Thermal Performance of Heat Exchanger Tube Installed with Triple Twisted-tapes

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***Abstract*— A current study deals with the numerical analysis of heat transfer intensification, flow-thermal fields, local heat transfer distribution, friction factor and thermal performance behavior of the heat exchanger tubes installed with triple twisted tapes. Each set of triple twisted tapes consists of three twisted tapes having an identical twist ratio. Twisted tapes of interest have three different twisted ratios (y/w = 1.0, 1.5 and 2.0). Simulation was performed for the turbulent flow with Reynolds numbers (*Re*) ranging from 5000 to 20,000. Numerical results indicate that the use of triple twisted-tapes leads to heat transfer enhancement. Nusselt number (*Nu*) and friction factor (*f*) increase while thermal performance factor (η) decreases with decreasing twist ratio of triple twisted tapes. The Nusselt numbers of the tubes installed with triple twisted-tapes having twist ratios of 1.0, 1.5 and 2.0 are higher than those of the plain tube by around 3.34-4.45, 2.75-3.65 and 2.56-3.41 times, respectively. Friction factors (*f*) of the tubes installed triple twisted-tapes with twist ratios of 1.0, 1.5 and 2.0 are increased up to 46.1-49.7, 24.16-26.04 and 15.86-17.09 times, respectively, as compared to those of the plain tube. The maximum thermal performance factors of 1.21, 1.23 and 1.32 are obtained by using triple twisted tapes with y/w = 1.0, 1.5 and 2.0 at the Reynolds number of 5000.**

***Index Terms*— heat transfer intensification, flow-thermal fields, heat exchanger, triple twisted-tapes**

I. INTRODUCTION

Heat exchangers have been widely applied in many industries and engineering applications such as thermal power plant, chemical reactors, refrigeration, heat exchangers, petrochemical and oil industries, *etc* [1-7]. Various heat transfer intensification techniques have been developed to improve the thermal performance and intensify the efficiency of heat exchangers [8-12]. Applying heat transfer intensification techniques usually causes friction loss penalty resulting in additional required pumping power. From an economic point of view, keeping pumping power in a proper level is one of the major concerns. Thus, optimizing geometries of heat transfer intensification devices and operating conditions is a very important issue.

Twisted tape is one of the most widely used devices in heat transfer intensification technique. Twisted tapes were also used with other enhancement techniques such as nanofluids [12] and twisted tubes [13]. Several modified twisted tapes have been invented for increasing the heat transfer rate and thermal performance of a heat exchanger. Among the modified twisted tapes, the multiple tapes (double tapes, twin tapes, quadruple tapes, etc.) show promising performance in heat transfer enhancement due to the effect of multiple swirl flows which promote the turbulent intensity of the fluid flow [6, 8, 10, 14-15]. These results have motivated the present work to conduct numerical analysis to gain a better understanding of flow and heat transfer behaviors in heat exchanger tubes installed with triple twisted tape. The main aims of the present numerical analysis are:

1. To study the influence twist ratios (y/w = 1.0, 1.5 and 2.0) of triple twisted tapes on the heat transfer rate, flow and temperature fields, friction factor, local heat transfer distribution and thermal performance factor.
2. To find out the optimum condition showing the maximum thermal performance factor under the constant pumping power constraint.

II. GOVERNING EQUATIONS

The numerical model of fluid flow and heat transfer in tubes with triple twisted tape inserts having various twist ratios (y/w = 1.0, 1.5 and 2.0) was conducted under the following assumptions: steady three-dimensional fluid flow and heat transfer, turbulent and incompressible flow, constant fluid properties, negligible body forces and viscous dissipation. Based on the above assumptions, the tube flow is governed by the continuity, the Navier-Stokes equations and the energy equations than can be written as follows:

Continuity equation:

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

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Momentum equation:

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

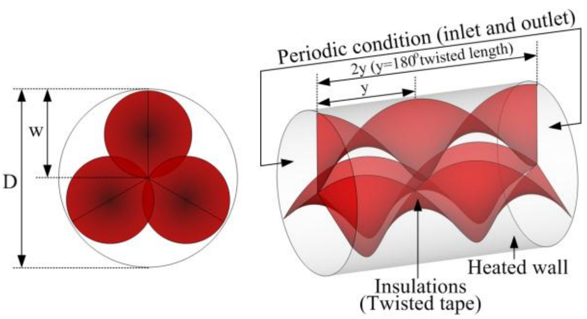
Energy equation:

