Heat Transfer Augmentation to Measure the Convective Heat Transfer Coefficient and Friction Factor of Stationary Square Duct with Various Angled Stagerred Ribs

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ABSTRACT
The thermal and hydraulic performances were examined experimentally for the square duct with arc of circular rib turbulators inclined at60^o. The inclined circular ribs are placed on opposite walls of the duct and the heat transfer coefficient and the frictional factor are calculated. The Stationary duct with aspect ratio (W/H=1) is considered for doing the analysis. The hydraulic and thermal performances aremeasured by calculating Frictional factor and the Nusselt number. Square ribs (w/e=1) are considered as the baseline configuration. The rib geometry configuration are having Blockage ratio of 0.083 and 0.125 and rib spacing (Pitch: Height ratio) is 10. The performance regarding heat transfer for the duct is calculated for varying Reynolds numbers. The results obtained for the duct with different ribs geometry configuration proved that as the rib width increases the thermal performance of the duct also increases. By combined effect of the rib width, rib spacing and flow parameter, the optimal cooling configuration was obtained.

Keywords
Convective heat transfer coefficient, Nusselt number, Friction Factor, Staggered position circular Ribs

1. INTRODUCTION
Heat exchangers have several engineering and industrial applications. The heat exchangers is quite complicated to design, as it requires exact analysis of pressure drop and heat transfer rate estimations apart from issues such as economic aspect and long term performances of the equipment. The major challenges in designing a heat exchanger is to make the equipment compact and obtain a high rate of heat transfer having minimum pumping power. In recent years, the high cost of energy and material has been resulted in an increased effort to produce more efficient heat exchange equipment. Sometimes there is a need for a compact heat exchanger in some applications, For example space vehicles, through heat transfer augmentation. For example, a heat exchanger in an OTEC (ocean thermal energy conversion) plant requires a heat transfer surface area ranging 10000 m^2/MW. Thus, an increasing the efficiency of the heat exchanger using an augmentation method may save the material cost considerably. Moreover, as a heat exchanger depreciates, the resistance to heat transfer increases due to scaling or fouling. These problems are common for heat exchangers used in chemical plants and marine application. Hence, to achieve a desired rate of heat transfer in aheat exchanger at an optimum pumping power, various techniques have been introduced in recent years and are discussed as follows.

2. HEAT TRANSFER AUGMENTATION
In general, some kinds of inserts are placed in the path of flow to enhance therate of heat transfer, and thus the hydraulic diameter of the passage is reduced. Enhancement of the heat transfer in a tube flow by specific inserts such as ribs, twisted tapes, dimples and wire coils and dimples is mainly due to blockage in the flow path, secondary flow and partitioning of the flow. Blockage in the flow path increases the pressure drop and increases viscous effects because of a reduction in the free flow area. Secondary flow provides an enhanced thermal contact between the fluid and surface because it creates swirling effect and results in improved temperature gradient due to the mixing of fluid, which leads to a high coefficient of heat transfer.

3. CLASSIFICATION OF AUGMENTATION TECHNIQUES
Heat transfer augmentation methods are classified in three categories [18]:

(1) Active method

(2) Passive method

(3) Compound method

3.1 Active method
This method involves some external power for the heat transfer enhancement. Moreover, it is not easy to provide external power in all applications. An example of active method is"induced pulsation by cams and reciprocating plungers".

3.2 Passive method
This method does not require external power.The heat exchanger industry has been craving for enhanced thermal contact and reduced pumping. A good heat exchanger design should have minimum generation of entropy or minimum exergy. It isimpossible to stop exergy loss, but it can be minimized by designing an efficient heat exchanger.

4. HEAT TRANSFER AUGMENTATION BY INSERTING RIBS

The use of ribs placing in the cooling channels or channel heat exchangers is widely used passive heat transfer enhancement method in single-phase internal flows. This heat transfer enhancement technique has been applied to various types of industrial applications such as shell-and-tube type heat exchangers, electronic cooling devices, thermal regenerators, and internal cooling systems of gas turbine blades. Periodically positioned ribs in the channels interrupt hydrodynamic and thermal boundary layers. For downstream of each rib the flow separates, recirculates, and impinges on the channel wall and these effects are the main reasons for heat transfer enhancement in such channels. The use of ribs increases the heat transfer rate and also substantial pressure loss. The rib geometry and arrangement also alter the flow field resulting in different convective heat transfer behavior. Particularly, the angled ribs, the rib to-channel height ratio, the rib cross-section, the rib pitch-to-height ratio are all parameters that influence the overall thermal performance and convective heat transfer coefficient.

