Experimental Investigation of Heat Transfer Analysis of Ribbed Duct for Thermal Performance Enhancement

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Abstract -In this paper a study of heat transfer enhancement of Divergent plain duct and divergent ribbed duct investigated by experimentally. The turbulence intensity of fluid flow increased due to obstacles present in the flow passage. For heat transfer enhancement active and passive methods are used. Passive method uses the insertion of fins, ribs, bumps, dimples, baffles, wires etc. in the flow passage to improve the heat transfer rate. In this paper rib is used as a tabulator. These ribs are arranged in staggered arrangement inner surface of duct .The divergent duct has an angle of 1.145 degree to give the minimum pressure drop and friction factor. The measurement was conducted within the range of velocity from 3.2 to 16 m/s. (Reynolds's No. 5000-25000).The thermal performance like Nusselt Number, Reynolds number and Heat transfer coefficient of divergent rib duct compared with plain divergent duct under the pressure drop, mass flow rate due to steam-wise acceleration or retardations. The thermal performance of divergent rib duct is higher than the plain divergent duct. Thermal Performance of Divergent Duct is increase by 34% due to Rib tabulator.

Key Words: Divergent Duct, Nusselt Number, Reynolds Number, Heat transfer coefficient, Ribs, Heat transfer Coefficient.

I. INTRODUCTION

Thermal performance of system and convective heat transfer is depends on the intensity of turbulence. Intensity of turbulence is increases with restriction in the flow path. Restriction is in the form of rough surfaces, rib, baffles, bumps, wires etc.Duct with artificial roughness in the form of rectangular ribs is one of the important and effective design improvements that has been proposed to improve the thermo hydraulic performance. Staggered arrangement is one of better arrangement than other arrangement.

. Enhancement of heat transfer rate is very important in all types of thermo technical applications for industry. Because of this there is savings in primary energy and also reduction in size and weight. Enhancement of heat transfer is essential in drying of food (food preservation) application. The drying time is dependent on the temperature and flow rate of the air. Because to reduce the drying time temperature of air or flow rate must be increased. Also, to increase the temperature if heater input is increased excessively, due to excess heating the system may not work efficiently or at the extreme it may fail. Therefore, the dissipation of the heat from the pipe surface to the flowing air through the pipe is very important for maintaining the efficient and reliable functioning of the plant.

Generally there are two main types of enhancing the heat transfer rate.

I. Passive method.
II. Active method.

PASSIVE TECHNIQUES: These techniques generally use surface or geometrical Modifications to the flow channel by incorporating inserts or additional devices. They Promote higher heat transfer coefficients by disturbing or altering the existing flow behavior (except for extended surfaces) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power.

ACTIVE TECHNIQUES: These techniques are more complex from the use and design point of view as the method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. It finds limited application because of the need of external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential as it is difficult to provide external power input in many cases.

M.Alhajeri*,et al.-[1]Fluid Flow and heat transfer results are presented from a ribbed U-tube, which models passages used to cool the blades in gas turbine engines. Computational fluid dynamics is used here to predict the air flow behavior and the surface Nusselt number distributions. The model of the coolant passage consists of two square legs that are connected by a sharp, 180degree bend with a rectangular outer wall. Four ribs are placed in each side of the leg and mounted in a staggered arrangement. The height and width of the rib are equal to 0.1 duct width, and the rib spacing is 10 times the rib height. Based on inlet flow conditions, the Reynolds number (Re) is 95000. It was found that, after the flow resettles from the disturbances created by the obstacle of the first rib or the effect of the bend, the flow forms two re-circulations, a large one behind the rib and a small one ahead of the rib. The maximum values of the Nusselt numbers are located at a distance of almost one rib height h ahead of the flow reattachment point

Liang-Bi Wang, Wen-Quan, et al.-[2] The local heat transfer and pressure drop characteristics of developing turbulent flows of air in three different types of ribbed ducts. These include the constant cross section square duct(straight duct), the ribbed diverging square duct and the ribbed converging square duct. The convergent/divergent duct has an inclination angle of 1 degree. The measurement was conducted within the range of Reynolds No. from 10000 to 77000.The heat

