Original Research Paper

Convective Heat Transfer and Thermal Performance in a Circular Tube Heat Exchanger Inserted with U-Shaped Baffle

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Abstract: Numerical investigations on flow structure and heat transfer characteristic in a heat exchanger tube fitted with U-shaped baffle are presented. The U-shaped baffle is created from the concepts of orifice and inclined baffle. The influences of the baffle height (b/D = 0.05, 0.10, 0.15, 0.20 and 0.25) and flow attack angle (α = 20°, 30° and 45°) of the U-shaped baffle on heat transfer, pressure loss and thermal performance are considered. The finite volume method with SIMPLE algorithm is selected to solve the present problem. The numerical results are reported in terms of flow visualization and heat transfer behavior. The performance analysis for the tube heat exchanger inserted with the U-shaped baffle is also reported. As the numerical result, it is found that the insertion of the U-shaped baffle help to improve the heat transfer rate and thermal performance higher than the smooth tube with no baffle. The increments of heat transfer rate and pressure loss are around 1–8.7 and 1–29 times over the smooth tube, respectively. Additionally, it is detected that the 45° U-shaped baffle with the BR = 0.2 gives the highest thermal enhancement factor around 2.9 at the highest Reynolds number, Re = 1200.

Keywords: Baffle, Thermal Performance, Heat Transfer Rate, Heat Exchanger, Finite Volume Method

Introduction

The thermal performance improvement in various types of the heat exchanger is frequently found in many industries. The performance improvement in the heat exchanger or heating/cooling channel can help to save production cost and thermal energy of the system. The thermal improvement can divide into two techniques; active and passive techniques. The passive technique is a popular way to improve the thermal performance and heat transfer rate in the heat exchanger due to the passive technique not requires the additional power into the system. The passive technique is the installation of the vortex generator in the tube/channel heat exchanger to generate the vortex, swirling flow and to disturb the thermal boundary layer on the tube wall. The selection for the type of the vortex generator depends on the application of the heat exchanger.

The circular ring (like orifice) is widely installed in the heating/cooling system to augment the heat transfer rate and performance. Kongkaitpaiboon *et al.* (2010a) experimentally studied on heat transfer and friction loss in a circular tube heat exchanger with circular-ring vortex generators. The effects of the diameter ratios (diameter of the orifice to the diameter of the tube) and pitch ratios (*P/D*) were investigated for turbulent region, *Re* = 4000–20,000. They reported that the heat transfer augmentation is around 57-195% when compared with the smooth circular tube without vortex generator. The selection of the conical-ring to improve the heat exchanger was also reported by many researchers (Yakut and Sahin, 2004a; 2004b; Durmus, 2004; Eiamsa-ard and Promvonge, 2006a; 2006b; Promvonge and Eiamsa-ard, 2007a; 2007b; 2007c; 2007d; Kongkaitpaiboon *et al.*, 2010b; Ozceyhan *et al.*, 2006; Akansu, 2006; Kiml *et al.*, 2003; 2004). They found that the conical-ring in the heat exchanger enhances heat transfer rate and thermal efficiency, but also increases very large pressure loss, especially, at low diameter ratio (low flow area).

The literature reviews for the numerical investigation on flow and heat transfer in the channel heat exchanger inserted with baffle are shown as Table 1.

As the previous works, the present generator is designed with the main aim to support the installation and maintenance methods in the real system and also

gives high heat transfer rate and thermal performance as the baffle turbulator. The U-shaped baffle is a combination of the inclined baffle and orifice plate (circular ring). The distinctive advantages for the inclined baffle are to give both high thermal performance and heat transfer rate, while the orifice plate allows easy implementation in real system.

In the current investigation, the effects of the baffle height and flow attack angle for the U-shaped baffle on heat transfer rate, pressure loss and thermal performance are considered for the laminar regime at $Re = 100-1200$. The numerical method (finite volume method) is selected to investigate the present problem. The numerical method may help to describe the mechanisms on the test section that

the way to develop the vortex generator in the system. The performance assessment for the tube heat exchanger inserted with the U-shaped baffle is also reported.

