ISSN 2395-1621

Performance Evaluation of Cross-flow Heat Exchanger using Plain and Almond Dimple Tubes

#1 S.K.Vyas, #2Senthil Kumar,#3A.M.Elgandelwar

[1]shivavyas12@gmail.com
[2]senthil.sankaralingam@thermaxindia.com
[3]atul.elgandelwar@mitpune.edu.in

#1Department of Mechanical Engineering, Savitribai Phule Pune University
Maharashtra Institute of Technology, Kothrud, Pune.

ABSTRACT

An objective of the present project work was to design, manufacture and evaluate the performance of a cross-flow heat exchanger using plain tube and almond dimple tubes having three different tube bank arrangements. Using air and water as test fluids a wide range of fluid flow condition were tested. The objective is to analyse the effect of gas flow over tubes of different configurations. The mass flow rate of air was varied such that the Reynolds number was in between 10000 to 37000. Since the objective is to analyse the performance of gas flow over tubes, the tube side fluid flow condition was kept constant.Experiments with Plain tube and dimple tubes having three different arrangements have been carried out. Experimental values obtained for Plain tube were compared with theoretical values of Plain tube for validation at the same flow condition. After the validation, experiments were conducted using Dimple tubes of three different arrangements.1) Dimple narrow end aligned to left.2) Dimple narrow end aligned to right.3) Dimple tubes were randomly arranged. Comparisons of experimental results for all three arrangements were compared with the Plain tube experimental results. The performance parameters related to heat exchanger such as pressure drop, overall heat transfer coefficient, heat duty have been reported in this work. The percentage increase in overall heat transfer co-efficient (Narrow end of dimple when aligned to left was found to be 7.02% to 19.23% when the Reynolds number range was between 10000 to 37000 respectively.Pressure drop found to be more in case of almond dimple tubes in all the three configuration when compare with plain tube.

Keywords— **Reynolds number, Pressure Drop, Almond Dimple, Overall heat transfer coefficient**

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. Introduction

We know that in our industries there is large consumption of energy, at the same there is depletion in our energy resources. So, it is necessary to conserve energy resources. Nowadays industries have started using high performance Heat Exchangers. High Performance Heat Exchangers use different techniques for enhancement of heat transfer, which results in reduction in heat transfer area and thereby reduction in energy required to manufacture. Increase in heat exchanger performance can lead to more economical design of the heat exchanger which can help to move energy, material and cost savings related to a heat exchange process.

A. DIMPLED TUBE

High heat transfer rates are obtained by the Dimple tubes. This is an important development for the energy conversion and process industries. Use of Dimple tubes provides more heat transfer and an earlier transition to high heat transfer .The enhanced structure for both the Almond and spherical dimple could disturb, swirl, break the boundary layer developing, and augment the mixing of the hot and cold fluid and then improve the heat transfer of the tubes. Almond Dimpled tube picture is shown in below:

Fig 1: Almond Dimple Tubes.

Numerical studies for each tube, covering a wide range of Reynolds numbers starting from 1000 and going as far as 5000 was carried out by K Senthil Kumar [7]. According to Nopparat [1] the air side heat transfer performance is augmented approximately 10–22% at all Reynolds Numbers and all dimple arrangements. For staggered arrangements, the dimples pitch of SL/D_{minor} ¼ 1.875 and ST/D_{minor} ¼ 1.875 yields the optimum thermal resistance values of about 21.7% better than flat plate. For the staggered configuration of the dimple and compare with the results of Plain tube, numerical investigation was carried out for similar experimental conditions with addition to align array of dimple [9]. Effect of the twisted tape insertion on heat transfer and flow friction characteristics in a concentric double pipe heat exchanger have been studied experimentally by using twisted tape placed inside the inner test tube of the heat exchanger with different twist ratios [10].

II. DESIGN METHODOLOGY

A. *Heat Duty Calculations*:

1) Air side Calculation [3]:

Heat Transfer (Heat Duty): Heat exchangers transfer heat from one working fluid to another. Here the working Fluid is Air which is flowing over the Tubes in shell side of the heat exchanger and water flowing inside the tubes.

