FORCED AND FREE CONVECTIVE HEAT TRANSFER ENHANCEMENT THROUGH RECTANGULAR DUCT OF PIN FIN APPARATUS USING WITH AND WITHOUT THREADS OF BRASS EXTENDED SURFACES

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Abstract: The present paper demonstrates experimental analysis of parameter aimed to enhance of heat transfer of a brass fin using pin fin apparatus. The present work focus on experimental investigation of heat transfer coefficient (h), mass flow rate (m·), Nusselt number (Nu), Efficiency & Effectiveness of fin of horizontal Rectangular pipe using with and without internal threads with air as a working fluid flow by a blower .The flow regime is selected for this study with the Reynold's number range 40 to 4000. The horizontal brass pipe was selected to constant and uniform heat flux. The experimental are observed that to study the effect of internal threads 0.5 cm brass material fin in a rectangular duct due to disorder effect of air. The heat transfer coefficient, Reynold's and Nusselt's number are analyse with by means of heat transfer enhancement method. Heat transfer coefficient increase with increasing the pitch of threads in brass fin due to turbulence action of air with surface material of fin and also Nusselt's number higher as match up to without threads of fin material as result of convective mode of heat transfer all the way through the fin.

Keywords: Fin, Extended surface, Heat exchangers, Heat transfer enhancement, free and force convection.

Table-1:- Nomenclature

D	Dia. Of duct in m
L	Length of duct in m
Tm	Average fin temperature in K
Tmf	Mean fin temperature in K
Q	Air flow rate in m3/sec
V	Velocity of air at ambient temperature in m/s
Vmf	Velocity of air at mean temperature in m/s
Re	Reynolds number
Nu	Nusselt number
h	Heat transfer coefficient in W/m2K
Kair	Thermal conductivity of air in W/m2k
m	Physical significance

1. INTRODUCTION

Heat transfer enhancement techniques are very important to save energy and using of optimal energy sources. It is the process of improving the performance of a heat transfer system. In the past decades, heat transfer enhancement technology has been developed and widely applied to heat exchanger applications such as automobiles, chemical industry and process industry. Much effort in the past decades has been aimed to provide economical methods for improving the performance of heat exchanger. Active, passive and compound techniques are used for the enhancement of heat transfer. Nowadays, there have been a large number of attempts to reduce size and cost of heat exchangers, in reducing size and cost of a heat exchanger are basically the heat transfer coefficient and pressure drop.

The performance of heat exchangers is essential for reducing size of the system and to make the system more compact and the performance depends on the rate of heat transfer. The high rate of heat transfer is desirable because, it reduces the fuel consumption. A wide range of experimental, theoretical and numerical studies has been performed on the effect of different parameters like Reynolds number, Nusselt number, Concentration of nano fluids and size of nano particle.

2. LITERATURE REVIEW

Fins are one of the most problems in the heat transfer to increase rate of heat transfer on a solid surface. For the cases of constant heat transfer coefficient, the analytical solution of temperature profile and rate of heat transfer can be easily obtained [1]. However in some problem such as boiling liquid, the heat transfer coefficient of fin no constant and varies with temperature difference between surface and the adjacent fluid in a nonlinear manner. The dependence of heat transfer coefficient on the local temperature difference can be governed by a power–law type form. Numerous studies have devoted to the analysis of fin performance of this type of problems due to its important application in engineering. Chang [2] solved a decomposition solution for temperature dependent surface heat flux. ADM results used for compared with results of the present study. Joneidi and Ganji [3] solved differential transformation method to determined fin efficiency of convective straight fins with temperature dependent thermal conductivity. Later Unal [4-7] made a series studies on an extended surface with no uniform heat transfer coefficient and showed that the equation can be integrated analytically in a closed form for a limited number of cases. Liaw and Yeh [8] used the same model and further studies all possible type of heat transfer including the cases of film and transition boiling with and without heat transfer at fin tip. They also conducted an analytical and experimental study for a fin with a various type of boiling occurring simultaneously at adjacent location on its surface [9].

