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Enhancement of Heat Transfer Coefficient through Forced Convection Apparatus by Using Circular and Elliptical Pipe

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***Abstract*— It is very essential to develop a forced convection system to carry out the analytical and experimental investigation of the heat transfer coefficient with the use of the elliptical and circular pipe. Convection is nothing but the transfer of heat through a fluid which is in bulk fluid motion. Depending on the motion of the fluid, it is classified as natural and forced convection. Buoyancy effect is the natural cause which occurs due to fluid motion in natural convection. On the other hand, in forced convection, the motion of the fluid is caused by external means like a pump or fan. In this system three, 130 Volt heaters are wounded over 800 mm test section of Circular pipe having a 25mm diameter and Elliptical pipe having 22 mm minor and 28 mm major diameter. Variable dimmer stat is used to control input to heater and control valve to control mass flow rate, to determine the average coefficient of heat transfer in turbulent flows inside smooth and straight Circular and Elliptical pipes. The research consists of a regression analysis performed between the Reynolds number and the Prandtl number finally to calculate the heat transfer coefficient of Circular and Elliptical pipes by experimental investigation. After calculating all the results of both the pipes finally we got that efficiency of Elliptical pipe is more than circular pipe also heat transfer coefficient is maximum in the elliptical pipe than the circular pipe and it also led to increasing the heat transfer rates. The deviation of the heat transfer coefficient is maximum as compared to Elliptical pipe.**

***Index Terms*—Elliptical pipe, Circular pipe, heat transfer, Forced Convection, Thermocouple, Augmentation.**

I. INTRODUCTION

Nowadays, heat exchange equipment is in demand due to material availability and the high cost of energy. The use of heat exchangers is extensively seen in radiators for space vehicles, air conditioning gadgets, thermal power plants, automobiles etc. Techniques to improve heat transfer are the major topics for the researchers today. As per the survey made the energy use is increased by 1.6 times in the world, from 4.03 x 1020 J in1999 to 6.4 x 1020 J in 2020.

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Hence, if the effectiveness of the energy uses could increase by 10%, by means of various heat transfer enhancement techniques, this will result in 6.4 x 1019 J of benefit in term of energy consumption to the society.

The economic operation and design of the process plant are ruled by the effective use of heat. With the continuous increase in volume and power in production, thedimensions and mass of heat exchangers also increase which in turn involves multi-million dollars investment for capital and operation cost. Hence it is a vital requirement to reduce overall dimension characteristics of heat exchangers.

Due to the needs like optimization and conservation, the development of heat exchangers is taking place. Different techniques are used to improve the heat transfer rates, which are normally referred to as heat transfer enhancement or heat transfer augmentation techniques. With the help of these techniques, the following experimentation is carried out.

II. LITERATURE REVIEW

Adegun [1] presented the numerical simulation of enhanced forced convective heat transfer in inclined elliptic ducts withmultiple internal longitudinal fins. Hajmohammadi [2] presented theoptimal design of tree-shaped inverted fins penetrated into heat generating bodies to enhance the heat transfer rate. Hajmohammadi [3] presented the optimal geometric structure of amicro-scale channel heat sink, by assuming slip boundary condition and investigated the 3D fluid flow and heat transfer phenomenainside the microchannel heat sink. Mapa and Mazhar [4] discussed the heat transfer using nanofluid in a mini heat exchanger utilizing commercially availableequipment. Faizal and Ahmed [5] investigated the channelheight between the plates to determine the configuration that givesthe optimum heat transfer. JozefCernecky [6], the paper deals withvisualization of temperature fields in the vicinity ofprofiled heat transfer surfaces and subsequentanalysis of local values of Nusselt numbers by forced air convection in an experimental channel. Priyank [7] studied the effect of Reynolds numberand relative roughness pitch on the heat transfer coefficientand friction factor has been studied. It is reported that the highest Nusselt number at a higher value of Reynolds number was provided by theroughened duct having circular and square rib with highestrelative roughness height. Squaresectioned rib provides a higher value of enhancement ascompared to circular rib at a higher value of Reynoldsnumber. Sagar [8] performed a numerical analysis of heattransfer for three different angles of w-shaped turbulatorssplaced at the bottom side wall of the square duct and reported that Nusselt number andfriction factor in a duct with W-rib insert increases ascompare to smooth duct without an insert. Pankaj [9] carried out an experimental analysis tostudy turbulent flow heat transfer in a rectangular duct withand without internal ribs and observed thatthe localNusselt number distribution is strongly depended on position, orientation, and geometry of the ribs. Also, thediscrete V-shaped ribs produce overall less heat transfer enhancement than continuous V-shaped ribs. Few researchers have carried out the study on micro pipes like micro channels and reported the performance of the microchannels [10-12].

