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STUDY ON HEAT TRANSFER AUGMENTATION IN AN AIR HEATER USING RECTANGULAR WAVY FIN TURBULATORS

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ABSTRACT

In the present study, the use of wavy fin turbulators on the annulus body of a double pipe air heater has been numerically investigated. The inner pipe consists of hot water whereas the annular section consists of cold air whose Reynolds Number (Re) ranged from 3000-15,000. Rectangular cross-sectioned wavy fin turbulators with various curvature ratios of 2, 3, 5, and 7.5 is numerically simulated to investigate the influence of curvature effects on turbulence. Results have been compared with the bare pipe and with rectangular straight fins. It is seen that wavy fin turbulators perform better (compared to (other two) bare and rectangular straight fin) than both of them by creating turbulence effects in the flow field. Nusselt number (Nu) is increased by 256% and 51% as compared to bare and rectangular straight fin type. Parametric studies revealed that as the curvature ratio (CR) is reduced, the turbulent intensity and turbulent kinetic energy increase with better heat transfer characteristics. The highest Thermo hydraulic performance index of 4.8 is noticed for the CR of 2 at a Re of 6000 and is 34% higher than the straight rectangular fin.

Keywords: *Wavy fin, curvature ratio, air heater, Nusselt number, Thermo hydraulic performance index.*

1. INTRODUCTION

Air heaters are devices that find wide applications in domestic as well as industries. In the HVAC systems, they serve as room heaters by heating the air to the required temperature to maintain thermal comfort. It finds application in various kinds of industries such as automobile, chemical, textile, refineries, power plants, and also electronic industries. Industrial applications may include drying, preheating for other chemical processes, process heating, etc. Based on the type of energy interaction, they may be classified as parallel, counter-flow, and cross-flow types. A suitable type of configuration of the air heater can be designed based on the type of heat input. Heat input may be from the waste liquids or gases such as exhaust from the boilers, furnaces, internal combustion engines. Heat input may also be from renewable energy-based sources such as solar energy, geothermal, etc. Extensive studies have been investigated by different researchers in exploring the suitability of solar energy as the heat input to heat air which usually occurs in the crossflow mode (Abed, Majdi, & Habeeb, 2021; Ibrahim & Kasem, 2021; Tian & Zhao, 2013; Tyagi, Panwar, Rahim, & Kothari, 2012; Yadav & Thapak, 2014). The main advantage of these types is solar energy is renewable and also available free of cost. The drawback includes as solar energy is available only during the day hours, it is required to store this energy for use at times when there is no sunshine (Manjunath, Karanth, & Sharma, 2017). Due to the lower conversion efficiency into heat, the surface area required for the solar collectors will be huge. To increase efficiency, some special arrangements like turbulators have been used by various researchers in the flow passage (Gawande, Dhoble, Zodpe, & Chamoli, 2016; Gilani, Al-Kayiem, Woldemicheal, & Gilani, 2017; Promthaisong & Eiamsa-Ard, 2019; Saini & Verma, 2008). But it is observed that the increase in the efficiency by using these turbulators is found to be very marginal, which cannot be justified by the initial investment required for the air heater and solar collectors.

Out of the various types of heat exchangers, a double pipe heat exchanger is a simple type whose construction is very simple, less expensive, and maintenance-free operation. Hence it can be fabricated and installed easily and can be used for heating air, either for HVAC systems or any other industrial applications. The major disadvantage of a double pipe heat exchanger is the lower heat transfer coefficient for a given dimension. Especially when air is the media used it will be further lowered due to the lower Cp value as compared to other fluids such as water. Several techniques have been used to increase the thermal performance of a double pipe heat exchanger. These methods may be classified as active and passive types. The active method involves external power input to augment the heat transfer such as induced vibrations, application of electrostatic fields, pulsating flow etc. (Fukue et al., 2022). They are not attractive as they require external power. Passive procedures involve flow modification techniques that will disturb the flow field and hence may generate turbulence and secondary flows. Due to this higher heat transfer rate could be achieved (Eiamsa-Ard, Wongcharee, Eiamsa-Ard, & Thianpong, 2010; El Maakoul et al., 2017).

Air being lighter fluid, the flow field can be easily modified by using simple fins or inserts that could change the flow path. The combined effects of added surface area and turbulence generated due to the flow modification yield better heat transfer coefficient and heat transfer rate. Several investigators have used various geometrical inserts in the flow field such as twisted tapes, helical coils, helical tapes, etc. to augment the heat transfer (Jedsadaratanachai & Boonloi, 2021). A twisted tape is a simple device placed in the flow field and when the fluid flows through it, follows a helical path. As a result, continuous flow separation and mixing take place alternatively and thereby increasing the heat transfer

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characteristics (Lin & Wang, 2009; Sarada, Raju, Radha, & Sunder, 2010). Helically coiled tubes are another passive technique used by the researchers to modify the flow field. Here a helical helically coiled tube is placed in the axial direction of fluid flow. Fluid, during its motion, creates a swirling action thereby due to continuous mixing and secondary flow generation, higher transfer coefficients will be noticed (Chang, Gao, & Shih, 2015; Verma, Kumar, & Patil, 2018). Helical tape insert is another technique used for heat transfer enhancement wherein a helical tape insert is introduced in the flow field. Fluid follows a helical path resulting in constant flow mixing, leading to improved heat transfer characteristics (Eiamsa-Ard, Yongsiri, Nanan, & Thianpong, 2013; Sivashanmugam & Suresh, 2006). The literature review suggest that such flow alteration devices are very effective in the laminar regimes. As the flow becomes more and more turbulent, their action will be gradually reduced and are found to be ineffective for highly turbulent flows. Moreover, they are found to be useful for denser fluids such as water. As the density reduces, swirling action will be reduced, and hence secondary flow generation and turbulence reduces. This severely affects the thermal performance of such flow augmentation devices.

Surface or geometrical modification is another technique used by researchers to augment the heat transfer characteristics in a double pipe heat exchanger. Geometries of different shapes are placed either in the inner pipe or in the annulus so that flow alteration takes place when a low-density fluid such as air passes through such geometries. Together with an additional area, they also contribute to the generation of vortices due to recirculation. This increases the thermal performance characteristics.

Prasad and Kumar (Prasad & Kumar, 2013) conducted studies using triangular fins on the inner and annulus of a double pipe heat exchanger. They obtained higher effectiveness with fins as compared to the bare pipe. They concluded that the arrangement of fins in the annulus is better than placing them in the inner pipe. Nieckele and Soboga (Nieckele & Saboya, 2000) investigated the rectangular configuration of the fins in the annulus body. Their results predicted that Nu is increased by 200% for the above configuration with the penalty of higher pressure drop. Kumar et al. (Kumar, Dinesha, Narayanan, & Nanda, 2020) noticed hemispherical turbulators placed on the outer body of the inner pipe enhanced the secondary flow and hence the heat transfer characteristics as compared to the bare pipe. Parametric studies suggested that decreased pitch ratio and increased diameter gave better results. Hussein et al. (Hussein, 2015) used semi-circular baffle structures on the annulus side of the heat exchanger. They obtained improved thermal performance with a modified configuration. They also claimed lowering pitch resulted in higher performance. Flow modification by the circular fins on the annulus body of a double pipe heat exchanger was studied by Agarwal et al (Agrawal & Sengupta, 1990). The heat transfer rate was increased by 300% with a simultaneous increase in ΔP by 23% as compared to that of the plain tube. Promvonge et al. (Promvonge, 2010) used conical turbulators in the annulus along with a twisted tape insert in the inner pipe. Higher performance was noticed for this arrangement due to the flow modification in the inner and also in the annulus. Kumar et al. (Kumar, Dinesha, & Sai Krishna, 2019) used cylindrical turbulators of constant height and diameter on the circumference of the outer body of the inner pipe at varied pitches. Smaller pitch ratios gave higher Thermohydraulic performance index (THPI) as compared to the higher pitch values.