Abbasbandy and shivanian [10] made exact analytical solution of a nonlinear equation arising in heat transfer that nonlinear equation same with equation that solved in this study with DTM. Of course, in the present study solution represented by an infinite series, but in the paper solved equation pure analytically. Sen Kou, Lee and Lai [11] made thermal analysis of a longitudinal fin with variable thermal properties by recursive formulation. Khani and abdul Aziz [12] made thermal analysis of a longitudinal trapezoidal fin with temperature dependent thermal conductivity and heat transfer coefficient, in this paper used of HAM for solved equation. In the present study for the comparison, problem solved numerically by Rang-Kutta fourth order method for N=1 and several assigned value of n. The concept of differential transformation method was first introduced by Zhou [13] in 1986 and it was used to solve both linear and nonlinear initial value problem in electric circuit analysis. One of most Advantages of this method reducing the size of computational work while the Taylor series method is computationally taken long time for large orders. Aziz and Hug [14] used the regular perturbation method and a numerical solution to compute a closed form solution for a straight convective fin with temperature-dependent thermal conductivity. The HAM was used by Domairry and Fazeli to solve rectangular purely convective fin with temperature dependent thermal conductivity [15]. Khani, ahmadzadeRaji, and HamidiNejad [16] used HAM to evaluate the analytical approximate solution and efficiency of the nonlinear fin problem with temperature dependent thermal conductivity and heat transfer coefficient. Mustafa Inc [17] used HAM to evaluate the efficiency of straight fin with temperature dependent thermal conductivity and to determine temperature distribution within the fin. Arslanturk [18] and Rajabi [19] used the ADM and HPM to evaluate the efficiency of straight fins with temperature dependent thermal conductivity and to determine the temperature distribution within the fin. Lesnic and Heggs [20] applied the ADM to determine the temperature distribution within a single fin with a temperature dependent heat transfer coefficient. Ching-Huang and Chen [21] used to adomain decomposition method to evaluate fin efficiency and the optimal length of convective rectangular fin with variable thermal conductivity, and to determine the temperature distribution within the fin. Kundu and Das [22] made the thermal analysis and optimization of straight taper fins has been addressed. In this paper has been observed that the variable heat transfer coefficient has a strong influence over the fin efficiency. Mokheimer [23] investigated performance of annular fins of different subject to locally variable heat transfer coefficient, in this paper performance of fin expressed in terms of fin efficiency as a function of the ambient and fin geometry parameters.

Recently, differential transformation method has been used to solve a wide range of physical problem. This method provides a direct scheme for solving linear and nonlinear deterministic and stochastic equation without the need for linearization and yield rapidly convergent series solution. Rashidi and Erfani [24] used DTM for solved fin efficiency of convective straight fins with temperature dependent thermal conductivity and comparison results with HAM. Chiam [25] used of perturbation method for solve heat transfer in a fluid with variable thermal conductivity over a linearly stretching sheet. Their results of this study showed that the differential transformation method has many merits including fast convergence and high accuracy. In this study is to apply differential transformation method to investigate a straight fin governed by power-law type temperature dependent heat transfer coefficient. Base of DTM, temperature on the fin surface can be expressed explicitly as a function of position along the fin. The effect of exponent value and fin parameter on temperature profile as well as fin tip temperature can also be obtained quickly. In addition to, heat transfer rate and fin efficiency are presented in detail. In the present study results are compared with [2, 10].

3. EXPERIMENTAL INVESTIGATION

3.1 Specifications of rectangular duct used in the analysis

The experimental study is done internal threads into the rectangular duct. Specifications as listed below:-

Specifications of Rectangular Duct:

Material of construction= Aluminium

Inner Diameter, ID= 10 cm

Outer Diameter, OD=11 cm

Length of duct = 50 cm

Air at atmospheric temperature was allowed to flow through the inner diameter of channel.

3.2 Details of experimental set-up

Figure shows the schematic diagram of the experimental set up. Test Pipe show in figure, the rectangular channel is used for this investigation and made up of Aluminium, Brass, Copper & Stainless steel material. All the geometrical dimensions are in term of channel height while the heat transfer coefficient are presented in term of channel hydraulic diameter (D_h=0.1m) A suction mode blower is used to draw the air from entrance to exit section. The heated test section is 500mm long and 100mm dia. The uniform heat flux plate type heater is fabricated from nichrome wire. This heater is connected in series with dimmer stat in order to supply the same amount of heat to heater. The heater is wounded on the surface of channel. Commercial fiber glass insulation is used on external surface to prevent the heat leakage due to convection and radiation. Nichrome bend heater encloses the test section to a length of 50 cm. Three thermocouples T_2, T_3 and T_4 at a distance of 15 cm, 30 cm and 45 cm from the origin of the heating zone are embedded on the walls of the pipe and two thermocouples are placed in the air stream, one at the entrance (T_1) and the other at the exit (T_5) of the test section to measure the temperature of flowing air as shown in Fig. 3 The pipe system consists of a valve, which controls the airflow rate through it and an orifice meter to find the volume flow rate of air through the system.