III. Experimental Setup

The schematic diagram of the open loop experimental setup is shown in Fig. 1 the loop consists of a blower unit fitted with a pipe in a horizontal orientation. The blower fan runs at constant speed. The outlet of a blower is connected to an MS pipe having inside diameter 25.4mm through a reducer. The U-pipe manometer is connected across the orifice meter to measure the flow rate of air flowing through the pipe.

Nichrome bend heater of 200W encloses the thermally developing section to a length of 300mm and the test section to a length of 500mm to cause electric heating. Supply to the heater is given from dimmer stat. Power input to the test pipe heater is varied using a dimmer stat, which is used to vary the voltage of the AC current passing through the heater and by keeping the current less than 2A. Two thermocouples are placed one at the entrance i.e. before the thermally developing section and the other at the exit i.e. after the test section to measure air inlet and outlet temperature (T_1 and T_{10}) respectively. Three thermocouples i.e. T2, T3, and T4 are placed on the thermally developing section and other five thermocouples i.e. T5, T6, T7, T8, and T9 are placed on the test section to measure the temperature at various points along the pipe surface. The outer surface of the test section is well insulated to minimize convective heat loss to the surrounding. The control valve at the exit section controls the airflow rate into the test section. Necessary precautions have been taken to prevent leakages from the system. The actual photograph and schematic of the experimental setup are as shown in the below Fig. 1 and Fig. 2 respectively.

Nomenclature

SYMBOLS	DESCRIPTION
A	Cross section area of pipe, (m^2)
As	Surface area of Test Section, (m^2)
T	Temperature (^{0}C)
T1, T10	Air temperature at inlet and outlet, (^{0}C)
T_2,T_3,T_4,T_5, T_6, T_7, T_8, and T_9	Test section temperatures, (^{0}C)
T_S	The average surface temperature of the working fluid, (^{0}C)
T_a	Bulk temperature, (^{0}C)
U	Air velocity through test section, (m/s)
K	Thermal conductivity, W/mK
H	Convective heat transfer coefficient, W/ m^2K
D	The inner diameter of the test section, (m)
C_P	Specific heat of air, J/kgK
L	Length of the test section, (mm)
M	The mass flow rate of air, (kg/sec)
N_u	Nusselt number
Pr	Prandtl number
Q_t	Total heat transferred to air, (W)
Q_D	Discharge through an orifice, (m^3/sec)
Re	Reynolds number
µ	Dynamic viscosity, (kg m/sec)
B	The ratio of Orifice Diameter to Pipe Diameter
P	Density, (kg/m^3)

