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Experimental study of Heat Transfer Enhancement due to Artificial Roughness in Annular Flow (Advanced manufacturing technology)

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Abstract: This paper presents results of an experimental investigation of heat transfer enhancement due to artificial roughness in annular flow. In this experiment we have taken two relative roughness pitch set up ($p/e=10$ and $p/e=5$) observations with helical springs wrapped on copper tube (Annular duct). The objective of the present experiment is basically to study the effect of use of perforated roughness elements on the heat transfer for the flow of air through an asymmetrically heated annular duct with helical coil spring wrapped in the form of helix on the copper tube. This study can be used as guidelines for better work on the annular heat exchanger with using helical springs wrapped over the duct.

Keywords: Heat transfer, annular duct flow, Roughness, Friction factor, Reynolds number

I. Introduction

Heat transfer to a fluid flowing in a circular tube annulus is a particularly interesting problem and one of considerable technical importance because of the surface can be heated independently. The duct wall roughness has a little effect on laminar flow however; it exerts a strong influence on the turbulent flow. The ratio of the surface roughness height to the laminar sub-layer thickness must be determining factor for the effect of roughness. For commercial pipes, the surface roughness height is very small as compared to the laminar sub-layer thickness and hence the roughness does not affect heat transfer.

Artificial roughness on the heat-transferring surface in the form of wires, integral ribs, or twisted tap inserts of different shapes and in various arrangements has been used in tubes and annular ducts.

The main thermal resistance to the convective heat transfer is due to the formation of a laminar sub-layer on the heat transferring surface. Efforts for enhancing heat transfer have been directed towards artificially destroying or disturbing this layer. This can be achieved by mechanical active, passive and combination of active and passive methods.

Active methods are:

- Mixing or scrubbing of fluid from the heat transferring surface by mechanical means
- Vibrating or rotating the heat transferring surface
- Suction of heated fluid through a porous surface

Passive methods are:

- Use of Fins
- Use of swirl flow devices
- Artificial roughness to create near wall turbulence to break the laminar sub-layer or to reduce its thickness.

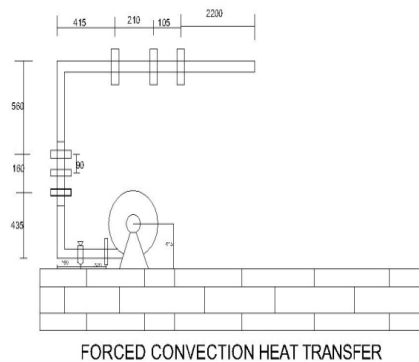
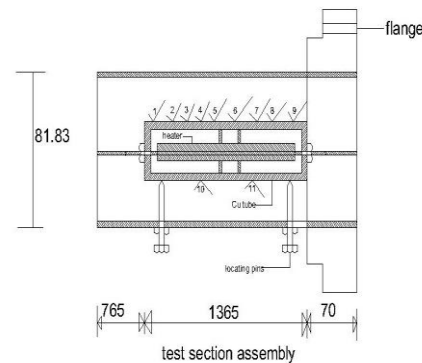
Artificial roughness on the heat transferring surface in the form of wires, helical springs or rib of different shapes and in various arrangements have been used in tubes and ducts.

For any basic type of roughness the key roughness parameters are

- a. **Relative Roughness Height** = $\frac{\text{Roughness Height (e)}}{\text{Hydraulic diameter of flow passage (D}_h\text{)}}$
- b. **Relative Roughness Pitch** = $\frac{\text{Distance between two adjacent element's pitch (p)}}{\text{Roughness height (e)}}$
- c. **Shape and type of roughness elements**

d. Angle of attack (α):

It is the angle of roughness element orientation with respect to the direction of flow. Gee and Webb investigated friction and heat transfer characteristics of rib roughened tubes having helix angle of 30° , 49° , 70° . The friction factor for a tube with ribs at 70° helix angle was found to be 60% higher than that with the smooth tube and friction factor decreased with the decrease in helix angle from 70° to 30° . The Stanton number was also found to decrease as helix angle decreases. Various researchers have reported optimum p/e ratio of 8 to 12 (Preferred value of 10). For $p/e > 8$ the flow reattaches after jumping the rib at a distance of 5 to 6 roughness height. It further develops before reaching next rib. The angle of inclination of helical coil spring with respect to flow direction is 66° (Approx)

II. Experimental setup**Figure-1 Experimental setup****Figure-2 Test section assembly**

The setup is shown in figure-1. The annular test section has been fabricated to determine the convective heat transfer coefficient for flow of air in annuli. The value of heat transfer coefficient ' h ', Nusselt number ' Nu ', has been found out at different flow rates for two configuration namely helical spring wrapped tube with $p/e=5$ and $p/e=10$.

