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COMPARATIVE ANALYSIS OF HEAT TRANSFER ENHANCEMENT WITH V-NOZZLE TURBULATORS

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Abstract

The effects of V-nozzle inserts on heat transfer and friction characteristics in a uniform heat flux tube are experimentally and numerically studied. The v-nozzle with pitch ratio 2.5 used. Experimental investigations have been carried out to study the effects of the V-nozzle turbulators on heat transfer augmentation, friction characteristics and enhancement efficiency, in a circular tube. It is found that using the V-nozzle can help to increase considerably the heat transfer rate at about 107% over the plain tube. A maximum gain of 1.25 on enhancement efficiency is obtained for the pitch ratio used, PR=2.5 this indicates that the effect of the reverse/re-circulation flows can improve the heat transfer rate in the tube. Experimental error recorded is 2.8% for friction factor when it's compared with Numerical result at the same Reynold No. In addition, correlations from the results are

1. INTRODUCTION

Heat transfer enhancement is the process of improving the performance of a heat transfer system. It generally means increasing the heat transfer coefficient. The performance of heat exchanger depends how effectively heat is utilized. The high performance of heat exchangers are very much essential in many practical applications such as aerospace, vehicles, refrigeration and air conditioning, cooling of electric equipment and so on. Reduction of the size of the heat exchanger may be possible due to improvement in the performance of heat exchanger[1][2].

On the other hand, a high performance heat exchanger of a fixed size can give a increased heat transfer rate and also there is decrease in temperature difference between the process fluids enabling efficient utilization of thermodynamic availability. The performance can be improved by using various augmentation techniques such as finned surfaces, integral roughness and insert devices [3][4]. A variety of different techniques are employed for the heat transfer process. Many active and passive techniques are currently being employed in heat exchangers, with twisted tape inserts providing a cost-effective and efficient means of augmenting heat transfer[6][7]. The reverse flow device or the turbulator is widely employed in heat transfer engineering applications. The reverse flow is sometimes called "re-circulation flow". The effect of reverse flow and boundary layer eruption (dissipation) is to enhance the heat transfer coefficient and momentum transfers. The reverse flow with high turbulent flow can improve convection of the tube wall by increasing the effective axial Reynolds number, decreasing the cross-section flow area, and increasing the mean velocity and temperature gradient. It can help to produce the higher heat fluxes and momentum transfer due to the large effective driving potential force but also higher pressure drop. The strengths of reverse flow and the reattached position are the main interest in many heat transfer applications such as heat exchangers, combustion chambers, gas turbine blades, and electronic devices. ANSYS-13.0 software tool is used for comparing the experimental result.

2. EXPERIENTSL SETUP

The experiments were carried out in an open-loop experimental facility as shown in Fig. 1. The loop consisted of a 0.75 kW blower, orifice meter to measure the flow rate, and the heat transfer test section. The copper test tube has a length of L=1250 mm, with 44 mm inner diameter (D), 51 mm outer diameter (Do), and 3.5 mm thickness (t) as depicted in Fig. 2. The tube was heated by continually winding flexible electrical wire to provide a uniform heat flux boundary condition. The electrical output power was controlled by a variance transformer to obtain a constant heat flux along the entire length of the test section and by keeping the current less than 3.5 A. The outer surface of the test tube was well insulated to minimize convective heat loss to surroundings, and necessary precautions were taken to prevent leakages from the system. The inner and outer temperatures of the bulk air were measured at certain points with a multichannel temperature measurement unit in conjunction with the PT-100 thermocouples as can be seen in Fig. 2. Eight thermocouples were tapped on the local wall of the tube and the thermocouples were placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The mean local wall temperature was determined by means of calculations based on the reading of PT-100 thermocouples. Fig. 2 represents the V-nozzle arrangement used in the present work. The V-nozzle was made of Mild steel 90 mm in length and its end and throat diameters were 36 mm and 20 mm, respectively.

