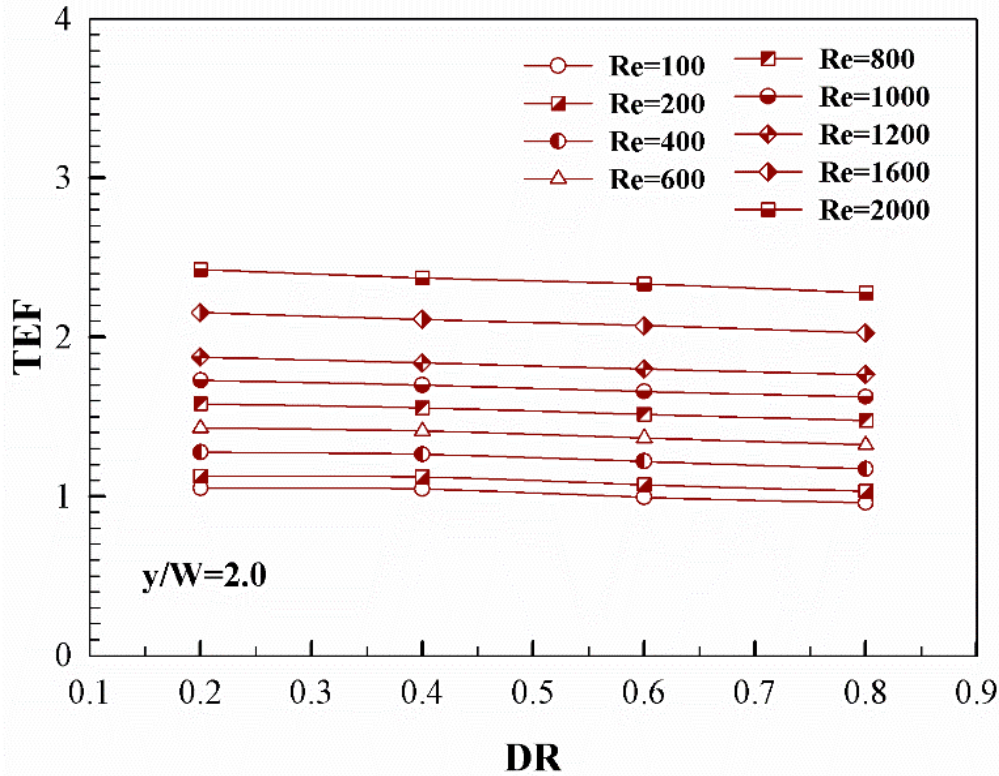


Figure 7: Variation of *TEF* with *DRs* for various Reynolds numbers.

[5] DISCUSSION AND CONCLUSION

As the numerical result, the use of the twisted-tape can generate the swirling flow in the test tube which help to improve heat transfer rate and thermal performance. The turbulence of the fluid flow is a key for enhancing the heat transfer rate. The higher diameter ratio and the Reynolds number result in the increase in turbulent level due to the higher pressure difference.

The order of heat transfer enhancement is about 1.31 – 5.44 times the smooth tube for using the twisted-tapes with $DR = 0.2 - 0.8$ and $Re = 100 - 2000$. However, the heat transfer augmentation is associated with enlarged pressure loss ranging from 1.95 to 13.68 times above the smooth tube. The highest thermal enhancement factor for the twisted-tapes with $DR = 0.20$ is found to be about 2.42 at $Re = 2000$.

NOMENCLATURE

A	heat transfer surface area, m^2
DR	diameter ratio, (W/D)
W	twisted-tape width/diameter, m
y	twisted length, m
D	tube diameter, m
f	friction factor
h	convective heat transfer coefficient, $W m^{-2} K^{-1}$
k	thermal conductivity, $W m^{-1} K^{-1}$
L	cyclic length of one cell (or axial pitch length, D), m
Nu	Nusselt number
p	static pressure, Pa
Pr	Prandtl number
Re	Reynolds number, $(\rho \bar{u} D / \mu)$
T	temperature, K
u_i	velocity in x_i -direction, $m s^{-1}$
\bar{u}	mean velocity in channel, $m s^{-1}$
Greek letter	
μ	dynamic viscosity, $kg s^{-1} m^{-1}$

Γ	thermal diffusivity
ρ	density, kg m^{-3}
TEF	thermal enhancement factor
Subscript	
in	inlet
0	smooth channel
w	wall
pp	pumping power

[6] ACKNOWLEDGEMENT

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