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CFD Analysis of Enhancement of Heat Transfer in a Tube with Rod Inserts

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Abstract:
The enhancement of heat transfer is very important many engineering applications to increase the performance of heat exchanger. The heat transfer augmentation techniques are classified as active and passive technique. If a proper configuration of the inserts is being selected depending on working condition that have been reported in literature. Due to advances in computer software, the CFD tool is very important and effective tool to understanding heat transfer applications. Incorporating the inserts, the heat transfer enhancement is increased due to its importance in different applications. In this paper includes the result of CFD analysis of Enhancement of Heat Transfer in a Tube with rod Inserts.

Keywords: *CFD Analysis, heat transfer enhancement, active technique, passive technique, inserts.*

1. Introduction

Heat transfer may be defined as the transmission of energy from one region to another region as a result of temperature gradient. The heat transfers to be maximized for the engineering applications for the effective working of the system. CFD analysis technique that not only predicts fluid flow behavior, but also the transfer of heat, mass, phase change, chemical reaction, mechanical movement, and stress or deformation of related solid structures. These characteristics have rapidly transformed CFD into a very popular tool in engineering analyses.

The word enhances means “to rise to a higher degree, to intensify, to increase the valve, attractiveness, or quality of, and to improve.” Hence, enhanced heat transfer is a heat transfer that has been improved. The study of improved heat transfer performance is referred to as the heat transfer enhancement, augmentation, or intensification. The enhancement of heat transfer has concerned researchers and practitioners since the earliest documented studies of heat transfer. In recent years, increasing energy and material costs have provided significant incentives for the development of energy efficient heat exchangers. An engineer utilizes his knowledge of heat transfer either to transmit heat in the most effective or economic way, or to protect his equipment against effective heat gains or losses.

In many engineering applications the enhancement of heat transfer from surface is must. As a result, considerable emphasis has been placed on the development of various augmented heat transfer surfaces and devices. Heat transfer enhancement today is characterized by rigorous activity both in research and industrial practice. This can be seen from the exponential increase in world technical literature published in heat transfer augmentation devices, growing patents and hundreds of manufacturers offering products ranging from enhanced tubes to entire thermal systems incorporating enhancement technology. Energy and materials saving considerations, space considerations as well as economic incentives, have led to the increased efforts aimed at producing more efficient heat exchanger equipment through the augmentation of heat transfer. Heat transfer augmentation techniques (passive, active, or a combination of passive and active methods) are commonly used in areas such as process industries, heating and cooling in evaporators, thermal power plants, air-conditioning equipment, refrigerators, radiators for space vehicles, automobiles etc.,

Passive techniques, where inserts are used in the flow passage to augment the heat transfer rate, are advantageous compared with active techniques, because the insert manufacturing process is simple and these techniques can be easily employed in an existing heat exchanger. Twisted tapes, wire coils, ribs, fins, dimples, mesh inserts etc., are the most commonly used passive heat transfer augmentation tools. Some of the first Experimental results obtained for fluid flow in porous medium. They measured Temperatures in turbulent region and they found that the enhancement of heat transfer using porous insert is more effective when compared to plain tube because Nussle number became double compared to that of plain tube and Pressure drop decreases 0.5 times more when compared with that of a plain tube at the same mass flow rate.

Bogdan I. Pavel & Abdulmajeed A. Mohammad [1] experimentally investigated the effect of metallic porous inserts in a pipe subjected to constant and uniform heat flux at a Reynolds number range of 1000-4500. The maximum increase in the length-averaged

Nu number of about 5.2 times in comparison with the clear flow case and a highest pressure drop of 64.8Pa were reported with a porous medium fully filling the pipe.

Angirasa [2] performed experiments that proved augmentation of heat transfer by using metallic fibrous materials with two different porosities namely 97% and 93%. The experiments were carried out for different Reynolds numbers (17,000-29000) and power inputs (3.7 and 9.2 W). The improvement in the average Nusselt number was about 3-6 times in comparison with the case when no porous material was used.

Fu et al [3] experimentally demonstrated that a channel filled with high conductivity porous material subjected to oscillating flow is a new and effective method of cooling electronic devices.

Mehmet Sozen & T M Kuzay [4] numerically studied the enhanced heat transfer in round tubes filled with rolled copper mesh at Reynolds number range of 5000-19,000. With water as the energy transport fluid and the tube being subjected to uniform heat flux, they reported up to ten-fold increase in heat transfer coefficient with brazed porous inserts relative to plan tube at the expense of highly increased pressure drop.

As Bogdan I. Pavel & Abdul majeed A. Mohamad carried out their work in a pipe with porous inserts in laminar and turbulent region with Reynolds number ranging from 1000-4500, the present work has been done similar lines but in turbulent region (Re number range of 7000-14000) as most of the flow problems in Industrial heat exchangers involve turbulent flow region.

In paper [5-8], Authors discussed CFD analysis of Enhancement of Heat Transfer in a channel with multiple longitudinal vortex generators.

2. Modeling Setup and Governing Equations

In order to shed some light on the experimental results presented, a CFD analysis of the heat transfer in the porous channel was completed. The first step in completing a CFD analysis of a system is to set up the governing equations. The equations involved in incompressible flow through porous channel include the continuity equation, the momentum equation, and the energy equation. The momentum and energy equations are, respectively

$$\rho \frac{D\vec{u}}{Dt} = -\nabla p + \rho \vec{g} + \mu \nabla^2 \vec{u}$$

$$= \left[\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)\right] + \left[u\frac{\partial p}{\partial x} + v\frac{\partial p}{\partial y} + w\frac{\partial p}{\partial z}\right] + \mu\varphi$$

Where g is the body force vector, $\partial i = cp\partial T + (\partial p/p)$ is the specific enthalpy, and p is pressure. For the specific case of heated flow through porous channel, the above equations can be further simplified. Assuming steady flow, constant properties, no body forces, only axial pressure variation, no viscous dissipation, and no axial heat conduction (valid for RePr > 100), the governing equations for heated flow through a rectangular porous channel are continuity

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

x-Momentum: $$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}\right) = -\frac{dp}{dx} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right)$$

y-Momentum: $$\rho\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z}\right) = \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right)$$

z-Momentum: $$\rho\left(u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z}\right) = \mu\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right)$$

Energy: $$\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z}\right) = \frac{1}{\alpha}\left(\frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right)$$

Given the complexity of these equations, computational methods of solving them are required.