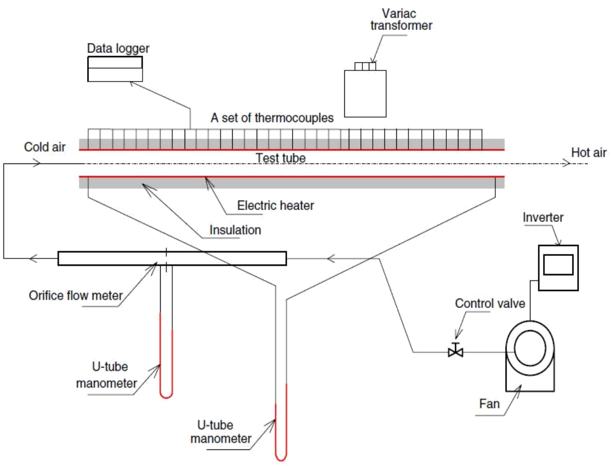


Fig-1: Schematic diagram of experimental heat transfer setup

The V-nozzles were placed with pitch ratio PR=2.5 with pitch lengths, l=90 mm for experiment. In the Fig. 2, the inlet bulk air at 25 °C from a 0.75 kW blower was directed through the orifice meter and passed to the heat transfer test section. The air flow rate was measured by an orifice meter, built according to ASME standard [10].Manometric fluid was used in U-tube manometers with specific gravity (SG) of 0.981 to ensure reasonably accurate measurement of the low pressure drop encountered at low Reynolds numbers. Also, the pressure drop of the heat transfer test tube was measured with inclined U-tube manometers. The volumetric air flow rates from the blower were adjusted by varying motor speed through the inverter, situated before the inlet of test tube. During the experiments, the bulk air was heated by an adjustable electrical heater wrapping along the test section. Both the inlet and outlet temperatures of the bulk air from the tube were measured by PT-100 thermocouple It was necessary to measure the temperature at 8 stations altogether on the outer surface of the heat transfer test pipe for finding out the average Nusselt number. For each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop across the test section and air flow velocity were measured for heat transfer of the heated tube with V-nozzles. The average Nusselt numbers were calculated and discussed where all fluid properties were determined at the overall bulk mean temperature.

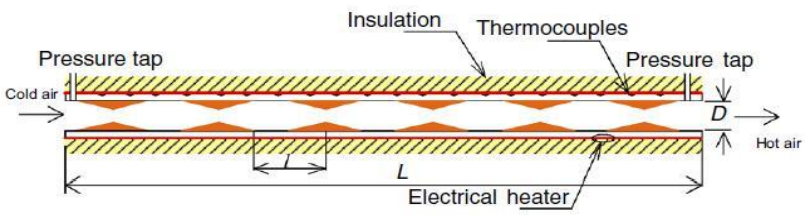


Fig-2: Test tube fitted with V-nozzle Turbulators

3. MATHAMATICAL CALCULATION

In the present work, the air is used as working fluid and flowed through a uniform heat flux and insulation tube. The steady state of the heat transfer rate is assumed to be equal to the heat loss from the test section which can be expressed as:

$$Q_{air} = Q_{Conv} \quad (1)$$

In which,

$$Q_{air} = mCpa(T_o - T_i) \quad (2)$$

The convection heat transfer from the test section can be written as:

$$Q_{conv} = hA(T_w - T_b) \quad (3)$$

Whereas,

$$T_b = (T_0 - T_i)/2 \quad (4)$$

And

$$T_w = \sum T_w/6 \quad (5)$$

Where T_w is the local wall temperature and evaluated at the outer wall surface of the inner tube. The averaged wall temperatures are calculated from 6 points, lined between the inlet and the exit of the test pipe. The average heat transfer coefficient, h and the mean Nusselt number, Nu are estimated as follows:

$$h = mCpa(T_0 - T_i)/A(T_w - T_b) \quad (6)$$

$$Nu = hD/k \quad (7)$$

The Reynolds number is given by

$$Re = UD/v \quad (8)$$

Friction factor, ƒ can be written as

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\rho\left(\frac{U^2}{2}\right)} \quad (9)$$

4. Experimental Results

The experimental results obtained under turbulent flow conditions for pitch ratio 2.5 of placing V-nozzles alone in a uniform heat flux tube are shown in Fig. 3. In the figure, the Nusselt numbers were related as a function of Reynolds number, using the mass-averaged velocity in the preliminary calculations. The results obtained for axial flow (plain tube) are also plotted for comparison. This technique results in an improvement of the heat transfer rate over the plain tube. The Nusselt number increases with increasing Reynolds number. The mean Nusselt number of using the V-nozzle increases at about 104% over the plain tube as shown in Fig 3. The increase in heat transfer rate with reducing pitch ratio is due to the higher turbulent intensity imparted to the flow between the V-nozzles. For the pitch ratio (PR=2.5), the increase in heat transfer rate is 107% over the plain tube for the Reynolds number ranging from 20000 to 50000

