Evaluation of Convective Heat Transfer Coefficient of Air Flowing Through an Inclined Square Duct

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***Abstract*—Liquid or gas flow through pipes or ducts is commonly used in heating and cooling applications and fluid distribution networks .the fluid in such application is usually forced to flow either by a fan or pump. It is a common practice at most of the fluid especially liquids are transported through circular pipes. This is because pipes with circular cross section can withstand large pressure difference between inside and outside without undergoing significant distraction .non circular pipes are usually used in applications such as heating and cooling of buildings ,where the manufacturing and installation costs are lower and the available space is limited for non circular duct work. As the fluid properties vary with temperature and location, the value of convective heat transfer coefficient (h) also varies .presently there are many correlations to predict heat transfer from heated vertical and horizontal pipes in both forced and natural convection situations. A review of literature on heat transfer coefficient (h) indicated that very little work has been done on inclined pipes in the recent past with little or no conclusive work reported for flow through an inclined pipe with square cross section.**

Using the fabricated experimental setup convective heat transfer coefficient of air passing through square duct was determined under the following conditions.

a) Constant heat input and varying the air velocity.
b) Constant air velocity and varying heat input
c) Constant heat input and air velocity by varying the angle of inclination of duct.
d)

***Keywords*—liquid, gas, convective heat transfer coefficient, temparеture, turbulent flow, reynolds number, nusselt number.**

I. INTRODUCTION

1. HEAT TRANSFER:
It is defined as “transmission of energy from one region to another as a result of temperature gradient”.

MODES OF HEAT TRANSFER

1) Conduction
2) Radiation
3) Convection

1.1. Conduction:
It is defined as the transfer of heat from one part of a substance to the of the same substance or one substance to the in physical contact with it without appreciable displacement of molecules forming the substance.

1.2. Radiation:
It is defined as the mode of heat transfer in which energy exchange takes place without the aid of intervening medium.

1.3. Convection:
It is a mode of heat transfer in which energy exchange takes place due to temperature difference by the macroscopic motion of fluid particles.

Convection is classified as follows
Free convection and forced convection.

a. Free convection:
It is mechanism of heat transfer in natural convection involves motion of fluid particles past a solid boundary which is a result of density differences resulting from energy exchange. As a result heat transfer coefficients will vary with geometry of a system.

b. Forced Convection:
Convection is the mechanism of heat transfer through a fluid in the presence of bulk fluid motion. Convection is classified as natural (or free) and forced convection depending on how the fluid motion is initiated. In natural convection, any fluid motion is caused by natural means such as the buoyancy effect, i.e. the rise of warmer fluid and fall the cooler fluid. Whereas in forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or fan.

Mechanism of Forced Convection:
Convection heat transfer is complicated since it involves fluid motion as well as heat conduction. The fluid motion enhances heat transfer (the higher the velocity , higher the heat transfer rate).The rate of convection heat transfer is expressed by Newton’s law of cooling:

$$Q_{Conv} = hA(T_S - T_a) \qquad \text{Equation (1)}$$

Where, Q= rate of heat transfer in watts
h= heat transfer coefficient in w/m^{2}c^o
T_S= Surface Temperature in c^o.
T_a=ambient temperature in c^o.

The convective heat transfer coefficient ‘h’ strongly depends on the fluid properties and roughness of the solid surface, and the type of the fluid flow (laminar or turbulent).

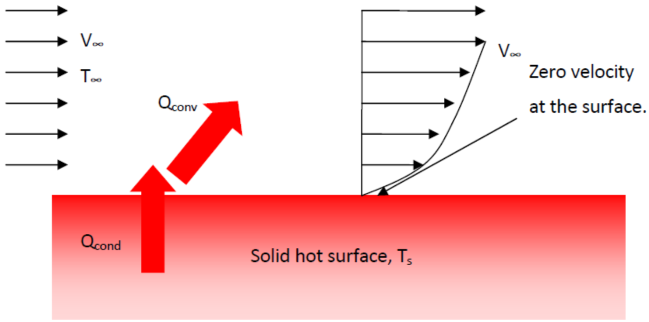

Fig. 1.1: Forced convection.