In this present study, we did the experimental investigation of friction factor and coefficient of heat transfer of stationary square duct with 60^0 angled arc of circle ribs. The blockage ratios (Rib height to hydraulic diameter ratio) are 0.083 & 0.125 andthe rib pitch to height ratio is 10.

Several investigations have been done to study the effect of all these parameters of ribs on heat transfer rate and friction factor for two opposite rough surfaces. J.C.Han [1] studied the heat transfer in a square channel duct with ribs on two walls for nine rib configurations. Average heat transfer rate and friction factor were recorded for P/e=10 and e/D_h=0.0625. They reported that the angled ribs and 'V' ribs enhances more the heat transferthan the continuous ribs. The heat transfer rates and the friction factor were highest for the 60° orientation amongst the angled ribs. Han [2] also investigated the effect of surface heat flux ratio on the heat transfer in a square ribbed channel with e/D_h=0.063 and P/e=10, by either heating only one of the ribbed walls or both of them, or all the four channel walls. They concluded that the former two conditions resulted in an increase in the heat transfer rate with comparison to the latter one and the averageNusselt number decreases for increasing Reynolds numbers and the thermal boundary condition becomes less relevant at higher Reynolds number.

J.C.Han and Y.M.Zhang [3] reported the augmentation of heat transfer in a square channel with seven different configurations of broken ribs and found that 60° broken 'V' ribs provided higher heat transfer at about 4.5 times of the smooth channel and perform better than the continuous ribs.Liou and Hwang [4,5] used real time Laser Holographic ratios with ribs on one wall. Liquid crystal thermography technique was employed for determining local temperature field. Different P/e ratios ranging 6 to 16 with only a single e/D ratio of 0.1 were used. They found that the heat transfer rate is highest at (P/e) ratios of 9 and 12.

5. EXPERIMENTAL SET UP

A photo of the experimental setup is presented in Figure 1 while the detail of 60° inclined ribs placed on the upper and lower channel walls is depicted in Figure 2. In Figure 1, a high pressure blower is connected to a settling tank with the help of circular pipe. The square channel configuration was characterized by the channel height, *H* of 54 mm and a baffle pitch (P) of ten times of channel height (pitch ratio, PR=P/H=0.83, 1.25) and the attack angle of 60^o. The overall length of the channel was 2000 mm. The test square channel made of 4 mm thick wooden plates has a cross section of 54×54 mm^2 and 650 mm long (L). The ribs dimensions were 4.482, 6.759 and 9.018 high (*e*) and 0.3 mm thick (*t*).

The plate-type heater is energized with help of AC power supply which is used for heating all walls of the test section in order to maintain a uniform surface heat flux. Air is used as a tested fluid in both the pressure drop and heat transfer experiments, is flows into the systems with the help of a 7.5 KW high-pressure blower. The variation in the operating speed of blower is done with the help of inverter to provide desired flow rates of air. The inclined manometer is used in order to measure pressure across the orifice. There are 11 thermocouples are used to measure temperature distribution along lower, upper and side walls. All thermocouples are*K*type, 1.5 mm diameter wire.

Fig 1. Experimental Setup

Fig2. 60^0 angled circular ribs with blockage ratio of 0.083

6. SELECTION OF VARIOUS COMPONENTS

Selection of Thermocouples

Chromel-Alumel thermocouple wire selected is calibration type is k-type. It is the most generally used type of thermocouple. Chromel is non-magnetic it is positive conductor yellow color code. Alumel is slightly magnetic; it is negative conductor, with red color code. Identification can be made by magnetic response. Its temperature range is -200 to 1250. Limit of error is ± 2.2. Color code for jacket of wires is yellow.