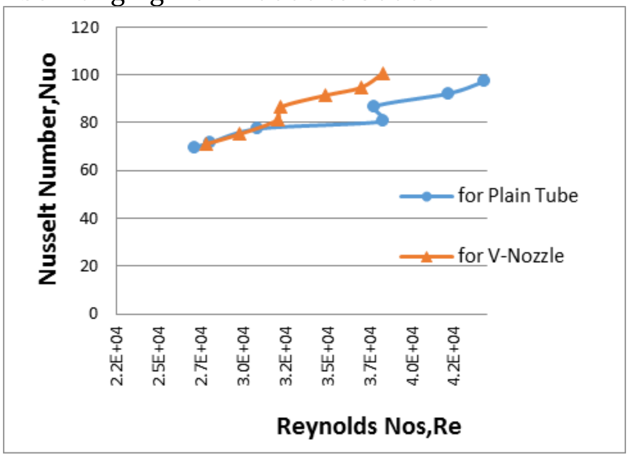


Fig-3: Reynolds number vs Nusselt number

The heat transfer coefficient increases with the increase in the Reynolds number is presented in Fig. 4.The mean heat transfer coefficient of using the V-nozzle increases at about 127% over the plain tube. The variation of the pressure drop in terms of friction factor across the test section as a function of Reynolds number for pitch ratio2.5 is presented in Fig. 5.The friction factor of axial flow (plain tube) is also plotted for comparison. It can be seen that the friction factors obtained from pitch ratio 2.5 are decrease with increasing the Reynolds number. The increase in friction factor with reverse/turbulent flow, in general, is much higher than that with axial flow.

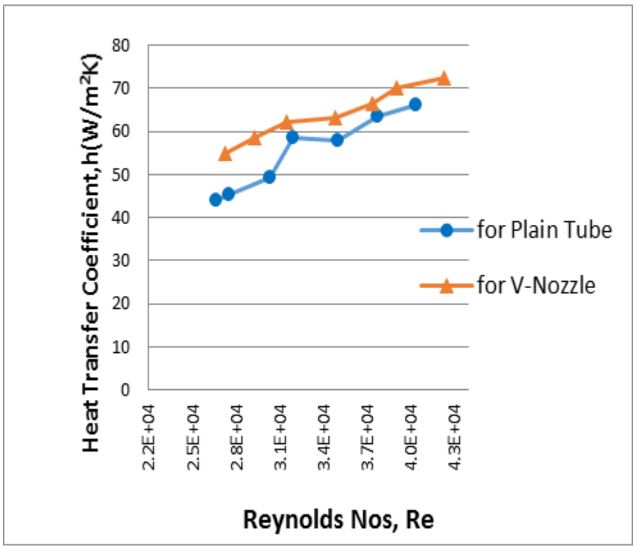


Fig-4: Reynolds number vs Heat transfer coefficient

This is because of the dissipation of dynamic pressure of the fluid due to higher surface area and the act caused by the reverse flow. Moreover, the pressure drop has the high possibility to occur by the interaction of the pressure forces with inertial forces in the boundary layer.

5. NUMERICAL METHOD

The commercial package of CFD (ANSYS - 13.0) was chosen as the CFD tool for this study to solve the above Mentioned governing equations accompanied with boundary conditions. Solution sequential algorithm (segregated solver algorithm) with settings including implicit formulation, steady (time-independent) calculation, viscous laminar model and energy equation, SIMPLE as the pressure-velocity coupling method, and first-order upwind scheme for energy and momentum equations was selected for simulation.

6. NUMERICAL RESULTS

Numerical results are compared with experimental results for the following parameters.
Fig. 7 shows the Nusselt number increases with increasing Reynolds number. In both the experimental and numerical method using the v-nozzle PR= 2.5.
Fig. 8 shows the friction factor across the test section as a function of Reynolds number for pitch ratio2.5 in both the experimental and numerical method.

