AUGMENTATION OF HEAT TRANSFER IN FORCED CONVECTION USING TWISTED TAPE INSERTS

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ABSTRACT
In the present-day scenario, the trend in heat exchanger design is towards higher heat transfer rates per unit volume. This is achieved by providing more surface area per unit volume of heat exchanger equipment or by decreasing the resistance to heat flow i.e. increasing the heat transfer coefficients.
The present paper discusses the various techniques for enhancement of heat transfer, fabrication of equipment for forced convection heat transfer, materials for fabrications and their specifications, the effect of insertion of twisted tapes (h/d = 1 and h/d = 2) on heat transfer coefficients.
The experiment revealed that at various power inputs the increase in heat transfer coefficient will be from 50% to 180% more when compared to that of the bare pipe. A comparison between the heat transfer coefficients with bare pipe and with twisted tapes is represented in the form of graphs. From the observations, a relation for Nusselts number with modified constant is derived using non-linear regression analysis for both bare pipes as well as twisted tapes.

Keywords: augmentation, heat exchanger, forced convection, bare pipe, twisted tapes, Nusselt number.

1. INTRODUCTION

The trend in heat exchanger design continues to be in the direction of higher heat transfer rates per unit volume. In the usual commercial fluid to the fluid heat exchanger, there is an obvious economic incentive in reducing equipment size. This can be accomplished by having more surface area per unit volume of heat exchanging equipment and/or decreasing the resistance to the heat flow i.e., increasing the heat transfer coefficient. The devices, which involve high heat generation rates, must dissipate it quickly. This should be accommodated in a relatively smaller surface area in equipment like furnaces, economizers, air-cooled condensers, air preheaters etc. In these high heat flux systems, the surface to fluid temperature differences should be kept moderate to avoid melting or structural failures of the heated surfaces. These situations have a stipulated interest in the development of techniques to augment heat transfer.
The study of improved or increased heat transfer performance is referred to as Heat Transfer Augmentation, enhancement or intensification. Use of artificially roughened surfaces, internal and external fins, inlet vortex generators, insertion of objects like mesh, brush or twisted tapes, coiled wires are few examples of augmentation techniques. Existing systems can often be improved by using an augmentation method while in other applications such as the design of heat exchangers for use in space vehicles, an augmentative scheme may be mandatory in order for the system to function properly and to meet the size limitations imposed. Twisted tape inserts are inexpensive and they can be employed to improve the efficiency of any heat transfer system. However, these inserts involve more pressure drop across the heat exchanger and thus increases the pumping power.
Augmentation techniques are classified as
(i) Active techniques: requires external power
(ii) Passive techniques: require no direct application of external power
The effectiveness of these both types of augmentation techniques is strongly dependent on the mode of heat transfer which may range from single phase free convection to dispersed flow film boiling. Two or more of the above techniques may be utilized simultaneously to produce an enhancement which is larger than either of the techniques operating separately. This is called as Compound augmentation.

E. Smith berg and F. Landi et.al. [1] found out velocity distribution, friction loss and heat transfer characteristics analytically and experimentally for a fully developed turbulent flow in tubes using tape swirl generators. They conducted experiments with air and water as working media under isothermal and forced convection heating conditions with pitch to diameter ratios ranging from 3.62 to 22.0. Experiments were conducted in the range of 6×10^3 to 100×10^3 Reynolds number.

R.F. Lopina and A.E. Bergles et.al. [2] conducted experiments on non-boiling heat transfer coefficient and pressure drops for the heating and cooling of water. They conducted experiments on electrically heated test section with distilled water. The experiments were conducted with the tight fitting full-length tapes with twist ratios ranging from 2.5 to 9.2 and with Reynolds number ranging from 5×10^3 to 5×10^5. It was found that percentage decrease in pressure drop with heat addition is somewhat less for swirl flow than for a comparable empty tube flow. At constant pumping power, the improvement is at least 20 percent for swirl flow than that of straight flow.