All of these equations are coupled, so a computational code must be used to solve equation simultaneous. For this analysis, the commercial code FLUENT was used to solve these equations.

2.1. Validation of Software for Plain Tube without Insert

For the validation of software plane tube without insert of 1m length and 1 inch dia is used. Constant heat flux of 126watt is maintained across the tube and the result obtained by software are compared with the result obtained by experiment. For heat transfer enhancement and pressure drop, color contour is obtained using gambit and fluent software for different Reynolds no.

Trial	Reynolds no.	ΔT °c (exp)	ΔT °c (cfd)	ΔP (exp) in Pa	ΔP (cfd) in Pa	% error (ΔT °c)	% error (ΔP)
1	11408	20	22	45.05	40.08	10	11.03
2	16037	19	21.5	68.67	71.99	13.15	4.83
3	19364	16.5	18	95	107	9.09	12.63
4	21560	14.5	16	114	119	10.3	4.38
5	24265	13	14	156	146	7.69	6.41

Table 1: study of pressure drop and heat transfer enhancement of plain tube

Reynold no	Nusselt no (cfd)	Nusselt no (exp)
11408	34.71	35.12
16037	45.58	46.12
19364	53.30	53.63
21560	57.76	58.45
24265	63.49	64.23

Table 2: Comparison of Nusselt no. with cfd and exp. Value

Reynold no	f (friction factor exp))$\times 10^{-5}$	f (friction factor cfd)$\times 10^{-5}$
11408	4.53	4.06
16037	3.21	3.37
19364	2.83	3.45
21560	3.05	3.05
24265	3.41	2.97

Table 3: Comparison of exp. Friction factor and cfd friction

As Reynolds no. is increasing pressure drop between tube is increasing. On the other hand, reynold no is increasing, temperature difference is also increasing. It means heat transfer rate is increasing. The value of pressure drop and temperature difference with experimental value is approximately same with cfd analysis. So software is validated for tube without insert and it is further used for the analysis of tube with insert in different cases. As Reynolds no increases, first the value of friction factor decrease then increases. Value of Nusselt number is approximately same in experimental and CFD method.

2.2. Result of CFD Analysis of Heat Transfer Enhancement Using Inserts

<u>2.2.1. Variation of Insert Angle Keeping Space between Insert Constant Angles</u>