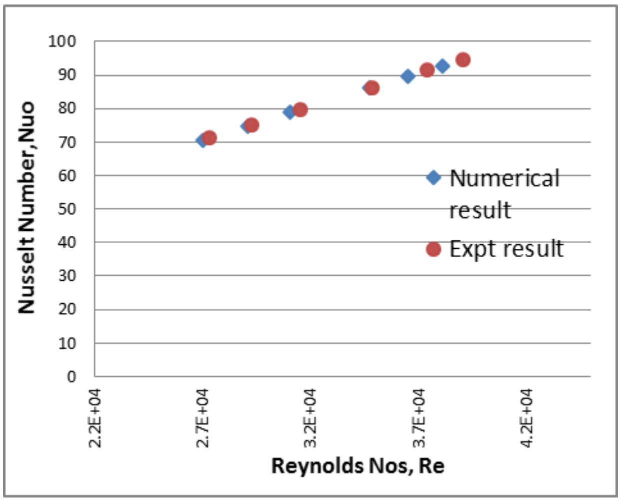


Fig-7: Reynolds number vs Nusselt number

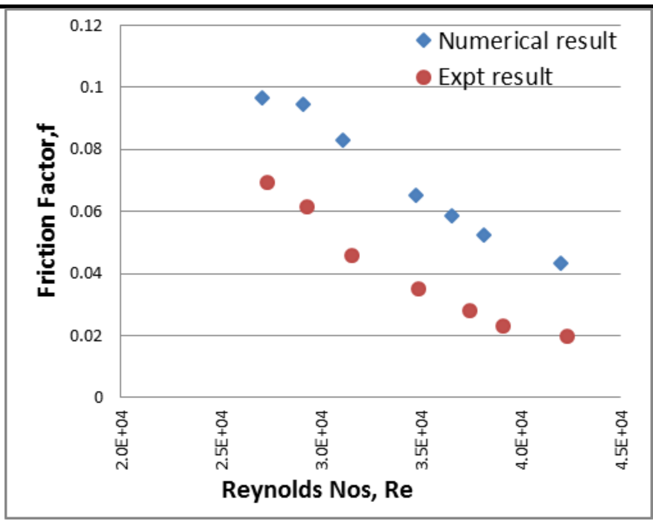
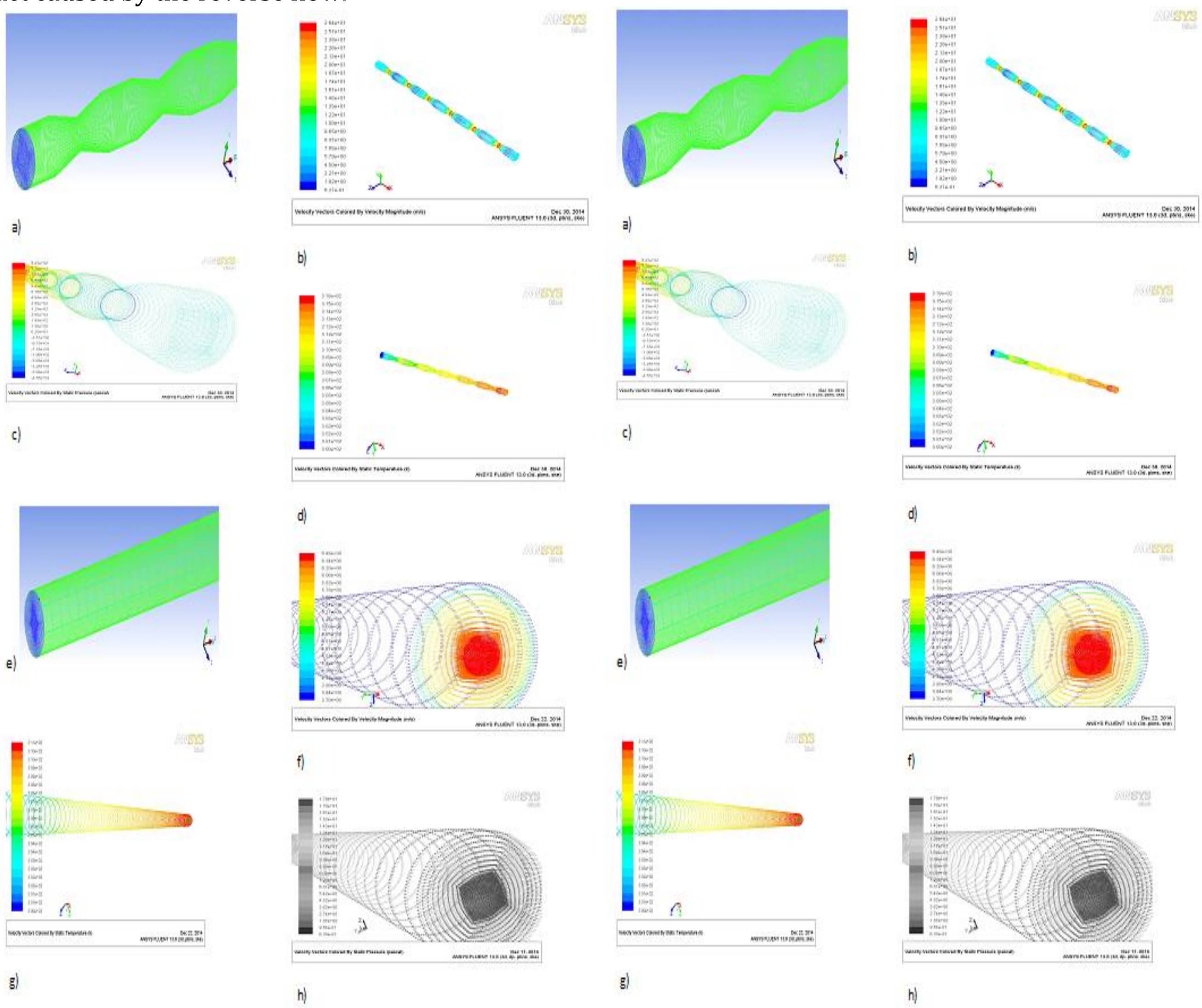
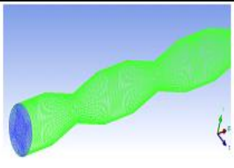