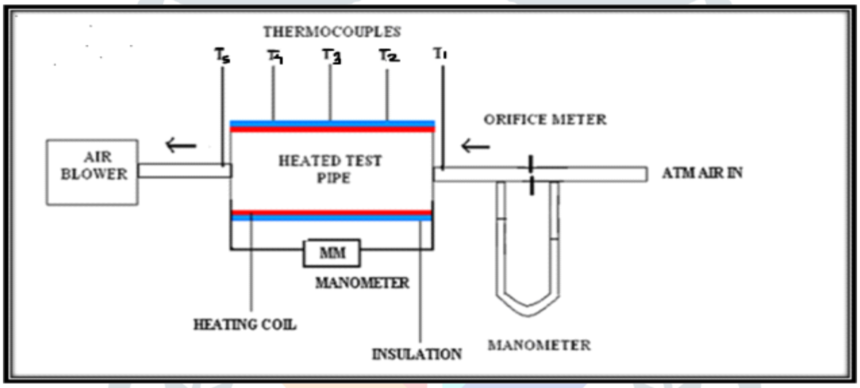

Figure-1:- Schematic diagram of the experimental setup

3.3 Design Parameters

3.3.1 Duct and Thread dimensions

- Diameter of Duct (D)= 10 cm
- Length of Duct (L)= 50 cm
- Heating Coil Length = 50 cm
- Thickness of channel (t1) = 0.5 cm
- Pitch of thread (P) = 1cm & 0.5 cm
- Material used for fin = Aluminium, Brass, Copper, Stainless steel

3.3.2 Constructional Features

The experimental setup consists of following components:-

Table-2:- Details of components/devices used in experimental setup and their description

The "J.P.T.I."unit consist of a experimental setup of different materials		
Sr.no.	Components	Technical Description
1	Centrifugal Blower	Specifications speed N = 2800 rpm.
2	Control Valve	Mass flow rate kg/sec
3	Thermocouples	Eight copper-constantan thermocouples having range of –50 to 1200 degree Celsius.
4	Heater	Plate type heater is used & length of heater is 500mm.
5	Rectangular channel	Rectangular duct made from aluminium sheet of thickness 0.5 cm, length 50 cm and Diameter 10 cm.

6	Digital Temperature Indicator	Digital temperature indicator is used in order to get the temperature readings from different place of duct.
7	Ammeter and Voltmeter	Ammeter and voltmeter are used to get the readings of current and voltage supplied to the heater.

Figure-2:- Pin Fin Apparatus (J.P.T.I)

Figure-3:-Test fin

4. OBSERVATIONS & CALCULATIONS (Brass Material)

Table-3:- Parameters

Dia. Of the fin	12.7mm
Length of fin	150mm
Dia. Of the orifice	14mm
Dia. Of the delivery pipe	36mm
Coefficient of discharge	0.65
Input voltage	100 V
Input current	1.02 A
Manometer reading	0.146

The experiments were conducted on the test rig initially without using any internal thread and the different heat transfer characteristics were calculated and then the same is done using internal thread with different pitch.

