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Thermo-hydraulic Performance of a Circular Tube Heat Exchanger with Combined Turbulators

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ABSTRACT— ***This article presents numerical investigations on heat transfer behaviors, flow configurations and thermal performance assessments in a circular tube heat exchanger with combined turbulators. The combined turbulators; wavy plate with 30^o V–baffle, are selected to enhance the heat transfer rate in the tube heat exchanger. The effects of the blockage ratios; b/D = BR, and flow directions; V–tip pointing downstream (called V–Downstream) and V–tip pointing upstream (called V–Upstream), for the baffles are investigated for the Reynolds number based on the diameter of the test tube, Re = 100 – 1200. The numerical results are presented in terms of flow configurations, heat transfer characteristics and performance evaluations and also compared with the smooth tube. It is found that the use of the combined turbulators can help to improve the thermal performance in the heating system due to the longitudinal vortex flows and the disturbance of the thermal boundary layer. In addition, the maximum TEF is found to be around 2.3 at the highest Reynolds number for the combined turbulators with BR = 0.20 and V-tip pointing downstream.***

Keywords— heat exchanger, combined turbulators, thermal performance, V-baffle, wavy surface

1. INTRODUCTION

Many types of turbulators or vortex generators have been used in heat exchangers to improve the heat transfer rate and thermal performance. The roughness surface, rib, baffle, groove are types of the turbulators, which are inserted into the heating channel or tube to create the vortex flows and longitudinal vortex flows through the test tube. Various types of the turbulators result in the differences on the flow configuration and heat transfer behavior. The selection for the type of the turbulators depended on the application of the heat exchanger. The investigations on the thermal performance in the tube or channel heat exchanger inserted with turbulators had been extensively reported on both numerical and experimental methods.

The V-shaped configuration of the turbulators are selected to improve the heat transfer rate and thermal performance for the channel or tube heat exchanger [1-12]. They reported that the V-shaped turbulators give higher thermal efficiency and heat transfer rate when compared with the smooth channel. Except for the V-types turbulators, the wavy surface is extensively used in the fin-and-tube heat exchanger and ther other types of the heat exchnager to enhance thermo-hydraulic performance of the heating system [13-24]. They found that the wavy surface can help to enhance the strength of the vortex flow that are reason for heat transfer augmentation.

As the literature reviews above, it is found that the use of the V–shaped baffle gives more efficiency than the other shaped baffles. The wavy surface is another type of the turbulators, which always inserts in the fin-and-tube heat exchamger. The wavy surface can enhance the heat transfer rate with moderate pressure loss penalty. In the present work, the combined turbulators between the wavy surface and V-baffle are selected to augment the heat transfer rate and thermal performance. The influences of the flow blockage ratios; *BR* = 0.05 – 0.20, flow directions; V–Downstream and V–Upstream, and the Reynolds numbers; *Re* = 100 – 1200, are investigated numerically. The combined turbulators with V–Downstream (*CVD*), combined turbulators with V–Upstream (*CVU*), single V–Downstream baffle (*VD*), single V–Upstream baffle (*VU*) and single wavy surface are inserted in the middle of the test tube with inline arrangement. The combined turbulators in the tube heat exchanger may generate stronger and longer vortex flows than the single turbulator.

2. COMPUTATIONAL MODEL AND NUMERICAL METHOD

2.1 Computational domain

Fig. 1 shows a circular tube heat exchanger with combined turbulators; wavy surface with V–baffle. The fully developed periodic flow and heat transfer are set for the inlet and outlet of the computational model. The air enters the circular tube at an inlet temperature, T_{in}, and flows over the turbulators. The baffle height is represented by *b*, while the tube diameter, *D*, is set equal to 0.05 m, *b*/*D* is known as the blockage ratio or *BR*. The axial pitch or distance between the V–baffle cells, *L*, is equal to *D* and *L*/*D* is defined as the pitch spacing ratio, *PR*. The grid cell around 120000 is selected for the current investigation due to the increasing grid cell form 120,000 to 240,000 has no effects on the Nusselt number and friction factor.