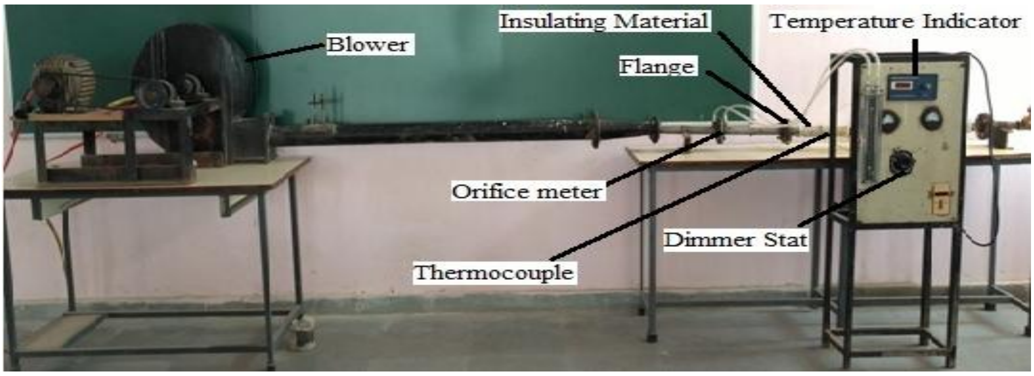

Fig. 1. Experimental Setup

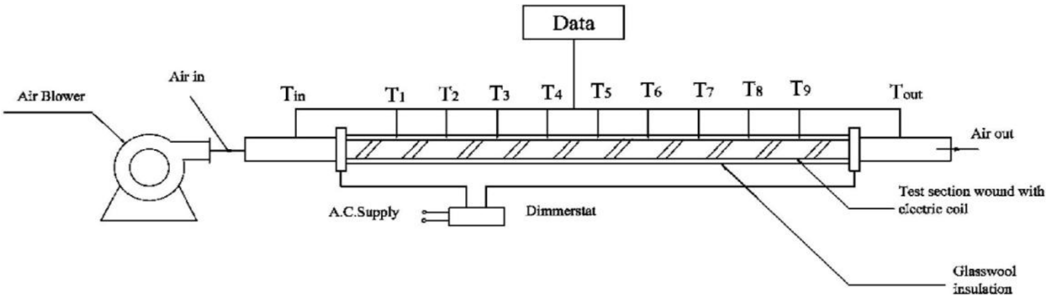

Fig. 2. Schematic Diagram of Experimental Setup

IV. Observations

A. Observations for Circular Pipe

i. Test Section Length, L=0.8m
ii. I.D of Test Section, D=0.0250m
iii. Cross-Sectional Area of Pipe, A=4.90625 × 10^{-4} m^2
iv. The surface areaof the test section, A_s =0.0628 m^2

B. Observations for Elliptical Pipe

i. Test section length, L=0.8m
ii. Hyd.D of the test section, D=0.0250m
iii. The cross-sectional area of a pipe, A= 4.90625 × 10^{-4} m^2
iv. The surface area of the test section, A_s =0.0628 m^2

TABLE I. Observation Table for Circular Pipe

SR.No	U(m/s)	T1	T2	T3	T4	T5	T6	T7	T8	T9	T10
1	7.3	37	82	84	84	86	89	82	90	89	53
2	5.6	33	84	88	87	90	93	86	95	94	51
3	4.3	34	92	92	96	98	102	106	110	103	55

TABLE II. Observation Table for Elliptical Pipe

SR.No	U(m/s)	T1	T2	T3	T4	T5	T6	T7	T8	T9	T10
1	7.3	34	63	69	68	61	74	66	78	85	50
2	5.6	36	65	72	71	65	79	69	83	90	53
3	4.3	35	68	75	74	68	84	71	85	92	54

V.Calculations

A. Sample Calculations for Circular Pipe

Calculation procedure for circular pipe of 25mm diameter at 130V voltage, 2A current and U=4.3 m/s.
Estimation of Reynolds Number and Prandtl Number

Mean bulk temperature,
$T_a = (T_1+T_{10})/2 = 44.5\ (^0C)$…(1)

Mean surface temperature,
$T_s = (T_1+T_2+T_3+T_4+T_5+T_6+T_7+T_8+T_9)/5 = 99.875\ (^0C)$ …(2)

Properties of air at 1atm at mean bulk temperature i.e. at $T_{a,}$

$Density_{air} = 1.1115(kg/m^3)$
$C_p = 1007$ (J/kg.k)
K = 0.027403 (W/m.k)
μ = 0.000019356 (kg/m.s)
$P_r = 0.71136$

Volume flow rate,
$Q_D = A \times U$(3)

$Q_D = 0.00211(m^3/s)$

Mass flow rate,
$\dot{m} = Density_{air} \times Q_D = 0.002346$ (kg/s) …(4)

Heat transferred to air,
$Q_t = \dot{m} \times C_p \times (T_{10} - T_1) = 49.60566$ (W) …(5)

Experimental convective heat transfer coefficient,
$h_{expt} = Q_t / (A_s \times (T_s - T_a))$ …(6)
$h_{expt} = 14.25773\ (W/m^2k)$

Experimental Nusselt number,
$Nu_{(expt)} = (h_{expt} \times L_c) / K = 13.00746$…(7)

Reynolds number,
$R_e = (Density_{air} \times U \times D) / \mu = 6173.086$…(8)

Theoretical Nusselt number by Dittus-Boelter equation,
$Nu_{(theo)} = 0.023 \times (R_e)^{0.8} \times (P_r)^{0.3} = 22.37466$…(9)

Theoretical convective heat transfer coefficient,
$h_{expt} = (Nu_{(theo)} \times K) / L_c = 14.25773\ (W/m^2k)$ …(10)

B. Sample Calculations for EllipticalPipe

Calculation procedure for Elliptical pipe of 25mm Hydraulic diameter at 130V voltage, 2A current and U=4.3 m/s.
Estimation of Reynolds Number and Prandtl Number

Mean bulk temperature,
$T_a = (T_1+T_{10})/2 = 44.5\ (^0C)$…(11)