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transfer performance of the convergent/divergent duct is compared with the ribbed straight duct under three constraints identical mass flow rate, identical pumping power and identical pressure drop. Because of the stream wise flow acceleration or deceleration. The local heat transfer characteristics of the convergent and divergent duct are quite different from those of straight duct. In the straight duct, the fluid flow and heat transfer become fully developed after 2-3 ribs, while in the convergent and divergent ducts there is no such trend. The comparison shows that among the three ducts,the divergent duct has the highest heat transfer performance, the convergent duct has the lowest, while the straight duct locates somewhere in between.

Anil P. Singh, Varun , Siddhartha, et al.-[3] In this present experimental investigation the effect of geometrical parameters of multiple arc shaped roughness element on heat transfer and friction characteristics of rectangular duct solar air heater having roughness on the underside of the absorber plate have been studied. The parameters were selected on the basis of practical considerations and operating conditions of solar air heaters. The experiments carried out encompasses Reynolds number (Re) in the range of 2200-22, 000, relative roughness height (e/D) range of 0.018–0.045, relative roughness width (W/w) ranges from 1 to 7, relative roughness pitch (p/e) range of 4–16 and arc angle (α) ranges from 30 to 75°.The thermo-hydraulic performance parameter was found to be best for relative roughness width (W/w) of 5.

Umesh potdar, Nilesh shinde, Manoj Hambarde , et al.-[4] The thermal & hydraulic performances were examined experimentally for the stationary square channel with V shaped & 45°inclined arc of circle rib tabulators. Ribs were placed on opposite walls and the heat transfer coefficient and frictional factor were calculated. Stationary channel with aspect ratio one (W/H=1) was considered for the analysis. The thermal & hydraulic performances were measured by calculating the Nusselt number and frictional factor. Square ribs (w/e = 1) were considered as the baseline configuration. Rib geometries, comprising three rib height-to-channel hydraulic diameter ratio (blockage ratio) of 0.083,0.125 & 0.167 as well as rib spacing (pitch to height ratio) is 10.The heat transfer performance for the channel was calculated for Reynolds numbers 45000 to 75,000. The results obtained for the channel with different ribs configuration proved that the increase in rib width increase the thermal performance of the channels. By combined effect of rib width, rib spacing and flow parameters, the optimal cooling configuration was obtained.

Sivakumar. K., Dr.E Natarajan, Dr.N. Kulasekharan, et al.-[5] In this present work, the local heat transfer and Nusselt number of developed turbulent flow in convergent/divergent square duct have been investigated computationally. Experimental results for this configuration are reported elsewhere in three different channels viz., smooth square duct, ribbed convergent square duct and ribbed divergent square duct. The angle of convergence of the duct is about 1°. Among the three channel shapes, the convergent square duct with ribs alone is considered for carrying out the present computational analysis. The computational analysis was conducted within the range of Reynolds number from10, 000 to 77,000. The heat transfer performance of the convergent ducts from the present analysis is compared with that of the experimental data reported and good agreement has been found .Because of the stream-wise flow acceleration, the local heat transfer characteristics of the convergent ducts are quite different from those of the straight duct. There is no trend of flow and heat transfer development or flow becoming fully developed in the case of convergent ducts.