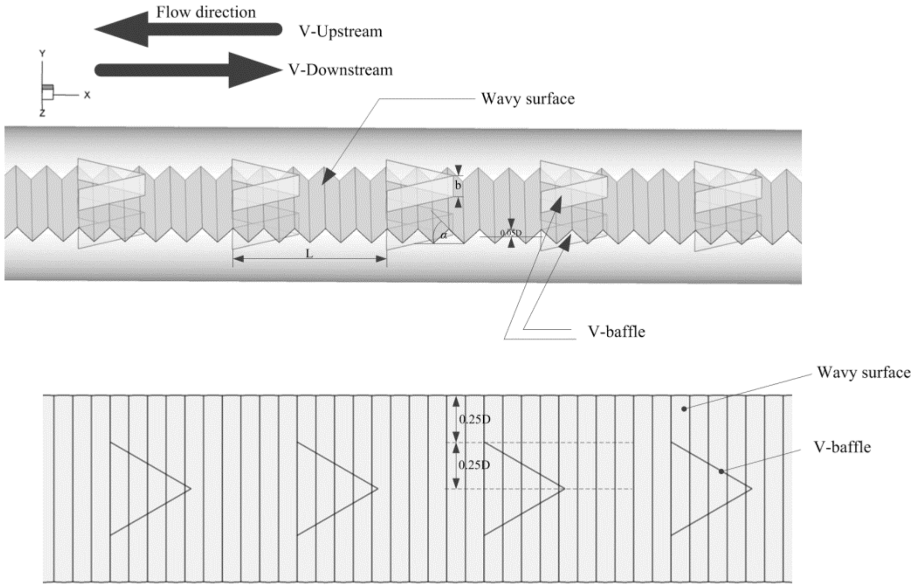


Figure 1 : Combined turbulators inserted in a circular tube.

2.2 Boundary conditions

Periodic boundaries are set for the inlet and outlet of the numerical domain. The test fluid is air with 300K. The physical properties of the air have been assumed to remain constant at average bulk temperature. Impermeable boundary and no–slip wall conditions have been implemented over the tube wall as well as the V–baffle and the wavy surface. The constant temperature of the circular tube wall is maintained at 310 K, while the V–baffle and wavy surface are assumed as adiabatic wall conditions (insulator).

2.3 Numerical models

The assumptions of the current investigation are as follows;

- 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.

3. MATHEMATICAL FOUNDATIONS

The numerical model is solved by the Navier–Stokes equations and energy equation. The Reynolds number, friction factor, Nusselt number and thermal performance enhancement factor are important parameters, which are calculated as follows;

$$Re = \rho \bar{u} D / \mu \quad (1)$$

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

$$Nu_x = \frac{h_x D}{k_a} \quad (3)$$

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

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

The numerical method for the present investigation is referred from *Ref.* [25].

4. RESULTS AND DISCUSSION

4.1 Validation of the smooth tube

The presented values are compared with the values form the correlations of the smooth circular tube with no turbulators at *Re* = 100 – 1200. The correlations of the Nusselt number and friction loss are displayed as equations 6 and 7, respectively.

$$Nu = 3.66 \tag{6}$$

$$f = 64/\mathrm{Re} \tag{7}$$

The results show agree well within ±0.06% and ±0.1% for the Nusselt number and friction loss, respectively. Therefore, the computational domain can assist to predict the behaviors in the tube heat exchanger.

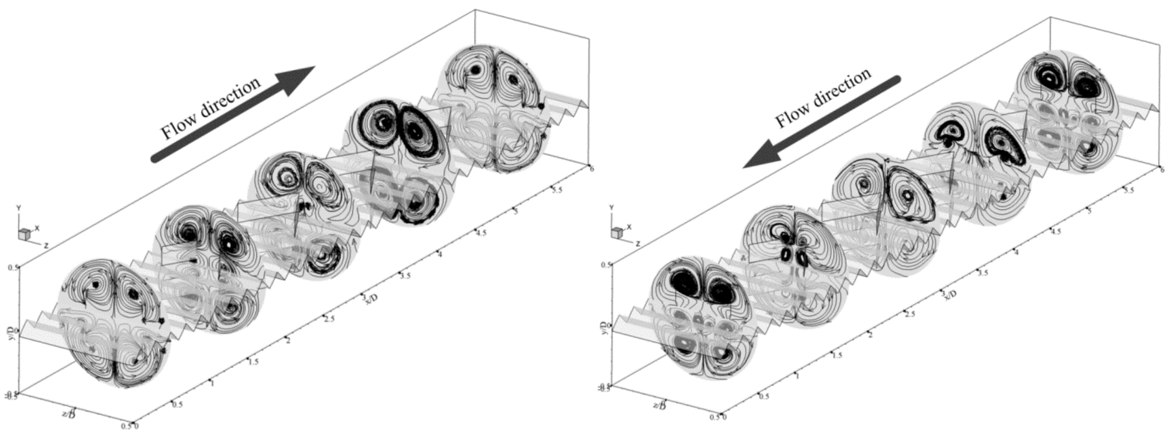
4.2 Flow configuration

The flow configurations in the tube are shown in two patterns; streamlines in transverse planes and the longitudinal vortex flows through the test tube. In general, the combined turbulators (wavy surface with V–baffle) and the single turbulator (only V–baffle) can produce the vortex flows and the longitudinal swirling flows over the test tube.