Mean surface temperature,
$T_s = (T_1+T_2+T_3+T_4+T_5+T_6+T_7+T_8+T_9)/5 = 77.125\ (^0C)$.(12)

Properties of air at 1atm at mean bulk temperature i.e. at $T_{a,}$

$Density_{air} = 1.1115(kg/m^3)$
$C_p = 1007$ (J/kg.k)
K = 0.0274 (W/m.k)
μ = 0.000019356 (kg/m.s)
$P_r = 0.7113$

Volume flow rate,
$Q_D = A \times U$…(13)

$Q_D = 0.00211(m^3/s)$

Mass flow rate,
$\dot{m} = Density_{air} \times Q_D = 0.00235$ (kg/s) …(14)

Heat transferred to air,
$Q_t = \dot{m} \times C_p \times (T_{10} - T_1) = 44.8813$ (W) …(15)

Experimental convective heat transfer coefficient,
$h_{expt} = Q_t / (A_s \times (T_s - T_a))$ …(16)
$h_{expt} = 21.8952\ (W/m^2k)$

Experimental Nusselt number,
$Nu_{(expt)} = (h_{expt} \times L_c) / K = 19.9751$…(17)

Reynolds number,
$R_e = (Density_{air} \times U \times D) / \mu = 6173.09$…(18)

Theoretical Nusselt number by Dittus -Boelter equation,
$Nu_{(theo)} = 0.023 \times (R_e)^{0.8} \times (P_r)^{0.3} = 22.3741$…(19)

Theoretical convective heat transfer coefficient,
$h_{expt} = (Nu_{(theo)} \times K) / L_c = 21.8952\ (W/m^2k)$ …(20)

VI. Results And Discussion

As the Reynolds number increased, the difference of temperature between inlet and outlet (T_1-T_{10}) was observed to be decreasing with increase in Reynolds number. This is due to a more mass flow rate of air supplied to the test at higher Reynolds number. T_2, T_3, T_4, T_5, T_6, T_7, T_8 and T_9 denote the wall temperature of the horizontal pipe at different locations. They are observed to decrease with the increase of Reynolds numbers. The result will be compared with Circular pipe to estimate the enhancement of heat transfer rate in the presence of Elliptical pipe.

It is observed from table III and IV that, experimentally obtained Nusselt number values are lesser than those obtained from using Dittus-Boelter correlation values for both pipes.

III. Result Table for Circular Pipe

Sr.No	U(m/s)	Ts	Ta	Re	Nu(expt)	Nu(theo)	h(expt)	h(theo)	Qt
1	4.3	99.875	44.5	6173.086	13.00746	22.37466	14.25773	24.52532	49.60566
2	5.6	89.625	42	8152.599	17.13111	27.95522	18.65373	30.43988	55.81715
3	7.3	85.75	45	10451.44	19.67436	34.09424	21.59457	37.42184	55.28907

IV. Result Table for Elliptical Pipe

Sr.No	U(m/s)	Ts	Ta	Re	Nu(expt)	Nu(theo)	h(expt)	h(theo)	Qt
1	4.3	77.125	44.5	6173.086	19.97514	22.3741	21.89515	24.5247	44.88131
2	5.6	74.25	44.5	7748.344	25.52515	26.83557	27.97863	29.415	52.29744
3	7.3	70.5	42	10538.76	33.17096	34.32872	36.11919	37.37986	64.67701

Fig. 3 shows the variation of heat transfer coefficient with Reynolds number for a circular pipe. The values obtained from experimentation of heat transfer coefficient are lesser than those obtained from using Dittus-Boelter correlation values i.e. theoretically calculated values.

Fig. 4 shows the variation of Nusselt No. with Reynolds number for a circular pipe. The experimentally obtained values of Nusselt No. are lesser than those obtained from using Dittus-Boelter correlation values i.e. theoretically calculated values.

Fig. 5 shows the variation of heat transfer coefficient with Reynolds number for Elliptical pipe. The experimental heat transfer coefficient values are lesser than those obtained from using Dittus-Boelter correlation values i.e. theoretically calculated values.

Fig. 6 shows the variation of Nusselt No. with Reynolds number for a circular pipe. The experimentally obtained values of Nusselt No. are lesser than those obtained from using Dittus-Boelter correlation values i.e. theoretically calculated values.

Fig. 7 presents the variation of heat transfer coefficient with Reynolds number for both Elliptical pipe and Circular pipe.The experimental heat transfer coefficient values for Elliptical pipe are more than Circular pipe.