It is seen from the literature that geometric modifications serve dual advantages based on the type of structure and arrangement. Initially, they add on to the exposed area for heat transfer and secondly they cause higher turbulence due to the flow modification. This also depends on the type of the turbulator such as cylindrical, conical, hemispherical, etc., and also their position and pitch. The influence of rectangular straight fins on the heat transfer characteristics of a double pipe heat exchanger has been studied by researchers (Agrawal & Sengupta, 1990; Soliman & Feingold, 1977). They act as fins without influencing the flow field. The use of wavy fins not only increases the area of heat transfer but also acts as turbulators causing swirl motion and fluid recirculation. The use of such wavy fins and their influence on the performance of a double pipe heat exchanger has not been explored in the published literature. Parametric studies related to the use of wavy fin turbulators and their influence on the flow field modification are also limited. Hence a detailed numerical study has been conducted using the wavy type of rectangular fins to observe the effect on the flow modification and heat transfer characteristics. The curvature ratio (CR) is varied from 2 to 7.5 and their influence on the heat transfer characteristics has been discussed to optimize the CR giving maximum benefits.

2. GEOMETRY AND NUMERICAL MODELLING

A double pipe heat exchanger with an inner pipe diameter of 15 mm and thickness of 1 cm and an annulus diameter of 40 mm is considered for the study. Four wavy-shaped turbulators with the dimensions mentioned in Table 1 are constructed on the outer surface of the inner pipe in the annulus region. The curvature ratio (CR) is defined by the ratio of the radius of the fin to the axial horizontal distance. The geometrical details of the wavy turbulator are shown in Fig. 1. Accordingly, four different curvature ratios of 2, 3, 5, and 7.5 have been selected in the present study.

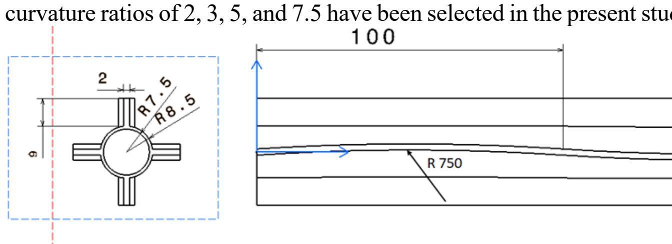

Fig. 1: Geometrical specifications of the wavy turbulator.

A bare pipe (without fins), straight finned, and wavy turbulator based double pipe heat exchanger is modeled using the Design Modeller tool of Ansys 2021 R2. Meshing is done by using tetrahedral elements. Figure 2 shows the meshed model of the wavy fin turbulator for a CR of 2. The circumferential separation of the wavy fins is taken as 90° which are maintained constant throughout the study. To capture the boundary effects inflation of 15 layers in the water domain and 15 layers in the air domain are given using the smooth transition technique. The meshed model of the bare, rectangular straight finned, and wavy model contains 1270250, 4133686, and 5158535 elements respectively. Hot water is the liquid in the inner pipe and air in the annulus region. At the inlet, the Re of water is maintained constant at 3000 and the temperature of 323 K. These conditions have been maintained constant throughout the study. Air Re is varied from 3000 to 15000 in increments of 3000 each time. The entry air temperature is kept at 293 K. Pressure outlet boundary condition is imposed at the outlet. The convergence limit for velocity is set to 10^{-3} in x, y, and z directions whereas for energy limit was set to 10-6. Kϵ turbulence model is used in the analysis. Conservation equations as shown by equations 1,2, and 3 are solved to obtain air outlet temperature for different Re.

Continuity Equation

$$\frac{\partial \bar{u}}{\partial x}+\frac{\partial \bar{v}}{\partial y}+\frac{\partial \bar{w}}{\partial z}=0 \tag{1}$$

Energy equation

$$\frac{\partial u_i T}{\partial x_i}=\rho\frac{\partial}{\partial x_i}\left\{\left(\frac{v}{Pr}+\frac{v_t}{Pr_t}\right)\frac{\partial T}{\partial x_i}\right\} \tag{2}$$

Momentum Equation

$$\frac{\partial u_i u_j}{\partial x_i}=-\frac{1}{\rho}\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left\{(v+v_t)\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\right\} \tag{3}$$

Table 1: Specifications of the heat exchanger

Specifications	*Conditions*
Inner Pipe Diameter	15mm
Outer Pipe Diameter	40mm
Inner Pipe Thickness	1mm
Length	1000 mm
Dimensions of the wavy turbulator	1mm×9 mm
Inner Pipe Diameter	15mm
Outer Pipe Diameter	40mm
Inner Pipe Thickness	1mm
Length	1000 mm
Dimensions of the wavy turbulator	1mm×9 mm
Inner Pipe Diameter	15mm
Outer Pipe Diameter	40mm
Inner Pipe Thickness	1mm

$$U_0 = \frac{1}{\frac{r_0}{h_i r_i} + \frac{1}{h_0} + ln\frac{r_o r_0}{r_i\ k}} \quad (9)$$

Inside heat transfer coefficient hi is calculated by using air velocity and fluid properties at the average temperature.

$$Re = \frac{\rho_i V_i d_i}{\Upsilon_i} \quad (10)$$

$$Nu_i = 0.023 Re^{0.8} Pr^{0.3} \quad (11)$$

Where D_i is the hydraulic diameter of the inner pipe.

$$N_{ui} = \frac{h_i d_i}{k_i} \quad (12)$$

Similarly, for the annulus region with the wavy fins, Re can be calculated by knowing the air velocity in the annulus. Knowing the values of U_o,

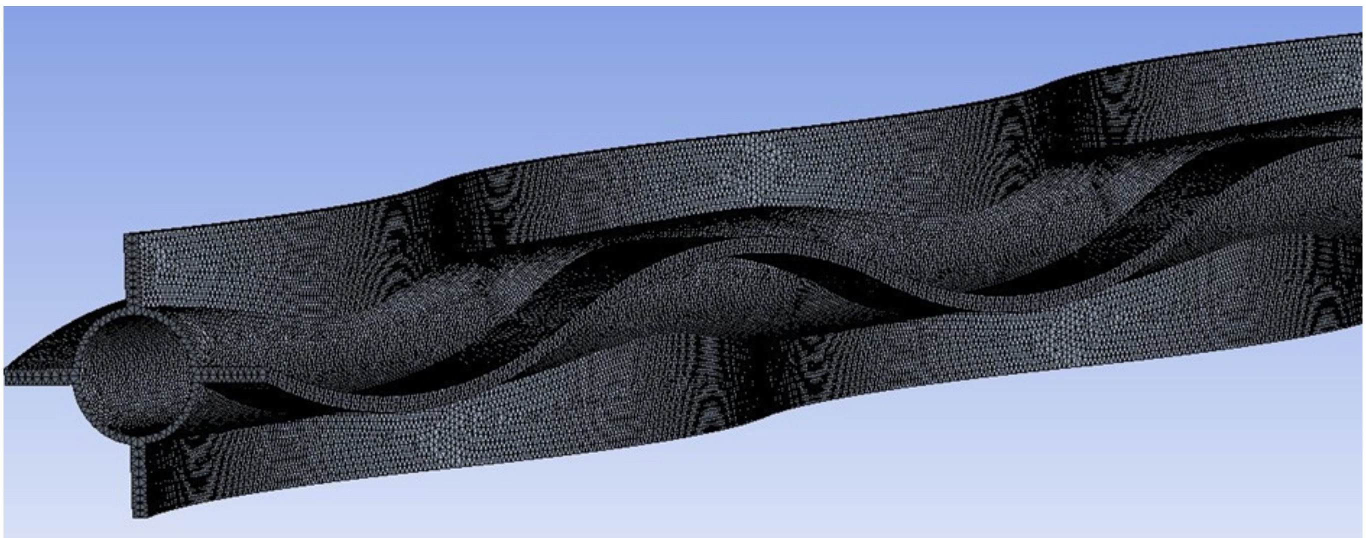

Fig. 2: Meshed model of the wavy fin turbulator (CR =2)