Physical Geometry of the Tube Heat Exchanger and Arrangement

The U-shaped baffles are inserted in the circular tube heat exchanger (Fig. 1) in order to increase heat transfer rate and thermal performance. The circular tube diameter, D, is around 0.05 m. The length of the periodic module is fixed at 0.05 m (=D). The U-shaped baffle is created from the concepts of the inclined baffle and orifice.

Table 1. The literature reviews for the numerical investigation on flow and heat transfer in the channel heat exchanger installed with baffle

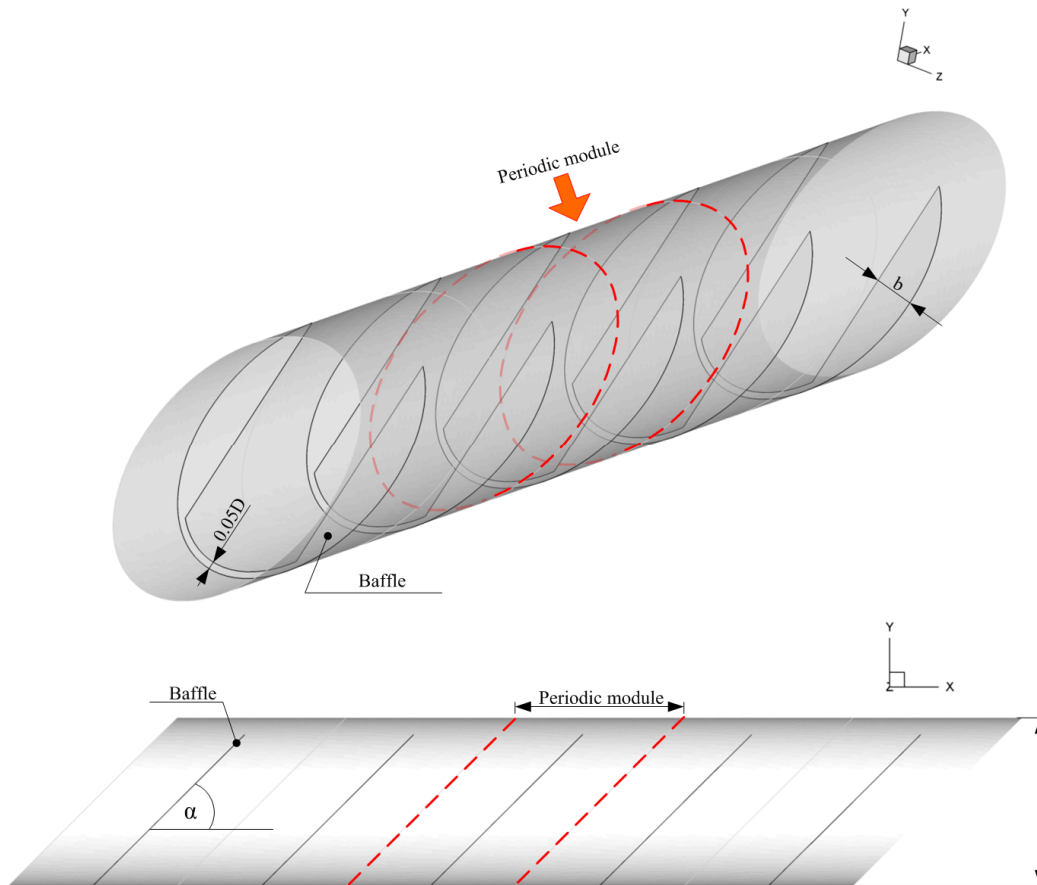| Authors	Studied cases	Nu/Nu_0	f/f_0	TEF
Jedsadaratanachai et al. (2011)	30° inclined baffle Inline, two opposite walls, square channel $BR = 0.2$ $PR = 0.5-2.5$ $Re = 100-2000$	1.00 – 9.20	1.00 – 21.50	3.78
Kwankaomeng and Promvonge (2010)	30° inclined baffle One side, square channel $BR = 0.1 - 0.5$ $PR = 1.0 - 2.0$ $Re = 100 - 1000$	1.00 – 9.23	1.09 – 45.31	3.10
Promvonge et al. (2010)	30° inclined baffle Inline, two opposite walls, square channel $BR = 0.1 - 0.3$ $PR = 1.0 - 2.0$ $Re = 100 - 2000$	1.20 –11.00	2.00 – 54.00	4.00
Promvonge and Kwankaomeng (2010)	45° V–baffle Staggered, two opposite walls, AR=2 channel $BR = 0.05 - 0.3$ $PR = 1.0$ $Re = 100 - 1200$	1.00– 11.00	2.00 – 90.00	2.75
Promvonge et al. (2010)	45° inclined baffle Inline–staggered, two opposite walls, square channel $BR = 0.05 - 0.3$ $PR = 1.0$ $Re = 100 - 1000$	1.50 – 8.50	2.00 – 70.00	2.60
Promvonge et al. (2012)	45° V–baffle Inline Downstream, two opposite walls, square channel $BR = 0.1 - 0.3$ $PR = 1.0 - 2.0$ $Re = 100 - 2000$	1.00– 21.00	1.10 – 225.00	3.80
Boonloi (2014)	20° V–baffle Inline Downstream–Upstream, two opposite walls, square channel $BR = 0.1 - 0.3$ $PR = 1.0$ $Re = 100 - 2000$	1.00– 13.00	1.00 – 52.00	4.20
Boonloi and Jedsadaratanachai (2013)	30° V–baffle Downstream, One side, square channel $BR = 0.1 - 0.5$ $PR = 1.0 - 2.0$ $Re = 100 - 1200$	1.00– 14.49	2.18 – 313.24	2.44
Jedsadaratanachai and Boonloi (2013)	45° Discrete–V–baffle Downstream, diagonally, square channel $BR = 0.05 - 0.20$ $PR = 1.0 - 1.5$ $Re = 100 - 1200$	1.40 – 8.10	2.50 – 36.00	2.50