It is assumed that the velocity of the fluid is zero at the wall; this assumption is called no slip condition. As a result, the heat transfer from the solid surface to the fluid layer adjacent to the surface is by pure conduction, since the fluid is motionless. Thus,

$$\left.\begin{array}{l} \dot{q}_{conv} = \dot{q}_{cond} = -k_{fluid} \left.\dfrac{\partial T}{\partial y}\right|_{y=0} \\ \dot{q}_{conv} = h(T_s - T_\infty) \end{array}\right\} \rightarrow h = \frac{-k_{fluid} \left.\dfrac{\partial T}{\partial y}\right|_{y=0}}{T_s - T_\infty} \qquad (W/m^2.K)$$

2. EXPERIMENTAL WORK

2.1. Experimental Setup of Square Cross Section Duct Without Insulation and Heater

Fig 2.1: Test section copper square duct without insulation and heater

The above figure shows an experimental setup which comprises of test specimen made up of copper having square cross section having dimensions 33x33x500mm, which is connected to a blower through a GI pipe and bellows. Mounting of the four thermocouples takes place on test section at equal distance.

2.2. Experimental Setup of Square Cross Section Duct With Insulation and Heater

Fig 2.2: Experimental Setup of Square Cross Section Duct with Insulation and Heater.

The equipment is mounted on a 2ft X3ft rectangular table which is made up of mild steel sheets. The various indicators like volt meter, ammeter, and temperature indicators are mounted on the panel board as shown in above figure 2.2, also the test specimen is surrounded by insulation and banded heater which is made up of Nichrome. Care should be taken in such a way that equally spaced banded heater should not be direct contact with test specimen; this can be achieved by proper wrapping of the insulation material on the test specimen.

2.3. Experimental Setup of Square Cross Section Duct at 0^0

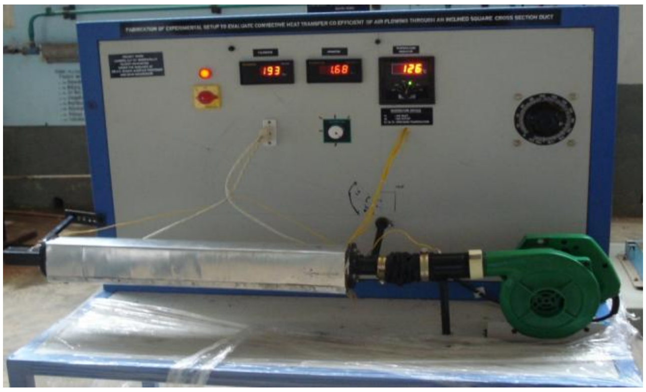

Fig. 2.3 Experimental Setup of Square Cross Section Duct at 0^0

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2.4 Experimental Setup of Square Cross Section Duct at 45^0

Fig. 2.4 Experimental Setup of Square Cross Section Duct at 45^0

2.5 Experimental Setup of Square Cross Section Duct at 60^0

Fig. 2.5 Experimental Setup of Square Cross Section Duct at 60^0

2.6 Experimental Setup of Square Cross Section Duct at 90^0

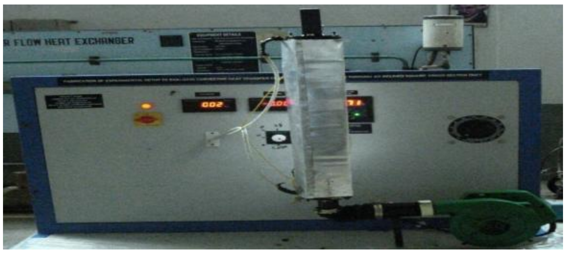

Fig. 2.6 Experimental Setup of Square Cross Section Duct at 90^0

The final experimental set up is as shown in above figures. The testing square tube is wounded with a band heater throughout the section at equal space, so that uniform heat distribution can takes place on surface of the square tube. Anemometer is used find the velocity of air flow through the duct, the velocity is directly digitally displayed in m/sec.