6.1 Selection of simple U tube manometer & inclined U- tube manometer

For plain tube, to calculate the friction factor, pressure drop is to be determined. The simple U-Tube manometer & 45˚ inclined U- Tube manometer are employed to measure losses in plain tube & test section respectively. Water of density 1000 Kg/m^3& Diesel of density 800 Kg/m^3 are taken as manometer liquid in simple U-Tube manometer & 45˚ inclined U- Tube manometer respectively.

6.2 Selection of orifice meter

For maximum limiting flow 0.06 m^3/s, selection of orifice meter is done &the coefficient of discharge is 0.627

7. CONSTRUCTION & ASSEMBLY

A photograph of the experimental apparatus is presented in Figure 1 while the details of 60^o inclined arc of circle ribs placed on the upper and lower channel walls are depicted in Figure 2. For connecting a high pressure blower having capacity of 7.5 HP to a flow straightner circular pipe is used shown in Figure 1. The square channel configuration was characterized by the channel height, Hof 54 mm and a rib pitch (P) of ten times of rib height (pitch ratio, P/e=10) and the attack angle of 60^o. The overall length of the channel was 1000 mm. The test square channel made of 20 mm thick wood has a cross section of $54x54mm^2$ and 650 mm long (L). The ribs dimensions were 4.482 , 6.759 mm height (e) and thickness (w) are 4.482, 6.759 mm according to the blockage ratio (e/D_h=0.083, 0.125) respectively. The test section consisted of the four walls. The air flow rate in the system is measured by an orifice plate meter. The pressure across the orifice is measured using differential water U -tube manometer. To measure the temperature distributions on the upper wall of duct, nine thermocouples are fitted. The thermocouples were inserted in holes drilled from the top face and axial separation is 65 mm apart. The thermocouples connected to the temperature indicator. Two static pressure taps are located at the top of the principal wall to measure axial pressure drops across the test section, which is used to evaluate average friction factor. The pressure drop is measured by a differential water U- tube inclined manometer.

8. DATA ANALYSIS

The aim of this experiment is to investigate the Nusselt number in the channel. The Reynolds number based on the duct hydraulic diameter, *Dh*, is given by[17],

$$\mathbf{Re} = U\,Dh / \nu \qquad (1)$$

where*U* and ν are the mean air velocity of the channel and kinematics viscosity of air, respectively. The average heat transfer coefficient "h" is evaluated from the temperatures measured and heat inputs. Heat is added uniformly to fluid (*Qair*) and the temperature difference of wall and fluid (*T*w−*T*b) is recorded, the average coefficient of heat transfer will be evaluated from the experimental data with the help of following equations:

$$Qair = Qconv = m\,\&Cp(To - Ti) = VI \qquad (2)$$

$$h = Q_{conv} / A\,(T_{S1} - T_b) \qquad (3)$$

in which,

$$Tb = (To + Ti) / 2 \qquad (4)$$

And

$$T_{S1} = \sum T_S / 10 \qquad (5)$$

The term *A* is the convective heat transfer area of the heated upper channel wall and $T_{S1}$is the average surface temperature obtained from local surface temperatures, *T*s, along the length of the heated channel. The terms *m* & ,*Cp*, *V* and *I* are the air mass flow rate, specific heat, voltage and current, respectively.

Then, average Nusselt number, *Nu*, is written as:

$$Nu = h\,D_h / k \qquad (6)$$

The friction factor, *f*, is evaluated by,

$$f = (2 / (L / D_H))\,(\Delta P / \rho\,U^2) \qquad (7)$$

where, ΔP is a pressure drop across the test section and ρ is density. The thermo-physical properties of the airare determined at the bulk air temperature, (T_b) from Eq. (4).

For equal pumping power,

$$(V\Delta P)_0 = (V\Delta P) \qquad (8)$$

in which *V* is volumetric air flow rate and the relationship between friction and Reynolds number canbe expressed as:

$$(f\,Re^3)_0 = (f\,Re^3),$$

$$Re_0 = Re\,(f / f_0)^{1/3} \qquad (9)$$

The thermal enhancement factor, η, defined as the ratio of coefficient of heat transfer of an augmented surface, *h*to smooth surface of duct , *h0*, at the same pumping power,

$$\eta = (h / h_0)_{pp} = (Nu / Nu_0)_{pp} = (Nu / Nu_0)\,(f / f_0)^{-1/3} \qquad (10)$$

9. RESULTS

The following results have been found out from the steam table taking into account the mean temperature (T_b) of the respective observations.