Voltage: 100 Volts

Current: 1.02 Amphere

Heat Generated: $Q = V * I$

Q=100 * 1.02 = 102 Watt

Table-4:- Observation Table for rectangular channel without internal thread

Sr no	V (Volts)	I (Amp)	Q (Watts)	V (m/s)	Inlet air temp Tb_1 (^{0}C)	Heater surface temp.			Exit bulk temp. Tb_5 (^{0}C)
						T_2 (^{0}C)	T_3 (^{0}C)	T_4 (^{0}C)	
1	100	1.02	102	3	40	88.3	83.2	89	41.8
2	100	1.02	102	3.3	40.1	87.1	82.2	87	42.4
3	100	1.02	102	3.6	40.2	86.8	81.3	86.5	42.9
4	100	1.02	102	3.9	40	84.9	80	85.1	43.3
5	100	1.02	102	4.2	40	83	79.1	83.9	44
6	100	1.02	102	4.5	40.1	82.2	78.8	82	45.3
7	100	1.02	102	4.8	40	81	77.8	81	45.5

Wall Temperature:

$$T_w = \frac{T_2+T_3+T_4}{3}$$

$$T_w = \frac{81+77.8+81}{3} = 79.93^0C$$

Properties at Mean Bulk Temperature: $$T_m = \frac{T_{b_1}+T_{b_5}}{2} = 42.75^0C$$

From Heat transfer Data Book – C. P. Kothandaraman and S. Subramanyan page no. 25

S.No.	**Temp. (^{0}C)**	**Density (kg/m^3)**	**Absolute viscosity (N-s/m^2)**	**Specific heat C_p (J/kg-K)**	**Thermal conductivity of air k (W/m-K)**
1	40	1.128	19.12 x 10^{-6}	1005	0.02756
2	50	1.093	19.61 x 10^{-6}	1005	0.02826

Various properties at mean bulk temp.

T_{mean} = 42.75^{0}C

ρ = Value + (42.75-40)/(50-40) x Value difference

= 1.128 + (42.75-40)/(50-40) x (1.128-1.093)

ρ = 1.1376 kg/ m^3

μ = 19.12 x 10^{-6} + (42.75-40)/(50-40) x (19.61 x 10^{-6} - 19.12 x 10^{-6})

μ = 19.25 x 10^{-6} N-s/m^2

C_p = 1005 J/kg-K

K = 0.02756 + (42.75-40)/(50-40) x (0.02756-0.02826)

K = 0.027752 W/m-K

Density ρ = 1.1376 kg/m^3

Absolute viscosity μ = 19.25 x 10^{-6} N-s/m^2

Specific heat C_p = 1005 J/kg-K

Thermal conductivity of air k = 0.027752 W/m-K

Hydraulic Diameter (D_h)

D_h = 10 cm= 0.1m

where, D_h = Hydraulic diameter in m

A = Area of channel m^2

$A = (\pi/4) \times (D_h)^2 = (\pi/4) \times (0.1)^2 = 0.0079\ m^2$

Velocity of air (V)

Discharge

$Q = 0.038\ m^3/s$

$Q = A \times V$

$0.038 = 0.0079 \times V$

V = 4.81 m/s

Mass flow rate (m)

$m = \rho AV$

m = (1.1376) x (0.0079) x (4.8)

m = 0.043 kg/sec

Heat transfer coefficient (h)

$Q = m \times Cp \times (T_{b5} - T_{b1}) = h \times A \times (T_w - (T_{b1}+T_{b5})/2)$

where, Q= heat transfer rate from heater

h= heat transfer coefficient W/m^2-K

A_s=Heat transfer area $m^2 = \pi D_h L = \pi \times 0.1 \times 0.5\ m^2 = 0.157\ m^2$

T_{b5}= bulk temperature at exit ^0C

T_{b1}=bulk temperature at inlet ^0C

Tw =avg. heater surface temp. = $(T_2+T_3+T_4)/3 = 79.93^0C$

Q= (0.043) x (1005) x (45.5- 40)

Q= 237.68 W

$Q = h \times A_s \times (T_w - (T_{b1}+T_{b5})/2)$

237.68 = h x (0.157) x (79.93-(40+45.5)/2)

h_f =40.90 W/m^2-K

Reynolds's Number (Re)

$Re = (\rho \times D_h \times v) / \mu$

$= \{(1.1376) \times (0.1) \times (4.8)\}/(19.25 \times 10^{-6})\}$

Re = 28366.129

Nusselt Number (Nu)

$Nu = (h \times D_h)/k$

$= \{(40.90) \times (0.1)\}/\{(0.027752)\}$

Nu = 147.37

(Test pipe) Manometer reading mm	T_w: wall temp (^0C)	(Orifice) Manometer Readings mm
No deflection	86.83	0.08
0.2	85.43	0.09
0.4	84.86	0.11
0.5	83.33	0.13
0.7	82	0.15
0.8	81	0.17
1	79.93	0.20