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

where Γ is the thermal diffusivity and is given by:

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

The energy equation was discretized by the QUICK scheme while the governing equations were discretized by the power law scheme, decoupling with the SIMPLE algorithm and solved using a finite volume approach. The solutions were considered to be converged when the normalized residual values were less than 10^{-5} for all variables and less than 10^{-8} for the energy equation.



(a) front view (b) isometric view

Figure 1. Schematic view of tube inserted with triple twisted-tapes.

III. Triple Twisted-Tape Geometry and Boundary Conditions

Fig. 1 depicts a round tube having an inside diameter (D) of 0.05 m integrated with triple twisted-tapes under constant wall temperature. Each set of triple twisted tapes consists of three twisted tapes having an identical twist ratio. Twisted tapes of interest have three different twisted ratios (y/w = 1.0, 1.5 and 2.0), where y is a twist pitch (180°/twist pitch) and w is a tape width. Air is selected as the working fluid which enters the tube with an inlet temperature (T_{in}) of 300 K (Pr = 0.7). Simulation was performed for the turbulent flow with Reynolds numbers (Re) ranging from 5000 to 20,000. Periodic boundaries were applied for the inlet and outlet of the flow domain. The inlet and outlet velocity profiles are identical. Impermeable boundary and no-slip wall conditions have been implemented over the tube wall as well as the triple twisted-tapes. The constant heat flux of the wall was maintained at 600 W/m^2 while the triple twisted-tapes assumed to be under adiabatic wall conditions. Four parameters of interest in the present work are Reynolds number (Re), Nusselt number (Nu), frictional factor (f) and thermal performance factor. Simulation for a plain tube without twisted-tape insert was also performed as the reference case.

Reynolds number is evaluated from

$$Re = \rho UD/\mu \quad (5)$$

Nusselt number is defined as

$$Nu = hD/k \quad (6)$$

Local Nusselt number at the tube wall is calculated from

$$Nu_x = h_x D/k \quad (7)$$

Average Nusselt number can be obtained by

$$Nu = 1/A \int Nu_x \mathrm{d}A \quad (8)$$

Friction factor can be written as

$$f = 2\Delta PD/(\rho U^2 L) \quad (8)$$

Optimization of triple twisted tape geometry is made by minimizing thermal performance based on the relationships.

$$(\dot{V}\Delta P)_p = (\dot{V}\Delta P)_t \text{ and } (f\,\mathrm{Re}^3)_p = (f\,\mathrm{Re}^3)_t.$$

Thermal performance factor (η) can be written as

$$\eta = (Nu_t/Nu_p)/(f_t/f_p)^{1/3} \quad (9)$$

Practically, thermal performance factor should be greater than unity, which reflects an overall energy saving by the heat transfer equipment.

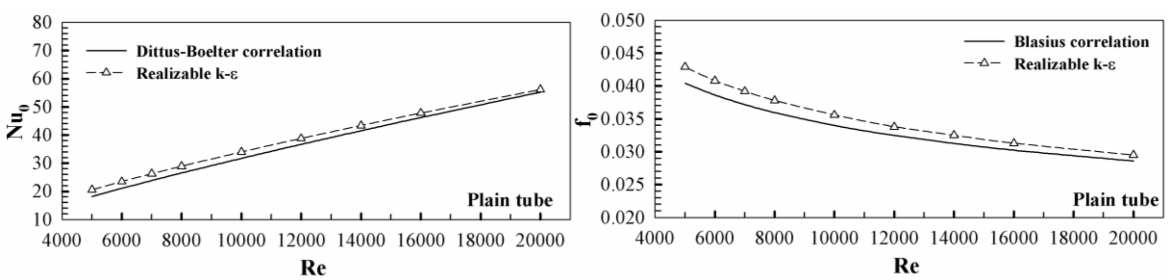


Figure 2. Validation of the plain tube alone.