Table 1:Readings and calculations of 60^0 angled circular ribs with blockage ratio 0.125

Sr. No	Head	P (kg/m3)	K (W/mK)	μ (Pa-s)	Cp (J/kgK)	Volume Flow Rate (m3/s)	Velocity U (m/s)	Re
1	5.5	1.093	28.26	0.1962	1005	0.02033	10.165	49204
2	6.8	1.0897	28.33	0.19669	1005	0.02266	11.33	54711
3	8.1	1.0864	28.4	0.19718	1005	0.02475	12.375	59712
4	9.5	1.0831	28.47	0.19767	1005	0.02686	13.43	64667
5	11.1	1.0798	28.54	0.19816	1005	0.02908	15.54	69901
6	12.7	1.0765	28.61	0.19865	1005	0.03115	15.575	74769

Table 2:Result of 60^0 angled circular ribs with blockage ratio 0.125

Sr. No.	Head	f	Mass Flow Rate (kg/s)	T_b (oC)	T_w (oC)	Q (Watt)	h (W/m2K)	Nu	Nu/Nu_0	f/f_0	THP
1	2.7	0.00933	0.038	50	71.556	152.1	100.5	206.3	3.1438	5.4882	1.782
2	3.2	0.00894	0.042	51	72.778	169.1	110.6	227.1	3.1124	5.2588	1.79
3	4	0.00938	0.046	52	73.333	184.5	123.2	253	3.156	6.7	1.674
4	4.8	0.0096	0.05	53	74.222	199.9	134.1	275.4	3.1608	6	1.74
5	5.4	0.00924	0.054	54	74.889	216	147.3	302.5	3.2307	6.66	1.717
6	6	0.00898	0.057	55	75.667	231.1	159.3	327	3.272	6.9076	1.718

Table 3:Readings and Calculations of Smooth Channel

Sr. No.	Head	ρ (kg/m3)	K (W/mK)	μ (Pa-s)	Cp (J/kgK)	Volume Flow Rate (m3/s)	Velocity U (m/s)	Re
1	5.5	1.07485	28.645	0.19889	1005	0.02051	10.25	49204
2	6.8	1.07155	28.715	0.19938	1005	0.02285	11.43	54711
3	8.1	1.06825	28.785	0.19987	1005	0.02498	12.49	59712
4	9.5	1.06495	28.855	0.20037	1005	0.02709	13.55	64667
5	11.1	1.06165	28.925	0.20085	1005	0.02919	14.59	69901
6	12.7	1.05845	28.995	0.20135	1005	0.03142	15.71	74769

Table 4:Result of Smooth Channel

Sr. No.	Head	Mass Flow Rate (kg/s)	T_b (oC)	T_w (oC)	Q (Watt)	h (W/m2K)	Nu_o (Channel)	f_o
1	0.5	0.038	55.5	106.33	114.05	31.93	65.6215	0.0017
2	0.6	0.042	56.5	107.33	126.81	35.537	72.9658	0.0017
3	0.6	0.046	57.5	108	138.41	39.04	80.1613	0.0014
4	0.8	0.05	58.5	109	149.89	42.281	86.8128	0.0016
5	0.8	0.054	59.5	110.11	162.02	45.603	93.633	0.0014
6	0.9	0.057	60.5	111.22	173.31	48.672	99.9349	0.0013

10. DISCUSSION ON RESULTS

The results that we are going to discuss are related to the various temperature readings that we have taken. From the temperatures we found out the Heat transfer Co-Efficient and Friction Factor.We can see that the nusselt number varies from 152.783 to 252.744 for staggered position of the rib with 60^0 angle and blockage ratio of 0.0.083 from Fig no. 3.We can see that the nusselt number varies from 206.3 to 327 for staggered position of the rib with 60^0angle and blockage ratio of 0.125 from Fig no. 3.We can see that the friction factor decreases from 0.0097 to 0.0085 for staggered position of the ribs with 60^0 angle and blockage ratio of 0.083 From Fig no. 4. We can see that the friction factor decreases from 0.00933 to 0.00898 for staggered position of the ribs with 60^0angle and blockage ratio of 0.125 From Fig no. 4.