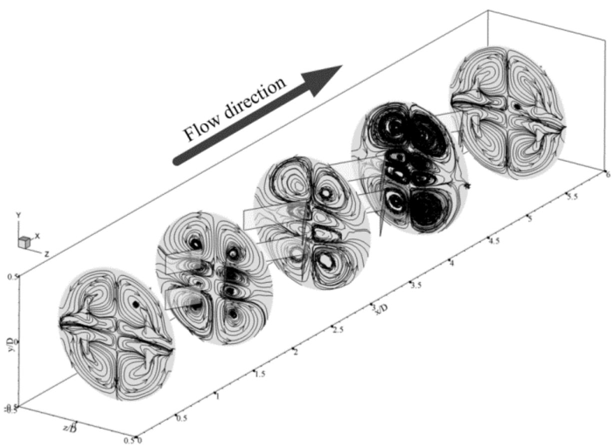
Figs. 2*a* and *b* present the streamlines in transverse planes for the *CVD* and *CVU* at *BR* = 0.15 and *Re* = 1000, respectively, while the single turbulators; *VD* and *VU*, are displayed as Figs. 2*c* and *d*, respectively. As the figures, all turbulators can generate four main vortex flows through the test tube. Considering at the lower part of the planes, the *CVD* and *VD* give counter rotating flows with common-flow-down, while the *CVU* and *VU* create counter rotating flows with common-flow-up. The differences of the flow structure leads to the difference of the heat transfer characteristics over the tube surface.

Figs. 3*a*, *b*, *c* and *d* illustrate the longitudinal vortex flows for the *CVD*, *CVU*, *VD* and *VU*, respectively, at *Re* = 1000 and *BR* = 0.15. The longitudinal vortex flows of the *CVD* and *VD* are found to be similar. The fluid flows impinge at the leading edges of the V–baffle and flow up to the upper part of the test tube and then the flows slide down to the left and right parts of the tube. This behavior appears repeatedly when the flows pass around six baffles (called "periodic flow profile"). The periodic flow profiles are found in cases of the *CVU* and *VU*. The air flows to the upper part of the test tube and then impinges at the V–tip of the baffle before slides to the next module.

In conclusion, the *CVD* and *CVU* give very close flow configurations as the *VD* and *VU*, respectively. The wavy surface has a few effect for the flow pattern. The vortex flows and the longitudinal vortex flow lead to a better fluid mixing and also reduce the thermal boundary layer. These phenomena are an important reason for the heat transfer augmentation.



(a) (b)



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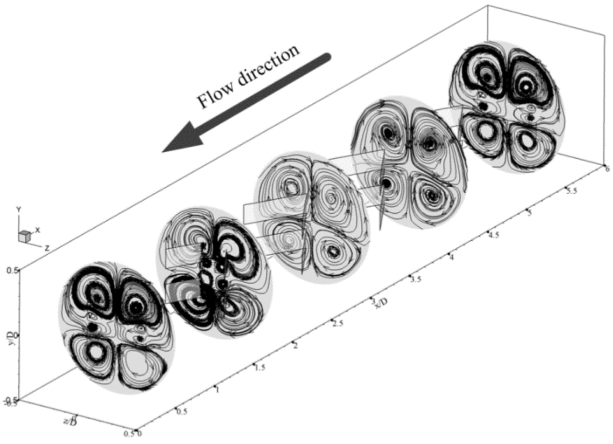
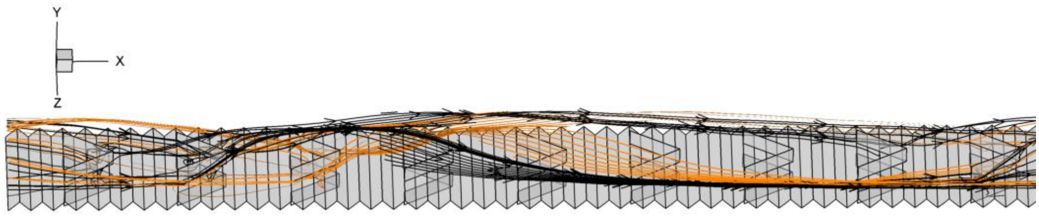
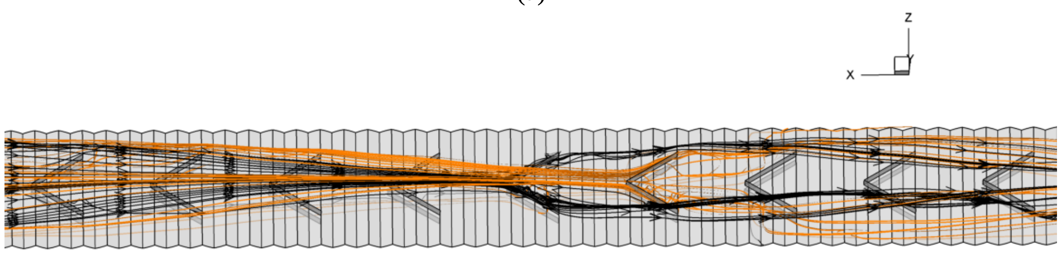


(d)

Figure 2 : Streamlines in transverse planes for (a) *CVD*, (b) *CVU*, (c) *VD* and (d) *VU* at *Re* = 1000 and *BR* = 0.15.