Lesley M.Wright, Wen-lung Fu, Je-chin Han, et al.-[6]An experimental study was performed to measure the heat transfer distributions and frictional losses in rotating ribbed channel with an aspect ratio of 4:1. Angled discrete angled, V-shaped and discrete V-shaped ribs investigated as well as the newly proposed W-shaped and discrete W-shaped ribs. In all cases, the ribs are placed on both the leading and trailing surface of the channel and they are oriented 45 deg to the mainstream flow. The rib height to hydraulic diameter ratio (e/D) is 0.078 and rib pitch to height ratio (P/e) is 10. The channel orientation with respect to the direction of rotation is 135 deg. The range of flow parameters includes Reynolds numbers (Re=10,000-40,000), rotation number (Ro=0.0 – 0.15) and inlet collant to wall density ratio ($\Delta\rho/\rho$=0.12). Both heat transfer and pressure measurements were taken, so the overall performance of each rib configuration could be evaluated. It was determined that the W-shaped and discrete W-shaped ribs had the superior heat transfer performance in both nonrotating and rotating channels. However these two configurations also incurred the greatest frictional losses while the discrete V-shaped and discrete angled ribs resulted in the lowest pressure drop. Based on the heat transfer enhancement and pressure drop penalty, discrete V-shaped ribs and discrete W-shaped ribs exhibit the best overall thermal performance in both rotating and nonrotating channels .These configurations are followed closely by the W-shaped ribs. The angled Rib configuration resulted in the worst performance of the six configurations of the present study.

Md.J.NINE, GyeongHwan LEE, HanShik CHUNG, Myoungkuk Ji, Hyomin JEONG, et al.-[7] The article represents an experimental investigation on friction and turbulent flow characteristics of free airflow through a rectangular duct fitted with semicircular ribs of uniform height (e = 3.5 mm) on one principle wall. The aspect ratio of the rectangular duct was AR= 5 where the duct height (H) was 30 mm. Four different rib pitches (P) of 28mm, 35mm, 42 mm and 49 mm were examined with constant rib height to hydraulic diameter ratio (e/Dh= 0.07) and constant rib height to channel height ratio (e/H = 0.116). The experimental results show some significant effects of pressure drop as well as turbulent characteristics at various configurations among different numbers of rib arrangements varying Reynolds number in the range of 15000 to 30000. Experimental results have been compared with numerical analysis and it can be seen a good agreement. The result explains the phenomena elaborately between two periodic ribs and enables to optimize the rib pitch ratio in terms of turbulence kinetic energy for maximum heat transfer.

P.R.Chandra, M.E.Niland, J.C. Han, et al.-[8] An experimental study of wall heat transfer and friction characteristics of a fully developed turbulent air flow in a rectangular channel with transverse ribs on one, two and four wall is reported. Tests were performed for Reynolds No. ranging from 10,000 to 80,000. The pitch-to-rib height ratio, P/e, was kept at 8 and rib height to channel hydraulic

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diameter ratio, e/Dh, was kept at 0.0625. The channel length to hydraulic diameter ratio, I/Dh, was 15. The heat transfer coefficient and friction factor values were enhanced with the increase in the number of ribbed walls. The heat transfer roughness function, G(e+), decreased with additional ribbed walls and compared well with previous work in this area. Friction dada obtained experimentally for the case with four ribbed walls compared well with the values predicted by the assumed theoretical relationship used in the present study and past publication .Results of this investigation could be used in various applications of internal channel turbulent flows involving different number of roughened walls.

Aman Sai ,Ranjit Singh, Brij Bhushan , et al.-[9] In the present paper CFD based investigation has been reported in order to study effect of roughness element pitch on heat transfer and friction characteristics of solar air heater duct for a range of system and operating parameters. It has been observed that roughened absorber plate results augmented heat transfer coefficient at the cost of frictional penalty. In order to predict performance of the system, Nusselt number and friction factor correlations have been developed by using the data generated under CFD based investigation.