Fig. 1. Circular tube heat exchanger fitted with modified baffles

Table 2. Boundary condition of the computational domain

Zone	Boundary condition
Inlet	Periodic
Outlet	Periodic
Tube wall	Constant temperature at 310K, no slip wall condition
U-shaped baffle	Insulator, no slip wall condition

The influences of the blockage ratio (b/D or BR = 0.05, 0.10, 0.15. 0.20 and 0.25) and flow attack angle (α = 20°, 30° and 45°) of the U-shaped baffle on flow configuration and heat transfer characteristic are investigated. The laminar flow with the Reynolds number in range 100-1200 is considered.

Boundary Condition and Assumption

The boundary conditions for the computational domain of the tube heat exchanger fitted with the U-shaped baffle are written as Table 2.

The assumptions for the present investigation are concluded as follows:

- Flow and heat transfer in the test tube are steady in three dimensions

- Flow is laminar and incompressible
- Body force, viscous dissipation, radiation heat transfer and natural convection are unnoticed.
- The test fluid is air with 300K (Pr = 0.707)
- The thermal properties of the air remain constant at average bulk mean temperature

Mathematical Foundation

The tube heat exchanger inserted with the U-shaped baffle is governed by the continuity, the Navier–Stokes equations and the energy equation.

The energy equation is discretized by the QUICK scheme, while the governing equations are discretized by power law scheme. The current investigation is resolved by finite volume method with SIMPLE algorithm. The solutions are measured 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.

The main parameters are Reynolds number, friction factor, local Nusselt number, average Nusselt number and thermal enhancement factor.

The Reynolds number is calculated as:

$$\text{Re} = \rho \overline{u} D / \mu \tag{1}$$

The friction factor, f, is measured by pressure drop, Δp, across the periodic module, L:

$$f = \frac{(\Delta p/L)D}{\frac{1}{2}\rho\bar{u}^2} \tag{2}$$

The local heat transfer is written as:

$$Nu_x = \frac{h_x D}{k} \tag{3}$$

The average Nusselt number can be obtained by:

$$Nu = \frac{1}{A}\int Nu_x \partial A \tag{4}$$

The Thermal Enhancement Factor (TEF) is calculated by the augmentations on both heat transfer and friction factor at similar pumping power:

$$TEF = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = (Nu/Nu_0)/(f/f_0)^{1/3} \tag{5}$$

The Nu_0 and f_0 is the Nusselt number and friction factor for the smooth circular tube, respectively.