A.E. Bergles, R.A. Lee and B. B. Mikic et.al. [3] In their experimental setup with water as working fluid and electrical heating, tried to take advantages of both roughness of tube and swirl generated by twisted tape to increase the heat transfer rate. They conducted the experiment with twist ratios ranging from 59-85 for smooth tubes and 54-79 for rough tubes with Reynolds numbers ranging from 10^3 to $10^{5.}$

E.E. Megerlin, R.W. Murphy And A.E. Bergles et.al. [4] conducted experiments with meshes of various gauges and brush inserts with water as working fluid. The brushes and mesh were developed for testing in electrically heated, circular tube test sections, there were 11 test sections used with a porosity of each section being 80+/-3 percent. The experiment was conducted with each test section from moderate power inputs, with small increments to burn out condition. Burn-out conditions were defined as data recorded just prior to the physical destruction of the test section.

A.W. Date et.al. [5] presented an analytical solution to predict the flow in an aqua tube with a twisted-tape. The author formulated partial differential equations of momentum and heat transfer. These partial differential equations were solved by existing numerical procedure. He also proposed laminar
flow predictions to demonstrate the influence of Reynolds number, twist ratio, Prandtl number and fin parameters on the flow characteristics. The transport equations of the twisted tape flow are strongly coupled.

Jose. L. Fernandes And Robert Poulter et.al. [6] used a Flag-type insert which was hinged on a diametral rod. This flag was capable to oscillate sideways. The oscillations were monitored by a piezoelectric probe at the surface. Instead of conventional electric heating, they used condensation of steam on the outside of the tube to heat water inside the tube. The overall heat transfer coefficient was calculated by dividing total heat flow by logarithmic mean temperature difference between water and steam and by heat transfer area.

1.1. Problem statement
A new method for the augmentation of heat transfer is used in this research work which involves the insertion of a twisted tape which acts as a swirl generator into the flow channel completely throughout its length.

1.2. Applications of Augmentation
a) Power Industry: Augmented heat transfer surfaces are widely used to improve gas side heat transfer in equipment like Furnaces, economics, air-cooled condensers and air preheaters.

b) Refrigeration Industry: The air conditioning and refrigeration industry have been a leader in the use of augmentation for single-phase forced convection.
Ex: Airside fin and waterside inserts. Internally finned tubes of various types are quite common for refrigerant evaporators of either shell and tube or coaxial type.

2. MATERIALS AND METHODS
2.1. Materials for Fabrication
1. G.I Pipe -25.4mm diameter
2. Gate valve
3. Copper tube-254mm length and 25.4mm diameter
4. Insulation rope -Asbestos
5. Twisted tape -Aluminum

2.2. Equipment used
1. Blower-1.55m^3/s discharge
2. Thermocouples- Chromel–Alumel
3. Digital temperature indicator
4. Electrical coil heater
5. Electrical supply board
6. Manometers
7. Orifice

2.3. Selection of Materials
Test Section
The test section is a copper tube of diameter 25.4mm has been selected due to the following advantages.
- It has high thermal conductivity
- It has smooth surface
- Thermocouple bead can be easily fixed on its surface.

Thermo Couple

A Chromel – Alumel Thermo-couple with composition:

Chromel – 90% Nickel, 10% Chromium

Alumel– 94% Nickel, 3% Manganese, 2% Aluminum, 1% Silicon

It is most widely used for industrial applications because of:

- It has fairly linear calibration curve.
- Good resistance to oxidation

3.EXPERIMENTAL SETUP & METHODOLOGY

A blower is taken and the mouth of the blower was reduced to 25.4mm diameter with the help of a reducer and then connected with a 25.4mm G.I. pipe. A copper tube is connected to the G.I. pipe with a flange on which electrical coils are wound. This serves as a heating section. Insulation rope is wound over the copper tube and a layer of plaster of Paris is coated on it so that the loss of heat transfer by convection to the surroundings can minimized. The other end of the copper tube is connected with a flange to a G.I. pipe on which gate valve is provided for varying the discharge of airflow. Other and of the gate valve is connected to an orifice which regulates the quantity of air leaving the section.