(a)



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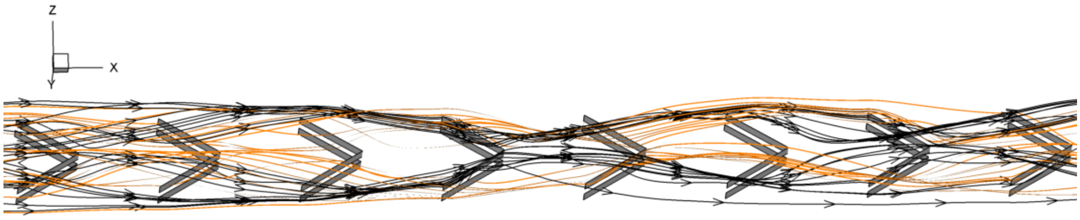
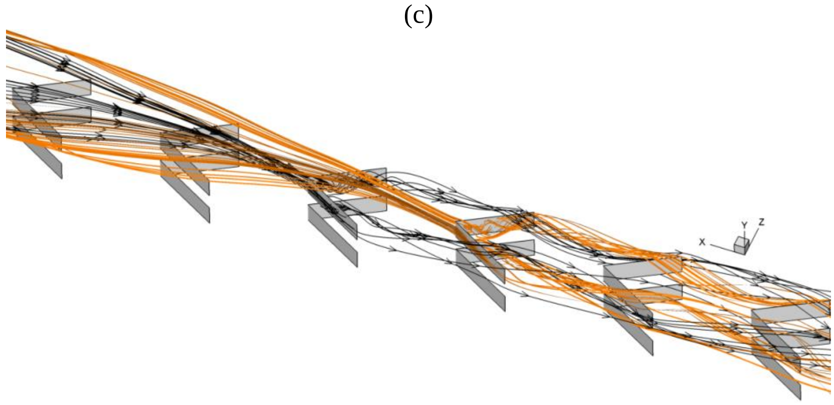


(c)



(d)

Figure 3 : Longitudinal vortex flows for (a) *CVD*, (b) *CVU*, (c) *VD* and (d) *VU* at *Re* = 1000 and *BR* = 0.15.

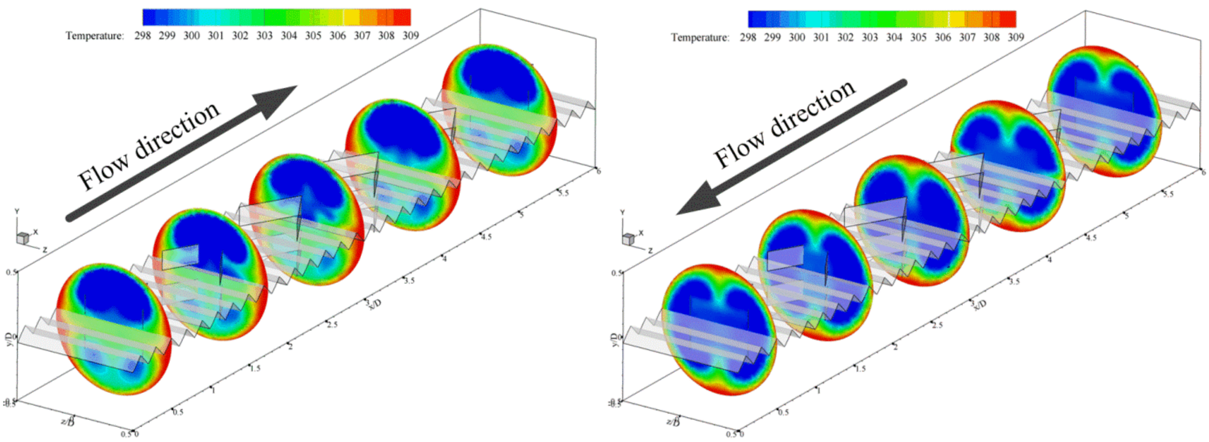
4.3 Heat transfer characteristic

The heat transfer behaviors are displayed in terms of temperature distributions in transverse planes and the Nusselt number distributions on the tube wall. The contours temperature in transverse planes can describe the better mixing of the fluid flow and structure of the thermal boundary layer. The contours Nusselt number can define the peak of the heat transfer regime on the tube wall.