Numerical Validation

The computational domain for the smooth circular tube heat exchanger is validated on both heat transfer rate and flow structure as reported in Fig. 2. The numerical schemes for the momentum; FOU, power law, SOU, are also compared. The numerical results reveal that the difference of the numerical scheme has no effect for the flow and heat transfer. The deviations on flow and heat transfer are found around ±0.3 and ±0.5%, respectively.

The difference of grid numbers; 80000, 120000, 180000, 240000 and 360000, for the computational domain of the tube heat exchanger inserted with the U-shaped baffle at $\alpha = 30°$ and $BR = 0.15$ is compared. Fig. 3a and b show the comparisons of the computational domain with various grid cells for heat transfer rate and pressure loss, respectively. The results show that the increment of grid cell from 180000 to 240000 has no effect for heat transfer and friction loss. Therefore, the grid around 180000 is applied for all cases of the computational model when considered on both time for investigation and accuracy result.

Numerical Result

The numerical results for the circular tube heat exchanger inserted with the U-shaped baffle are reported in terms of mechanisms on flow and heat transfer. The performance assessment for the present investigation is also concluded.

Flow and Heat Transfer Mechanisms

The flow configuration in the circular tube heat exchanger fitted with the U-shaped baffles is presented in term of streamline in transverse planes as depicted in Fig. 4. As the figure, it is found that the U-shaped baffle can produce the vortex flow through the test section. The symmetry vortex flow with common-flow-down is detected due to the baffle symmetry. The position of the vortex core depends on the location in the test tube. The vortex flow in the heating section help a better fluid mixing that is the reason for heat transfer and performance augmentations. The similar flow structure is found in all cases of the present investigation, but the strength of the vortex flow is not equal.
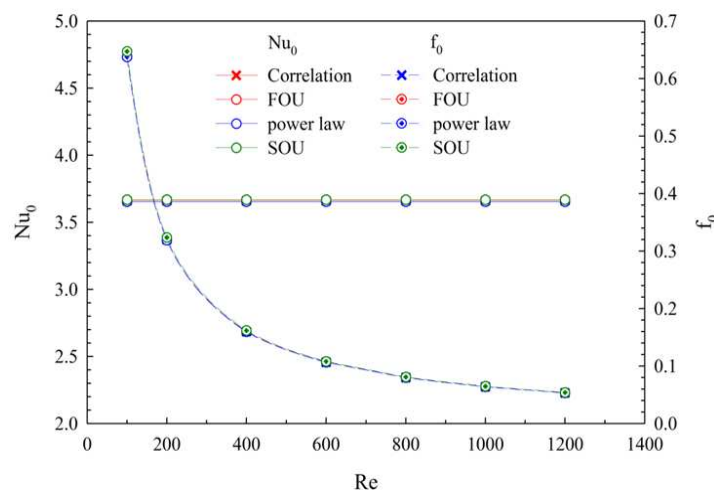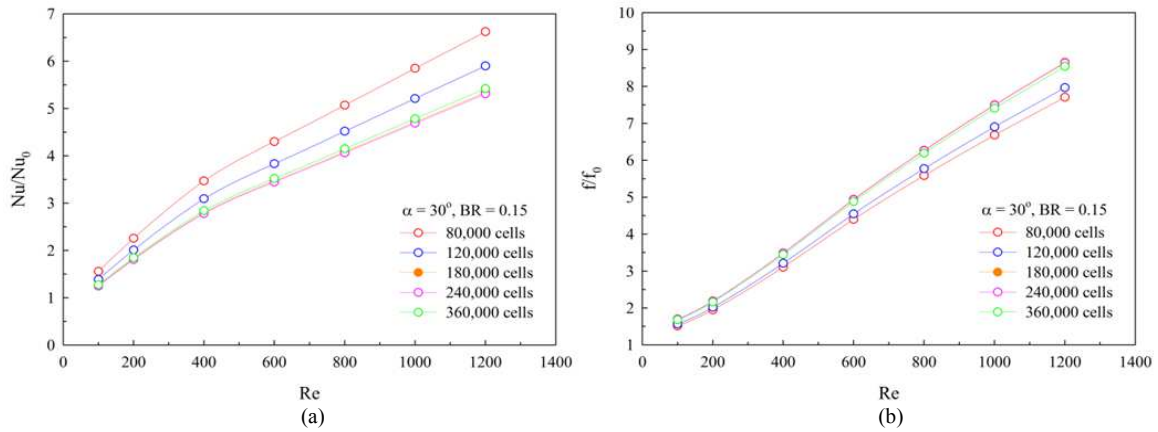