3. THERMOCOUPLES

Thermocouples are used to sense various temperatures along the square tube.

TYPE B

Type B thermocouples can be used up to 1600°C with short term cursions up to 1800°C. They have a low electrical output, therefore are rarely used below 600°C. In fact the output is virtually negligible up to 50°C, therefore cold junction compensation is not usually required.

TYPE E

Type E thermocouples are often referred to as Chromel-Constantan thermocouples. They are regarded as more stable than Type K, therefore often used where a higher degree of accuracy is required.

Type J

Type J thermocouples degrade rapidly in oxidising atmospheres above 550°C. Their maximum continuous operating temperature is around 750°C though they can with stand short duration excursions to 1000°C. They are generally not used below ambient temperature due to condensation forming on the wires leading to rusting of the iron.

Type K

Type K are the most widely used thermocouples in the Oil & Gas, and refining industries due to their wide range and low cost. They are occasionally referred to as Chromel-Alumel thermocouples. Note that above about 750°C oxidation leads to drift and the need for recalibration.

Type N

Type N thermocouples can handle higher temperatures than type K, and offer better repeatability in the 300 to 500°C range. They offers many advantages over Type R & S at a tenth of the cost, therefore prove to be popular alternatives.

Type R

Type R thermocouples cover similar applications as Type S but offers improved stability and a marginal increase in range. Consequently, Type R tend to be used in preference to Type S.

Type S

Type S thermocouples can be continually at temperatures up to 1450°C. They can with stand short duration excursions up to 1650°C. They need protection from high temperature atmospheres to prevent metallic vapor ingress to the tip resulting in reduction of emf generated. Protection commonly offered is high purity recrystallised alumina sheath. For most industrial

Applications, thermocouples are housed in a thermo well.

Type T

Type T thermocouples are rarely used in industrial applications, and lend themselves more to use in laboratory situations.

3.1. Layout of experimental set up

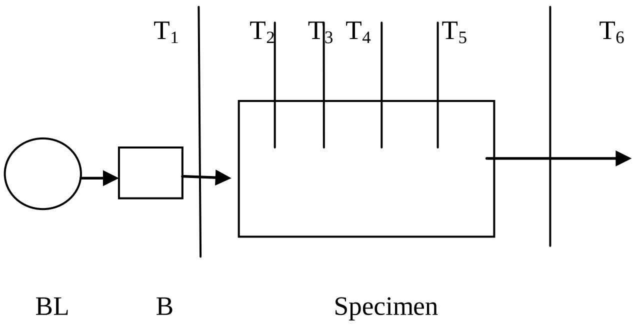

Fig.3.1 Layout of experiment setup

Where BL = Blower
B= Bellows
T1= inlet temperature of air
T6= outlet temperature of air
T2,T3,T4,T5= Specimen surface temperature

The above figure shows a neat block diagram of a experimental setup, in which test specimen is placed at zero degree. Angle of inclination can be varied in terms of $45^{0,}$ 60^{0} and 90^{0} with the help of lever and nut-bolt arrangement. Here we are using six thermocouples of k-type, two for inlet and outlet i.e. T1 and T6, rest of the four (i.e. T2,T3,T4 and T5) thermocouples are placed at equal distance on the surface of test specimen. With the help of blower regulator, velocity of air can be set and same can be readout digitally with the help of anemometer .Heat input can be set with the help of variac provided on control panel and same can be read out digitally with the help of voltmeter as volts and ammeter as amps.

4. METHODOLOGY

The methodology consists of detailed operating procedure of experiment set up and how to take the various readings like temperature and heat input.

4.1 Description Of The Apparatus

The apparatus consists of a blower for forced circulation. The air from the blower passes through a flow passage, heater and then to the test section. Air flow is measured by an anemometer which is placed opposite to the test section at exit side. A heater placed around the tube heats the air, heat input is controlled by a dimmer stat. Temperature of the air at inlet and at outlet are measured using thermocouples. The surface temperature of the tube is measured at different sections using thermocouples embedded on the square duct. Test section is enclosed with wool and rope, where the circulation of rope avoids the heat loss to outside.