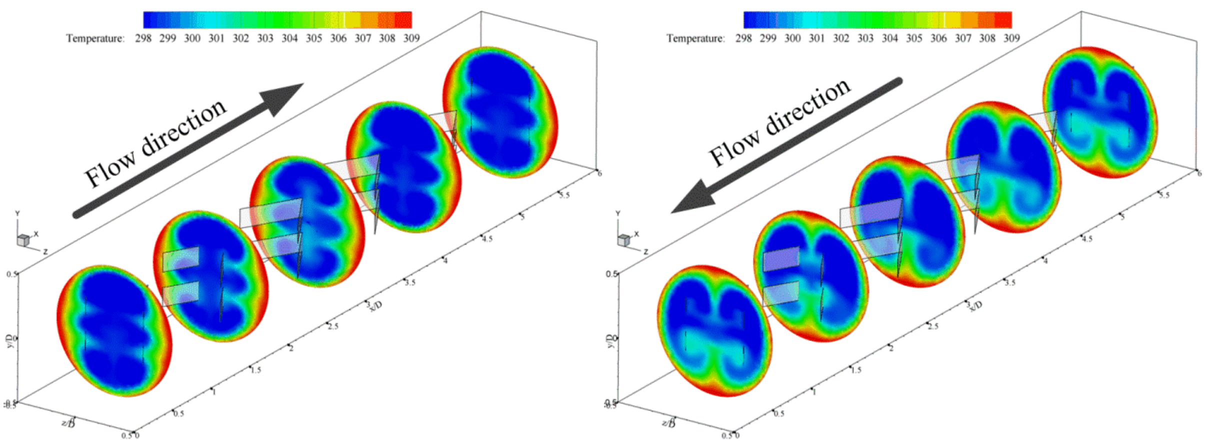
Figs. 4*a*, *b*, *c* and *d* show the temperature contours in transverse planes for the *CVD*, *CVU*, *VD* and *VU*, respectively, at *Re* = 1000 and *BR* = 0.15, while the temperature contours in transverse planes for the single wavy surface is displayed as Fig. 4*e*. It is seen in the figure that the single wavy surface gives the contour temperature close to the smooth tube with no turbulators. The single wavy surface in the tube heat exchanger slightly helps the fluid mixing. Considering at the color layer, the red layer (hot fluid) is found in large area for all planes of the tube inserted with the wavy surface. This means that the wavy surface can generate the weak vortex flow over the test tube.

For the *CVD*, *CVU*, *VD* and *VU*, the turbulators can help to improve the behavior of the fluid mixing. Considering at the contours temperature, the thin layer of the hot fluid (disturbing the thermal boundary layer) is found at the upper-lower parts of the *CVD* and *VD* and at the left-right parts of the *CVU* and *VU*. The reason of this may be that the difference of the flow configurations; common-flow-down and common-flow-up, effects for the difference of the impinging area.

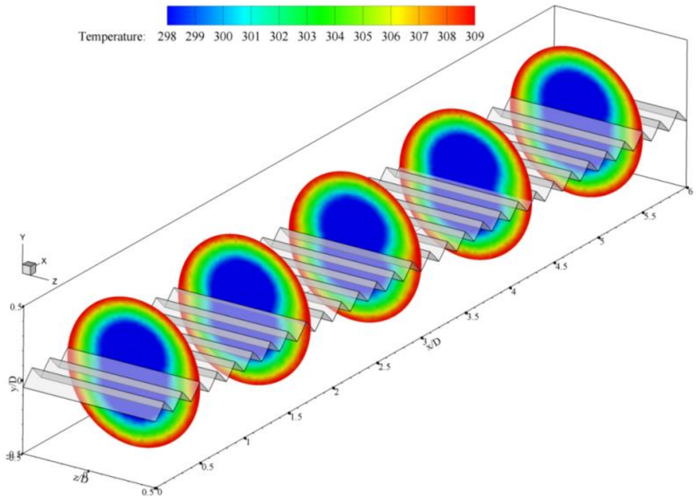
Figs. 5*a*, *b*, *c* and *d* present the Nusselt number distributions on the tube surface for the *CVD*, *CVU*, *VD* and *VU*, respectively, at *Re* = 1000 and *BR* = 0.15, while the Nusselt number distributions on the tube wall for the wavy surface are shown in the Fig. 5*e*. The Nusselt number distributions for the smooth tube with no turbulators are shown with blue layer (Nu = 3.66). As the figures, the use of the turbulators leads to higher heat transfer rate than the smooth circular tube for all cases. The peaks of the heat transfer area are found at the upper-lower parts for *CVD* and *VD*, while the *CVU* and *VU* give the highest heat transfer region on the left-right parts of the test tube. The V–Downstream give higher heat transfer rate than the V–Upstream on both the combined turbulators and single turbulator.



(a) (b)

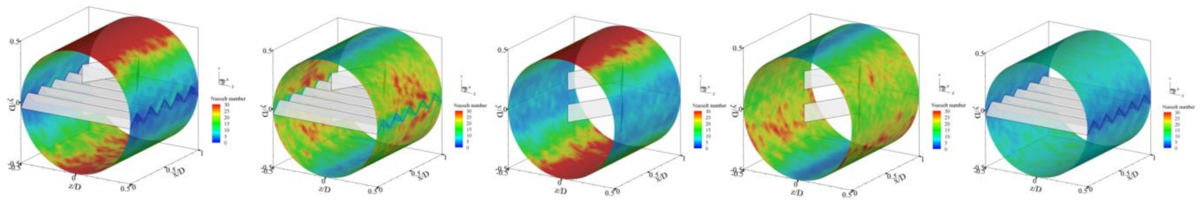


(c) (d)



(e)

Figure 4 : Contours temperature in transverse planes for (a) *CVD*, (b) *CVU*, (c) *VD*, (d) *VU* and (e) wavy surface at *Re* = 1000 and *BR* = 0.15.