Table-5:- Observation Table for rectangular channel with internal thread (P=0.5)

Sr no	V (Volts)	I (Amp)	Q (Watts)	V (m/s)	Inlet air temp Tb_1 (^0C)	Heater surface temp.			Exit bulk temp. Tb_5 (^0C)
						T_2 (^0C)	T_3 (^0C)	T_4 (^0C)	
1	100	1.02	102	3	39.3	88	89.2	86.3	44.3
2	100	1.02	102	3.3	39.1	86.9	87.8	85	46.3
3	100	1.02	102	3.6	39.3	85.5	86.3	84.9	48.3
4	100	1.02	102	3.9	39.3	83.6	85.9	83.6	49.5
5	100	1.02	102	4.2	39.3	80.9	83.8	82.6	50.1
6	100	1.02	102	4.5	39	79.2	81.6	80.5	50.9
7	100	1.02	102	4.8	39.3	76.8	75.1	76.8	50.8

(Test pipe) Manometer reading	Tw wall temp (^0C)	(Orifice) Manometer Readings mm
1.5	87.83	0.08
2	86.56	0.09
2.6	85.26	0.11
3	84.36	0.13
3.3	82.43	0.15
3.8	80.43	0.17
4	76.23	0.20

Table-6:- Comparison in heat transfer coefficient for different test pipe

S. No.	Heat transfer coefficient without internal threads	Heat transfer coefficient with internal threads (p=0.5 cm)
1	6.52	18.07
2	9.66	30.47
3	12.77	44.14
4	17.73	57.18
5	23.68	67.79
6	34.76	85.87
7	40.90	101.52

Above Table show comparison of the heat transfer coefficient between the various tests pipes used for experimental work. It is find that heat transfer coefficient for test pipe with internal threads of pitch (p= 0.5 cm) is maximum as compared to other test pipes. So the heat transfer is maximum in test pipe with internal threads of pitch (p= 0.5cm).

5. RESULT

The experimentation is carried out with the rectangular duct with and without threads using heat transfer enhancement methods. Heat transfer coefficient, Reynold's number and Nusselt's number are calculated for all conditions.