4.2 Operating Procedure

1. Start the blower after keeping the valve open, at desired rate.
2. Put ON the heater and adjust the voltage to a desired value and maintain it constant.
3. Allow the system to reach a steady state.
4. Note down all the temperatures from T_1 to T_6, voltmeter and ammeter readings.
5. Repeat the experiment for different heat input and flow rates.

4.3 Pre Cautions To Be Taken

Do's:

1. Before switching ON the unit, make sure that the variec is in Zero position.
2. Operate thermocouple selector switch gently.
3. Operate the unit minimum twice a week.
4. Increase the voltage very slowly by using variec knob.

Don'ts:

1. Do not go above 200 volts power input to heater.
2. Do not operate the equipment if line voltage is less than 200 volts.

4.4 Specifications

Specimen : Copper square Tube.
Size of the Specimen : 33 mm x 33 mm x 500 mm long.
Heater : Externally heated, Nichrome wire Band Heater 500W.
Ammeter : Digital type,0-20amps, AC.
Voltmeter : Digital type, 0-300volts, AC.
Dimmer stat for heating Coil : 0-230v, 2amps.
Thermocouple Used : 6 no. k-type, range: 0 to 400^{0}c.
Centrifugal Blower : Single Phase 230v, 50 Hz, 13000rpm.
G. I pipe diameter, 'd_p': 33 mm.
Outer duct : Aluminum.

4.5. Procedure For Calculation

1) Note down all the parameters which are displayed on control panel, which includes voltage, current & all temperatures.
2) Calculate the surface temperature and ambient temperatures by using the following formulae
3)

$$TS = \frac{(T2 + T3 + T4 + T5)}{4}$$

$$Ta= (T1+T6)/2$$

$$T_{film} =(Ts+Ta)/2$$

4) Find the properties of air at film temperature like kinematic viscosity (v), prandtl number (Pr), thermal conductivity (K).
5) Calculate the Reynolds number (Re), and formula is given as Re= (V Dh)/ν
6) Based on the Reynolds number, select the Hilpert's constants like C and m.
7) Calculate Nusselts number by using parameters like Pr,C,m,Re_D and the formula is given by Nu = Cx $Re_D{}^{m}$x $pr^{0.333}$.
8) Calculate the convective heat transfer coefficient by using the formula

$$Nu=(hxDh)/K$$

$$So\ h=(KxNu)/ Dh$$

9) Repeat the calculation part for following different situations.
 a) By keeping the heat input as constant and varying the velocities in terms of 10,20,and 30 m/sec ,for various angle of inclinations like 0^{0},45^{0},60^{0} and 90^{0}
 b) By keeping the air velocity as constant and varying the heat input in terms of 100v ,150v and 200v , for various angle of inclinations like 0^{0},45^{0},60^{0} and 90^{0}

10)Tabulate all the calculations for separate angle of inclinations.

5. RESULTS AND DISCUSSIONS

5.1. Results for constant heat input 100v and varying air velcity at 0^0

Sl No	Velocity M/S	Heat Input Watts	Reynolds No. Re_d	Nusselts No. Nu	h W/M^2 k
1	10	94.5	19013	75.53	63.72
2	20	88.74	37,811	115.14	97.69
3	30	88.74	56,091	148	125

5.2. Results for constant velocity 20m/s and varying heat input at 0^0

Sl No	Velocity M/S	Heat Input Watts	Re_d	Nu_d	h W/M^2k
1	20	87.72	37,812	116	98.107
2	20	199.12	35,363	111	96.70
3	20	322.31	34,792	106	96.18

5.3. Results for constant heat input 100v and varying air velocities at 45^0

Sl No	Velocity M/S	Heat Input Watts	Reynolds No. Re_d	Nusselts No. Nu	h W/M^2k
1	10	88.5	37,811.51	115.51	97.69
2	20	88.58	35,752	111.50	96.67
3	30	87.72	33,316.50	106.61	95.37