(1) At angle 90^o

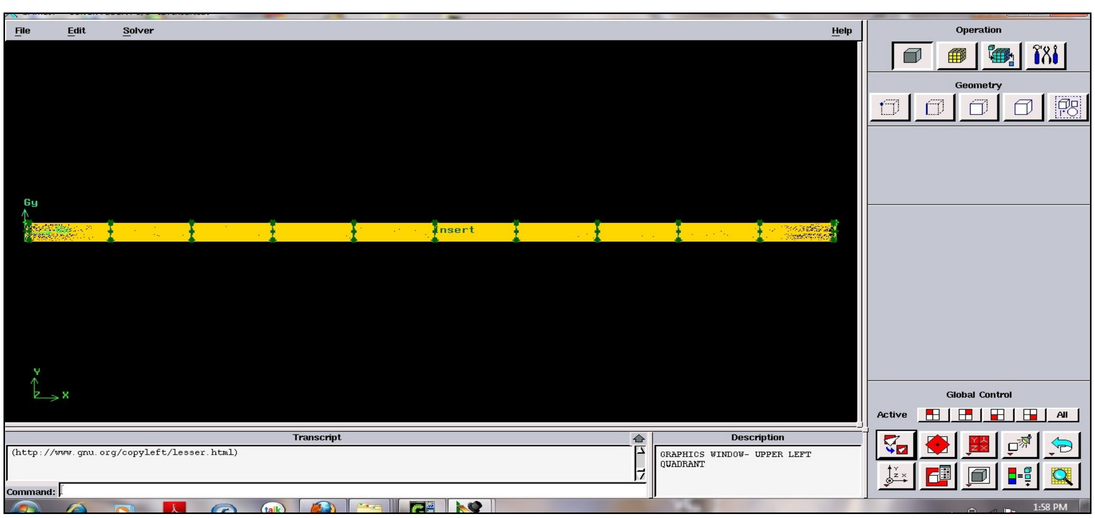

Figure 1: Meshing of tube for insert with angle 90^o

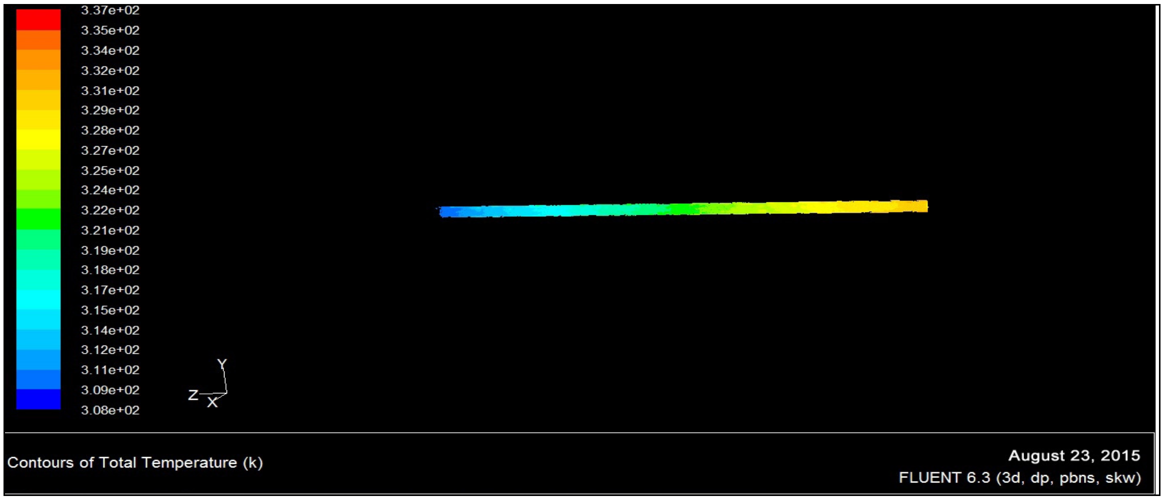

Figure 2: Temperature contour

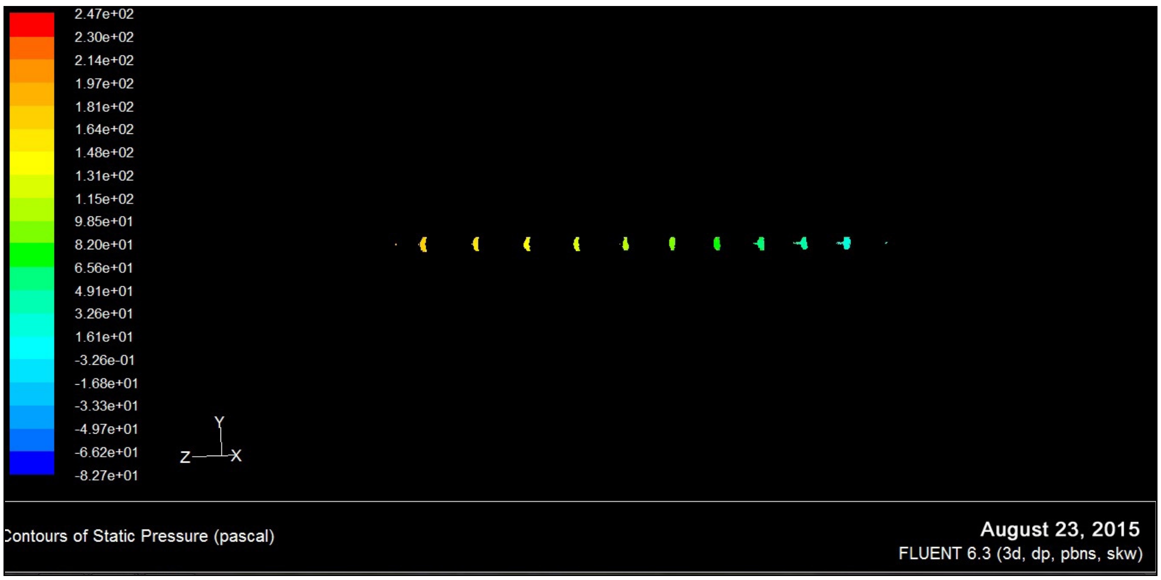

Figure 3: Static pressure contour

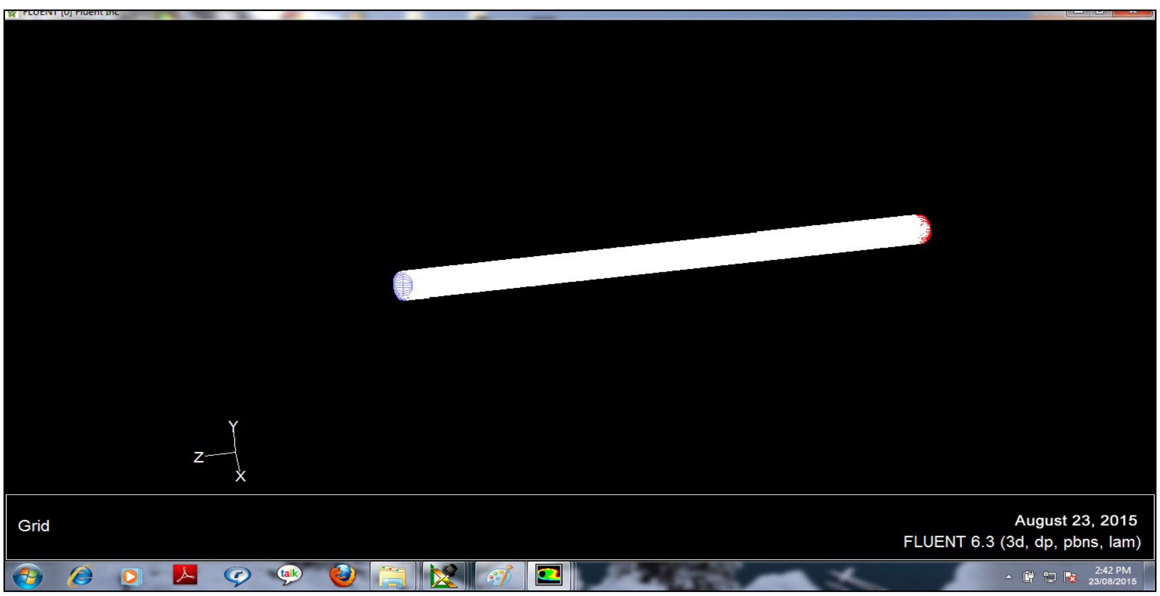

Figure 4: Grid generation

(2) At angle 60^o

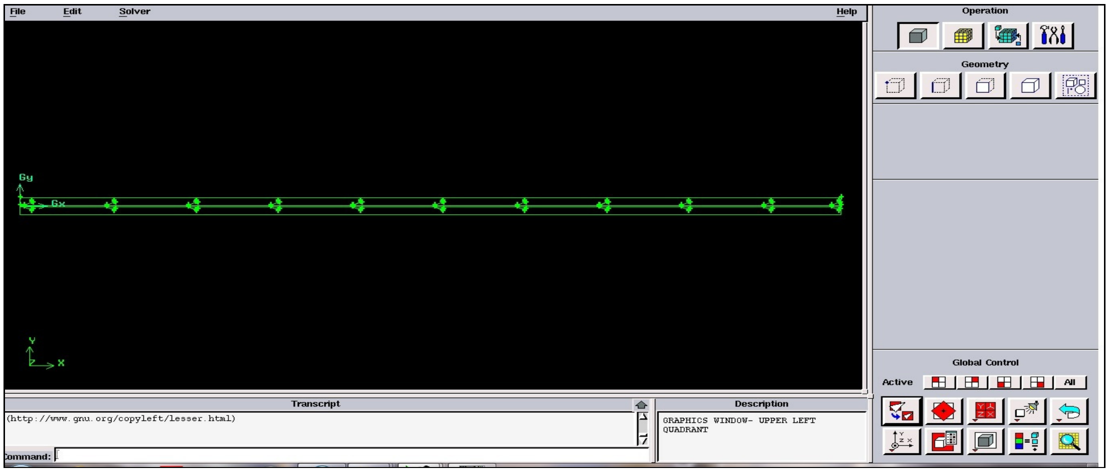

Figure 5: Modeling of tube for insert at angle 60^o

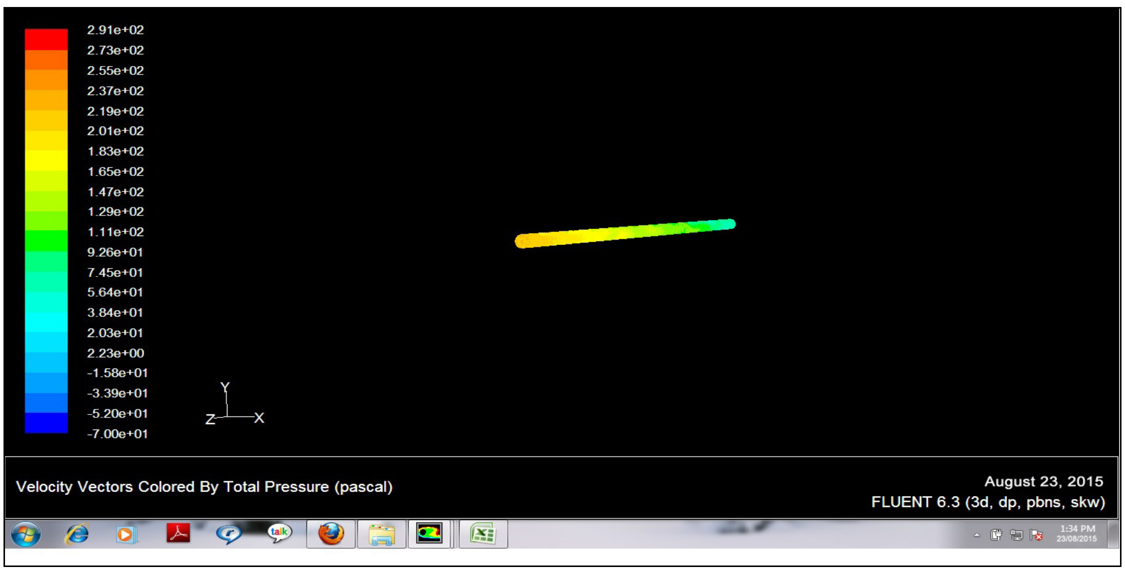

Figure 6: Total pressure contour

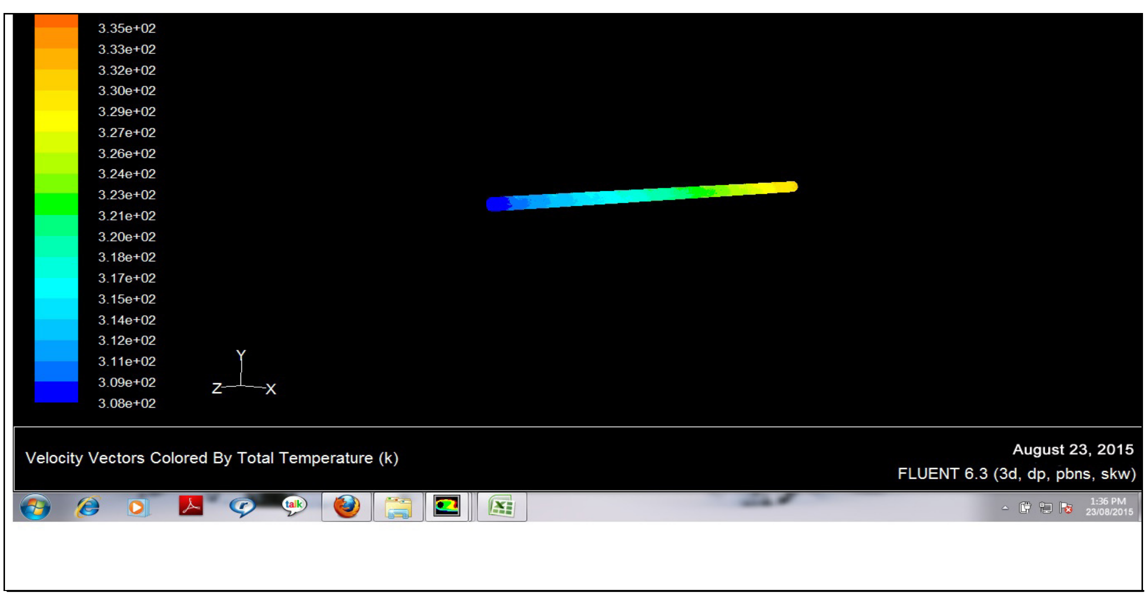