Fig-8: Reynolds number vs Friction factor

It can be seen that the friction factors obtained from pitch ratio 2.5 are decrease with increasing the Reynolds number. The increase in friction factor with reverse/turbulent flow, in general, is much higher than that with axial flow. This is because of the dissipation of dynamic pressure of the fluid due to higher surface area and the act caused by the reverse flow.





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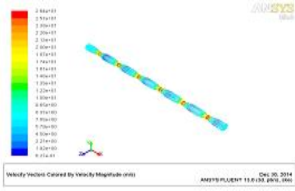

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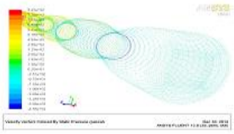

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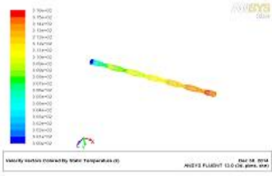

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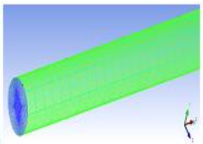

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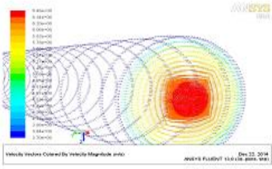

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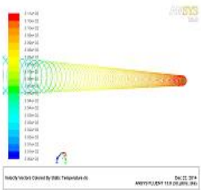

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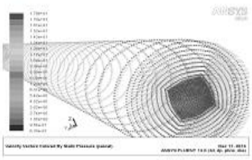

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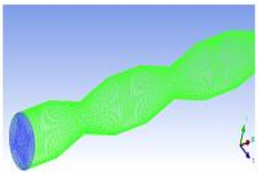

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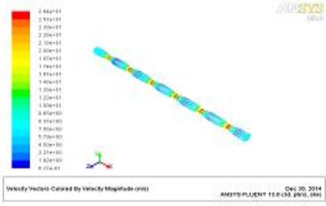

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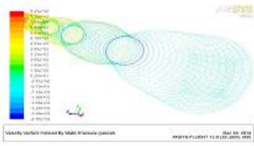

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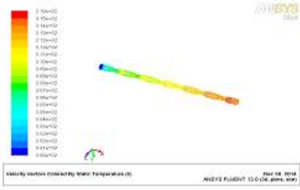