IV. Numerical Results and Discussion

Validation test of plain tube alone

The numerical results of heat and fluid flow behaviors in tubes with triple twisted-tapes are reported in terms of (1) heat transfer rate (Nusselt number, *Nu*) and (2) flow (friction factor, *f*). To evaluate the accuracy and precision of the present numerical analysis, the present Nusselt number (*Nu*) and friction factor (*f*) results of a plain tube alone are compared with those of standard previous works including Dittus-Boelter correlation for Nusselt number (*Nu*) and Blasius correlation for friction factor (*f*) as shown below.

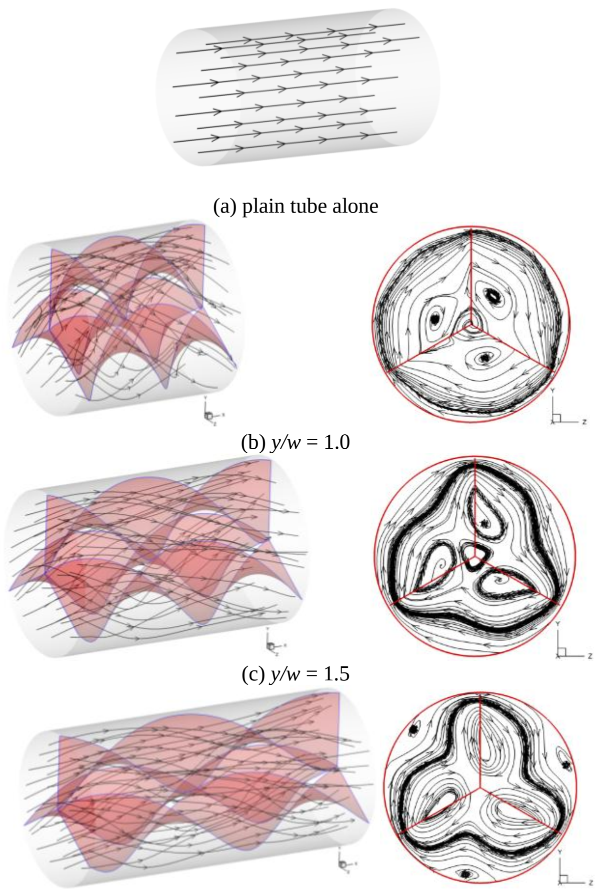
Correlation of Dittus-Boelter,

$$Nu = 0.023Re^{0.8}Pr^{0.4} \tag{10}$$

Correlation of Blasius,

$$f = 0.079Re^{-0.25} \tag{11}$$

Evidently, the maximum deviations of the present numerical Nusselt number (*Nu*) and friction factor (*f*) from the standard correlations of Dittus-Boelter and Blasius are ±3.2% and ±3.4%, respectively. The comparisons show that the numerical results are sufficiently accurate.



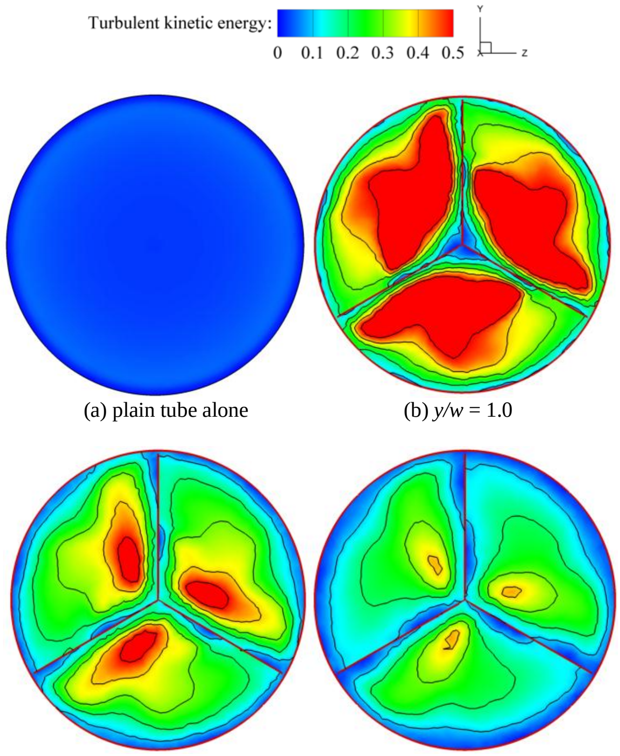
(a) plain tube alone

(b) y/w = 1.0

(c) y/w = 1.5

(d) y/w = 2.0

Figure 3. Streamlines and velocity vectors through tube inserted with triple twisted-tapes at Re = 5000.