(a) (b) (c) (d) (e)

Figure 5 : Contours Nusselt number for (a) *CVD*, (b) *CVU*, (c) *VD*, (d) *VU* and (e) wavy surface at *Re* = 1000 and *BR* = 0.15.

4.4 Performance assessment

The variations of the Nu/Nu_0 with the Reynolds number at various cases are shown in the Figs. 6*a* and *b* for the combined turbulators and single turbulators, respectively, at *BR* = 0.05 – 0.20 and *PR* = 1.0. In general, the Nu/Nu_0 tends to increase with the rise of the Reynolds number and the blockage ratio in all cases. The *BR* = 0.20 gives the highest heat transfer rate, while the *BR* = 0.05 performs the reverse results.

For combined tabulators, the *CVD* performs higher heat transfer rate than the *CVU* for all *BR* values. At *Re* = 1200, the *CVD* gives the highest Nusselt number ratio around 7.3, 5.8, 4.5 and 3.3 for *BR* = 0.20, 0.15, 0.10 and 0.05, respectively, while the *CVU* provides around 6.2, 5.2, 4.3 and 3, respectively.

For *CVD* at *Re* = 1200, the *BR* = 0.20 gives higher heat transfer rate than the *BR* = 0.15, 0.10 and 0.05 around 20.55 %, 38.36 % and 54.79 %, respectively. The *BR* = 0.15 provides higher heat transfer rate around 22.41 % and 43.10 % than the *BR* = 0.10 and 0.05, respectively, while the *BR* = 0.05 performs the Nusselt number around 26.67% lower than the *BR* = 0.10.

For *CVU* at *Re* = 1200, the *BR* = 0.20 gives higher heat transfer rate than the *BR* = 0.15, 0.10 and 0.05 around 16.13%, 30.65% and 51.61%, respectively. The *BR* = 0.15 performs higher heat transfer rate around 17.31% and 42.31% than the *BR* = 0.10 and 0.05, respectively, while the *BR* = 0.10 provides higher Nusselt number ratio than the *BR* = 0.05 around 30.23%. In addition, the use of the combined turbulators gives the Nusselt number around 1.5 – 7.3 times higher than the smooth circular tube with no turbulators for *Re* = 100 – 1200 and *BR* = 0.05 – 0.20.

The Nu/Nu_0 of the single wavy surface slightly increases when increasing the Reynolds number. The maximum Nusselt number of the wavy surface is found to be around 2.0 times higher than the smooth tube. For single baffle, the *VD* performs higher heat transfer rate than the *VU* for all the *BR*s around 10%. The optimum friction factor ratios are around 6.1, 5.2, 4.7 and 3.7 for the *VD* at *BR* = 0.20, 0.15, 0.10 and 0.05, respectively, while the *VU* gives the friction factor ratio around 5.5, 4.8, 4.2 and 3.5, respectively, for *BR* = 0.20, 0.15, 0.10 and 0.05. For *VD*, the *BR* = 0.20 provides around 14.75%, 22.95% and 39.34% higher than the *BR* = 0.15, 0.10 and 0.05, respectively, on the heat transfer rate. For *VU*, the *BR* = 0.20 gives the heat transfer rate higher than the *BR* = 0.15, 0.10 and 0.5 around 12.72%, 23.63% and 36.36%, respectively. In the range investigates, the use of the single turbulators provides the Nusselt number around 1.5 – 6.2 times in comparison with the smooth tube.

In comparison, it is found that the Nusselt number for the combined turbulators is higher than the single turbulators when *BR* > 0.1. At the *BR* = 0.05 and 0.1, the combined turbulators and single turbulator give nearly values of the Nusselt number for all *Re* values. The reason of this may be that the assist flow from the wavy surface and baffle has high effectiveness at the *BR* = 0.15 and 0.20.

Figs. 7*a* and *b* present the variations of the friction factor ratio, f/f_0, for the combined turbulators and single turbulators, respectively, at various *BR*s and flow directions. In general, the turbulators give higher friction loss than the smooth tube with no turbulators. The f/f_0 for all cases increases when increasing the Reynolds numbers and *BR* values. The *BR* = 0.20 provides the highest values of the friction loss, while the *BR* = 0.05 shows the reverse results.

The f/f_0 values for the *CVD* and *CVU* are seen to be nearly values for all *BR*s. The friction factors of the *BR* = 0.20, 0.15, 0.10 and 0.05 are around 32, 23, 16 and 11 times higher than the smooth tube, respectively. The *BR* = 0.20 performs the higher friction factor than the *BR* = 0.15, 0.10 and 0.05 around 28.13%, 50% and 65.63%, respectively. Additionally, the use of the combined turbulators gives the friction loss around 4 – 32 times higher than the smooth tube.