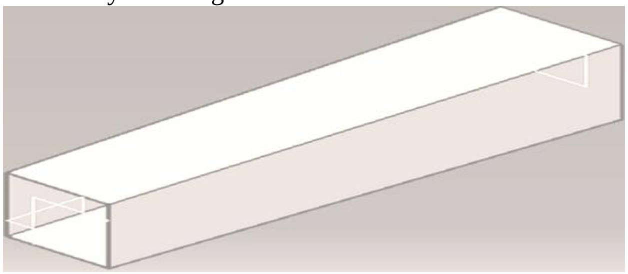
Sivakumar. K., Dr.E Natarajan, Dr.N. Kulasekharan, et al.-[10] The work reported in this paper is a systematic experimental heat transfer and pressure drop comparison between smooth and three different sized square ribbed divergent rectangular ducts. The heights of the rib tabulators (*e*) were 3, 6 and 9 mm. This yields a rib height (*e*) to mean hydraulic diameter of the duct (*Dm*) ratio of 0.035, 0.0697 and 0.1046 respectively with a fixed rib pitch (*p*) to test section inlet width (*w*) ratio of 0.6, and to maintained identical mass flow rate. The results obtained from the ribbed ducts were compared with that of the same parameter smooth (without ribs) divergent rectangular duct. The enhanced heat transfer rate for the 3 mm height rib divergent rectangular duct is more than 6, 9 mm rib height rectangular divergent duct and smooth duct. For pressure drop point of view 6 and 9 mm rib height is higher than 3 mm and smooth duct respectively.

II. EXPERIMENTAL APPARATUS:

Fig 1. Actual setup of Experiment

A actual setup of the system is shown in fig 1, This is an indoor setup consists of Band heaters of 500 watts capacity Heater regulator to supply the regulated power input to the heater. Digital Voltmeter and Ammeter to measure power input to the heater. Thermocouples (k- type) at suitable position to measure the temperatures of body and the air. Digital Temperature Indicator with channel selector to measure the temperatures .Blower unit is used to generation of air. A control valve is provided to regulate the air flow. Anemometers are used to measure the air velocity. Control panel to house all the instrumentation.

Divergent duct 4 m long. These are inlet section in $0.02*0.02m^2$, outlet section is $0.036*0.036m^2$.This geometry makes the divergent duct having 1.145 ^{o} inclination along the direction in the cross section. The Divergent plain duct with uniform cross-section as shown in fig 2. and The Divergent Ribbed duct with uniform cross-section as shown in fig 3.The Rib arrangement is staggered in inner all surfaces of ducts. The rib size & arrangement are shown in fig4. with a fixed e/Dh = 0.045 and p/e= 8.To get a detetailed distribution of the local heat transfer coefficient, 4 Thermocouples(k-type) are imbedded in one of the ribbed wall with specific distance 0.08m,0.16m.0.24m,0.32m. along it’s centreline to measure the surface temperatures .The ducts are well insulated by an insulation strip/material. The divergent duct was made of stainless steel .the wall thickness of duct is 2mm .The thermal conductivity of divergent duct material is 16.27W/mk.



.Fig 2. Divergent Duct

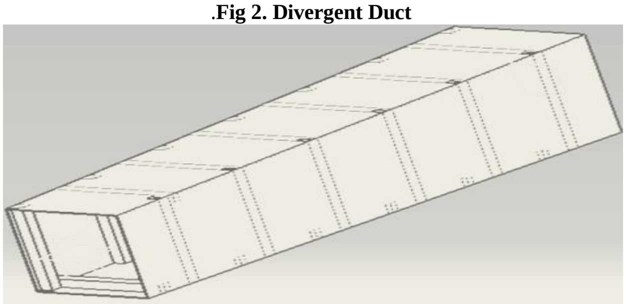


Fig 3. Divergent Ribbed Duct

Fig 4. Staggered Arrangement of ribs

III. EXPERIMENTAL PROCEDURE:

Instrument should check whether all equipment is in proper working before starting the setup. Also check all the Instruments are in proper way & correct. Switch on the MCB and then console on switch to activate the control panel. Anemometer is used to measure the air velocity. Thermocouples are used to measure the surface temp. of ribs. Flow of air can be control with the help of control valve for the proper value of Reynolds no. Switch on the blower unit and heater. Wait for reasonable time to allow temperatures to reach steady state. Collect all the relevant data concern with setup & required data for various Reynolds’s no. 5000-25000. Measure inlet temp, surface temp (T1, T2, T3, and T4) to outlet temp. at known time interval. .Repeat the

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experiment for different values of power input to the heater and blower air flow rates.