3. THEORITICAL BACKGROUND

The average Nusselt number (Nu) in the annulus is determined by knowing the overall heat transfer coefficient and effective temperature difference. The following set of calculations are performed to determine the N_{uo} and h_o . Similarly, the friction factor is calculated by using the values of pressure drop between the inlet and exit of the fluid in the annulus.

$$Q_h = m_h c_{ph}(T_{hi} - T_{ho}) \quad (4)$$

$$Q_c = m_c c_{pc}(T_{co} - T_{ci}) \quad (5)$$

$$Q_{avg} = Q = (Q_h + Q_c)/2 \quad (6)$$

$$Q_{avg} = U_o A_o LMTD \quad (7)$$

$$LMTD = \frac{(T_{hi} - T_{co}T_{co}) - (T_{ho} - T_{ci})}{\ln\left[\frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}\right]} \quad (8)$$

The average heat transfer rate is calculated by equation 6. By knowing the values of Q_{avg}, A_o, and LMTD, the overall heat transfer coefficient U_o can be calculated.

Overall heat transfer area based on the outer diameter is given by

h_i, k, outside heat transfer coefficient ho can be calculated.

$$R_{e\ air} = \frac{\rho_o V_o D_{air}}{\Upsilon_o} \quad (13)$$

Where D_{air} is the hydraulic diameter of the annulus given by

$$D_{air} = \frac{16LH}{L+H} \quad (14)$$

where L and H are the length and height of the fin respectively.

The above relations can be substituted in (9) outside heat transfer coefficient ho can be calculated. By substituting the value of ho in equation (15) , Nuo values can be obtained.

$$N_{uo} = \frac{h_0 D_{air}}{k_0} \quad (15)$$

Annulus pressure drop values are used to calculate the friction factor of the fluid in the annulus. It is calculated as

$$f = \frac{2\Delta P D_{air}}{\nu L V^2} \quad (16)$$

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where ΔP is the pressure drop between the entry and exit of the fluid in the annulus, and L is the length of the heat exchanger and V is the air velocity in the annulus section.

To optimize the number of elements of the wavy model, a grid independence study is undertaken considering the air outlet temperature as the parameter as shown in Fig. 3. Simulation has been conducted by varying the number of elements of the wavy model with CR 2 and Re_{air} 6000. When the mesh is course, the outlet temperature is lower, and it increases with the number of elements. It is found that by increasing the mesh elements from 515835 to 541456, the increase in the outlet temperature is very marginal. Hence 515835 is being considered as the optimum number of elements for further numerical simulation.

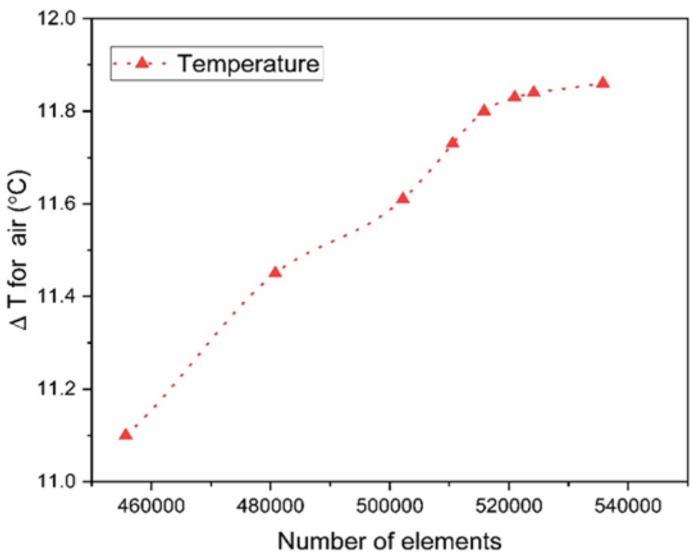

Fig. 3: Grid independence test for the wavy fin turbulator (CR =2, Re=6000)

4. VALIDATION

Airside Nu for the bare tube is determined by using the temperature difference for air noticed from the CFD values. The inlet mass flow rate of the hot water flowing in the inner pipe is kept constant at 0.028 kg/s. Whereas air side Re ranging from 3000 to 15000.They are compared with the values obtained by the Dittus bolter equation given by

$$Nu = 0.023Re^{0.8}Pr^{0.4} \tag{17}$$

where Nu is the Nusselt number, Re is Reynold's number and Pr is the Prandtl number.

These values are plotted against the simulated values and are shown in Fig. 4. It is found that the average difference between the analytical and simulated results was within 4.1% for Nu which was well within the limits from the literature. This indicates that the model is capable of predicting the results with high accuracy. Hence the developed model can be used for further analysis using wavy fin turbulators.

5. RESULTS AND DISCUSSION

Numerical simulation is performed for the wavy fin turbulator for different CR 2, 3, 5, and 7.5. Rewater, mass flow rate and temperature is kept constant at 2500, 0.028 kg/s and 333 K respectively. Re of the hot water is kept constant for the entire simulation. Cold fluid outlet temperatures are measured and the heat transfer characteristics such as Nuo and ho are calculated using the equations 4 - 13 as discussed in the previous section. The predicted values for the wavy-type turbulators are also compared with the values obtained for straight rectangular fins.