Fig. 2. Validation of the smooth circular tube

Fig. 3. Grid independence for (a) Nu/Nu_0 and (b) f/f_0 of the circular tube heat exchanger fitted with modified baffle at $\alpha = 30°$ and $BR = 0.15$
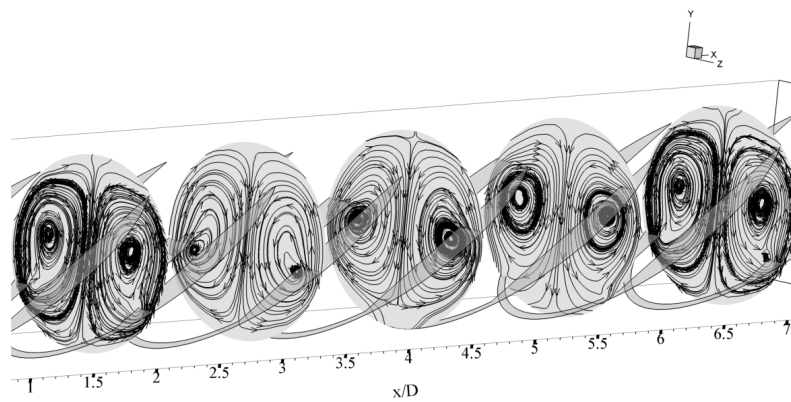

Fig. 4. Streamline in transverse planes for the circular tube heat exchanger fitted with modified baffle at $\alpha = 30°$, $BR = 0.15$ and $Re = 600$
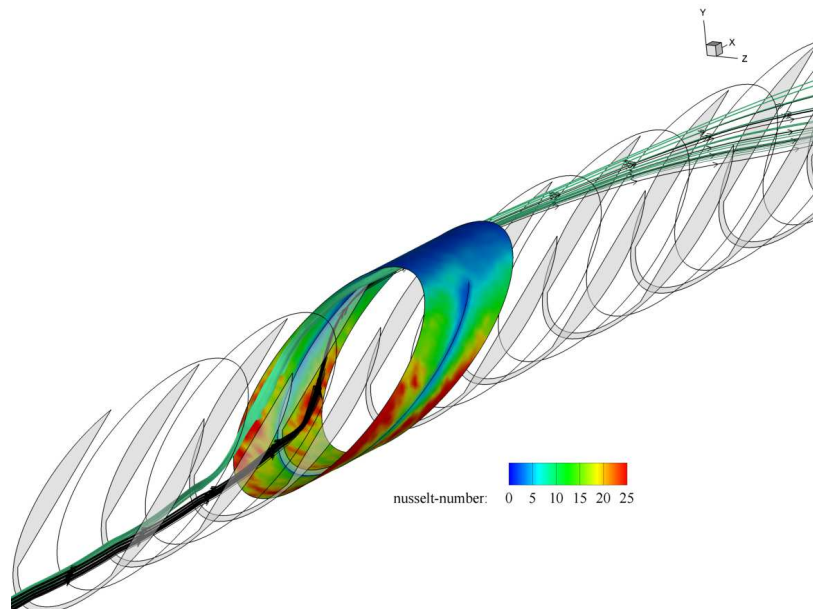

Fig. 5. Longitudinal vortex flow with local Nusselt number distribution on the tube wall for the circular tube heat exchanger fitted with modified baffle at $\alpha = 30°$, $BR = 0.15$ and $Re = 600$

Fig. 5 presents the local Nusselt number with longitudinal vortex flow in the tube heat exchanger fitted with U-shaped baffle at $\alpha = 30°$, $BR = 0.15$ and $Re = 600$. The local Nusselt number distribution is an indicator to describe the disturbance of the thermal boundary layer on the tube wall. The thermal boundary layer disturbance in the circular tube heat exchanger is cause for heat transfer rate and thermal performance improvements.