Figure 7: Total temperature contour

(3) At angle 45^o

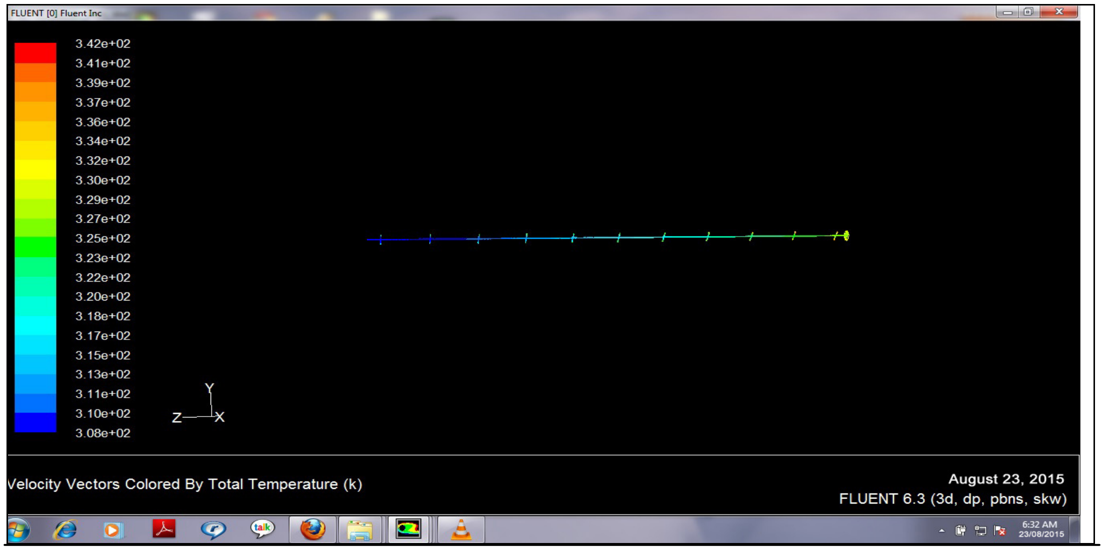

Figure 8: Total temperature contour

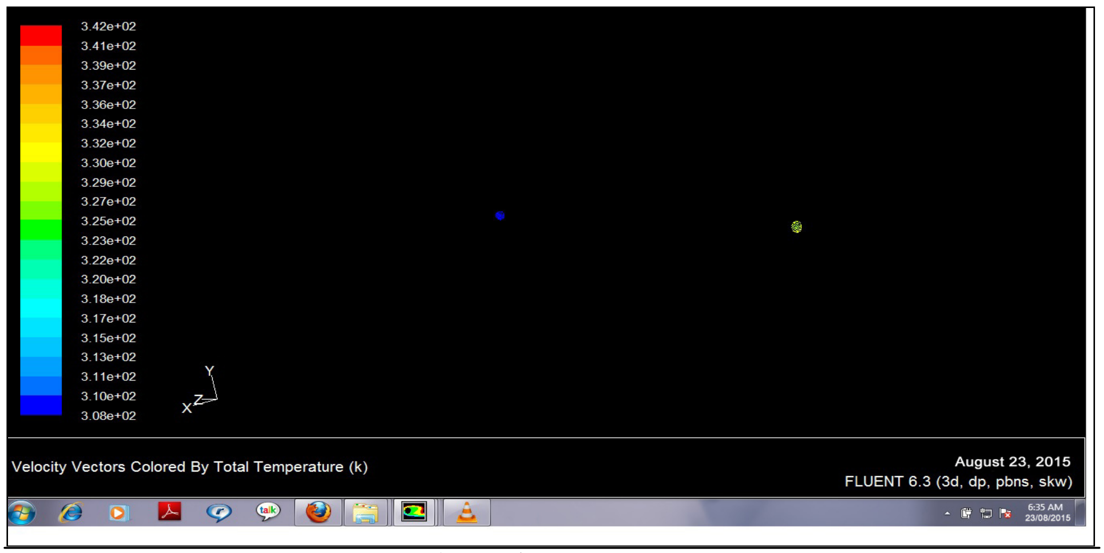

Figure 9 : Total temperature contour

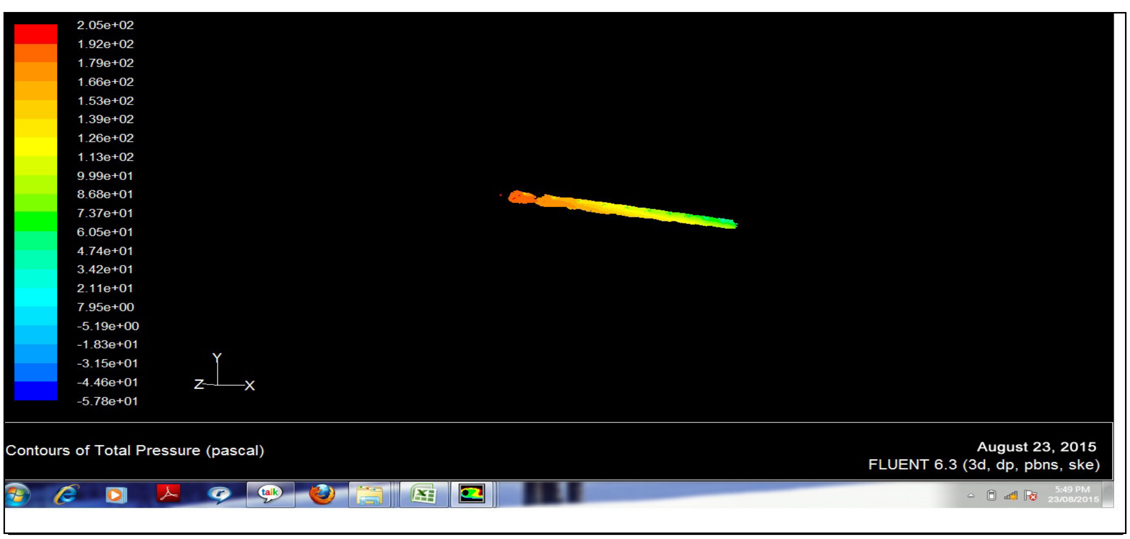

Figure 10: Total pressure contour

Angle of insert	ΔT °c	ΔP	Nu.	Re
90	27	243	91.72	16037
60	25	225	109.8	16037
45	34	227	138.4	16037
30	29	190	98.32	16037

Table 4: Study of pressure drop and temperature change

2.3. Due to Angle Variation

As angle of insert increases, the pressure drop of tube decreases. But temperature chance of air has no regular trend. Pressure drop is minimum for angle 30 degree and maximum for angle 90 degree. Temperature difference is maximum for 45 degrees and minimum for 60 degrees. Even pressure drop is almost equal in the case of 60 degrees and 45 degrees. Main objective of project is to enhance heat transfer rate with minimum pressure, so angle 45 degree is suitable for angle aspect. Even pressure drop is minimum at angle 30 degree but temperature difference is less compared to angle at 45 degrees. so 30 degrees is the most suitable angle in angle aspect.