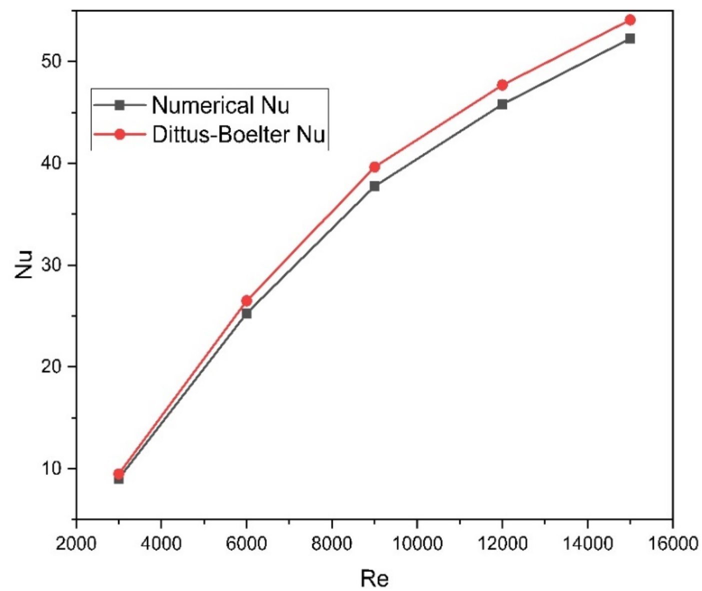

Fig. 4: Comparison between the CFD predicted and analytical values for Nu

5.1 Flow physics

Simulations have been performed by varying the CR values which is defined as the ratio of the curvature radius to the axial distance of the curvature along the flow length. Simulations have been carried out with CR ranging from 2 to 7.5. When the air passes through the curvature portion, the flow field will be disturbed. Flow mixing occurs leading to flow recirculations and turbulence and a sort of flow recirculation takes place. This increases the flow turbulence. Hence the bulk fluid velocity will be increased as seen from Fig. 5 colored by velocity values. It compares the velocity values plotted for CR =7 and CR=2. It is seen that for CR =2, the flow disturbance or flow mixing will be higher. This can be visualized with higher velocity values near the wavy construction as depicted in the Fig. 5. Recirculation or vortex formation can also be visualized from Fig. 6 which has been taken on an isoplane perpendicular to the flow direction. This reveals that the wavy nature of fins induces vortex creation on either side of the wavy fins, thus improving the overall flow pattern and hence average fluid bulk velocity.

Fig. 7 and 8 compare the temperature distribution contour along the length of the heat exchanger for the bare as well as with wavy fin turbulator. In a bare pipe, air passes along the length without undergoing any flow treatments thus limiting the energy transfer and subsequently lowering temperature rise as clearly seen from figure 7. Introducing a wavy fin in the annulus body will induce flow disturbance in the air path. Hence flow mixing takes place on passing through the continuous wavy type structures on either side. This phenomenon improves the flow turbulence and accordingly more amount of heat will be absorbed from the hot fluid. Hence in comparing Fig. 7 and 8, it is clear that introducing a wavy type of fin effectively increases the outlet temperature of the fluid.

On comparing the temperature contours for CR 7.5 and CR 2, it is also clear that the lower the CR, the higher will be the bulk temperature rise. As the CR is made smaller, the curvature part will be higher, which further enhances the fluid mixing and hence recirculation. This contributes to the higher energy transfer and hence as the fluid moves along the length of the heat exchanger, the bulk air temperature rises and is higher as compared to the higher CR. This effect of CR can be observed from Fig. 8.

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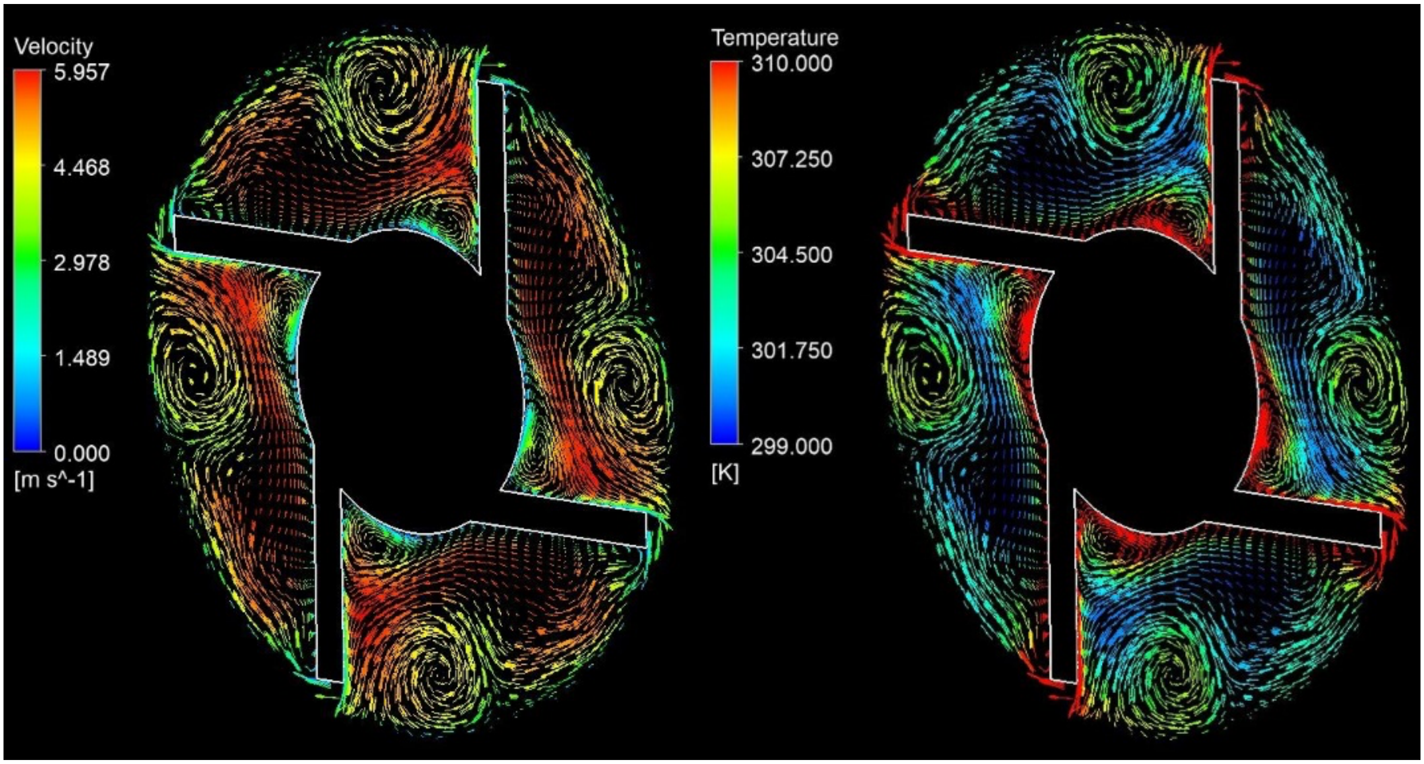

Fig. 5: Velocity and temperature contour for the air domain in a vertical plane for the wavy fin turbulator (CR=2)