IV. DATA REDUCTION:

The goals of the experiments are calculated by following equations:

1. $T_b = (T_i + T_o)/2$
2. $T_w = (T_1 + T_2 + T_3 + T_4)/4$
3. $m = (\rho * A_c * V)$
4. $A_c = a+((b-a)/L)*(L/2)$
5. $A_s = \Pi * D_h * L$
6. $Q_s = m*C_p*(T_i - T_o)$
7. $Q_{loss} = (0.05*Q_s)$
8. $Q_{net} = (Q_s - Q_{loss})$
9. $h = Q_{net}. (A_s*(T_b - T_w)$
10. $Re = (V*D_h) / v$
11. $Nu = (h*D_h)/K$

V.RESULTS AND DISCUSSION:

1	2	3	4	5
RE	Tb	Tw	Tw-Tb	ΔT
5000	49.25	43.75	5.5	1.5
10000	49.5	44	5.5	1
15000	49.5	44.25	5.25	1
20000	49.5	44.88	4.625	1
25000	49.5	45.38	4.125	1

Table 1.a.Divergent Duct without Ribs

6	7	8	9	10
m	CP	Q	hexp	Nuexp.
0.002782	1006	3.988508	20.6107	23.84709
0.005565	1006	5.31801	27.4809	31.79611
0.008347	1006	7.977015	43.18431	49.96532
0.011129	1006	10.63602	65.3600	75.62319
0.013911	1006	13.29503	91.6030	105.987

Table 1.b.Divergent Duct without Ribs

1	2	3	4	5	6
RE	Tb	Tw	Tw-Tb	ΔT	% ENH
4638.46	48.25	41.75	6.5	3.5	
9276.91	49	43.375	5.625	2	
13915.4	49.25	44.875	4.375	1.5	34.05
18553.8	49.25	45.5	3.75	1.5	

Table 2.aDivergent Duct with Ribs:

7	8	9	10	11	12
M	CP	Q	hexp	Nuexp.	%ENH
0.0024	1006	8.02449	35.0873	37.697	36.74
0.0048	1006	9.17085	46.3375	49.784	36.13
0.0072	1006	10.3172	67.0239	72.009	30.61
0.0096	1006	13.7562	104.259	112.01	32.48

Table 2.bDivergent Duct with Ribs

1	2	3	4	5
(Tw-Tb)1	(Tw-Tb)2	(Tw-Tb)3	(Tw-Tb)4	h1
8.25	6.25	5.25	2.25	89.06
8.5	6.5	4.5	2.5	115.3
7.5	6.5	4.5	2.5	195.9
7.5	5.5	3.5	2	261.3
6.5	4.5	3.5	2	376.8

Table 3.a. Divergent Duct without Ribs (Heat transfer coefficient & Nusselt no. variation along the Length)

6	7	8	9	10	11	12
h2	h3	h4	Nu1	Nu2	Nu3	Nu4
37.77	26.74	42.2	103.1	43.70	30.93	48.85
48.42	41.6	50.6	133.3	56.03	48.12	58.62
72.63	62.4	76.0	226.7	84.04	72.18	87.
114.4	107	126.7	302.3	132.4	123.7	146
174.9	134	158.3	435.9	202.3	154.7	183

Table 3.b. Divergent Duct without Ribs (Heat transfer coefficient & Nusselt no. variation along the Length)

1	2	3	4	5	6
(Tw-Tb)1	(Tw-Tb)2	(Tw-Tb)3	(Tw-Tb)4	h1	h2
11.25	7.25	5.25	2.25	131.39	65.508
9	6	5	2.5	187.71	90.463
7.25	5.25	3.25	1.75	262.15	116.31
6.25	4.25	2.75	1.75	405.45	191.57
5.25	3.25	2.75	1.75	603.35	313.14

Table 4.a. Divergent Duct with Ribs (Heat transfer coefficient & Nusselt No variation along the Length)

7	8	9	10	11	12
h3	h4	Nu1	Nu2	Nu3	Nu4
53.789	84.94	141.1	70.38	57.79	91.26
64.547	87.37	201.7	97.19	69.34	93.87
111.72	140.4	281.6	124.9	120.0	150.8
176.04	187.2	435.6	205.8	189.1	201.2