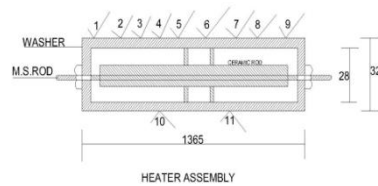
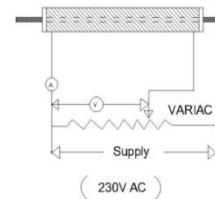
The setup consists of a centrifugal blower connected with a flow regulating valve and an annular test section through an orifice plate along with an inclined 'U' tube manometer. The test section is an annular duct, consisting of a heater placed inside the inner copper tube through its length. The specifications of the equipments used are given. Some modifications have been made in heater, entrance length, and pressure tap locations.

List of Equipments and Specifications:

- (1) .Blower- Motor: Power-2KW, RPM-2880, Voltage- 415V, 3 ϕ
- (2) .Air Heater: Nichrome wire of 450 Ω wound on ceramic tube insulators with Bakelite discs at ends, which Resists the heat loss from the ends of copper tube and they keep the heater concentrically in the Copper tube.
- (3). Test Section: $L=2200$ mm (4). Control valve (5). Orifice plate: Dia. 30 mm (6) .Ammeter: 0-10 amps.
- (7) Voltmeter: 0-300 volts (8) Multimeter (9) Wattmeter (10) Electronic micro manometer L.C. =0.01 and U-tube manometer (1:4.82)

III. Heater Assembly and heater circuit

The heater assembly consists of hollow ceramic rods. Nichrome resistance wire is wound around the ceramic rods. These ceramic rods are then slipped on mild steel rod, ceramic rods being separated with Bakelite washers in between them, as shown in figure attached. The resistance placed in series is capable of generating 117.6 Watt heat at full voltage supply. Teflon tape has been wound around the mild steel rod to avoid any short circuit or electric shock possibility. The mild steel rod has been threaded at its end to hold the ceramics rods firmly in position with the help of nuts. A variac has been used to control the supply of electric energy to the heater. The copper tube (1365mm long) which works as heat transferring surface is slightly longer than the length of heater assembly (1250 mm) the assembly is placed centrally inside the Cu tube. Two Bakelite sheet washers of diameter equal to inner diameter of the copper tube are screwed on the threads at the end of the MS rod. These washers are insulators, which resist the heat loss from the ends of the copper tube and also, they keep the heater concentrically in the copper tube.

Figure-3 Heater Assembly

Figure-4 Heater Circuit


Calibrated copper-constantan thermocouples have been used to measure various temperatures. A thermocouple located mid way in the pipe before the inlet to the test section measure the inlet air temperature. The outlet air temperature has been measured by transverse a thermocouple in the annular space as shown in figure. The inner tube surface (Heat transferring surface) temperature has been measured by fixing 11 thermocouples along the length as depicted in the figure attached. The electric energy supply to heaters has been measured with the help of an ammeter, a voltmeter and a Wattmeter.

Test Section Assembly:

The heater assembly along with the copper tube is placed inside a steel pipe (81.83 mm inside diameter). Three locating pins, provided on the periphery of the GI pipe at each end, located the assembly concentrically within the GI pipe. The heat generated in the heater is transferred to the flowing air by convection.

Orifice Plate Assembly:

The orifice plate assembly for the measurement of air flow rate through the test section is shown in the figure attached. A vertical/Inclined water manometer has been used. The coefficient of discharge C_d for $d/D=0.6$, has been taken to be 0.65 specified in Kent's hand book.

Roughened tubes:

The roughened tubes for the present investigation have been prepared by wrapping a helical springs of 4.65mm diameter in the form of helix with pitch of 43mm and 21.5mm. The helical spring has coils at pitch of 1.67mm and wire diameter of 0.5mm.

IV. Experimental Procedure

- Connections for the heater circuit are made (As in figure)
- The main supply is switched 'ON' and variance is adjusted, keeping the voltmeter and the ammeter reading constant, heater is allowed to work for some time.
- The blower is switched 'ON' and the control valve is adjusted to give desired flow rate. Orifice manometer and micro manometer readings are recorded.
- The manometer is recorded when steady state is reached. The steady state condition has been assumed to reach when the temperature does not deviate for 20 minutes period. The steady state condition was achieved in 1.5 to 3.0 hrs.