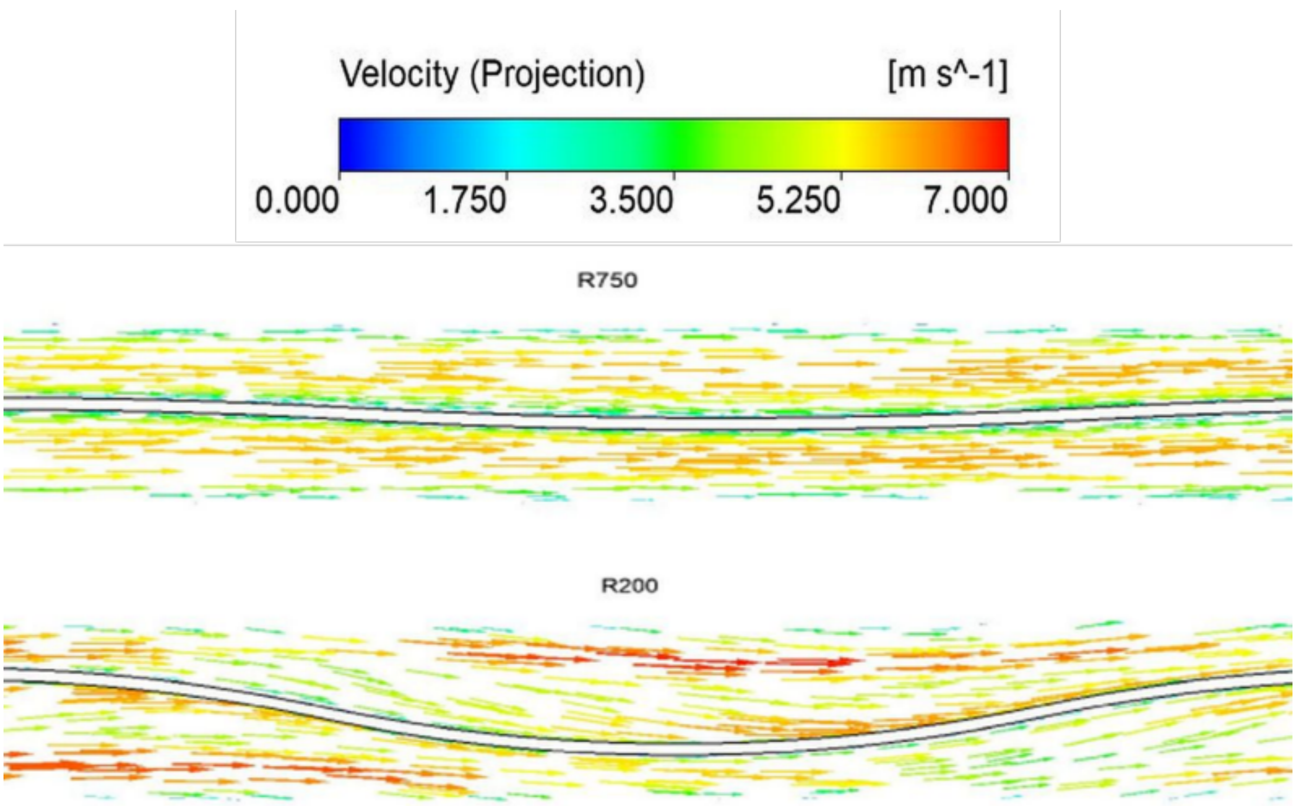

Fig. 6: Velocity plots for wavy fin turbulators in an isoplane coloured with velocity values

Fig. 7: Temperature distribution of air along the length for the bare pipe

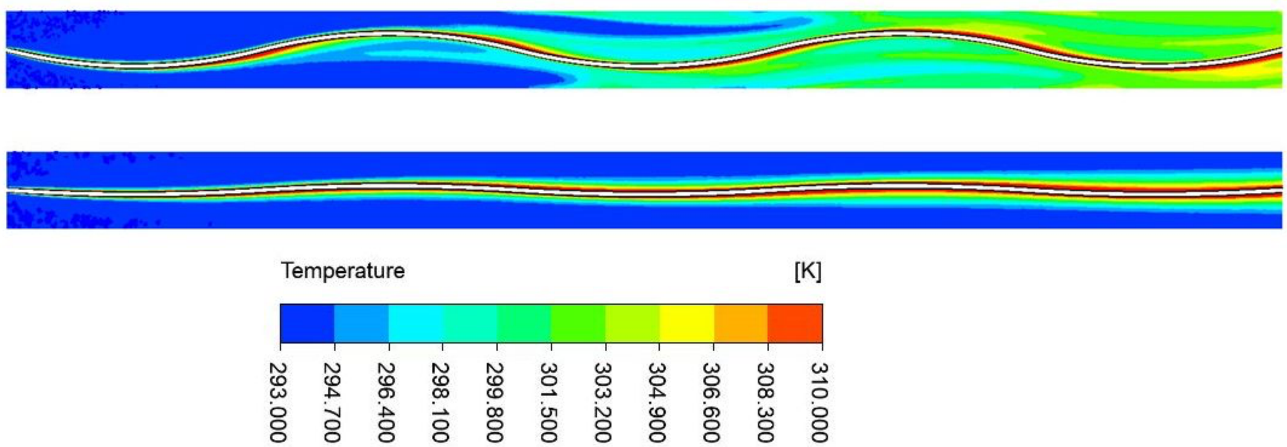

Fig. 8: Temperature distribution of air along the length for the wavy turbulator for CR =2 and CR =7.5

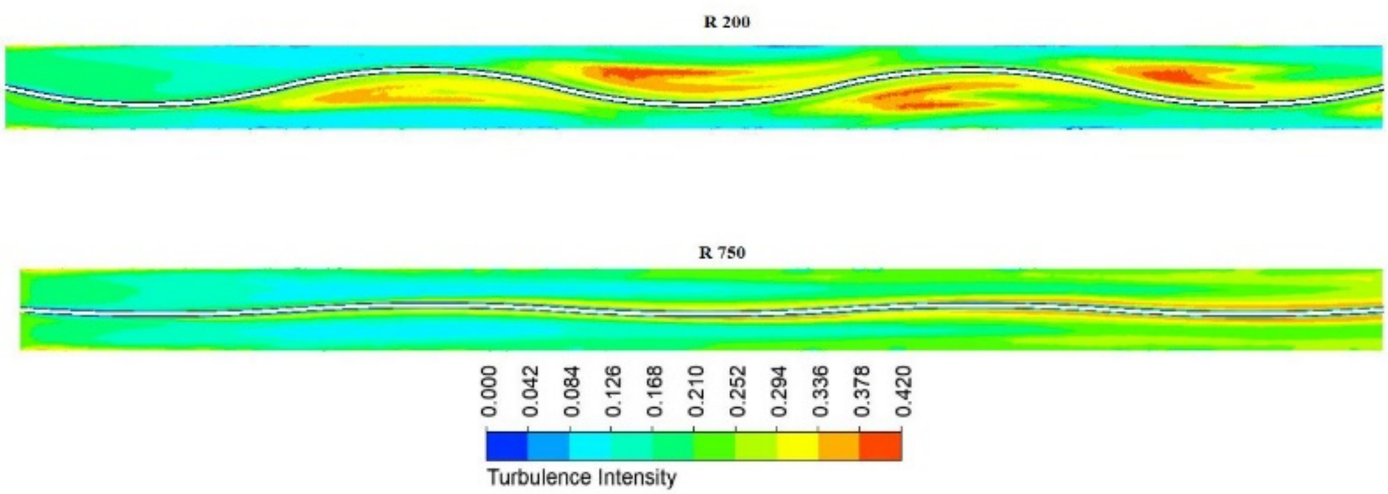

Fig. 9: Distribution of turbulent intensity of air along the length for the wavy turbulator for CR =7.5 and CR =2