2.3.1. Variation of Distance between Insert Keeping Angles 90^{o}

H (1) Distance = 7.5 cm

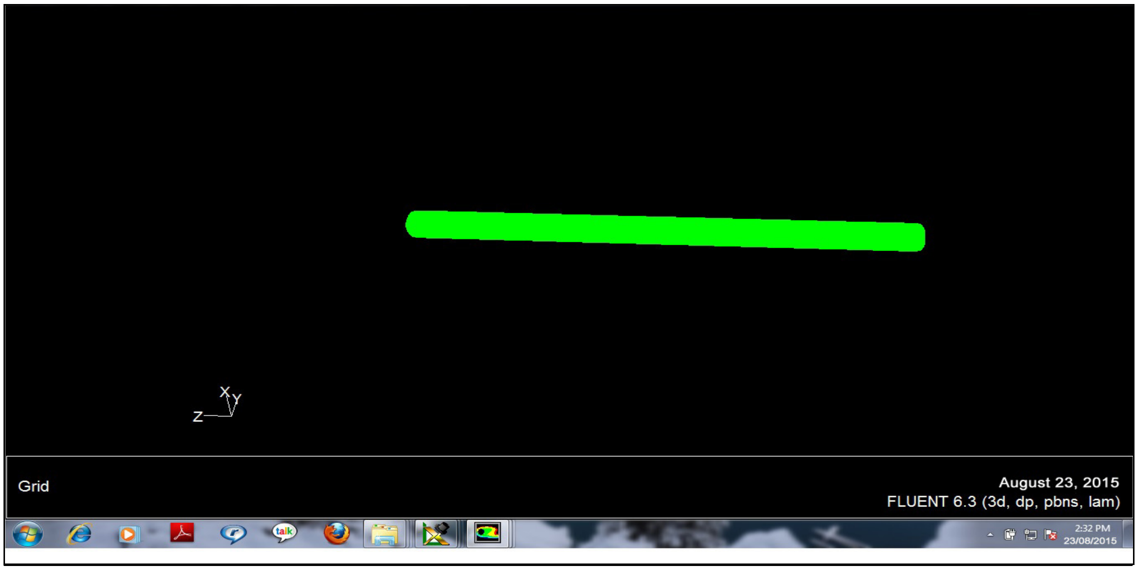

Figure 11: Grid generation

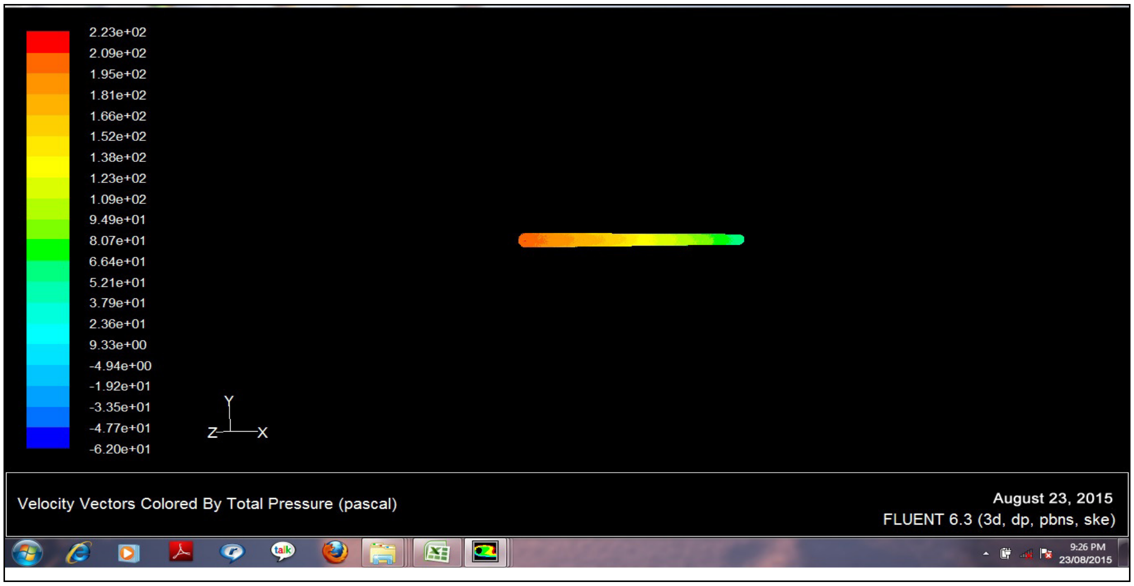

Figure 12: Total pressure contour

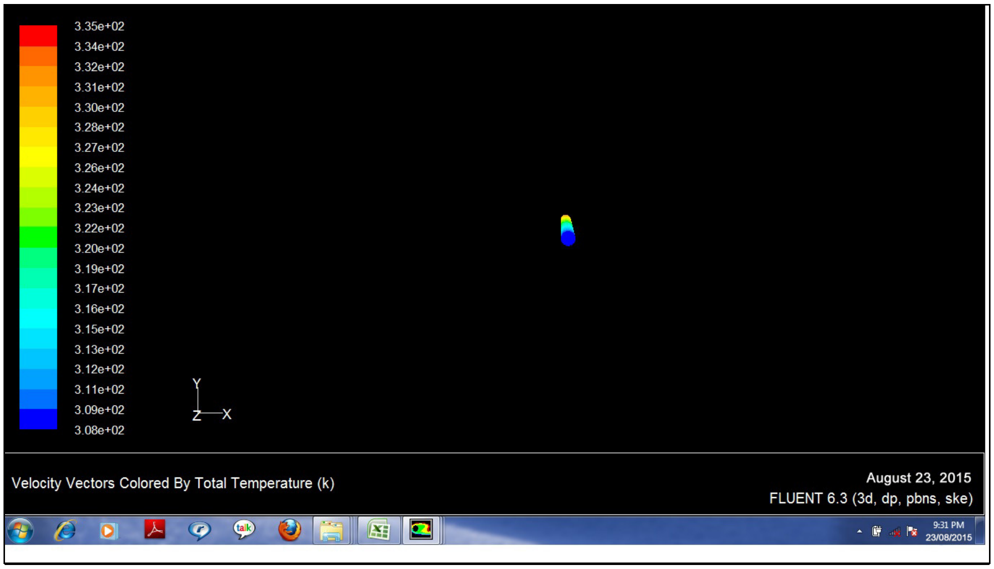

Figure 13: Total temperature contour

(2) Distance = 10 cm

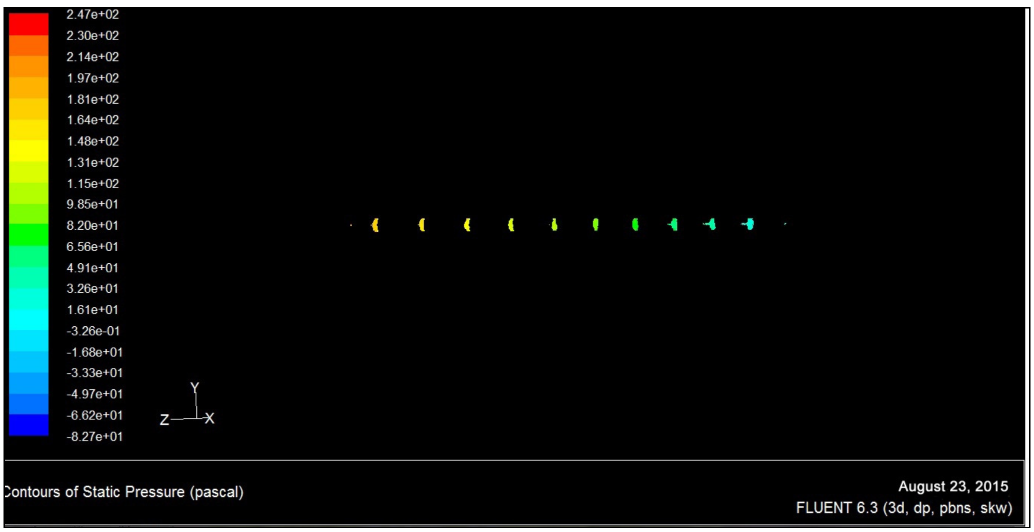

Figure 14: Static pressure contour

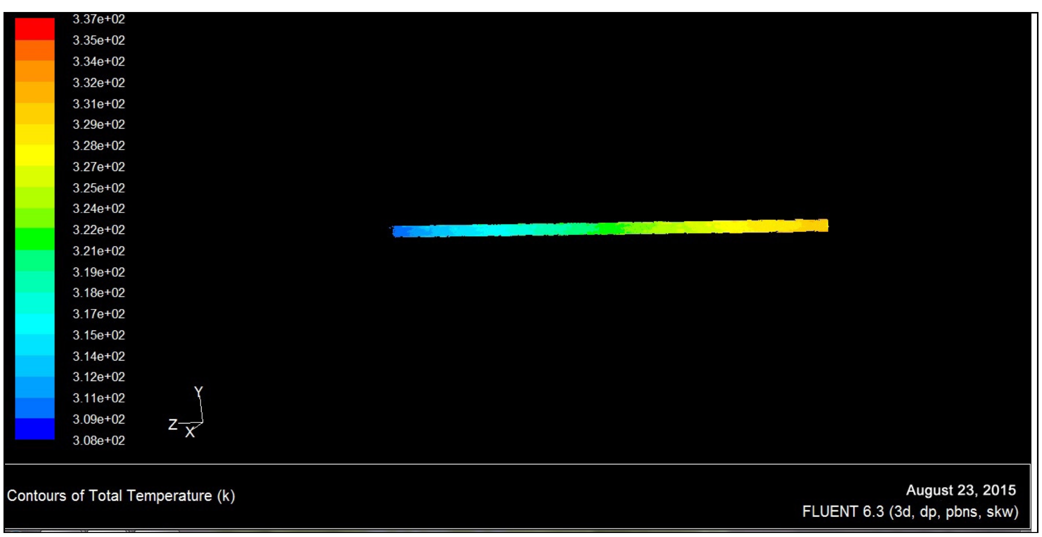

Figure 15: Total temperature contour

(3) At distance = 15 cm

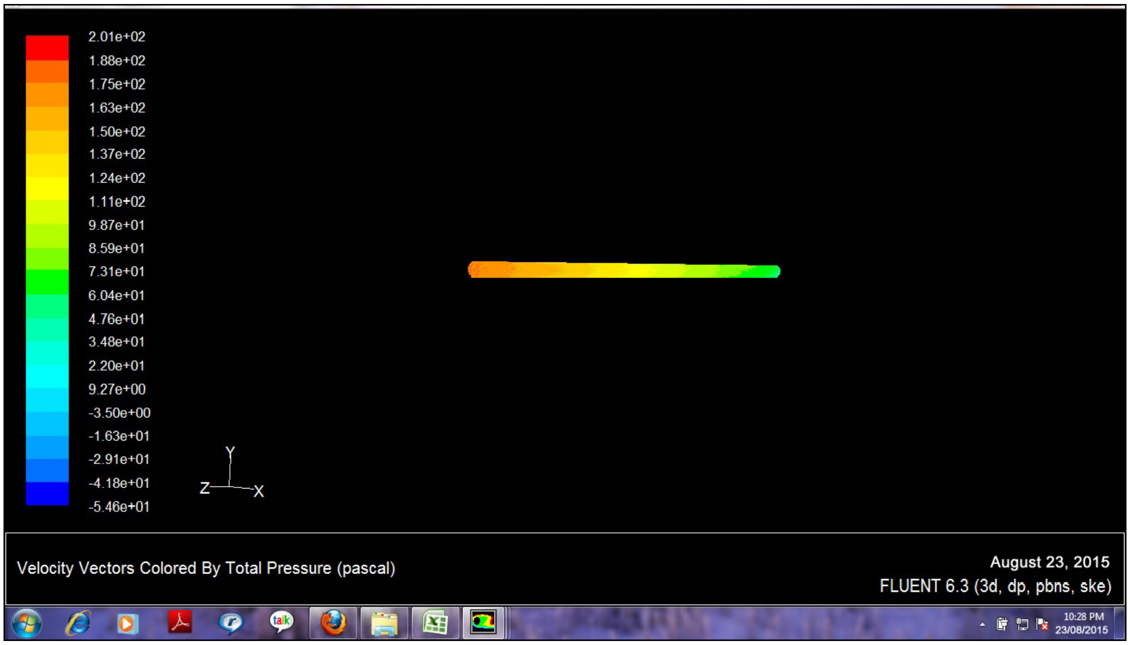

Figure 16: Total pressure contour

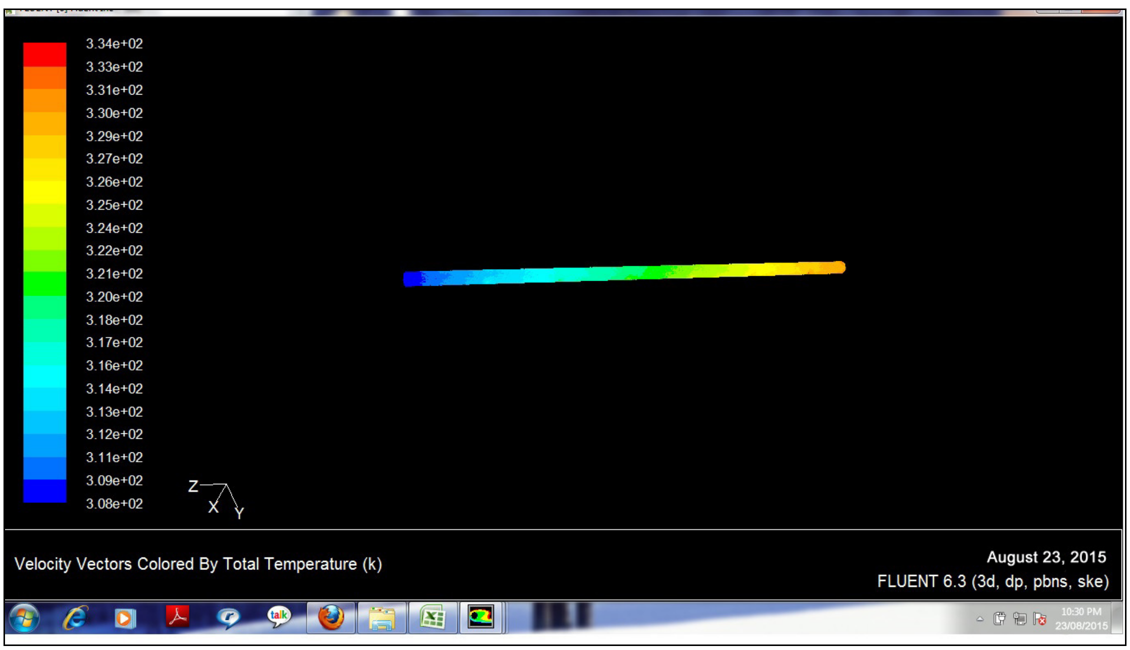

Figure 17: Total temperature contour

Spacing	ΔP in Pa	ΔT ºc
7.5	225	27
10	212	26
15	200	25

Table 5: Study of pressure drop and temperature change at different spacing between insert

2.4. Due to Spacing between Inserts