Fig. 8 shows the variation of Nusselt No. with Reynolds No. for both Elliptical pipe and Circular pipe. The experimentally obtained values of Nusselt No. for Elliptical pipe are more than Circular pipe.

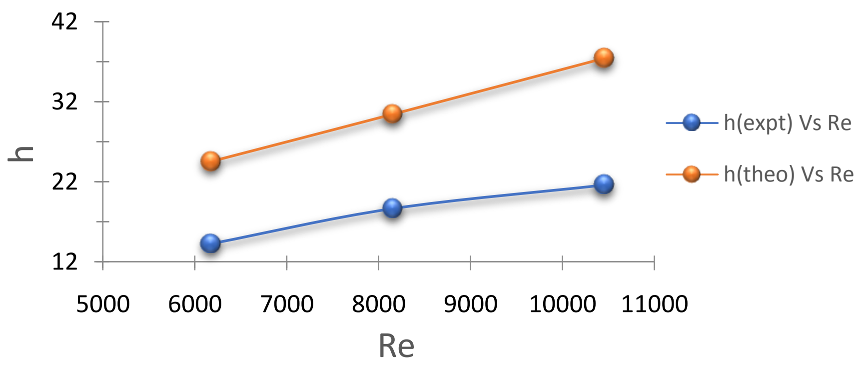

Fig. 3. Variation of Heat Transfer Coefficient with Reynolds No. for Circular Pipe

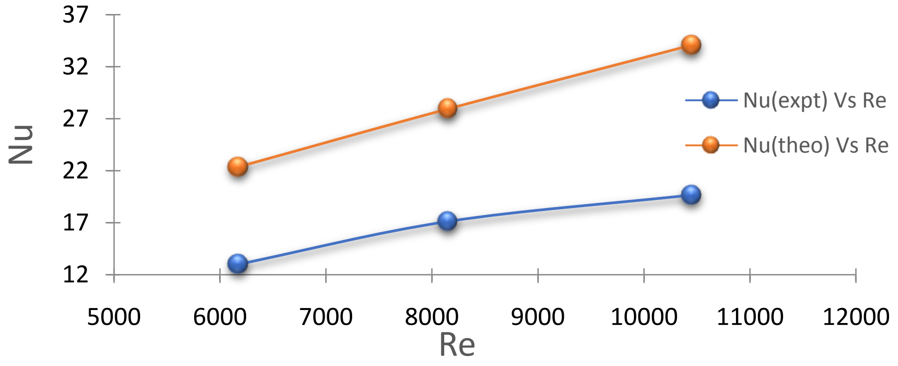

Fig. 4. Variation of Nusselt No. with Reynolds No. for Circular Pipe

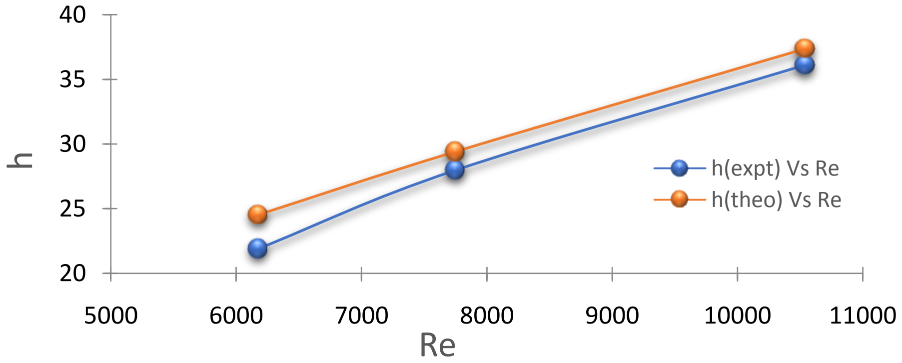

Fig. 5. Variation of Heat Transfer Coefficient with Reynolds No. for Elliptical Pipe