We can see that the nusselt number varies from 380.505 to 506.892 for inline position of the rib with 60^0 angle and blockage ratio of 0.0.083 From Fig no. 5.We can see that the nusselt number varies from 252.073 to 391.59 for inline position of the rib with 60^0 angle and blockage ratio of 0.125 From Fig no. 5.

We can see that the friction factor decreases from 0.0187 to 0.0171 for inline position of the ribs with 60^0 angle and blockage ratio of 0.083 From Fig no. 6.We can see that the friction factor decreases from 0.0176 to 0.0174 for inline position of the ribs with 60^0angle and blockage ratio of 0.125 From Fig no. 6

From these above given points we can say that the Nusselt number for Inline Position is greater than that for the Staggered Position. Thus the Heat Transfer Co-efficient for the Inline Position is also greater than that for the Staggered Position.

From above given points we can say that the Friction Factor for Inline Position is greater than that for the Staggered Position.

Table No. 5:Validation of set up for friction factor

Re	49204	54711	59712	64667	69901	74769
Exp values	0.0017	0.0017	0.0014	0.0016	0.0014	0.0013
$f = 0.046Re^{-0.2}$	0.0053	0.00519	0.00509	0.00502	0.00494	0.00487

11. CONCLUSIONS

Experimental study has been carried out to investigate airflow friction and heat transfer characteristics in a channel fitted with different inclined arc of circle rib heights for the turbulent regime, Reynolds number of 40,000 to 75,000. The use of the rib turbulators causes a very high pressure drop increase, especially for high flow blockage rib, e/D_h=0.167 and also provides considerable heat transfer enhancement, Nu/Nu0=2.93-6.73, depending on rib height. Nusselt number augmented with the rise of Reynolds number. The 60˚ angled inclined arc of circle ribs with e/D_h=0.125 should be applied to obtain higher thermal performance, resulting more compact heat exchanger. The best operating region for all rib turbulators is found at higher Reynolds number values.

12. FUTURE SCOPE

Experiment can be conducted for higher Reynolds number, higher heat flux, different aspect ratio, blockage ratio & pitch to height ratio.Experiment can also validate with CFD analysis

13. GRAPHS

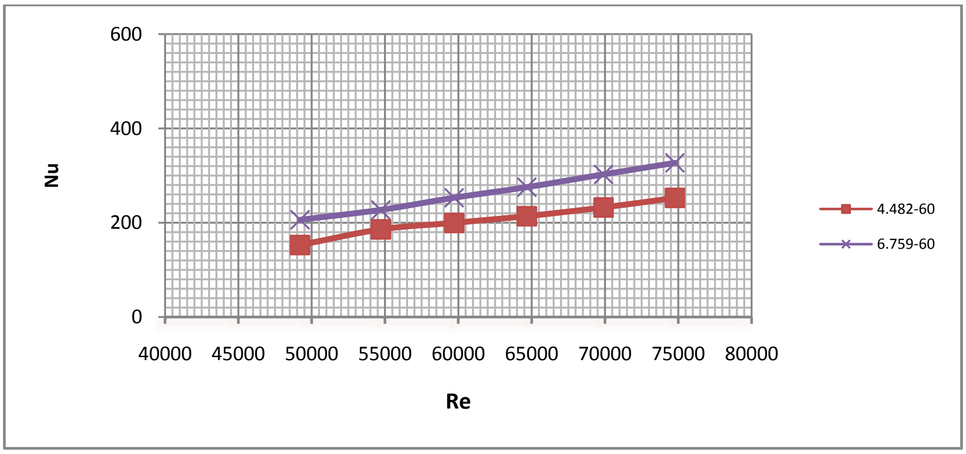

Fig No. 3: Nusselt Number vs Reynolds Number For Staggered Position.