The second parameter of project is changing the spacing between insert. As the spacing between insert increases pressure drop first increases then decreases. on the other hand, as spacing increases the temperature difference first increases the decreases. From the above it indicates that spacing has no significant role for heat transfer enhancement. Variation of temperature difference is nearly 2 degrees. So now pressure drop is the main criteria. Pressure drop is minimum in case of spacing 15 cm .so 15 cm is suitable spacing for heat transfer enhancement.

Reynolds no	Nusselt no
11408	35.12
16037	46.11
19364	53.62

Table 6: Changing Reynolds number at angle 90^o

As Reynolds no increases, the value of Nusselt no also increases and pressure drop also increases .so fluid at low Reynolds no much more preferred for heat transfer enhancement.

3. Conclusion

In this paper, analysis is done by varying parameter. In first case, variable is angle of insert. Insert is arranged symmetrically on a thin wire. wire is centrally placed inside the tube. All these cases are considered modeling the geometry using gambit software. As angle of changes from 90 degrees to 60 degrees, pressure drop decreases with 18 Pa and temperature difference decreases by 2 degrees. Again angle changes from 60 degrees to 45 degrees, pressure drop has a little change of 2 Pa but temperature changes significant.

Second variable is spacing between insert. As spacing between insert increases from 7.5 cm to 10 cm, pressure drop increases by 18 Pa and temperature rise increases by 2 degrees. Again spacing is increased by5 cm, pressure drop decrease by43 Pa and temperature increment decrease by2 degree. So by analyzing data with both variables, it concludes that 45 degree is the suitable angle of insert and 15 cm is the suitable spacing between insert.

4. References

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