A U-tube water manometer is connected across the orifice. A thermocouple is used to measure the temperature of the air at the inlet to the blower and a thermocouple attached near the gate valve is used to measure the temperature of air leaving the section. The experiments were conducted with bare pipe and twisted tape insertion at various power inputs by varying the flow through the pipe. The experimental set up and twisted tape inserts are shown in figure 3.1 and 3.2.

Fig.3.1: The Experimental setup

a) Bare Pipe:

- Initially, the gate will be in the closed position.
- Start the blower and the gate valve is slowly opened to adjust the required flow rate with the help of manometer connected across the orifice.
- The power supply is given to the heater.
- For this fixed flow rate, by varying the power input experiment is conducted.
- Steady-state conditions are required to prevail in order to take the readings.
- Steady state temperatures of the four thermocouples connected to the heating section as well as the inlet and outlet temperatures of air are to be noted.
- Manometer head for flow rate is to be noted.
- By adjusting the flow rate to a new value and by varying the power input, the above procedure is to be repeated.

b) Twisted Tape Insertion

A flat plate made of aluminium is twisted into the form of a tape by fixing one end of the plate in a vice and applying a twist at the other end, making sure that a twisted tape of required h/d ratio is obtained. The twisted tape covers the entire length of the test section and creates turbulence.

Procedure with Twisted Tape

After placing the twisted tape, the gate valve is closed completely and the blower is switched on. Then the gate valve is opened slowly to adjust the flow rate at required Reynolds number. The power supply is given to the heater and the time lapse is allowed for the steady state to be reached.

Fig.3.2: Twisted Tape Inserts

After steady state is attained the temperatures of the six thermocouples fixed on the surface of the test section and air inlet and outlet temperatures are noted. The manometer heads for flow rate are also noted. Then the gate valve is adjusted to the next manometric head and the above procedure is repeated.

4. RESULTS AND DISCUSSIONS

4.1. With Bare Pipe, Power Input = 50 Watts

S.NO	HEAD in mm	SURFACE TEMPERATURE				AVG TEMP (T_s) ^0C	AIR INLET TEMP (T_i)^0C	AIR OUTLET TEMP (T_o) ^0C	AVG TEMP (T_b) ^0C
		T_1	T_2	T_3	T_4				
1	50	53	55	55	53	54	32	37	34.5
2	100	53	51	54	52	52.5	32	36	34
3	150	52	51	50	53	51.75	32	35	33.5
4	200	51	51	50	49	50.25	32	35	33.5
5	250	50	49	50	46	48.75	32	34	33

Heat Transfer Co-efficient (h) W/m^{2}K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
71.66	69.82	67.27	66.51	14086	0.7001
87.13	83.29	82.96	122.12	20572.4	0.7002
102.79	91.7	97.84	87.29	25280.9	0.7003
119.69	115.39	110.13	109.83	29308.9	0.7003
148.37	142.03	138.33	135.15	32729.5	0.7004

4.2. With Bare Pipe, Power Input = 90 Watts

HEAT TRANSFER CO-EFFICIENT (h) W/m^{2}K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
89.87	87.64	86.26	84.44	14415.1	0.6999
125.34	122.12	118.62	114.4	20439.3	0.6998
158.64	152.96	151.26	145.76	25060	0.700
171.47	166.23	163.44	158.35	29137.6	0.7001
179.71	173.12	172.24	166.48	32540	0.7002

S.NO	HEAD in mm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	50	55	58	58	58	57.25	32	39	35.5
2	100	54	56	54	57	55.25	32	39	35.5
3	150	53	55	52	54	53.5	32	38	35
4	200	52	55	51	53	52.75	32	37	34.5
5	250	51	51	52	51	51.25	32	36	34

4.3. With Bare Pipe. Power Input = 140 WATTS

S.NO	HEAD in Cm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	5	66	72	69	53	70	33	43	38
2	10	61	68	69	65	65.75	33	42	37.5
3	15	62	67	65	66	65	33	41	37
4	20	62	65	64	64	63.75	33	39	36
5	25	62	63	63	61	62.25	33	38	35.5