Performance Analysis

The performance analysis of the tube heat exchanger inserted with the U-shaped baffle on the Nusselt number ratio, friction factor ratio and thermal enhancement are reported as Figs. 6a, b and c, respectively.

In general, the addition of the U-shaped baffle in the tube heat exchanger provides higher heat transfer rate, pressure loss and thermal performance than the smooth circular tube with no baffle. The Nusselt number ratio, friction factor ratio and thermal enhancement factor increases when increasing the Reynolds number for all cases.

Considering at similar flow attack angle, the heat transfer rate and friction loss increase when enhancing the baffle height. This is because the augmentation of the baffle height leads to increase vortex strength. The $BR = 0.25$ of the U-shaped baffle performs the highest heat transfer rate and friction loss, while the $BR = 0.05$ provides the opposite result.
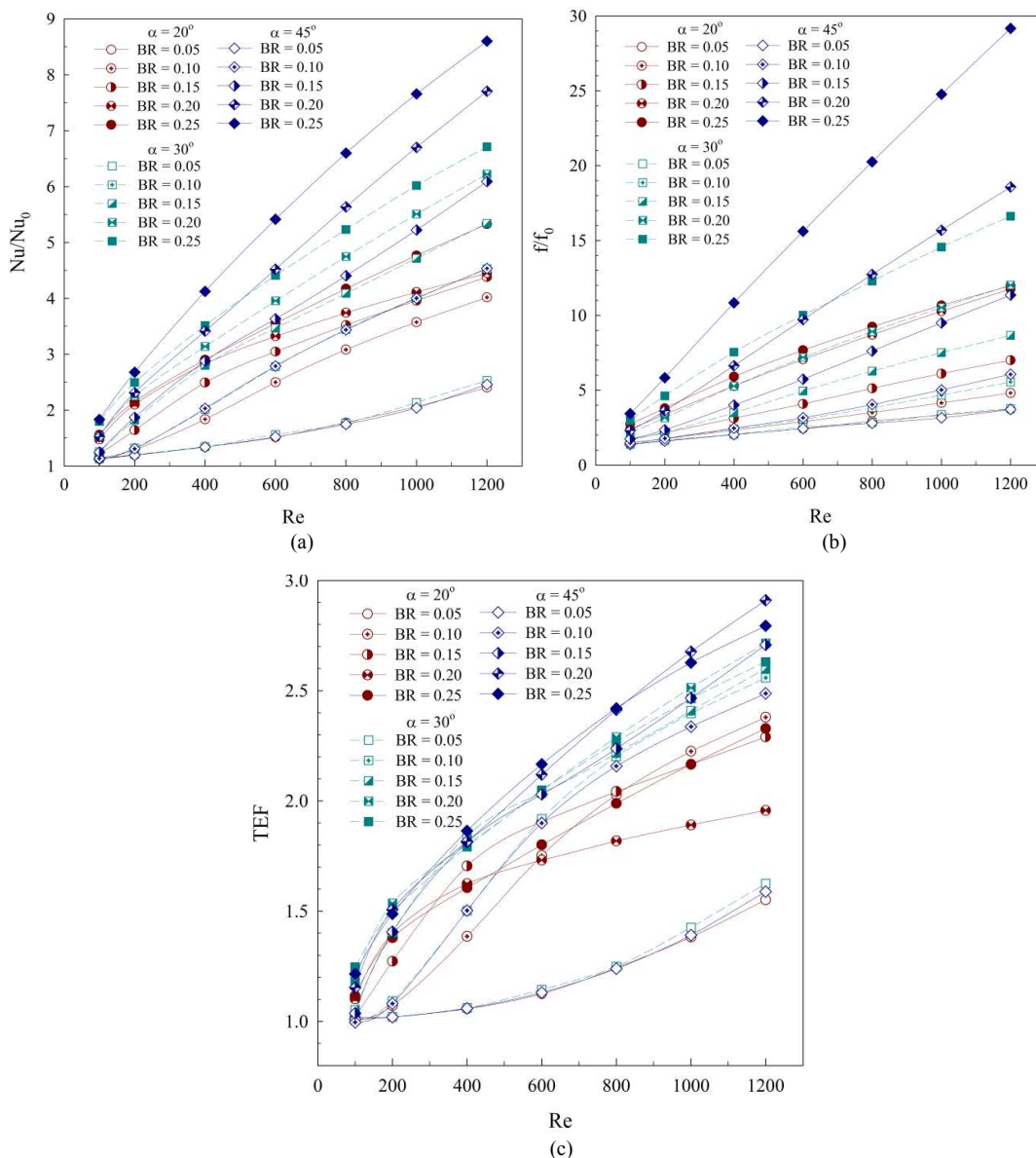