5.4. Results for constant air velocity =20 m0sec and heat input at 45^0

Sl No	Velocity M/S	Heat Input Watts	Re_d	Nu	h W/M^2k
1	20	88.58	37,811.51	115.51	97.69
2	20	192.21	35,752	111.50	96.67
3	20	351.75	33,316.50	106.61	95.37

5.5. Results for constant heat input and varying velocities at 60^0

Sl No	Velocity M/S	Heat Input Watts	Re_d	Nu	hW/M^2k
1	10	85.85	18800	75.04	63.62
2	20	88.74	37,599	115.36	97.812
3	30	91.52	56398	148	125.48

5.6. Results for constant velocity 20 m/sec and varying heat input at 60^0

Sl No	Velocity M/S	Heat Input Watts	Re_d	Nu	h W/M^2k
1	20	88.74	37,598	115.17	97.651
2	20	197.81	36,561	113.10	96.86
3	20	322.31	34,978	109.96	96.26

5.7. Results for constant heat input100v and varying velocities at 90^0

Sl No	Velocity M/S	Heat Input Watts	Re_D	Nu	h w/m^2k
1	10	85.85	18077	73.17	63.126
2	20	89.61	36,768	113.50	97.202
3	30	85	55,460	146.33	125.00

5.8. Results for constant velocity=20 m/sec and varying heat input at 90^0

SL NO	VELOCITY m/s	HEAT INPUT Watts	Re_D	Nu	h w/m^2k
1	20	89.61	36,768	113.50	97.20
2	20	190.72	34,979	109.96	96.27
3	20	329.28	32,967	105.96	95.18

5.9 GRAPHS

i. Comparison between inclination of square duct and heat transfer co efficient for different velocities.

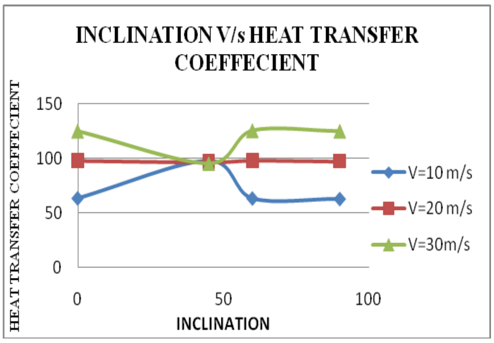

Fig.5.1.Inclination v/s heat transfer coefficient for all velocities

a) When velocity =10 m/sec, the heat transfer co efficient (h) is increasing ,when the inclination of square duct is changed from 0^0 to 45^0 and ' h " is starts decreasing .And when inclination is changed to 60^0 and "h" will become linear, when inclination is changed to 90^0 .

b) When Velocity = 20 m/sec , the heat transfer co efficient is almost linear for all angle of inclinations .

c) When velocity = 30 m/sec, the heat transfer co efficient (h) is starts decreasing when angle of inclination changed from 0^0 to 45^0. And value of "h" will starts increasing when the angle of inclination changed from 45^0 to 60^0. And the value of "h' is almost linear when angle of inclination changed from 60^0 to90^0.

ii. Comparison between velocity and heat transfer co efficient for 0^0 inclination

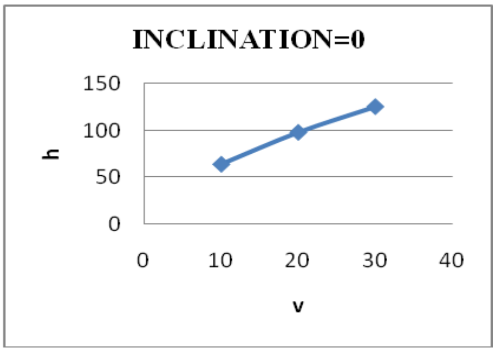

Fig.5.2. velocity v/s heat transfer co efficient at 0^0

From the above graph it is clear that for 0^0 inclination, the heat transfer co efficient (h) is directly proportional to velocity of air i.e. here the increase in the heat transfer co efficient h takes place by increasing the velocity of air and vice versa too.

iii. Comparison between velocity and heat transfer co efficient for 45^0 inclinations.