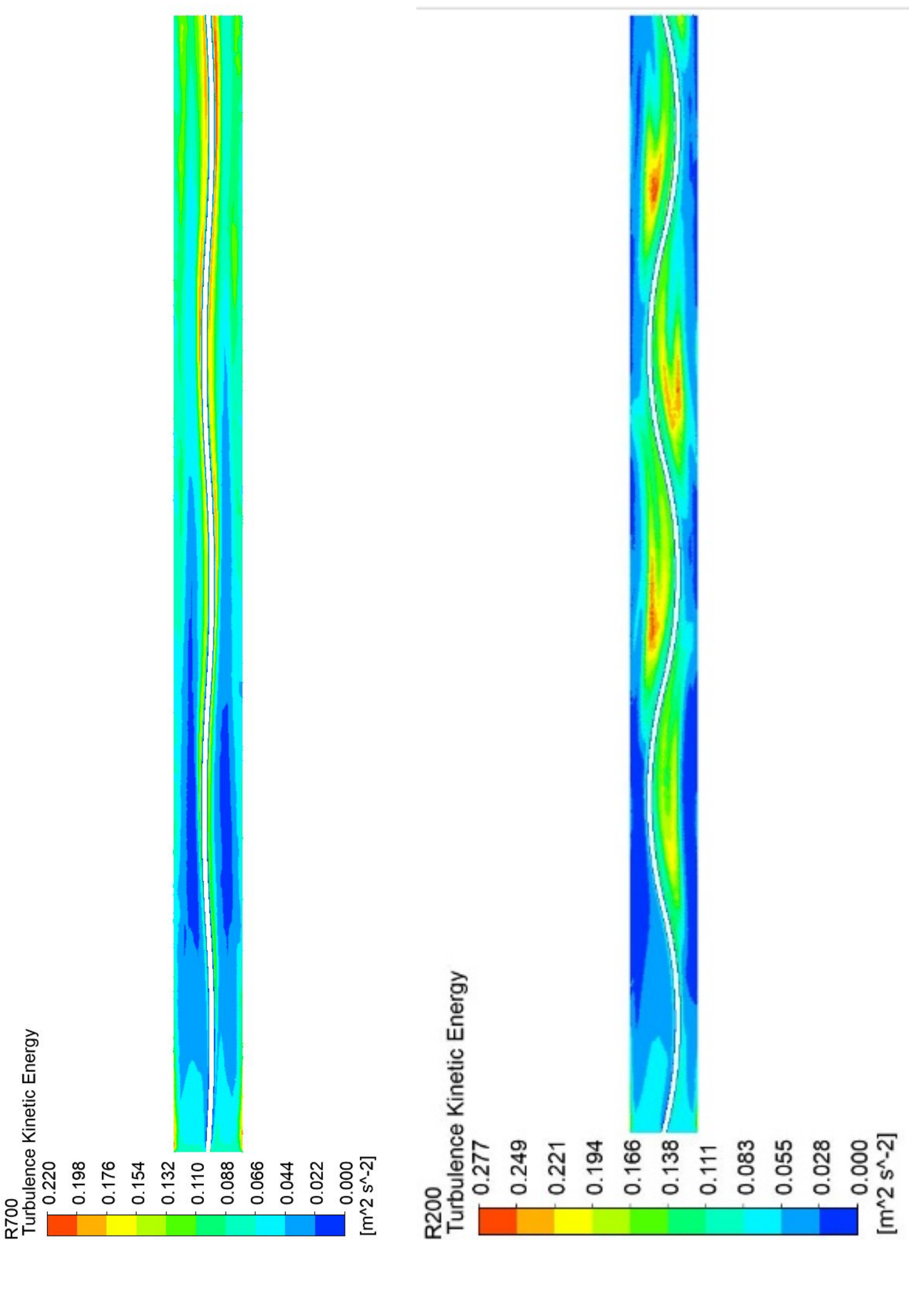

Fig. 10: Distribution of turbulent kinetic energy of air along the length for the wavy turbulator for CR =7 and CR =2

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The effect of introducing a wavy finned turbulator can also be explained by the turbulent intensity comparison. Turbulence intensity in general defined as the ratio of the root-mean-square of the velocity fluctuations to the mean flow velocity. Adding a wavy fin-type configuration, the flow gets disturbed which also causes the velocity fluctuations. Hence turbulent intensity will be increased as the flow progresses towards the outlet from the inlet as clearly indicated in Fig. 9. It is also observed that as the CR is reduced, for a given axial distance, the curvilinear path will be increased. Hence more fluid mixing takes place or recirculation of air particles will occur. When air experiences similar treatments at different curvature locations, the bulk turbulent intensity will rise and hence will contribute to the enhancement of thermal performance.
An increase in fluid turbulence also tends to increase the turbulent kinetic energy of the fluid as seen from Fig. 10. Comparison of turbulent kinetic energy for CR =7.5 and CR=2 reveals that as the CR value decreases, turbulent kinetic energy also increases due to the higher flow turbulence. It is seen from the figure that the maximum turbulent kinetic energy for CR = 2 is 0.277 m^2s^{-2} and for CR=7.5 is 0.22 m^2s^{-2}. Hence it clearly shows the advantage of providing curvature to the straight rectangular fin to improve the thermal performance.

5.2 . Heat transfer Characteristics

Numerical simulation has been conducted by varying the air Re ranging from 3000 - 15000 in increments of 3000 each time, with Re of water being held constant at 3000 for the sake of comparison. The difference between the air outlet and inlet temperature for various cases have represented in Fig. 11.

5.2.1. Outlet air temperature

Fig. 11 shows the variation of ΔT for bare, straight rectangular fin type and also for wavy turbulator cases. It can be seen that for all the wavy profiles, ΔT is higher than the bare and straight finned type. In the case of wavy finned type turbulator, due to flow mixing, air turbulence will be improved. This causes higher energy transport thus increasing the bulk outlet temperature. Hence for the same air inlet temperature, a higher difference will be noticed. In the case of straight rectangular form, due to the absence of the turbulence effects, outlet temperature is inferior compared to the wavy fin. It is also seen that as Re is increased, the difference in air temperature reduces as the higher air velocity reduces the retention or contact time of air with the wavy turbulator. This minimizes the flow modification and hence outlet temperature drops.

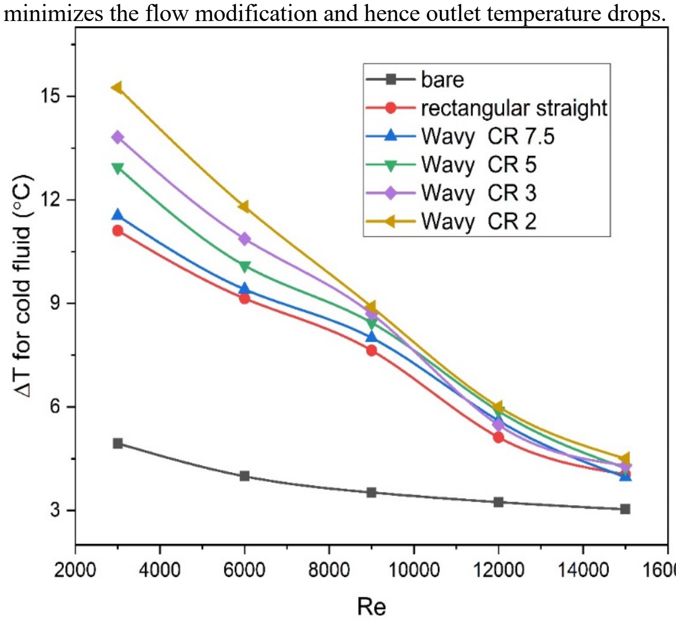

Fig. 11: Variation of ΔT for the air with Re for various turbulator configurations

5.2.2 Nusselt number (Nu).

Nu is the non-dimensional number that is a measure of heat transfer rate from the heat exchanger. It is defined as the ratio of convective heat transfer to conductive heat transfer under identical conditions. Figure 12 represents the variation of Nu for bare, rectangular straight fin and wavy fin turbulated cases for different values of Re. It is observed that the wavy type outperformed the other two types by showing higher Nu for all configurations and Re. An increase in the heat transfer characteristics due to the higher turbulence levels induced for the wavy type turbulators is visible in the higher Nu for these cases. Further, it is also seen that as CR reduces, the Nu value increases and the highest value of Nu is observed for CR =2. This may be due to the higher turbulence effects generated for wavy type with CR =2 as compared to other configurations with higher CR. For CR =2, the increase in Nu for Re of 6000 is 43% higher than with CR =7.5 and 48% and 261% higher than rectangular straight fin type and bare tube.