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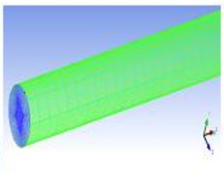

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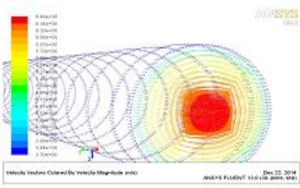

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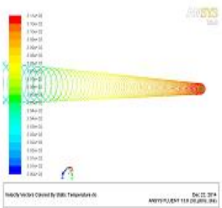

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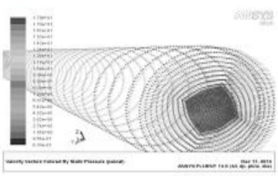

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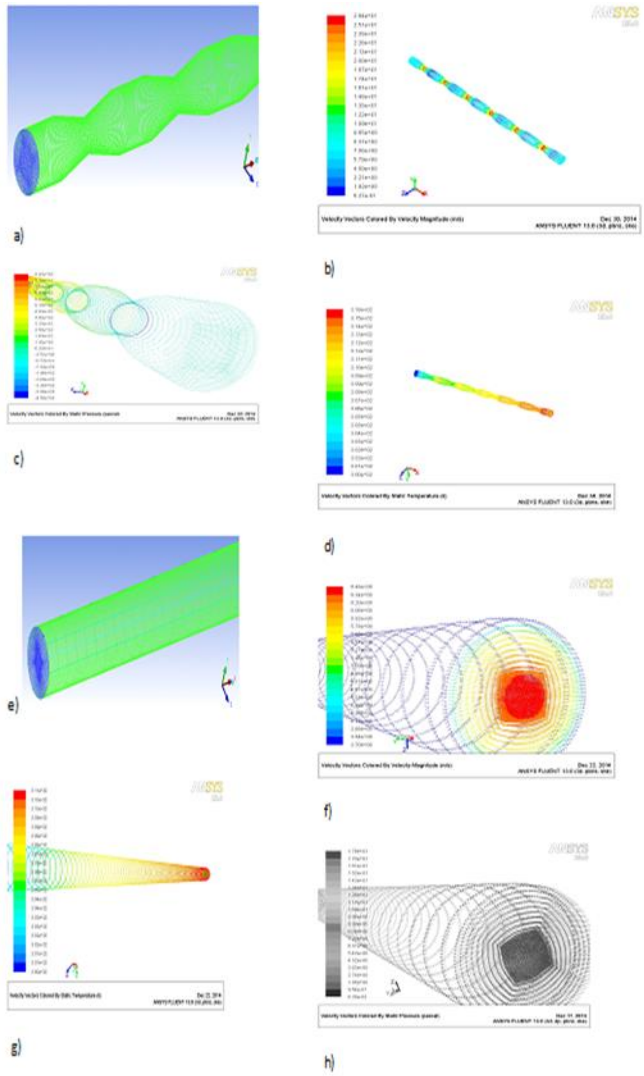

Fig-6: a) meshing using ICEM CFD software (V-nozzle , PR=2.5) b) velocity vector colored by velocity magnitude for V-nozzle PR=2.5 c) velocity vector colored by pressure magnitude for V-nozzle PR=2.5 d) velocity vector colored 6by static temperature for V-nozzle PR=2.5 e) meshing using ICEM CFD software(plain tube.) f) Velocity vector colored by velocity magnitude for plain tube. g) Velocity vector colored by pressure magnitude for plain tube. h) Velocity vector colored by static temperature for plain tube.

7. CONCLUSION

Experimental investigations have been carried out to study the effects of the V-nozzle turbulators on heat transfer, friction factor and enhancement efficiency, in a circular tube. Major findings can be summarized as follows:

• The heat transfer in the circular tube could be promoted by fitting with V-nozzles while it brings about the energy loss of the fluid flow. The mean heat transfer rates obtained from using the V-nozzles with PR=2.5 found to be 107% over the plain tube. However, the increase in friction factor is much higher than the increase in Nusselt number at the same Reynolds number.

• The enhancement efficiency decreases with increasing Reynolds number. The maximum value of enhancement efficiency obtained from using the PR=2.5 found to be 1.25.

The friction factor is decreased with increased the Reynolds number as experimental error recorded is 2.8% when its compared with Numerical results.

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