1. Heat Duty for Air side $Q = m_a \times C_p \times (\Delta t)_a$
2. Mass flow rate of Air,(ma)= $A_d \times v \times \rho$
3. Air Flow area (Fa)= (length × height) –((Tube OD× length) ×no of rows)
4. Mass Velocity for air (Ga)=$\left(\frac{m_a}{F_a}\right)$
5. Volume flow rate = $\frac{m_a}{\rho}$
6. Air velocity Va = $\frac{\text{volume flow}}{\text{Flow Area}}$
7. Reynolds no(Re) = $\left(G_a \times \frac{do}{\mu_{air}}\right)$
8. Prandtl No. Pr = $(\mu_{air} \times C_p)/k_{air})$
9. Nusselt no (N_u) = 0.33× $(R_e)^{0.6}$ × $(P_r)^{0.33}$ (correlation)
10. Convective heat transfer co-efficient(h_{air})= $\frac{N_u \times k_{air}}{do}$

Similarly Water side Calculations can be obtained using above formulas.

Overall heat transfer calculations (Theoretically):

$U = \frac{h_{air} \times h_{water}}{h_{air} + h_{water}}$

2) Overall heat transfer calculations (Experimentally):

$$\frac{Q}{(A \times (\Delta T)_{LMTD} \times Ft)}, (\Delta T)_{LMTD} = \left(\frac{\Delta T1 - \Delta T2}{\ln\left(\frac{\Delta T1}{\Delta T2}\right)}\right)$$

Where, $\Delta T_1 = (T_1 - t_2)$ and $\Delta T_2 = (T_2 - t_1)$, External surface area of tube: $A_s = \pi \times do$, Heat transfer area: $A = A_s \times L \times N$

3) *Pressure Drop Calculations for air side* [3]:

Dimensions taken in Feet:

Volume of duct=L×S×H where,

Width of heat exchanger

$(S) = (\text{no of columns} - 1) \times \text{Pitch} + 2 \times \text{gap} + do$

Height of heat exchanger:

$(H) = (\text{no of rows} - 1) \times \text{Pitch} + 2 \times \text{gap} + do$

Volume of total tube:$(Vt) = \pi \times (r_i)^2 \times L \times n$

Free volume for air (F_v) = $V_t - V$,$A_s = 2 \times \pi \times (r_i) \times L \times N$

Equivalent volume diameter: $E_{vd} = 4 \times \left(\frac{F_v}{A_s}\right)$

Pressure Drop:

$$(\Delta P)a = \left(\frac{f \times G \times G \times L}{(5.22 \times 10^{10} \times E_{vd} \times S_g) \times \left(\frac{E_{vd}}{P_t}\right)0.4 \times \left(\frac{P_L}{P_t}\right)0.6}\right)$$

4) *Design Specification of Air to Water Heat exchanger*:

Matrix for the Arrangement of tubes: 3×4(3 rows × 4 columns)

Width of heat exchanger = $(S) = (\text{no of columns} - 1) \times \text{Pitch} + 2 \times \text{gap} + do$

Height of heat exchanger=

$(H) = (\text{no of rows} - 1) \times \text{Pitch} + 2 \times \text{gap} + do$

Available Heat transfer area = $\pi \times do \times L \times N$

=π×.0381×.375×12 = 0.538 m^2

By taking it into design considerations (i.e. Air side Reynolds number 10000 to 40000) the air-water heat exchanger configuration is arrived as mentioned.

Table I: Design Parameters of air-water heat exchanger

Tube ID	0.0316	m
Thickness	0.00325	m
Tube OD	0.0381	m
no of tubes	12	
Tube length	375	mm
Transverse Pitch	57.15	mm
Longitudinal Pitch	57.15	mm
Gap between last tube and wall of heat exchanger	10	mm
Box Dimension		
L	280	mm
H	175	mm

W	375	mm
Hydraulic Diameter	300	mm
no of column	4	
no of rows	3	
Heat transfer area	0.538	m^2
Length of Tube	375	mm