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
139.56	126.42	135.27	120.73	14243.5	0.6994
147.23	144.3	139.38	135.72	20203.5	0.6995
171.54	169.3	161.28	154.29	24818	0.6996
179.59	176.59	174.83	168.86	28823	0.6998
197.43	192.8	189.74	184.47	26747	0.6993

4.4. With Bare Pipe, Power Input = 204 Watts

S.NO	HEAD in mm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	50	84	83	81	85	83.25	35	48	41.5
2	100	81	83	82	84	82.5	35	47	41
3	150	81	80	82	83	81.50	35	46	40.5
4	200	80	79	78	81	79.50	35	45	40
5	250	76	79	75	80	77.50	35	43	39

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K	NUSSELTS NUMBER (Nu)	REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)

THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
151.26	149.07	139.68	132.26	14008	0.69885
172.83	168.58	163.22	157.88	19800	0.6989
183.74	181.78	171.41	165.52	24330.6	0.69895
202.32	198.44	182.68	175.27	28166.9	0.699
214.68	211.45	203.62	197.89	31670.8	0.7028

4.5. With Twisted Tape Δ /D Ratio = 1, Power Input = 50 Watts

S.NO	HEAD in mm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	50	46	45	43	47	45.25	33	41	37.5
2	100	45	44	42	46	44.25	33	41	37
3	150	44	44	41	45	43.5	33	41	37
4	200	44	43	40	44	42.75	33	40	36.5
5	250	44	43	40	44	42.75	34	40	37

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
166.82	162.32	161.26	154.68	14279.9	0.6995
187.23	184.53	182.86	177.62	20262.7	0.6996
215.38	212.4	209.67	202.29	24828.3	0.6996
251.02	246.68	243.14	238.49	28748.6	0.6997
291.21	288.4	284.92	279.62	32039.48	0.6995

4.6. With Twisted Tape Δ /D Ratio = 1, Power Input = 90 Watts

S.NO	HEAD in mm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	50	52	51	47	53	50.75	34	45	39.5
2	100	50	49	45	51	48.75	34	44	39
3	150	50	48	44	50	48	34	43	38.5
4	200	49	48	44	50	47.75	34	43	38.5
5	250	48	47	43	49	46.75	34	42	38

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		

199.51	193.54	191.26	188.64	14196.73	0.7029
227.48	247.51	221.24	236.21	20032.02	0.7028
251.62	225.05	243.63	214.62	24605.5	0.6993
262.71	257.32	254.32	248.29	28396.7	0.6993
289.38	286.84	281.49	272.39	31856.08	0.6994

4.7. With Twisted Tape Δ /D Ratio = 1, Power Input = 140 Watts

S.NO	HEAD in mm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	50	59	58	52	59	56.75	35	48	41.5
2	100	57	56	50	57	55.5	35	47	41
3	150	56	55	50	57	54.5	35	46	40.5
4	200	55	53	49	55	53	35	45	40
5	250	54	51	48	54	51.75	35	45	40

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORETICAL VALUE	EXPERIMENTAL VALUE	THEORETICAL VALUE	EXPERIMENTAL VALUE		
215.62	211.12	209.86	203.24	13955.22	0.69885
249.86	245.12	241.43	236.28	19803.036	0.6985
275.83	273.98	269.71	262.49	24259	0.69885
302.9	298.66	291.23	269.72	28260.9	0.699
338.62	333.26	316.08	298.42	31480.42	0.699

4.8. With Twisted Tape Δ/D Ratio = 1, Power Input = 204 Watts

S.NO	HEAD in mm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	50	67	69	60	71	66.5	35	53	44
2	100	66	66	57	68	64.5	35	52	43.5
3	150	65	64	56	67	63	35	51	43
4	200	63	63	55	66	61.75	35	50	42.5
5	250	62	62	54	65	60.75	35	49	42

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
239.24	233.32	231.68	227.49	23913.11	0.6986
261.29	257.23	253.41	249.68	23981.5	0.69865
282.16	278.04	275.32	264.9	24060.05	0.6987
315.96	311.12	307.62	302.46	24118.78	0.69875