Table 4.b. Divergent Duct with Ribs (Heat transfer coefficient & Nusselt No variation along the Length)

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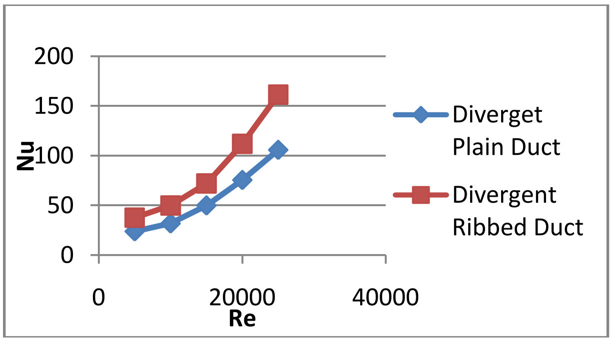
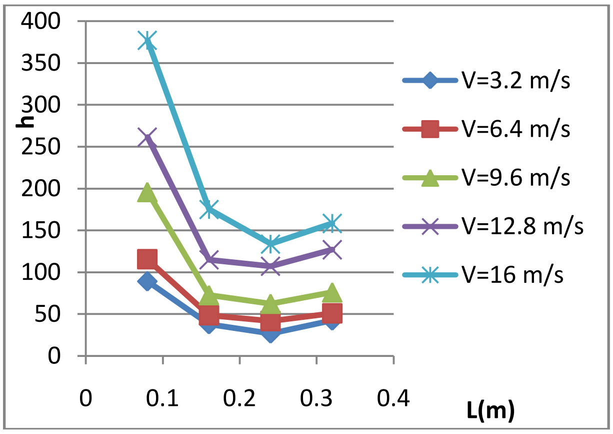
a) Plain Divergent Duct & Divergent Ribbed duct:



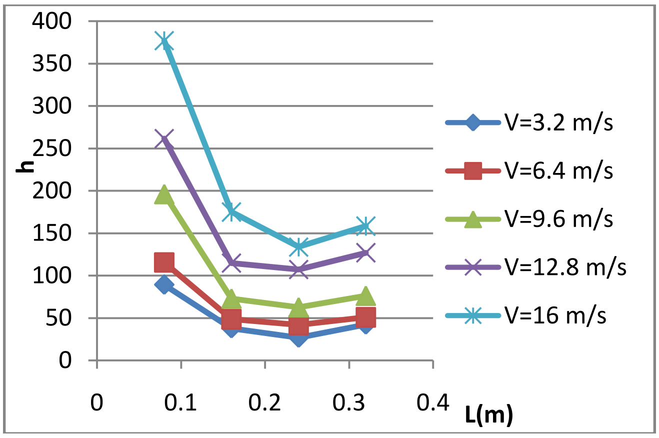
Graph 1: Reynold's No. Vs Nusselt No.

b) Divergent Duct Without Ribs (Heat transfer coefficient variation along the Length)



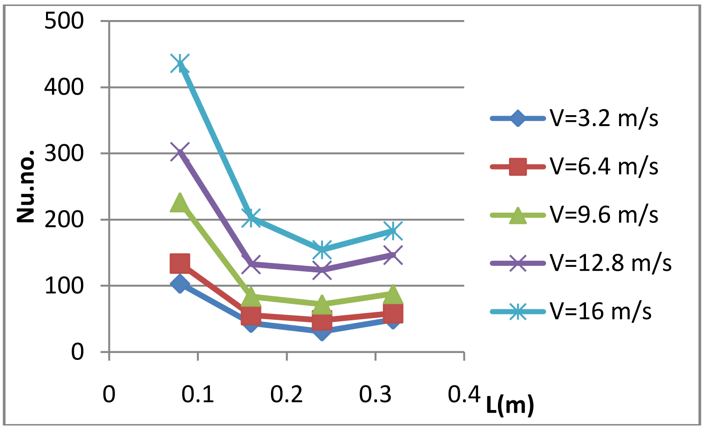
Graph 2: Length Vs Heat transfer Coefficient.