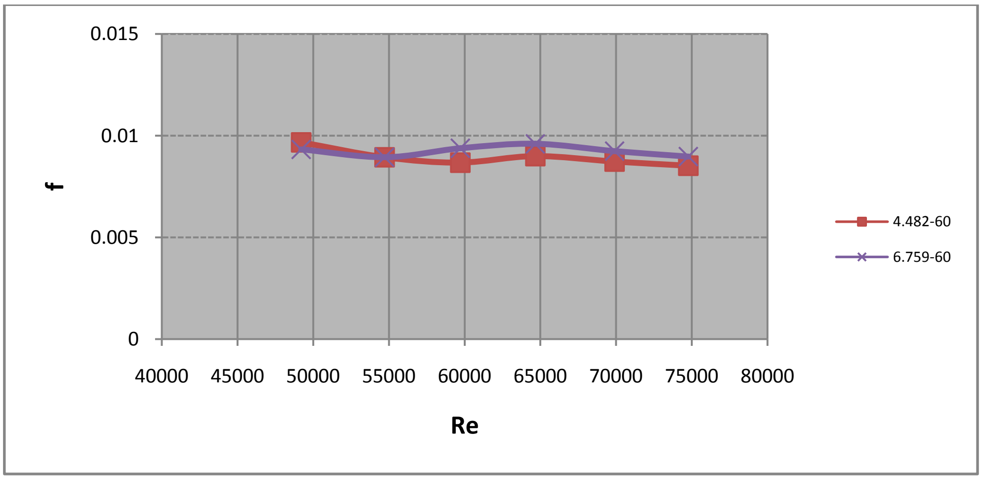

Fig No. 4: Friction Factor vs Reynolds Number for Staggered Position.

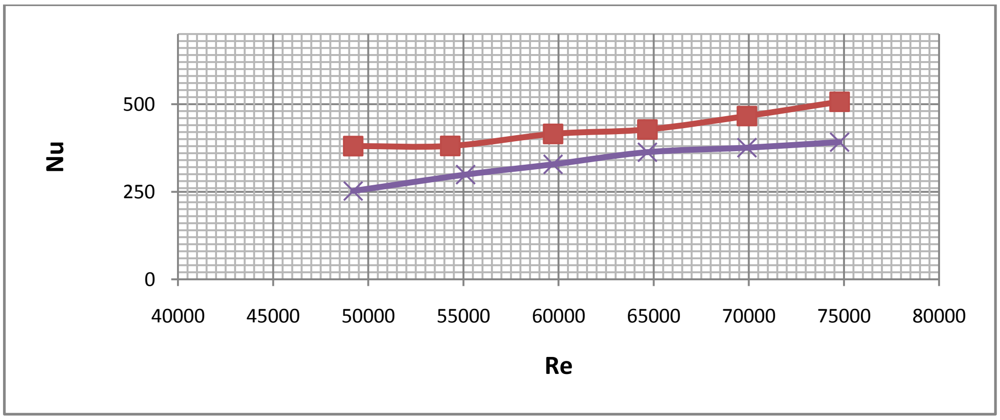

Fig No. 5 : Nusselt Number vs Reynolds Number for Inline Position.