Table II: Theoretical Design Results for Air to Water Heat Exchanger

Sr. No	Items	Units				
1	Air flow rate (outside)	kg/hr.	432	828	1199	1549
2	Inlet Temp	deg c	99.2	76.11	69.45	68.3
3	Outlet Temp	deg c	80	67.3	62.3	57.4
4	Velocity	m/s	5.43	6.03	6.78	7.35
5	Reynolds no		10000	20000	28000	37000
6	Water flow rate (inside)	kg/hr.	204	204	204	204
7	Inlet Temp	deg C	32	31.1	32.4	32.2
8	Outlet Temp	deg C	37.5	35.8	37.3	37.2
9	Velocity	m/s	0.1	0.1	0.1	0.1
10	Reynolds no		3202	3202	3202	3202
12	Overall U	W/m2k	54	62.42	70.42	77.2
13	Heat Duty	kW	1.17	1.21	1.22	1.24
14	Pressure Drop outside	mmWC	1	2	6.5	10
15	LMTD	deg C	56.05	38.5	29.91	28.46

III.EXPERIMENTAL SETUP

Consist of fan which is connected to the VFD which controls the frequency for obtaining the mass flow rate of air. Heater is connected after the fan which increases the temperature of air indicated by red colour arrows.

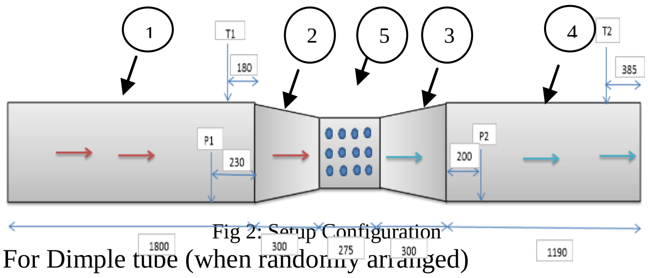


Fig 2: Setup Configuration

For Dimple tube (when randomly arranged)

A. Setup basically consists of four Ducts labelled by 1,2,3,4.

*Duct 1= (300*1800) mm², Duct 2= (300*175) mm², Duct 3= (175*300) mm² Duct 4= (300*1190) mm²*

T1 = Thermocouple for inlet temperature measurement, T2 = Thermocouple for outlet temperature measurement. Label 5 indicates Heat Exchanger: It has 12 numbers of tubes having length=375 mm

Dimension of Heat Exchanger indicated by 5: (175*387*280)

B. Experimental Procedure:

1. Before starting the fan note down the water level inside the U- tube Manometer. Check the inlet temperature of water. Turn on the fan which is connected to the VFD (Variable Frequency Drive).The motor was adjusted using VFD.

2. Test was taken at different flow rates of air. After switching on the fan let the flow get stabilised for a period of 5 minutes also note down the Pressure Drop readings. Now turn on the Heater. At the same time open the valve to allow water to flow through tubes.

3. Measure the Water inlet temperature using Thermometer. Wait until the system reaches a maximum Temperature which is time required to achieve the steady state. Time for steady state was kept 30 min.

4. After 30 min again note down the inlet and outlet temperature readings of air displayed on Digital Meter. Also measure the inlet and outlet temperature of water. Write down Pressure difference present on U-tube Manometer. Repeat the same procedure three times after every 10 min. Turn off the heater and water supply.

C. Experimental Evaluation*:*

A. For Plain Tube

Table III: Experimental observation for Plain Tube.

Reynolds no (air side)	Heat Duty for air side (kW)	Heat Duty for water side (kW)	Heat loss (kW)	Heat loss %	Overall heat transfer coefficient (w/m^2k)	Pressure Drop (mmWC)
10000	2.19	1.17	1	46	40.09	1
20000	2.15	1.09	1.05	44	55.54	2
28000	1.99	1.17	0.82	41	72.31	5.5
37000	2.41	1.24	1.17	49	81.65	8