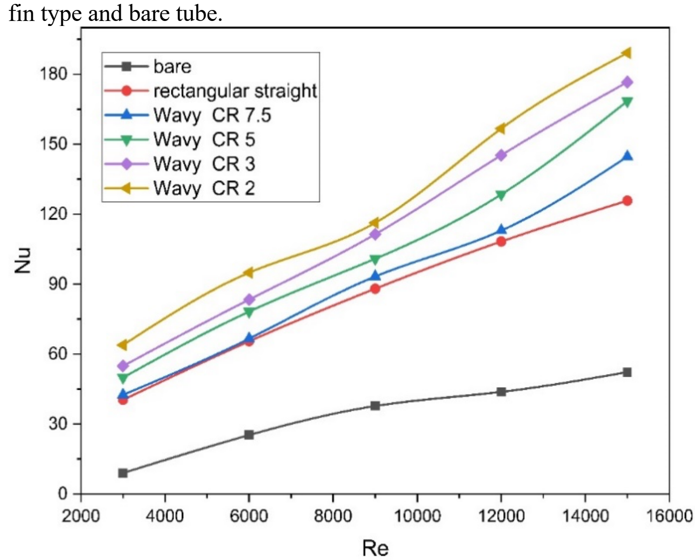

Fig. 12: Variation of Nu with Re for various turbulator configurations

5.2.3 Ratio of heat transfer rate

Fig. 13 depicts the ratio of heat transfer rate from the turbulator condition to that of the bare tube (without turbulator). Q_t/Q_b represents the ratio where Q_t represents the heat transfer rate from the turbulator configuration and Q_b represents the heat transfer rate from the bare pipe. All the ratios calculated are higher than 1 revealing the better performance of finned over the bare pipe. When a special surface such as rectangular straight or wavy fins is introduced in the heat transfer zone, it increases the surface area accounting higher convective heat transfer rate. From the graph, it is evident that the ratio is lowest for the rectangular straight fined case and is highest for the wavy case with CR =2. This justifies the advantage of the wavy fin over the straight one by showing a higher heat transfer rate. As discussed in the previous section, when CR is smaller higher flow distribution takes place enhancing the flow turbulence. Hence higher energy transfer takes place and accordingly higher heat transfer rate will be observed. Their effect reduces as the CR value is increased. For a straight rectangular fin, such flow modification will not happen at all, ending with a reduced heat transfer rate as seen from the figure. For CR =2, their ratio will be 37.5% higher than the rectangular straight finned case for Re of 3000. It reduces to 11.3% as the Re is increased to 15000

5.2.3 Heat transfer coefficient

Heat transfer coefficient is the proportional constant between the heat flux and the thermodynamic driving force for the flow of heat which is the temperature difference. It is an indication of the heat transfer capacity for the given condition of the heat exchanger.

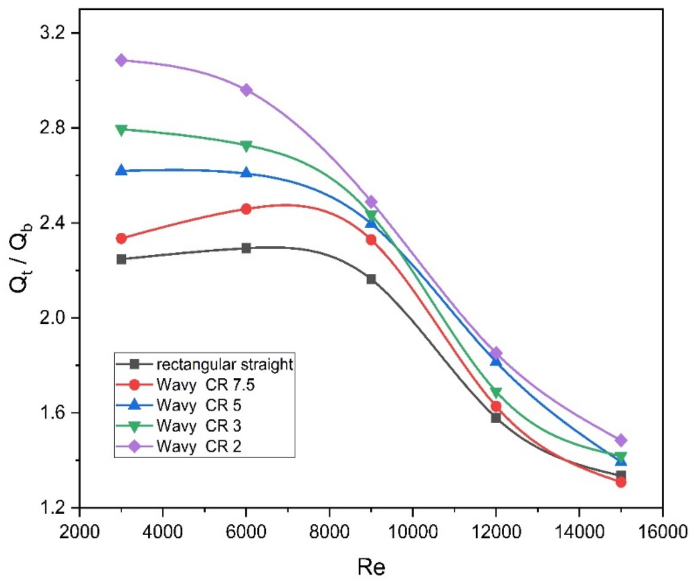

Fig. 13: Variation of Q_t/Q_b with Re for various turbulator configurations

Higher heat transfer coefficients represent higher heat transfer capacities of the fluid. For the various simulated configurations, heat transfer coefficients are represented in figure 14. It is witnessed that higher values could be noticed for the wavy type as compared to that of either bare or rectangular straight finned type. Flow modification sufficiently helps to increase the air turbulence and hence contributes to the enhancement of the heat transfer coefficient. Further lower CR shows higher values as compared to the values obtained with higher CR. When the CR is lowered from 7.5 to 2, for Re of 6000, the heat transfer coefficient is increased by 42.1%. Similarly, these values are 51% and 271% higher than the rectangular straight fin type and the bare type. When the Re is increased to 15,000 the above values will be increased by 116% and 260%.

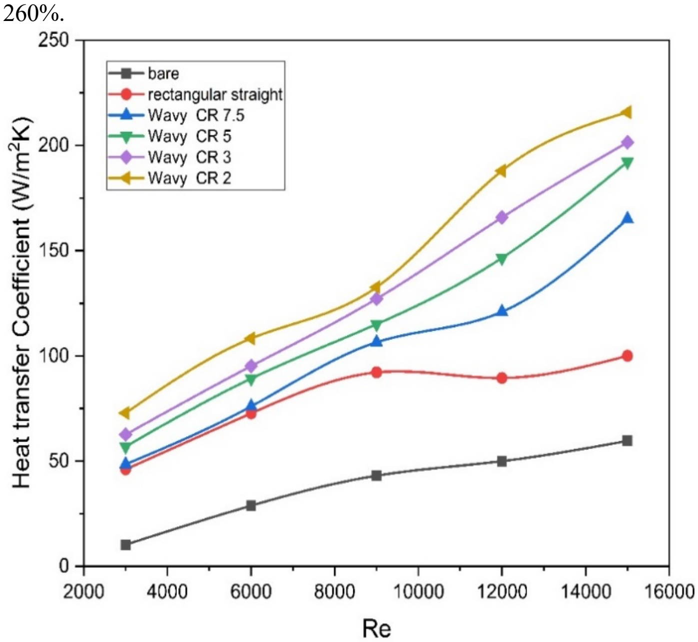

Fig. 14: Variation of convective Heat transfer coefficient with Re for various turbulator configurations

5.2.4 Friction factor

Friction factor (f) is a dimensionless quantity that provides the relation between the pressure loss due to friction along the length of the pipe and the average velocity. It is calculated by using equation 13. When wavy turbulators are provided in the annulus, higher friction factors will be noticed as compared to the bare pipe as seen in Figure 15. The values are also higher than the rectangular straight finned type. Turbulators are the obstacles placed along the flow path which partially block the flow. Hence higher frictional resistance will be induced and hence pressure loss will be higher. These effects will be observed as higher friction factor values for the wavy type than the straight and bare case. Their value reduces as the Re is increased for all tested cases. An increase in velocity overcomes the frictional resistance and hence reduces the f values. Further, it is noticed that as the CR ratio is decreased, f values tend to rise. When the CR is decreased from 7.5 to 2, for Re of 3000, f values increased by 67% and for 15,000 they increase by 62%. Similarly, for Re of 15,000, the f values are 199% higher than the rectangular straight finned pipe.