Throughout the time of taking observations, manometer, ammeter and voltmeter readings are maintained constants. A number of observations have been taken for different heat input and flow rates. The observations recorded are given in the tables (Observation tables). However, the micro monometer went out of order and pressure drop data for friction factor calculation could not be recorded.

V. Data Reduction and Observations

A. Data Reduction:

1. Mean tube surface temperature:

$$T_s = T_1 + T_2 + T_3 + T_4 + T_5 + T_6 + T_7 + T_8 + T_9 + T_{10} + T_{11}$$

Where $T_1, T_2, T_3, T_4, T_5, T_6, T_7, T_8, T_9, T_{10}$ and T_{11} are temperature measured from the thermo couples placed on the heat transfer surface.

2. Mean outlet temperature: Mean of the readings taken by traversing thermocouple radially at outlet of pipe.

3. Mass flow rate:

$$m = \rho (\pi/4 d^2) C_d \sqrt{2gh_a}$$

where $h_a = (\rho_w h_w) / \rho_a$

4. Heat gained by the air: $Q = mC_p(T_o - T_i)$

5. Heat Transfer coefficient: $h = Q/A_s(T_s - T_m)$, where $T_m = (T_i + T_o)/2$

6. Hydraulic Diameter:

$$D_h = 4 \times \text{Flow Area} / (\text{wetted perimeter})$$

7. Nusselt number: $Nu = (D_h \times h) / K$, Where k = thermal conductivity of fluid

Nusselt number equation for circular duct from literature is:

$$Nu = 0.023 (Re)^{0.8} (Pr)^{0.4}$$

8. Reynolds number:

$$Re = (GD_h) / \mu_a \text{ where } G = m/A_c (K_g/m-s) \mu_a = \text{Dynamic viscosity of air } (K_g/m-s)$$

The properties of air k, μ_a, ρ_a etc. have been taken at mean air temperature t_m .

9. Coefficient of friction:

$$f = \frac{2gD_h(\Delta p)}{4l\rho V_p^2}$$

Where Δp is pressure drop in the test section

B. Formule used in calculations:

Heat transfer coefficient:-

$$h = Q_u / A(T_s - T_m)$$

where $A = \pi DL, D=0.0319, L=1.364m$

$$T_m = (T_i + T_o)/2$$

T_s = mean of all temperatures corresponding thermocouple readings at surface of copper tube.

Nusselt No.:- $Nu = hD_h/k$, Where $D_h = 4A/P = D_o - D_i$

$K=0.0274$ at T_m temperature, So $Nu = 1.8285h$

Table-I Shows the Nusselt Number corresponding to $P/e=5$ and $P/e=10$

S.No.	Nusselt No.		(Nu) ₅ / (Nu) ₁₀
	$P/e=10$	$P/e=5$	
1.	45.876	49.219	1.073
2.	57.992	69.022	1.190
3.	68.309	869.893	1.272

Table-II Shows the observation of temperature for particular flow rate (For $P/e=5$)

Table-II OBSERVATION TABLE (For $P/e=5$)																			
S. No.	Flow (?p) mm	Applied Voltage (V)	Air inlet temp T_i (mv)	Tube surface temperature (mv)											Air outlet temp T_o (mv)				
				T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{o1}	T_{o2}	T_{o3}	T_{o4}	T_{o5}
1.	70.0	44x4	1.6	2.2	2.3	2.4	-	2.5	-	2.7	2.7	-	2.7	-	1.9	1.9	2.0	2.0	2.0
2.	70.0	44x4	1.6	2.3	2.3	2.4	-	2.6	-	2.7	2.8	-	2.7	-	1.9	2.0	2.0	2.0	2.0
Av g	70.0	44x4	1.6	2.25	2.3	2.4	-	2.55	-	2.7	2.75	-	2.7	-	1.9	1.95	2.0	2.0	2.0
3.	200.00	50x4	1.7	2.3	2.4	2.5	-	2.7	-	2.9	2.9	-	2.9	-	2.0	2.1	2.1	2.1	2.1
4.	200.00	50x4	1.8	2.3	2.4	2.5	-	2.7	-	2.8	2.9	-	2.8	-	2.0	2.1	2.1	2.1	2.1
5.	200.00	50x4	1.8	2.4	2.5	2.6	-	2.8	-	2.9	3.0	-	2.9	-	2.1	2.1	2.1	2.1	2.1
Av g	200.00	50x4	1.8	2.33	2.43	2.53	-	2.83	-	2.87	2.93	-	2.87	-	2.03	2.1	2.1	2.1	2.1
6.	380.00	56x4	1.8	2.4	2.5	2.6	-	2.8	-	2.9	3.0	-	3.0	-	2.1	2.2	2.2	2.2	2.2