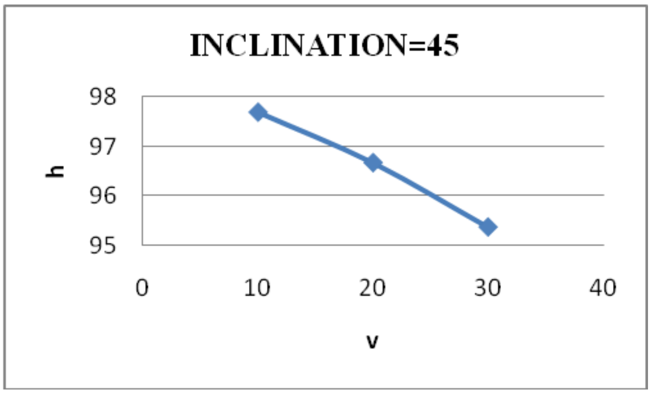

Fig.5.3. velocity v/s heat transfer co efficient at 45^0

From the above graph it indicates that for 45^0 angle of inclination , convective heat transfer co efficient (h) is inversely proportional to the velocity of air i.e. there is a decrease in the value of h takes place by increasing the air velocity.

iv. Comparison between velocity and heat transfer co efficient for 60^0 inclination

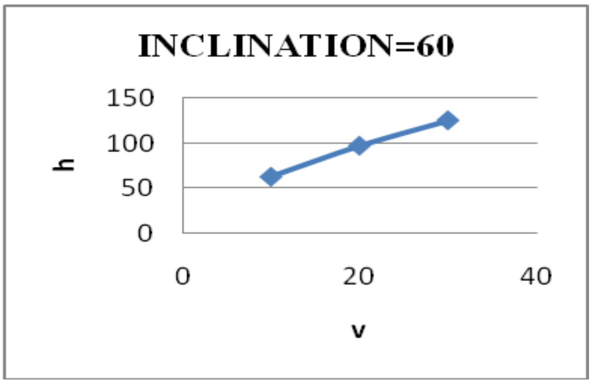

Fig.5.4. velocity v/s heat transfer coefficient at 60^0

From the above graph it is clear that for 60^0 angle of inclination, the heat transfer co efficient (h) is directly proportional to velocity of air i.e. here the increase in the heat transfer co efficient h takes place by increasing the velocity of air.

v. Comparison between velocity and heat transfer co efficient for 90^0 inclination.

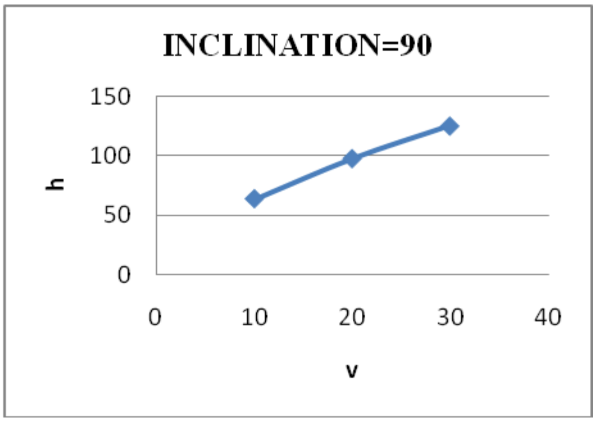

Fig.5.5.velocity v/s heat transfer co efficient at 90^0

From the above graph it is clear that for 90^0 angle of inclination, the heat transfer co efficient (h) is directly proportional to velocity of air i.e. here the increase in the heat transfer co efficient h takes place by increasing the velocity of air.

vi. Comparison between heat input and heat transfer co efficient for different angles