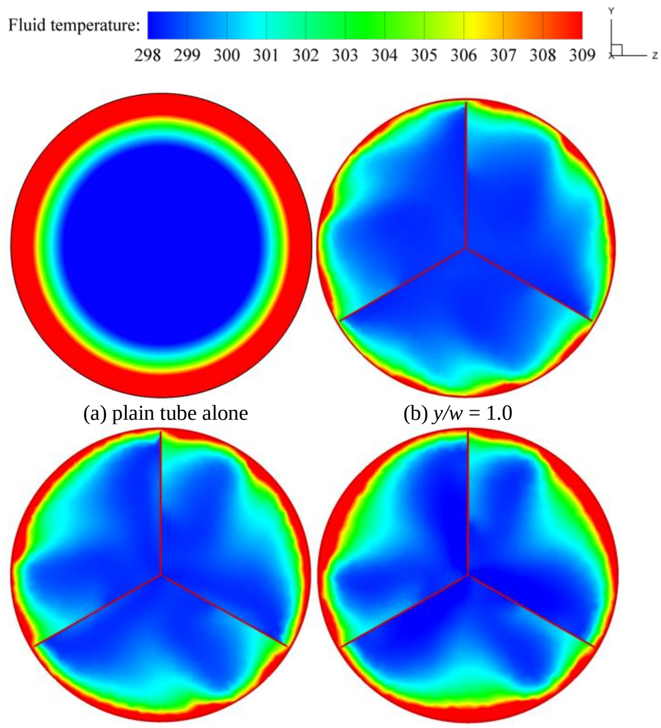
(a) plain tube alone

(b) y/w = 1.0

(c) y/w = 1.5

(d) y/w = 2.0

Figure 4. Turbulence kinetic energy (TKE) contours in tube inserted with triple twisted-tapes at Re = 5000.



(a) plain tube alone

(b) y/w = 1.0

(c) y/w = 1.5

(d) y/w = 2.0

Figure 5. Temperature fields in tube installed with triple twisted-tapes for Re = 5000.

Flow structure and thermal behavior

Contour plots of streamlines and velocity vectors through the tubes installed triple twisted-tapes having twist ratios (y/w) of 1.0, 1.5 and 2.0 and also a plain tube alone are demonstrated in Fig. 3. Axial flow is found in the plain tube while swirling flow is observed in the tubes installed triple twisted-tapes. The induced swirling flow helps in enhancing fluid mixing between the core regions and the tube walls. The stream lines show that swirling flow becomes stronger as twist ratio decreases.

Fig. 4 depicts the turbulent kinetic energy (TKE) contours of tubes with and without twisted tapes at Re = 5000. Obviously, TKE of the tubes with the triple twisted-tapes is higher than that of the one without twisted tape (the plain tube). It is found that high TKE intensity is found around the locations between tapes and along tape edges where shear stress is high. The results indicate that flow disturbance caused by the triple twisted-tapes helps in intensifying TKE and thus heat transfer rate.

Fig. 5 demonstrated the fluid temperature contours of tubes with and without twisted tapes at Re = 5000. The predicted results show that the hot fluid can move into core region with presence of triple twisted-tapes. As twist ratio decreases, hot fluid moves further to the middle of tube. This can be explained that the tapes with shorter twist length induced stronger swirling flow having higher tangential velocity which gives higher shear stress and more efficient in disrupting thermal boundary layers.