7.	380.00	56x4	1.8	2.4	2.5	2.6	-	2.8	-	2.9	3.0	-	3.0	-	2.1	2.2	2.2	2.2	2.2
Avg	380.00	56x4	1.8	2.4	2.5	2.6	-	2.8	-	2.9	3.0	-	3.0	-	2.1	2.2	2.2	2.2	2.2

Table-III Shows the observation of temperature for particular flow rate (For P/e=10)

Table-III OBSERVATION TABLE (For P/e=10)																			
S. No.	Flow (?p) mm	Applied Voltage (V)	Air inlet temp T_i (mv)	Tube surface temperature (mv)											Air outlet temp T_o (mv)				
				T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{01}	T_{02}	T_{03}	T_{04}	T_{05}
1.	70.0	44x4	1.80	2.5	-	-	-	2.8	2.8	2.9	3.0	3.1	-	2.9	2.1	2.2	2.2	2.2	2.2
2.	70.0	44x4	1.80	2.5	-	-	-	2.8	2.8	2.9	3.0	3.0	-	2.9	2.1	2.2	2.2	2.2	2.2
Avg	70.0	44x4	1.80	2.5	-	-	-	2.8	2.8	2.9	3.0	3.05	-	2.9	2.1	2.2	2.2	2.2	2.2
3.	200.00	50x4	1.80	2.4	-	-	-	2.7	2.8	2.9	2.9	3.0	-	2.9	2.1	2.1	2.1	2.1	2.1
4.	200.00	50x4	1.70	2.3	-	-	-	2.6	2.7	2.8	2.8	2.9	-	2.8	2.0	2.0	2.0	2.0	2.0
Avg	200.00	50x4	1.75	2.35	-	-	-	2.65	2.75	2.85	2.85	2.95	-	2.85	2.05	2.05	2.05	2.05	2.05
5.	380.00	56x4	1.80	2.3	-	-	-	2.7	2.8	2.9	2.9	3.0	-	2.9	2.1	2.0	2.0	2.0	2.1
6.	380.00	56x4	1.80	2.3	-	-	-	2.7	2.7	2.9	2.9	3.0	-	2.9	2.0	2.0	2.1	2.1	2.1
Avg	380.00	56x4	1.80	2.3	-	-	-	2.7	2.8	2.9	2.9	3.0	-	2.9	2.05	2.0	2.05	2.05	2.1

Table-IV Shows the observation of Average value of data (For P/e=10)

Table-IV OBSERVATION TABLE (For P/e=10)(Average values)															
S. No.	Flow (?p) mm	Applied Voltage (V)	Air inlet temp T_i (°C)	Tube surface temperature (°C)											Air outlet temp T_o (°C)
				T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	
1.	70.0	44x4	44.80	61.2	-	-	-	68.1	68.1	70.3	72.6	72.6	-	70.3	48.9
2.	200.0	50x4	43.45	27.7	-	-	-	64.65	66.95	69.2	69.2	71.45	-	69.2	50.7
Avg	380.0	56x4	44.80	56.5	-	-	-	65.8	66.95	68.1	70.3	72.6	-	70.3	50.7

Table-V Shows the observation of Average value of data (For P/e=5)

Table-V OBSERVATION TABLE (For P/e=5)(Average values)															
S. No.	Flow (?p) mm	Applied Voltage (V)	Air inlet temp T_i (°C)	Tube surface temperature (°C)											Air outlet temp T_o (°C)
				T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	
1.	70.0	44x4	40.00	55.35	56.5	58.9	-	62.35	-	65.8	66.95	-	65.8	-	51.70
2.	200.0	50x4	44.00	57.22	59.88	62.42	-	66.5	-	69.64	71	-	69.64	-	51.5
Avg	380.0	56x4	44.80	58.9	61.2	63.5	-	68.1	-	70.3	72.6	-	72.6	-	54.2

Table-VI Shows the Final value after calculations (For P/e=10)