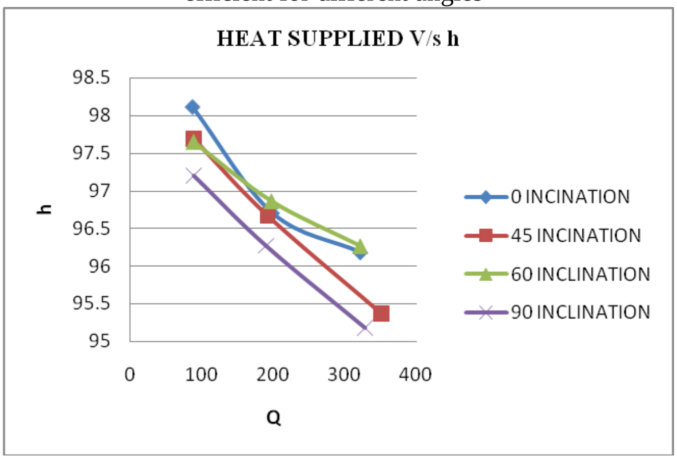

Fig.5.6. heat input v/s heat transfer co efficient all angle of inclinations.

The above graph shows the variation of heat transfer co efficient h with respect to heat supplied at various inclinations. It can be seen that the value of heat transfer co efficient (h) decreases with increase in amount of heat supplied at any inclination. At 0^0 inclination , the heat transfer co efficient is maximum at its low heat input (100w) and it decreases drastically with increase in amount of heat supplied . However it can be concluded that , at 60^0 inclination gives the best value of "h" irrespective of any amount of heat supplied. Where as inclinations of $45^0$and 90^0 gives minimum heat transfer co efficient compared to 0^0 and $60^{0.}$ Finally it is concluding that , it is convenient and comfortable to inclined the duct by 60^0 instead of $45^0$and 90^0.

vii. Comparison between Reynolds number and Nusselt's number

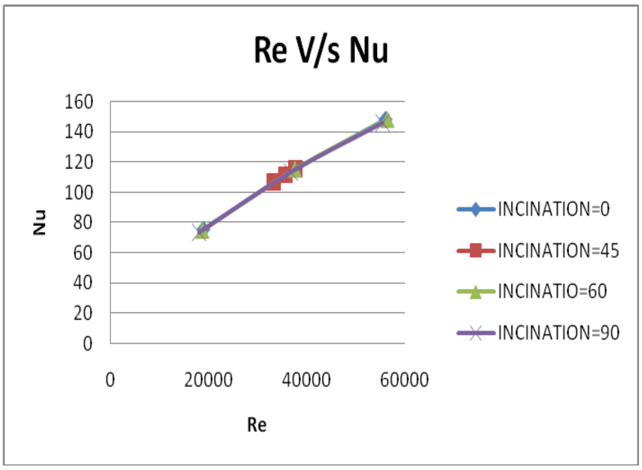

Fig.5.7. Reynolds no. v/s Nusselts no.

From the above graph it is clear that , irrespective of angle inclination of duct , there is a direct proportionality exist between the Reynolds number (Re) and Nusselt number. i.e.by increasing the velocity of air there must be an increase in the Reynolds number in turn Nusselt number.

6. CONCLUSIONS

1. There is an urgent need for the formulation of correlations for convective heat transfer under internal flow conditions with inclined pipe orientation.

2. The literature survey shows that the use of horizontal pipe correlations for calculating heat loss from inclined pipe orientation yields erroneous results of significant magnitude.

3. Designers and engineers need to be guided when using horizontal pipe correlations for inclined pipe calculations as there may be significant errors.

4. The maximum velocity will be occurring at the center line of the test specimen where profile obtained is parabolic, which indicates fully developed flow.

7. SCOPE FOR FUTURE WORK

1. Work for non circular ducts with oriented at various angle can be carried to the horizontal.
2. Work for different fluid (both liquid and gas) can be carried out.
3. Work for different geometry can be carried out.
4. Exact heat transfer coefficient can be carried through dimensional analysis and boundary layer analysis.

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