Fig. 6. Performance analysis: (a) Nu/Nu_0 Vs Re, (b) f/f_0 Vs Re and (c) TEF Vs Re

Considering at similar blockage ratio, the enhancements of heat transfer rate and friction loss are detected when increasing the flow attack angle of the U-shaped baffle, except for $BR = 0.05$. The 45° U-shaped baffle gives the highest heat transfer rate and pressure loss, while the 20° U-shaped baffle provides the reverse result. The values on heat transfer rate and friction loss at $BR = 0.05$ for all flow attack angles are closely found. In range studies, the heat transfer rate and friction loss are around 1–8.7 and 1–29 times above the smooth tube, respectively, depended on flow attack angle, blockage ratio and Reynolds number.

The addition of the U-shaped baffle in the circular tube heat exchanger enhances thermal enhancement factor higher than the smooth tube in all cases ($TEF > 1$). The optimum thermal enhancement factor is around 2.9 at $BR = 0.2$ and $\alpha = 45°$ when considering at the highest Reynolds number, $Re = 1200$. In range investigates, the thermal enhancement factor is found to be around 1–2.9 depended on the blockage ratio, flow attack angle and Reynolds number.

Conclusion

The article presents the numerical studies on flow and heat transfer characteristics in a circular tube heat exchanger inserted with the U-shaped baffle. The effects of height and flow attack angle for the U-shaped baffle on heat transfer, pressure loss and thermal performance are considered at laminar regime, $Re = 100$-1200. The major findings are concluded as follows:

- The insertion of the U-shaped baffle can improve heat transfer rate and performance in the test section due to the vortex flow and impinging flow which created by the baffle
- The vortex strength increment is due to the augmentations on height and flow attack angle for the U-shaped baffle. The high vortex strength leads to higher heat transfer rate and pressure loss
- The heat transfer rate and friction loss are around 1–8.7 and 1–29 times over the smooth circular tube, respectively, when inserted the U-shaped baffle in the circular tube. The optimum thermal enhancement factor is detected at 45° U-shaped baffle with the $BR = 0.2$ around 2.9 at $Re = 1200$

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Author's Contributions

First Author: Designed the research plan and organized the investigation.

Second Author: Participated in all experiments and contributed to the writing of the manuscript.

Ethics

This article is original and contains unpublished material. The corresponding author confirms that all of the other authors have read and approved the manuscript and no ethical issues involved.

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Nomenclature

BR	flow blockage ratio ($=b/D$)
b	baffle height, m
D	diameter of tube
f	friction factor
h	convective heat transfer coefficient, W m^{-2} K^{-1}
k	thermal conductivity, W m^{-1} K^{-1}
Nu	Nusselt number ($=hD/k$)
p	static pressure, Pa
Pr	Prandtl number ($Pr = 0.707$)
Re	Reynolds number
T	temperature, K
u_i	velocity in x_i-direction, m s^{-1}
$\bar{u}$	mean velocity in channel, m s^{-1}

Greek letter

α	angle of attack, degree
TEF	thermal enhancement factor ($= (Nu/Nu_0)/(f/f_0)^{1/3}$)
ρ	density, kg m^{-3}

Subscript

in	inlet
0	smooth tube
pp	pumping power