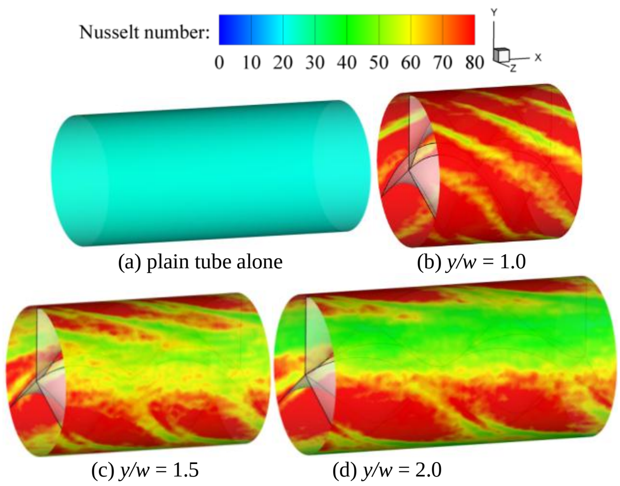


Figure 6. Distribution of local wall Nusselt number (Nu) in tube inserted with triple twisted-tapes at Re = 5000.

Contour plots of local wall Nusselt number of tubes with and without twisted tapes at Re = 5000 are demonstrated in Fig. 6. Evidently, high wall Nusselt number areas (in red color) are found along the tape edges due to the good fluid mixing near the tube walls. Local Nusselt number increases with decreasing twist ratios (y/w) due to the better fluid mixing. The triple twisted tapes with the smallest twist ratios (y/w = 1.0) give the highest local wall Nusselt number with the most uniform distribution. On the other hand, the plain tube alone offers the lowest local wall Nusselt number with the poorest uniformity.

Heat transfer

Heat transfer in term of average Nusselt number ratio or $Nu/\ Nu_0$ (Nu_0 is Nusselt number of the plain tube), is reported in Fig. 7. Average Nusselt number (Nu) enhances with increasing Reynolds number (Re). All Nusselt number ratios are above unity reflecting the heat transfer enhancement by triple twisted tapes due to the reasons mentioned above. However, Nusselt number ratio decreases with increasing Reynolds number. Because thermal boundary layer at higher Reynolds number is thinner, therefore, the effect of flow disturbance becomes less significant. As expected, the use of the tapes with smaller twist ratio results in higher Nusselt number. This is directly related to the strength of swirl intensity. The Nusselt numbers of the tubes installed with triple twisted-tapes having twist ratios of 1.0, 1.5 and 2.0 are higher than those of the plain tube by around 3.34-4.45, 2.75-3.65 and 2.56-3.41 times, respectively. In other words, the tapes with the smallest twist ratio (y/w = 1.0) offer higher Nusselt number (Nu) than the ones with twist ratios of 1.5 and 2.0 up to 21.7% and 30.3%, respectively.

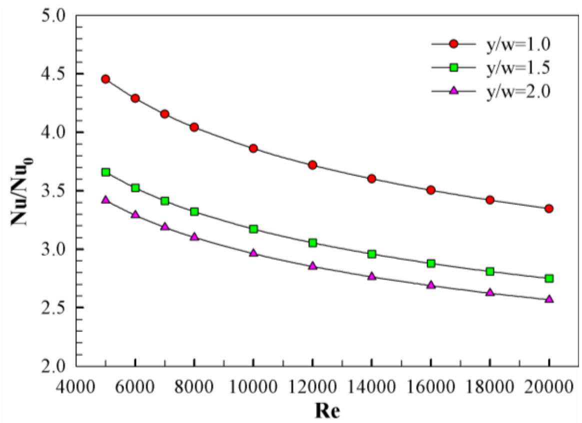


Figure 7. Variation of Nusselt number ratio with Reynolds number for tube inserted with triple twisted-tapes

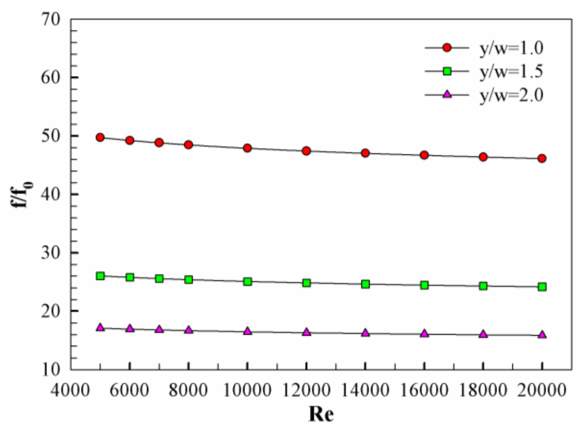


Figure 8. Variation of friction factor ratio with Reynolds number for tube inserted with triple twisted-tapes.