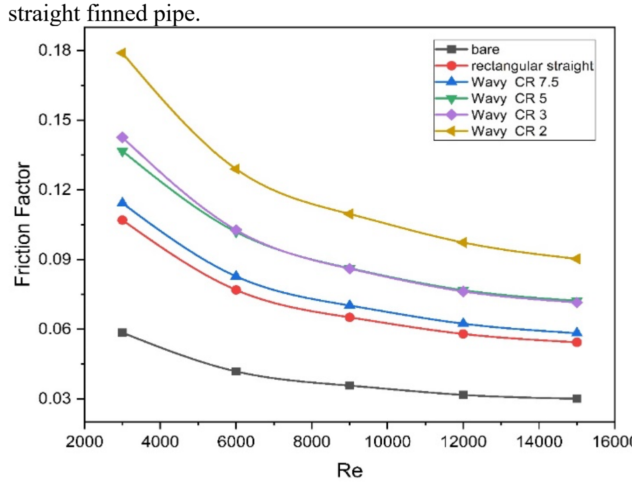

Fig. 15: Variation of friction factor f with Re for various turbulator configurations

5.2.5 Thermo hydraulic performance index (THPI)

When the wavy turbulators are provided in the annulus region it is seen from the previous section that Nu and heat transfer rate will be increased. Similarly, pressure drop and f value also increase. An increase in ΔP value requires more pumping power to push the fluid from the inlet to the outlet. Hence a performance parameter called THPI is used to compare the thermal performance of heat augmentation devices considering the same pumping power. It is defined by the relation

$$\text{THPI} = \frac{\frac{Nu_T}{Nu_b}}{\left[\frac{f_T}{f_b}\right]^{\frac{1}{3}}}.$$

Where NuT and Nub are the Nu for the wavy turbulator and bare pipe respectively. fT and fb are the friction factors for the turbulator and bare pipe respectively. For the different simulated conditions THPI is found and the results reveal that the highest value of 4.8 is noticed for the wavy fin with CR = 2 occurred for a Re of 3000 as seen from figure 16. For the same Re, THPI of 3.6 is noticed for the rectangular straight fin. As the Re is increased it is observed that THPI dropped. For higher Re, both Nu as well as ΔP increases. But the increase of ΔP is higher than the increase in Nu. HenceTHPI drops for higher Re. But from the simulated values, it is understood that THPI values for the wavy fins are higher than straight rectangular fin justifying the better thermal performance of wavy type turbulators. As the CR ratio is increased, THPI again shows a reducing trend. This may be due to the reduced benefits obtained in terms of an increase in Nu for higher CR. Hence it is concluded that by constructing a wavy type of turbulator, in addition to increased convective heat transfer area, it also acts as a turbulator which increases

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flow mixing and hence turbulence which is not visible in the case of straight rectangular fins. Hence wavy fin turbulators can be considered as one of the heat augmentation techniques for air heating applications to increase the thermal performance of a double pipe heat exchanger.

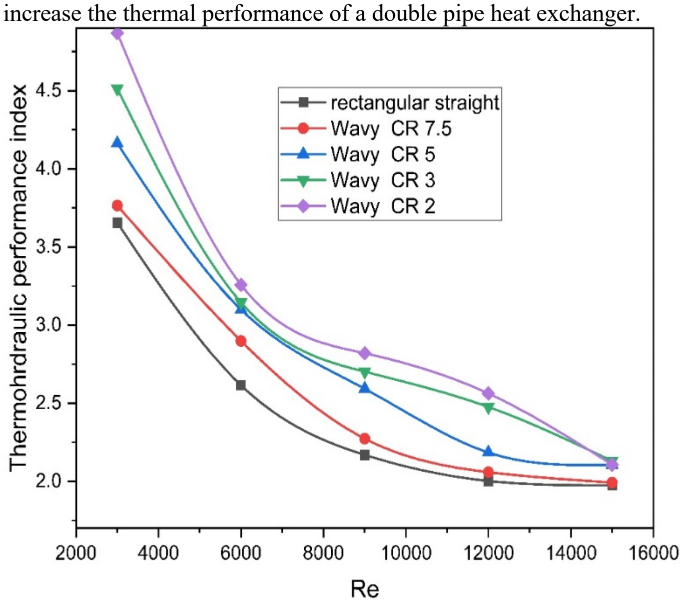

Fig. 16: Variation of THPI with Re for various turbulator configurations

6. CONCLUSIONS.

In the current work, the use of wavy type fin turbulators has been investigated for air heating applications in a double pipe heat exchanger. The influence of such turbulators on the heat transfer characteristics has been numerically studied with different CR ratios. The results can be summarised as below.

1. The wavy fin turbulator configuration showed better thermal characteristics than the bare and straight rectangular finned configuration.

2. For a configuration of CR = 2, the average bulk temperature at the outlet is increased by 29% and 195 % as compared to the rectangular straight and bare pipe respectively for the Re of 6000.

3. For the same configuration, under similar conditions, Nu is increased by 48% and 261% whereas pressure drop also increased by 65% and 207% as compared to the bare and straight finned type.

4. On increasing the Re from 3000 to 15000 ΔT dropped by 67% whereas Nu and ho increased by 207 % and 195% for the wavy configuration with CR =2

5. Decreasing CR from 7.5 to 2, improved the thermal performance of the wavy type turbulators. On reducing from 7.5 to 2, Nu, ΔT, and h0 increased by 42.4%, 28.2%, and 42.1% respectively for Re of 6000.

6. The highest value of THPI, 4.8 is noticed for the wavy fin turbulator with CR =2, and the value deteriorated with the increase in Re. When the Re is increased from 3000 to 15000 it reduced by 47%.

Wavy-type turbulator is simple in construction, easy to fabricate, and is also cost-effective. Hence the drawback of the use of double pipe heat exchanger for air heating due to the constraints such as lower heat transfer coefficient and lower thermal efficiency can be improved by using the wavy type turbulators. This will increase the thermal performance of the heat exchanger considerably.

Nomenclature

Q_h	rate of heat transfer from hot fluid (W)
Q_c	rate of heat transfer from cold fluid (W)
m_h	mass flow rate of the fluid (kg/s)
m_c	mc mass flow rate of cold fluid (kg/s)
C_{ph}	specific heat of hot fluid (kJ/kgK)
C_{pc}	specific heat of cold fluid (kJ/kgK)
Q_{avg}	average heat transfer rate (W)
T_{hi}	hot fluid inlet temperature (°C)
T_{ho}	hot fluid outlet temperature (°C)
T_{ci}	cold fluid inlet temperature (°C)
T_{co}	cold fluid outlet temperature (°C)
U_o	overall heat transfer coefficient (W/m2K)
LMTD	log mean temperature difference
A_o	outer surface area of the heat exchanger (m2)
h_i	inside convective heat transfer coefficient (W/m2K)
h_o	outside convective heat transfer coefficient (W/m2K)
R_{ei}	reynolds number of the inner fluid
V_i	velocity of the fluid in the inner pipe (m/s)
d_i	hydraulic diameter of the inner pipe (m)
L	length of the wavy turbulator (m)
h	height of the wavy turbulator
N_{ui}	nusselt number of the inner fluid
N_{uo}	nusselt number of the annulus fluid
R_{eo}	reynolds number of the outer fluid
D_{air}	hydraulic diameter of the annulus (m)

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