c) Divergent Duct with Ribs (Heat transfer coefficient variation along the Length)



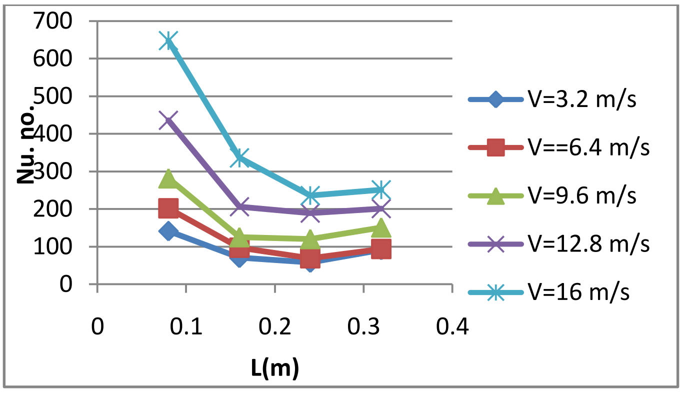
Graph 3: Lenght Vs Heat transfer Coefficient.

d) Divergent Duct without Ribs (Nusselt no. variation Along the Length)



Graph 4: Lenght Vs Nusselt number

e) **Divergent Duct with Ribs (Nusselt no. variation Along the Length)**



Graph 5: Length Vs Nusselt number

From Graph 1 it is shown that Nusselt no. increases with increasing Reynolds no. for Plain Divergent Duct & Divergent Ribbed duct.

From Graph 2 it is shown that Heat transfer coefficient variation along the Length is decreases up to a particular point & then increase for plain duct.

From Graph 3 it is shown that Heat transfer coefficient variation along the Length is decreases up to a particular point & then increase for ribbed duct.

From Graph4 it is shown that Nusselt no. variation Along the Length is decreases upto a particular point & then increase for plain Divergent duct.

From Graph 5 it is shown that Nusselt no. variation Along the Length is decreases upto a particular point & then increase for Divergent ribbed duct. Heat transfer enhancement in Divergent Ribbed Duct is more as compared to Plain Divergent Duct. In divergent ribbed duct boundary layer separation is very good and also it can achieve high Reynolds number in the order of **5,000 to 25,000.** Divergent ribbed Duct can create more turbulence than Plain Divergent Duct. Pressure drop in case of Divergent ribbed Duct is more at the outlet as compared to Plain Divergent Duct. The outlet temperature of Divergent ribbed Duct is Decreases. Thermal

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Performance of Divergent Ribbed Duct is increase by **34.05%** due to Rib Turbulator

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Nomenclature:

T_b = Bulk Temperature , ^{o} C
T_i = Inlet Temperature of Duct, ^{o} C
T_o= Outlet Temperature of Duct, ^{o} C
T_w =Surface Temperature, ^{o} C
m =Mass Flow Rate of Air, kg/s
ρ = Density of Air at 50 ^{o} C, kg/m^3
A_c = Cross Section Area of Duct,m^2
V = Velocity of Ai, m/s.
a = Hydraulic Diameter at Inlet, m.
b = Hydraulic Diameter at Outlet, m.
L=Total Length of Duct, m.
A_s = Surface Area of Duct, m^2
D_h = Hydraulic Diameter Of Duct, m.
L = Length Of Duct, m.
Q_{net} = Net Heat Transfer Rate, W
h = Heat Transfer Coefficient, , W/m^{2}k
Re= Reynolds's Number
v = Kinematic Viscosity at 50 ^{o}C, m^2/s
Nu = Nusselt Number
K= Thermal Conductivity of Air
Cp= Specific heat, J/kg K
k=Thermal conductivity, W/mk
ENH=Enhancement, %
e= Rib Height, m
Δ**T=** Temp Difference, ^{o} C

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