372.18	363.22	345.28	331.24	24189.06	0.6988

4.9. With Twisted Tape Δ /D Ratio = 2, Power Input = 50 Watts

S.NO	HEAD in mm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	50	50	51	49	51	50.25	34	44	39
2	100	47	46	48	49	47.5	34	43	38.5
3	150	45	47	46	46	46	34	42	38
4	200	44	45	45	44	44.5	34	41	37.5
5	250	43	43	43	44	43.25	34	40	37

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
119.68	114.35	115.72	109.62	14188.1	0.7028
129.28	126.4	124.68	118.72	20130.9	0.6993
154.27	151.6	149.27	144.29	24726.2	0.6994
179.47	174.5	173.69	167.42	28635.76	0.6995
197.79	195.87	193.26	188.26	32106.6	0.6996

4.10. With Twisted Tape Δ/D Ratio = 2, Power Input = 90 Watts

S.NO	HEAD in Cm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	5	53	53	54	56	54	33	45	39
2	10	52	50	50	53	51	33	43	38
3	15	50	50	49	51	50	33	42	37.5
4	20	49	47	48	49	48.25	33	41	37
5	25	49	47	48	48	47.25	33	40	36.5

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
143.32	140.42	138.86	128.34	14157.96	0.7028
165.27	163.4	159.64	151.9	20146.71	0.6994
191.22	189.3	185.81	171.2	24746.7	0.6995
216.39	212.43	211.33	209.64	28660.7	0.6996
228.64	223.14	221.96	218.96	32007.6	0.7023

4.11. With Twisted Tape Δ/D Ratio = 2, Power Input = 140 Watts

S.NO	HEAD in Cm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	5	58	57	57	57	58	34	46	40
2	10	57	55	56	58	56.5	34	44	39
3	15	53	54	56	55	54.5	34	43	38.5
4	20	51	53	53	52	52.25	34	42	37
5	25	50	49	49	51	49.75	34	41	37.5

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
174.26	170.44	168.14	158.11	13995.22	0.69885
195.74	191.45	188.09	180.44	19803.03	0.6985
215.73	211.23	209.62	199.21	24259	0.69885
241.62	238.56	236.28	221.32	28260.9	0.699
255.93	250.55	248.63	241.35	31480.42	0.699

4.12. With Twisted Tape Δ/D Ratio = 2, Power Input = 204 Watts

S.NO	HEAD in Cm	SURFACE TEMPERATURE				AVG TEMP $^0C(T_s)$	AIR INLET TEMP $(T_i)^0C$	AIR OUTLET TEMP $(T_o)^0C$	AVG TEMP $^0C(T_b)$
		T_1	T_2	T_3	T_4				
1	5	67	66	66	69	67	35	49	42
2	10	65	64	63	67	64.75	35	47	41
3	15	63	62	61	64	62.5	35	45	40
4	20	61	59	58	61	59.75	35	45	40
5	25	58	57	60	61	59	35	44	39.5

HEAT TRANSFER CO-EFFICIENT (h) W/m^2K		NUSSELTS NUMBER (Nu)		REYNOLDS NUMBER (Re)	PRANDTL NUMBER (Pr)
THEORITICAL VALUE	EXPERIMENTAL VALUE	THEORITICAL VALUE	EXPERIMENTAL VALUE		
195.66	193.2	189.26	181.33	23913.11	0.6986
221.38	219.52	215.33	198.22	23981.51	0.69865
241.77	236.38	233.62	219.32	24060.05	0.6987
263.88	258.61	255.83	241.2	24118.78	0.69875
291.62	284.91	281.29	276.24	24189.06	0.6988

4.13. GRAPHS

Comparison of heat transfer coefficients with twisted taped inserts at various power inputs are shown below