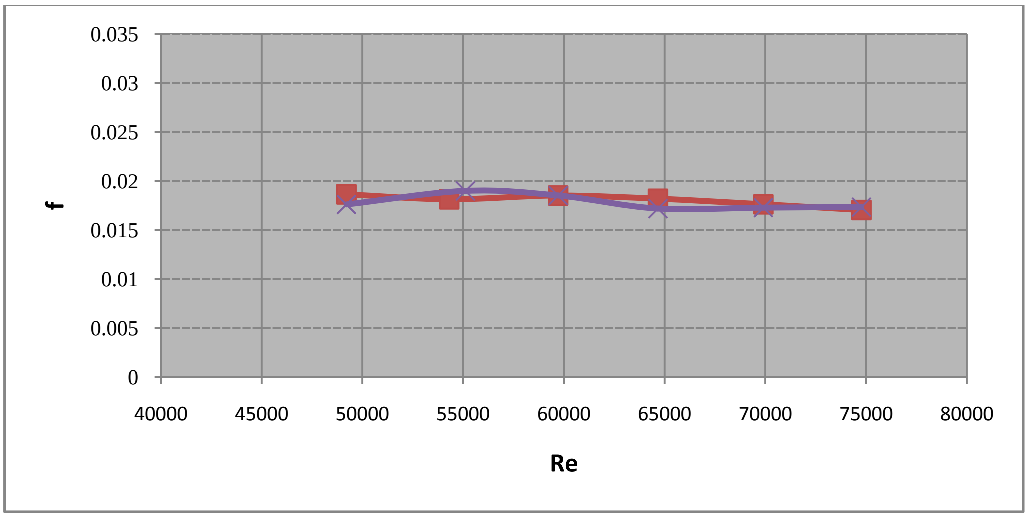

Fig No. 6 : Friction factor vs Reynolds Number for Inline Postion.

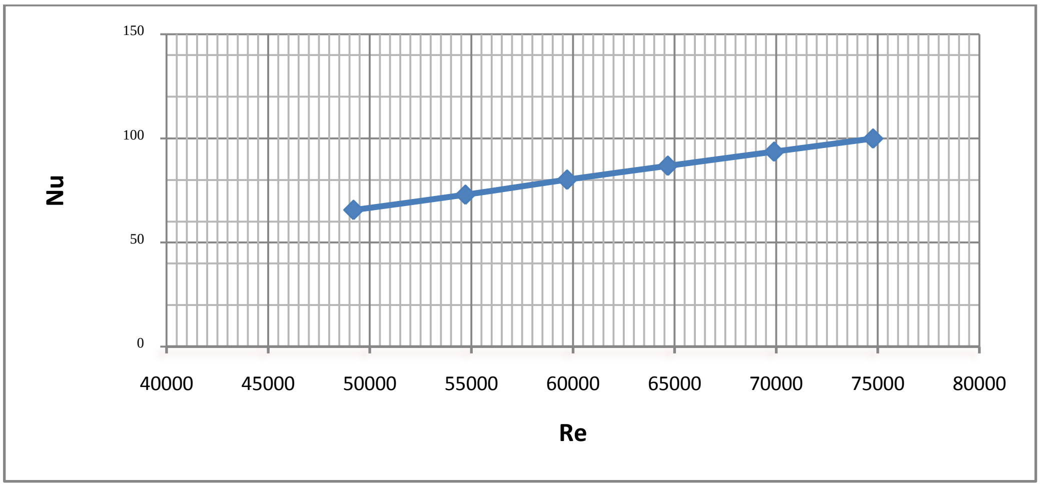

Fig No. 7 :Nusselt Number vs Reynolds Number for smooth Channel.

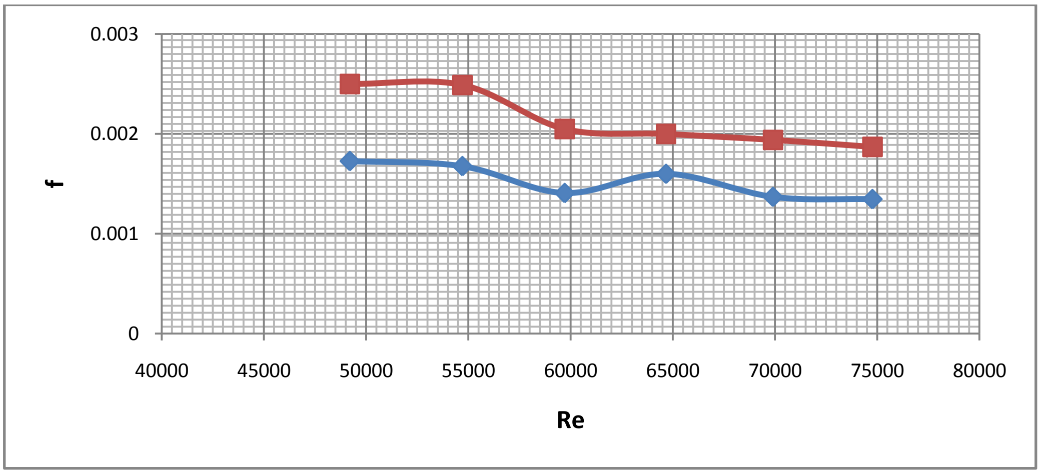

Fig No. 8: Friction factor v/s Reynolds number graph for smooth channel

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