Friction factor

1. Friction loss in term of average friction factor ratio or f/f_0 (f_0 is friction factor of the plain tube), is shown in Fig. 8. Friction factor decreases with increasing

Reynolds number (*Re*) from 5000 to 20,000, which the typical trend found in moody chart for turbulent regime. All friction factor ratio are considerably higher than 1.0 due to the pressure drop penalty caused by triple twisted tapes. Friction loss increases with decreasing twist ratio, attributed to the stronger flow disturbance and the longer flow along the twisted tape surface. Friction factors (f) of the tubes installed triple twisted-tapes with twist ratios of 1.0, 1.5 and 2.0 are increased up to 46.1-49.7, 24.16-26.04 and 15.86-17.09 times, respectively, as compared to those of the plain tube. In other words, the tapes with the smallest twist ratio (y/w = 1.0) cause higher friction loss than the ones y/w = 1.5 and 2.0 by around 90.87% and 190.8%, respectively.

Thermal performance

Influence of triple twisted-tapes on thermal performance factor (η) is presented in Fig. 9. In general, thermal performance factor increases with decreasing Reynolds number, indicating that the use of the twisted tapes is more beneficial at lower Reynolds numbers. Although, the tapes with larger twist ratio give poorer heat transfer, the tapes give higher thermal performance factor, attributed to better tradeoff between enhanced heat transfer and increased friction loss. At Reynolds number of 5000, the triple twisted-tapes with y/w = 1.0, 1.5 and 2.0 give the maximum thermal performance factors (η) of 1.21, 1.23 and 1.32, respectively.

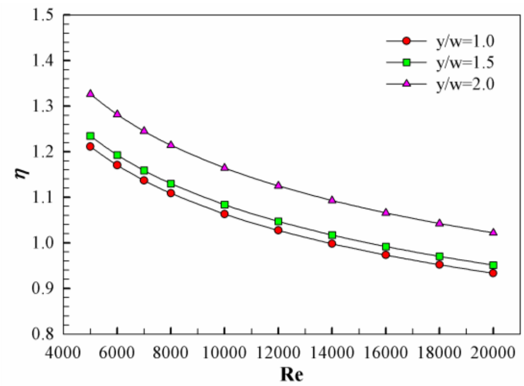


Figure 9. Variation of thermal enhancement factor with Reynolds number for tube inserted triple twisted-tapes.

V. SUMMARY

The effects of triple twisted-tape with twist ratios of y/w = 1.0, 1.5 and 2.0 on heat transfer intensification of turbulent fluid flow through a round tube are numerically studied. The main findings are summarized as follows:

- Swirl flow generated by the triple twisted-tape helps in improving fluid mixing between the core region and the tube walls and enhancing heat transfer rate.
- With an increase of Reynolds number (*Re*), Nusselt number (*Nu*) rises while friction factor (f) and thermal performance factor (η) drop.
- Nusselt number (*Nu*) and friction factor (f) increase while thermal performance factor (η) decreases with decreasing twist ratio of triple twisted tapes.
- The maximum thermal performance factors of 1.21, 1.23 and 1.32 are obtained by using triple twisted tapes with y/w = 1.0, 1.5 and 2.0 at the Reynolds number of 5000.
- The heat transfer and flow behaviors of the same system for laminar and transition flow regimes is being numerically investigated.

CONFLICT OF INTEREST

The authors declare no conflict of interest.

AUTHOR CONTRIBUTIONS

B. Samutpraphut; conducted the research, wrote the paper

P. Promthaisong; wrote the paper

K. Wongcharee; analyzed the data

S. Eiamsa-ard; analyzed the data

V. Chuwattanakul; analyzed the data and wrote the paper

All authors had approved the final version.

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