B. For Dimple Tube When Direction of Dimple (Configuration 1):

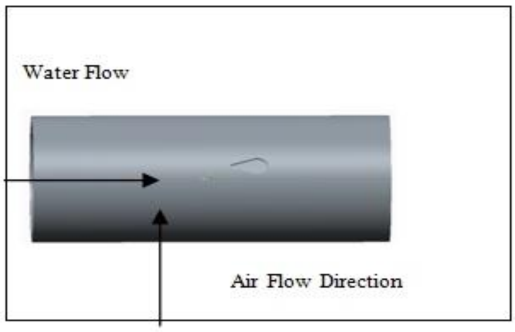


Fig 3: Showing Direction of Dimple

Table IV: Experimental observation for Dimple tube when narrow end align to left

Reynolds no (air side)	Heat Duty for air side (kW)	Heat Duty for water side (kW)	Heat loss (kW)	Heat loss %	Overall heat transfer coefficient (w/m^2k)	Pressure Drop (mmWC)
10000	1.94	1.28	0.74	38	43.12	2
20000	1.95	1.31	0.64	37	64.04	3.5
28000	1.63	1.29	0.56	21	86.02	7

37000	1.65	1.36	0.32	19	101.23	11

C. *When Direction of Dimple (Configuration 2):*

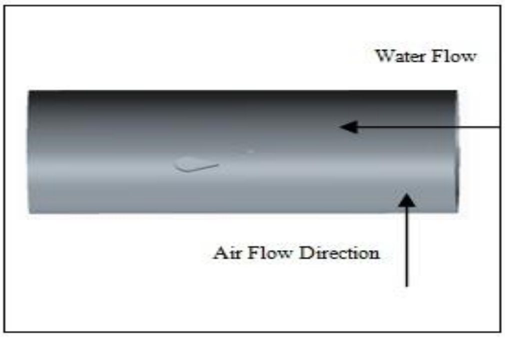


Fig 4: Showing Direction of Dimple

Table V: Experimental Observation for Dimple tube when narrow end align to Right

Reynolds no (air side)	Heat Duty for air side(kW)	Heat Duty for water side(kW)	Heat loss (kW)	Heat loss %	Overall heat transfer coefficient (w/m^{2}k)	Pressure Drop (mmWC)
10000	1.97	1.17	0.6	31	53.07	1.5
20000	1.74	1.26	0.4	26	65.03	3
28000	1.51	1.19	0.32	21	78.23	6.5
37000	1.49	1.21	0.28	19	94.53	10

D. *For Dimple tube (when randomly arranged)*

Table VI: For Dimple tube (when randomly arranged)

Tube No	Wide End	Narrow End
1	B	A
2	A	B
3	B	A
4	A	B
5	A	B
6	B	A
7	B	A
8	B	A
9	B	A
10	A	B
11	A	B
12	A	B

Table VII: Experimental Observation for Dimple tube when narrow tubes randomly arranged.

Reynolds no(air side)	Heat Duty for air side (kW)	Heat Duty for water side (kW)	Heat loss (kW)	Heat loss %	Overall heat transfer coefficient (w/m^{2}k)	Pressure Drop (mmWC)
10000	1.84	1.12	0.72	39	35.86	1.5
20000	1.68	1.14	0.54	32	57.86	3
28000	1.83	1.35	0.58	32	71.35	6.5
37000	1.49	1.21	0.28	41	119	10

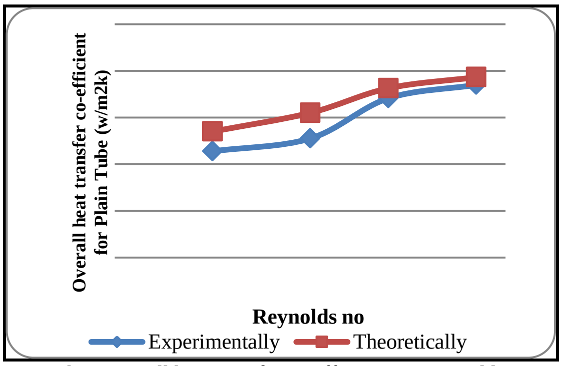
IV. RESULT & DISCUSSION

a Note 1: Compared three configurations of almond dimple tube with the Plain Tube.
Note 2: Depth of Dimple=1.5mm, distance between the dimple=10mm, dimple length=10mm, dimple radius=2mm (dimensions of dimple tube).

A. Comparison between Experimental and theoretical values. *(validation)*

Table VIII: Experimental and theoretical values of Overall heat transfer Co-efficient of Plain tube.