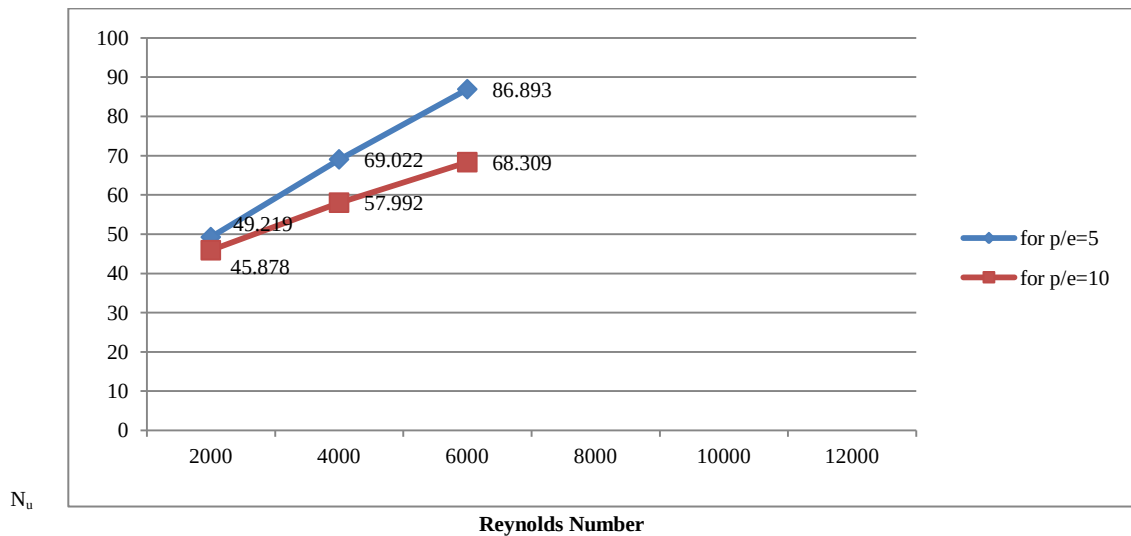
Table-VI CALCULATION TABLE (p/e=10)												
S.No.	T_i	T_o	T_s	T_m	flow	()	Q_u	m	R_e	f	h	Nu
1.	44.8	54.2	69.03	49.5	70	-	66.985	0.00708	4040	-	25.0903	45.878
2.	43.45	50.7	66.91	47.075	200	-	85.98	0.0118	6790	-	31.71	57.992
3.	44.8	50.7	67.22	47.75	380	-	99.43	0.0166	9710	-	37.358	68.309

Table-VI I Shows the Final value after calculations (For P/e=5)

Table-VII CALCULATION TABLE (p/e=10)												
S.No.	T_i	T_o	T_s	T_m	flow	()	Q_u	m	R_e	f	h	Nu
1.	40	48.9	61.66	44.45	70	-	63.327	0.00708	4040	-	26.918	49.219
2.	44	51.5	65.18	47.75	200	-	89.942	0.0118	6790	-	37.748	69.022
3.	44.8	51.9	66.74	48.35	380	-	119.45	0.0116	9710	-	47.516	86.893

VI. Result and Discussion

The experimental analysis always shows that Nusselt no. increases by 7% to 27% when the pitch is changed to half, i.e. from $p/e=10$ to $p/e=5$. The enhancement increases with Re . It is to be noted that $p/e=10$, Nusselt no. values obtained here are in reasonable agreement with the earlier studies. Despite the fact that there was some error in correct measurement of surface temperature. This enhancement shows that in case of perforated roughness a lower pitch is preferable as against $p/e=10$ has been reported to be nearly optimum for solid rib roughness.



The result can be increased turbulence due to closure arrangement of spring coils. However due to known measurement of pressure loss data, it is not possible to predict the correct effect on the friction factor. Basically any enhancement scheme success depends on heat transfer enhancement greater than friction factor enhancement. The work can be extended/repeated in future with better instrumentation and pressure loss data recording, for friction factor calculation.

VII. Conclusion

In this paper, an experimental investigation has been carried out for heat transfer and friction factor in annular duct with helical spring wrapped on the inner heated tube of annulus flow. Reynolds no. extends from 4040 to 9710 i.e. in the transition region. Two p/e values have been selected for the study. The heat transfer enhancement is 7% to 27% increasing with Reynolds no. which can be termed as quite significant. This enhancement can be attributed to the increased turbulence level at the heated surface. The real advantage of the enhancement scheme can be ascertained by taking friction factor data.

VIII. References

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