The friction factor ratios of the *VD* and *VU* are seen to be nearly values for *BR* = 0.05, 0.10 and 0.15. The *VD* performs slightly higher friction factor ratio than the *VU* at *BR* = 0.20. The single wavy surface shows the friction factor ratio close to the V–baffle at *BR* = 0.15. The f/f_0 of the *BR* = 0.05, 0.10 and 0.15 is around 5, 8 and 14, respectively, at the highest Reynolds number. At *BR* = 0.20 and *Re* = 1200, the f/f_0 is around 21 and 19 for *VD* and *VU*, respectively. In range studies, the friction loss appears around 3 – 21 times higher than the smooth tube for the single baffle. In addition, the combined turbulators perform higher friction factor than the single turbulators around 25.00 – 34.38%.

Figs. 8*a* and *b* present the *TEF* for the combined turbulators and single turbulators, respectively. The *TEF* of the smooth tube with no turbulators is equal to 1 (dash line). In general, the *TEF* tends to increase with the rise of the Reynolds number for all turbulators. The use of the single wavy surface provides lower thermal performance than the smooth tube (almost points are lower than 1), while the use of the combined turbulators and single V–baffle gives higher *TEF* than the smooth tube. The *TEF* of the *CVD* is higher than the *CVU* for all *BR*s. In range investigates, the *TEF* is found to be around 0.9 – 2.3 depended on *BR*, *Re* and turbulators configuration. The optimum *TEF* is around 2.3 for the *CVD* at *BR* = 0.20 and *VD* at *BR* = 0.10.

5. CONCLUSION

Numerical investigations on flow configurations, heat transfer behaviors and thermal performances in a circular tube with the combined and single turbulators are presented for the Reynolds number, *Re* = 100 – 1200. The combined turbulators; wavy surface with V–baffle, and single turbulators; wavy surface and V–baffle, are inserted in the middle of the circular tube. The influences of the blockage ratios; *BR* = 0.05 – 0.20, and flow directions; V–Downstream and V–Upstream, are investigated. The main findings are as follows;

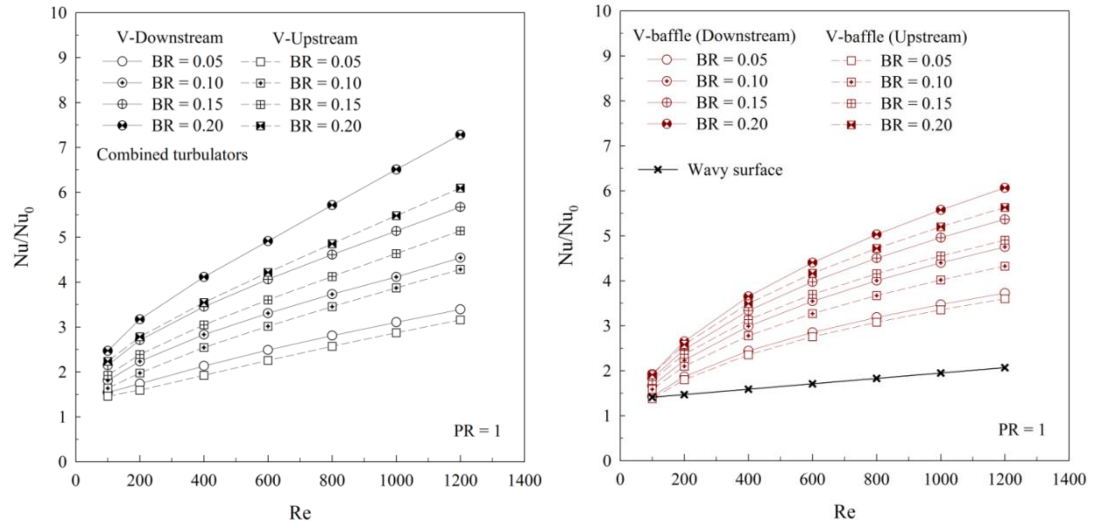
The improvements of the heat transfer rate and thermal performance in the tube with turbulators are due to the better mixing of the air flow and disturbing thermal boundary layer. The enhancements of the Reynolds number and blockage ratio result in the rise of the vortex intensity or vortex strength.

The four main vortex flows are created by the combined turbulators and single V–baffle, but the vortex flow is not found in case of the single wavy surface. The counter rotating flows with common-flow-down and common-flow-up are found in the case of V–Downstream and V–Upstream, respectively, for both the combined and single V–baffle turbulators.

The difference of the flow structure leads to the difference of the heat transfer behavior. The peak of heat transfer area is found at the upper-lower parts and left-right parts for the V–Downtream and V–Upstream, respectively.