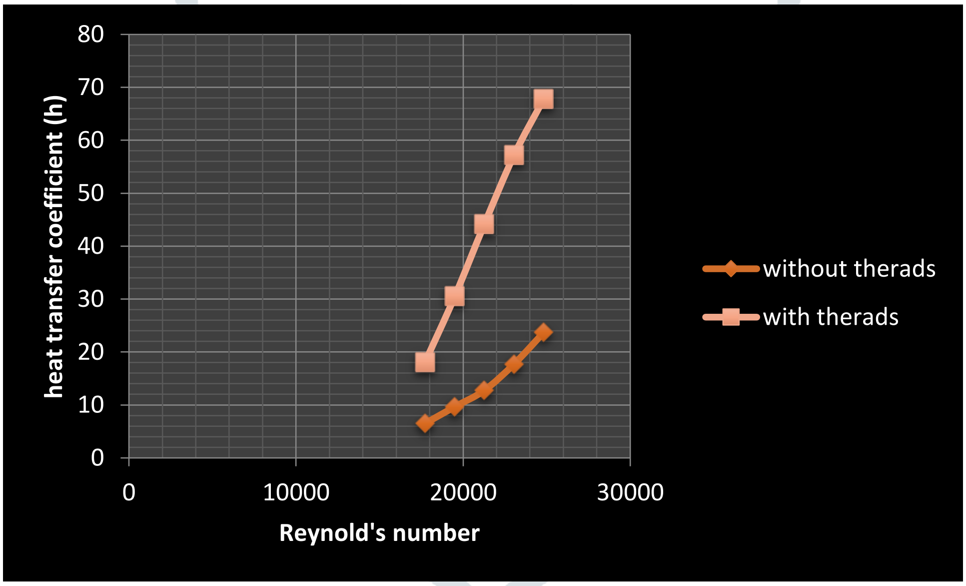

Figure-4:- Heat transfer coefficient Vs Reynold's number

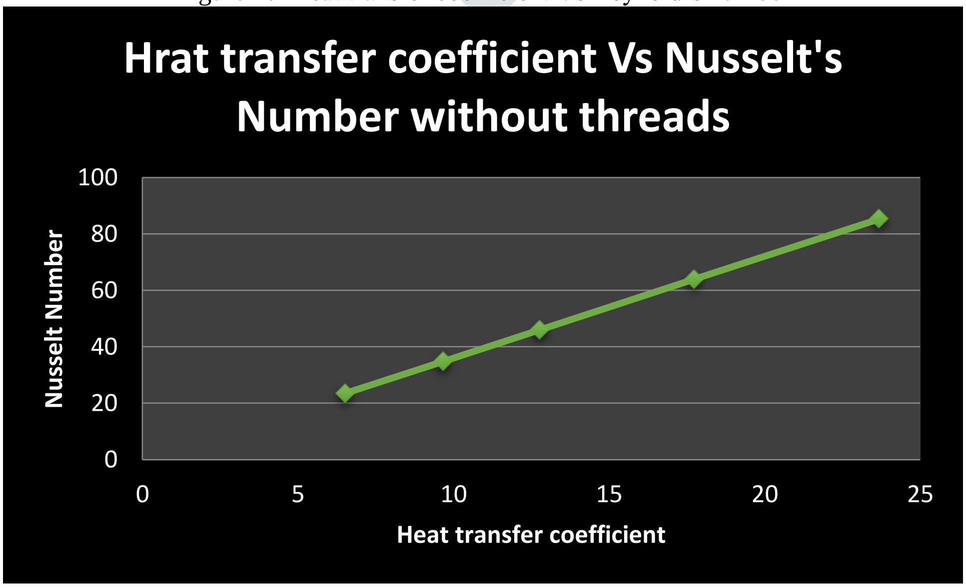

Figure-5:- Heat transfer coefficient Vs Nusselt's number without threads

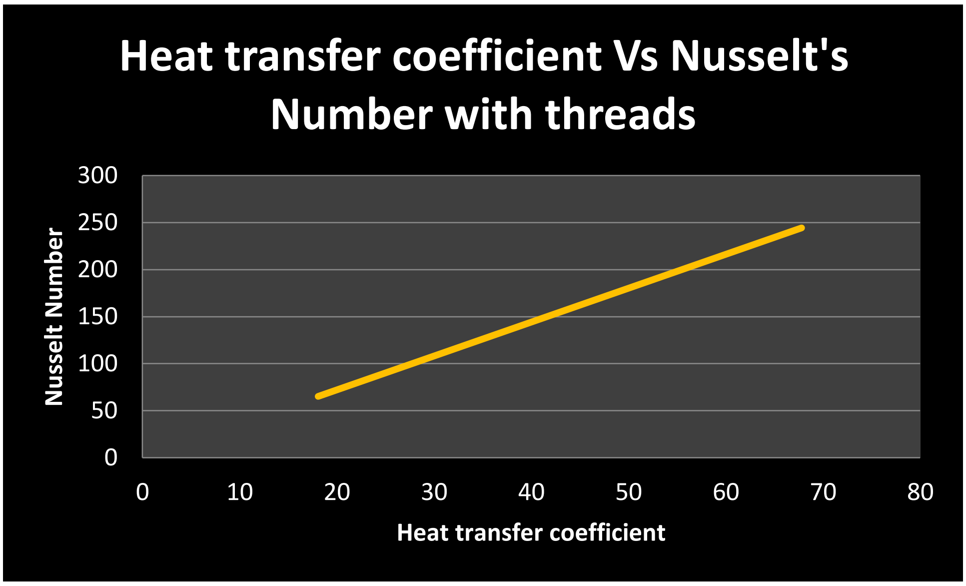

Figure-6:- Heat transfer coefficient Vs Nusselt's number with threads

6. CONCLUSION

The experimental investigation are observed that to study the effect of internal threads 0.5 cm brass material fin in a rectangular duct due to turbulence effect of air. The heat transfer coefficient, Reynold's and Nusselt's number are analysed with using heat transfer enhancement method. Heat transfer coefficient increases with increasing the pitch of threads in brass fin due to turbulence action of air with surface material of fin and also Nusselt's number higher as compare to without threads of fin material as result of convective mode of heat transfer through the fin.

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