Air side Reynolds no	Overall heat transfer co-efficient for Plain Tube (w/m2k)	
	Experimentally	Theoretically
10000	45.54	54
20000	51	62
28000	68.25	72.56
37000	74	77.3



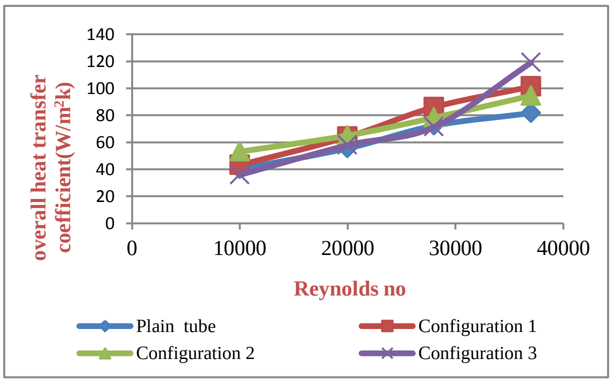
Graph 2: Overall heat transfer co-efficient Vs. Reynolds No

In order to validate the heat transfer measurement system in the experiments, the Overall Heat transfer coefficient of the Plain tube obtains experimentally were compared with the theoretical values of Overall heat transfer coefficient. Experimental results obtained of the Overall Heat transfer coefficient of the Plain tubes agree reasonably well with the theoretical values.

B. Comparison of Overall heat transfer Coefficient:

Table IX: Comparing of Overall heat transfer Coefficients of Plain & Dimple Tube with different arrangements.

Air Reynolds no	Overall heat transfer coefficient (w/m^{2}k)			
	Plain tube	Configuration 1	Configuration 2	Configuration 3
10000	40.09	43.12	53.07	35.86
20000	55.54	64.04	65.03	57.86
28000	72.31	86.02	78.23	71.35
37000	81.65	101.23	94.53	119



Graph 3: Overall heat transfer co-efficient Vs. Reynolds number

1. The overall heat transfer characteristics for the Air-flow in the staggered Almond Shape Dimples formed on the tube with different configuration and

Plain tube has been measured respectively within the Reynolds no range of 10000 to 37000.

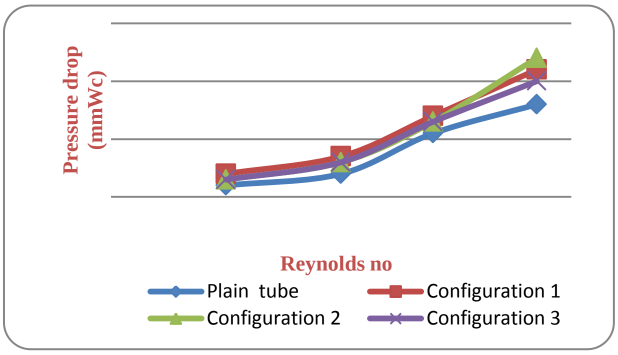
2. We can infer from the Graph 3 that overall heat transfer coefficient for Plain Tube is less than almond Dimple tubes when dimples are align left as well as right.

3. Discussion: The main reason for the heat transfer enhancement should be that the almond shapes on tube wall surface further increase the turbulent mixing in the flow near the wall by producing multiple vortex pairs, which enhance the turbulent flow heat transfer from the wall.

C. Comparison of Pressure Drop:

Table X: Comparing of pressure drops of Plain & Dimple Tube with different arrangements

Air Reynolds no	Pressure Drop(mmWC)			
	Plain tube	Configuration 1	Configuration 2	Configuration 3
10000	1	2	1.5	1.5
20000	2	3.5	3	3
28000	5.5	7	6.5	6.5
37000	8	11	12	10



Graph 4 : Pressure Drop Vs. Reynolds no.