Fig 1 : COMPARISON OF HEAT TRANSFER CO-EFFECIENTS WITH TWISTED TAPE INSERTS AT PI = 50W

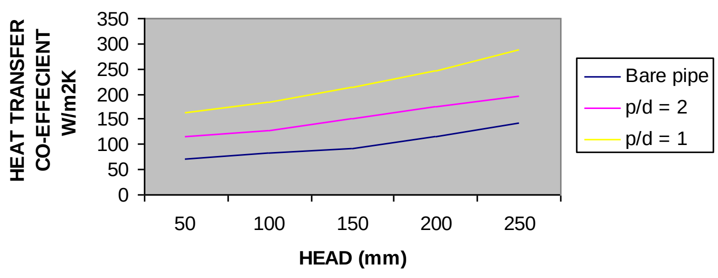


Fig 2 :COMPARISON OF HEAT TRANSFER CO-EFFECIENTS WITH TWISTED TAPE INSERTS AT PI = 90W

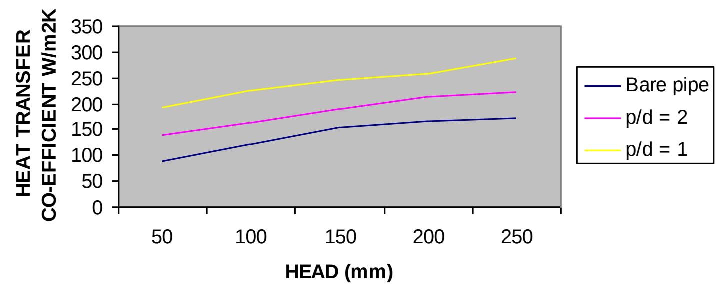


Fig 3 : COMPARISON OF HEAT TRANSFER CO-EFFECIENTS WITH TWISTED TAPE INSERTS AT PI = 140W

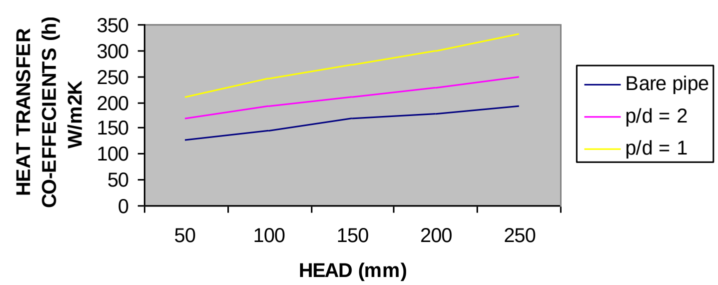
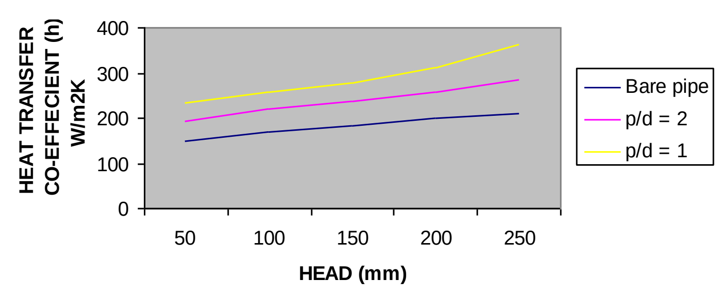


Fig 4 : COMPARISON OF HEAT TRANSFER CO-EFFECIENTS WITH TWISTED TAPE INSERTS AT PI = 204W



From the Fig 1, it is evident that the heat transfer coefficients are increasing with increasing manometric head. The reason might be the transition of flow from Laminar to turbulent. It is also clear that heat transfer coefficient is increasing with decreasing δ/d ratio & hence more for tape with δ/d ratio =1. The rate of increase of heat transfer coefficient is far more for with twisted tape inserts when compared to the bare pipe. A maximum of 75% enhancement in heat transfer coefficient can be observed by inserting twisted tapes. From the figures 2, 3, 4 similar results can be observed with twisted tape inserts at various power inputs.