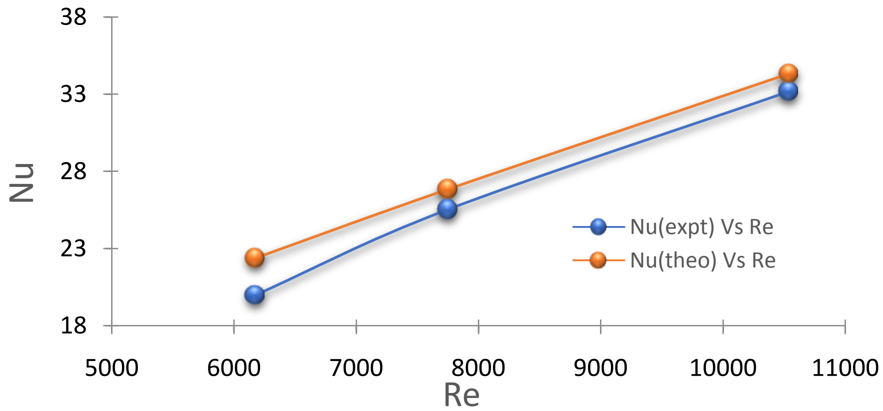

Fig. 6. Variation of Nusselt No. with Reynolds No. for Elliptical Pipe

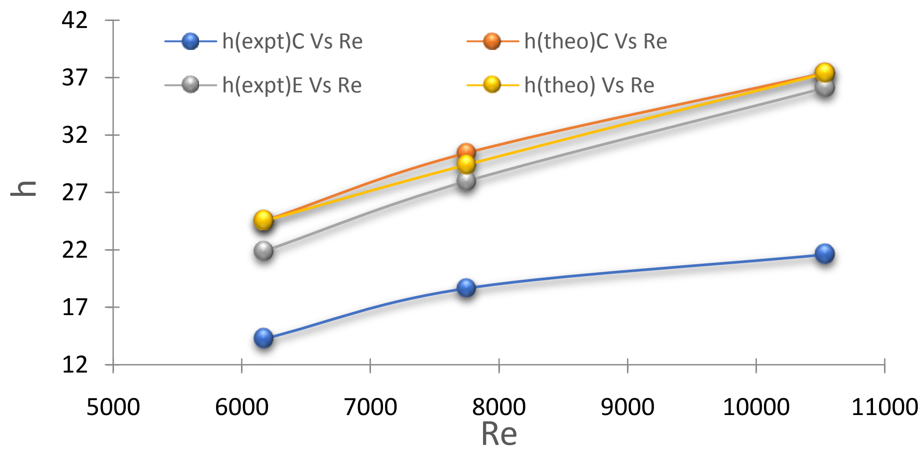

Fig. 6. Variation of Nusselt No. with Reynolds No. for Elliptical Pipe

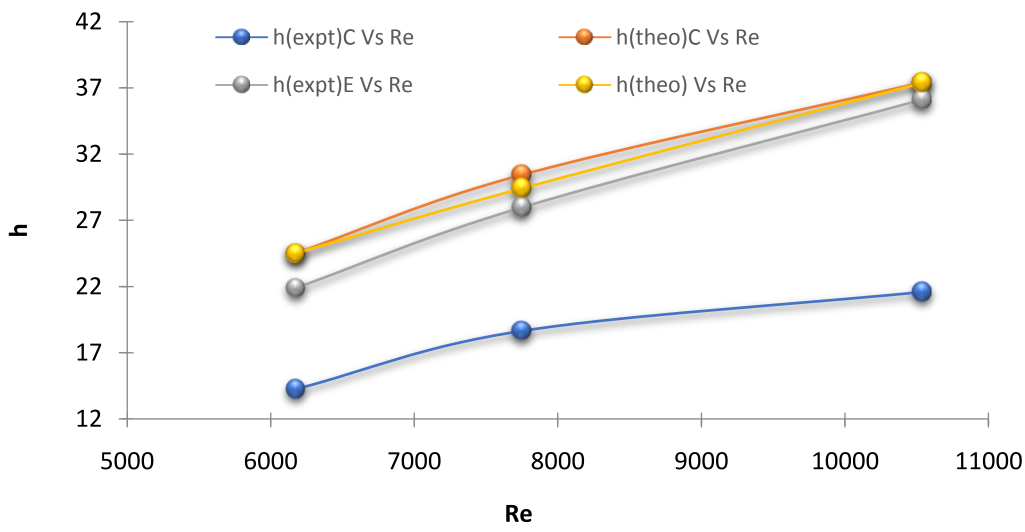

Fig. 7. Variation of Heat Transfer Coefficient with Reynolds No. for both Pipes

VII. Conclusion

Experimental investigations were performed to investigate the heat transfer characteristics in an externally heated horizontal circular pipe and elliptical pipe. It is found that the heat transfer coefficient is maximum in an elliptical pipe than a circular pipe and led to increase heat transfer rates. Also, from the above charts and result table III and IV, it is clear that the theoretical values of heat transfer coefficient for a different reading of elliptical pipe are nearly the same to the experimental values. To conclude, an elliptical pipe is more efficient over a circular pipe in forced convection apparatus.

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