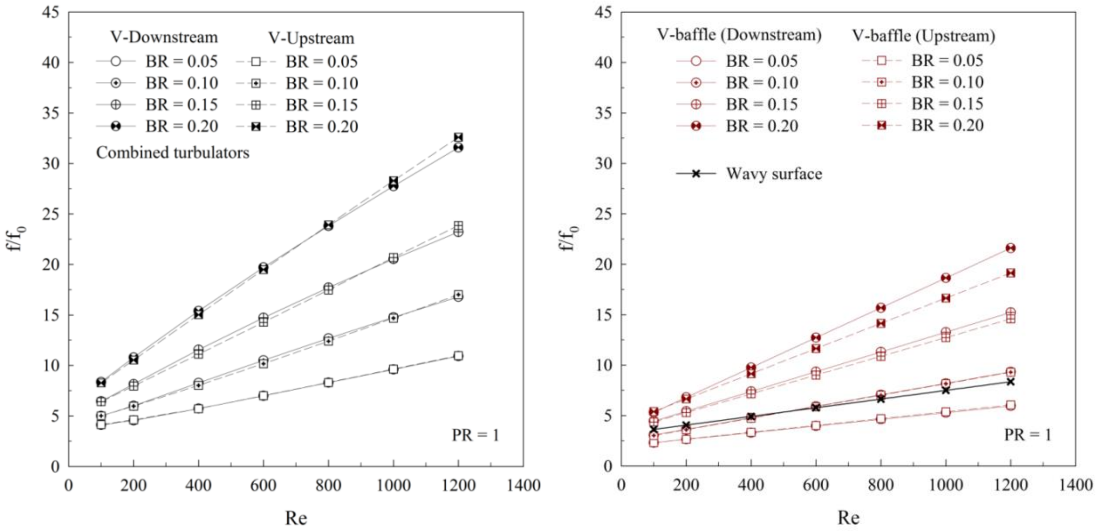
The V–Downstream performs higher heat transfer rate than the V–Upstream on both turbulators. The combined turbulators provide higher heat transfer rate than the single turbulators at high blockage ratio, BR = 0.15 and 0.20. The Nu/Nu_0 is around 1.5 – 7.3 and 1.5 – 6.2 for the combined turbulators and single turbulators, respectively, at *Re* = 100 – 1200 and *BR* = 0.05 – 0.20.

The thermal enhancement factor is around 0.9 – 2.3 and 1 – 2.3 for the combined turbulators and single turbulators, respectively, at *Re* = 100 – 1200 and *BR* = 0.05 – 0.20. The optimum point appears at *BR* = 0.20 and 0.10 for the *CVD* and *VD*, respectively.



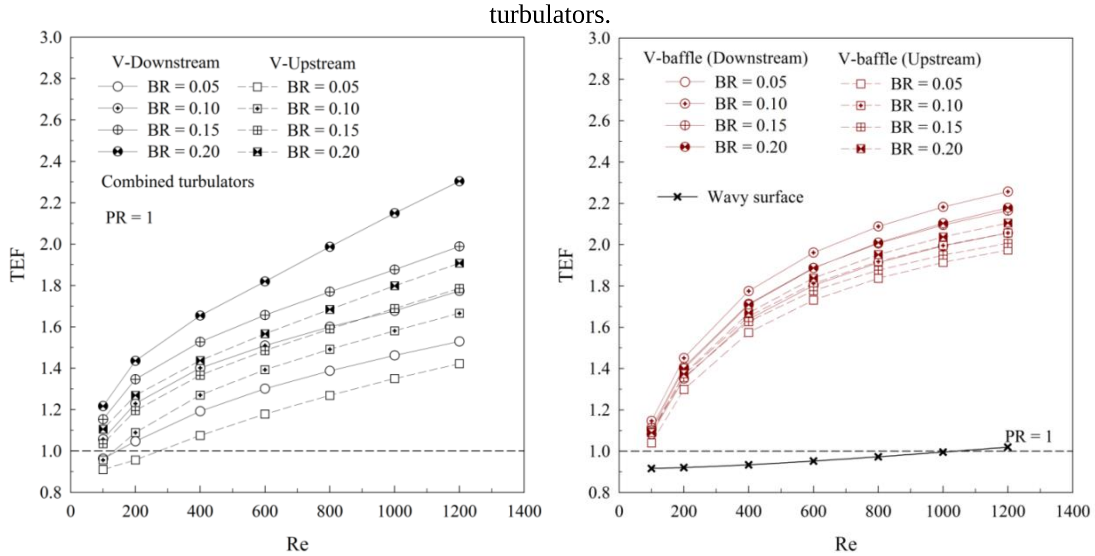
(a) (b)

Figure 6 : Variation of the Nusset number ratio with the Reynolds number for (a) combined turbulators and (b) single turbulators.



(a) (b)

Figure 7 : Variation of the friction factor ratio with the Reynolds number for (a) combined turbulators and (b) single turbulators.



(a) (b)

Figure 8 : Variation of the thermal exhancement factor with the Reynolds number for (a) combined turbulators and (b) single turbulators.

NOMENCLATURE

BR	flow blockage ratio, (b/D)
b	baffle height, m
D	diameter of the tube
f	friction factor
GCI	grid convergence index
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
PR	pitch or spacing ratio, L/D
Re	Reynolds number,
T	temperature, K
TEF	thermal enhancement factor, $(=(Nu/Nu_0)/(f/f_0)^{1/3})$
u_i	velocity in xi-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
α	baffle inclination angle or angle of attack, degree
ρ	density, kg m^{-3}

Subscript

in	inlet
0	smooth tube
w	wall
pp	pumping power

6. ACKNOWLEDGEMENT

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