4.14. MODEL CALCULATIONS

For a power input of 50 watts, and a manometric head of 50 mm, the calculations are as follows:

1. Average Surface temperature (T_s) = $(T_1+T_2+T_3+T_4+T_5)/5$ = **54^0C**
2. Bulk Mean Temperature (T_b) = $(T_0+T_i)/2$ = **34.5^0C**
3. Density of Air (ρ_a) = $(P_a)/(RT_a)$ = **1.158 Kg/m^3**

   Where,

P_a = Atmospheric Pressure = 1.01325 N/m^2
R = Gas Constant = 0.287 KJ/Kg-K
T_a = Ambient Temperature = 32^0C = 305K

4. Head Of Air (ha) = $(\rho_w h_w)/(\rho_a)$ = **43.18 m**

   Where,

   ρ_w = Density of Water = 1000 Kg/m^3
   h_w = Manometric Head in terms of water = 50 mm
   ρ_a = Density Of Air = 1.158 Kg/m^3

5. Discharge (q) = C_d x a x $(2gh_a)^{0.5}$ = **$4.75\text{x}10^{-3}$ m^3/s**

Where,

C_d = Co-efficient of discharge for orifice = 0.6
a = Area of Orifice = $(\pi d^2)/4$ = $2.72\text{x}10^{-4}$ m^2
d = Diameter of Orifice = 18.6 mm
g = Acceleration due to gravity = 9.81 m/s^2

6. Velocity of air = (q/A) = **9.39 m/s**

   Where,

   A = Area of pipe = $(\pi D^2)/4$ = 5.06x10-4 m^2
   D = Diameter of pipe = 0.0254 m

7. Heat Transferred (Q) = m x C_p x (T_o-T_i) = **27.64 W**

   Where,

   m = Mass flow rate = ρ_a x q = $5.50\text{x}10^{-3}$ Kg/s
   C_p = Specific heat of air = 1.005 KJ/Kg-K
   T_o = Outlet Temperature of air = 37^0C = 310K
   T_i = Inlet Temperature of air = 32^0C = 305K

8. Experimental value of heat transfer co-efficient

(h_{exp}) = Q/ ((A_sx (T_s-T_b)) = **69.82 W/m^{2}K**

Where,

As = Surface area of pipe = π x D x L = 0.0203 m^2
L = Length of Heating Section = 0.254 m

9. Experimental value of Nusselts number (Nu_{exp}) = (h x D)/ K = **66.51**

   Where,

   K = Thermal Conductivity of air = $26.665\text{x}10^{-3}$ W/m-K

10. Reynolds Number (Re) = (VD)/ν = **14086**

Where,

ν = Kinematic Viscosity of air = 16.04 m^2/s

11. Prandtl Number = **0.7001** (From Data Book)

12. Theoretical Value of Nusselts Number (Nu_{th}) = C $(Re)^m(Pr)^n$

    = $0.031(14086)^{0.8}(0.7001)^{0.4}$

    = **67.27**

13. Theoretical Value of Heat Transfer Co-efficient (h_{th}) = (Nu_{th} x K)/ D

    = **70.62 W/m^{2}K**

5.CONCLUSIONS

A new method for the augmentation of heat transfer is used in this research which involves the insertion of a twisted tape which acts as a swirl generator into the flow channel completely throughout its length.

The equation governing heat transfer is,

$Nu = 0.023(Re)^{0.8}(Pr)^{0.4}$ DITTUS-BOELTER EQUATION

By non-linear regression analysis, we got the governing equation as

$Nu = 0.031(Re)^{0.8}(Pr)^{0.4}$ For Bare Pipe.

$Nu = 0.035(Re)^{0.8}(Pr)^{0.4}(1+d/\delta)^{1.5}$ For Twisted Tapes.

The following conclusions are drawn:

It is clear from tables 4.1 to 4.12 that there is not much deviation from a practical Nusselts number with theoretical Nusselts number and the analysis has shown that the percentage deviation is within 19%.

Similar analysis has been carried out by comparing the practical Nusselts numbers with that obtained by using Dittus-Boelter equation. The deviation is found to lie between 21% to 24%. Finally, it can be concluded that twisted tapes can best be utilized for enhancing heat transfer

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