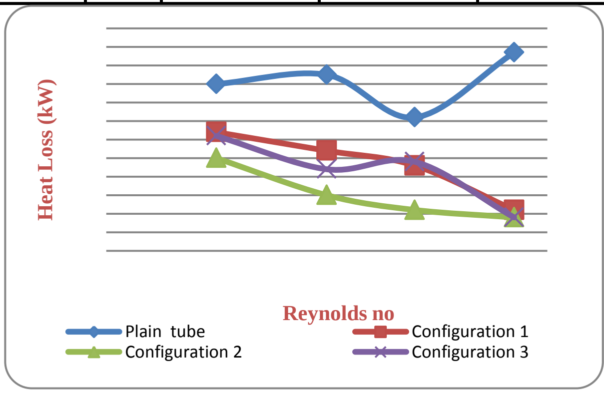
1. From Graph 4 it can be seen that as the Reynolds no increases there is increase in the pressure drop. We can infer from the Graph 4 that Pressure Drop in Plain Tube is less than almond Dimple tube when dimples are aligned left as well as right and also that pressure drop values are more for Dimple Tube when Dimple narrow end align to left. This may be due to some angle orientation in dimple alignment.

2. From the above discussion in case of overall heat transfer characteristics, almond Shapes on tube wall surface further increase the turbulent mixing in the flow near the wall and produce multiple vortex pairs, which enhance the turbulent flow and thus there is increase of Pressure Drop

D. Comparison of Heat loss:

Table XI: Comparing of Heat Losses in Plain & Almond Dimple Tube

Air Reynolds no	Heat Loss(kW)			
	Plain tube	Configuration 1	Configuration 2	Configuration 3
10000	1	0.74	0.6	0.72
20000	1.05	0.64	0.4	0.54
28000	0.82	0.56	0.32	0.58
37000	1.17	0.32	0.28	0.28



Graph 5: Heat Loss Vs. Reynolds no.

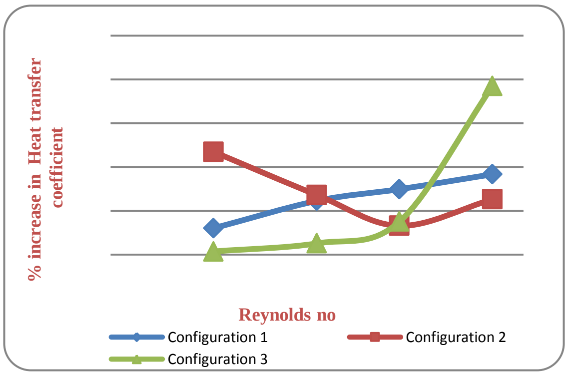
1. We can infer from the Graph 5 that Heat loss for Plain Tube is more than Dimple Tube when dimples are aligned left as well as right. Also we can infer from the Graph that Heat losses are less for Dimple Tube when Dimple narrow end align to right.

2. As per the above Discussion in case of overall heat transfer characteristics there is more turbulent mixing due to the vortices formed from secondary flows for Almond Dimple tubes which enhances the heat transfer coefficient and thus results in less heat loss.

E. Percentage increase in heat transfer co-efficient:

Table XII: Comparison of Percentage increase in heat transfer co-efficient with Plain tube

	Percentage increase in heat transfer co-efficient(w/m^2k)		
Air Reynolds no	Configuration 1	Configuration 2	Configuration 3
10000	7.02	24.45	1.71
20000	13.2	14.59	3.56
28000	15.93	7.56	8.55
37000	19.34	13.62	39.51



Graph 6: % incrrease in heat transfer co-efficient Vs. Reynolds no.

V.CONCLUSION

a An investigation was carried out for inferring the performance of the cross-flow heat exchanger using Plain and Almond Dimple Tubes.Based on the results obtained from the analysis, the following conclusions have been drawn out.

1. The overall heat transfer co-efficient is found to increase while using almond shaped dimpled tubes over Plain tubes. The percentage increase in overall heat transfer co-efficient (Narrow end of dimple when aligned to left is 7.02 to 19.23 when the Reynolds number range was between 10000 to 37000.

2. Heat Loss is less in case of dimple tubes as compare to Plain tubes this is due to there is more turbulent mixing due to the vortices formed from secondary flows for Almond Dimple tubes which enhances the heat transfer coefficient and thus results in reduction in heat loss.

3. The secondary vortices generated because of the dimple also help in enhancing convective heat transfer coefficient as the vortices help in mixing the hot and cold fluids. Thus, the Dimple on the tube found to enhance heat transfer over a Plain tube for turbulent air Flows.

ACKNOWLEDGMENT

Thanks to Mr. P. Charles, Head of ORC Dept. in Thermax India Ltd. for giving the opportunity to work on this challenging topic.

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