EXPERIMENTAL INVESTIGATION OF EFFECT OF AIR VELOCITY ON FIN PERFORMANCE BY FORCED CONVECTION

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***Abstract*— This investigation deals with the performance prediction of the cross flow air cooled circular fin tube heat exchanger. Experimental and theoretical studies were conducted to perform the optimization of the thermal design of the heat exchanger. The experimental work was carried out by two different types of air cooled heat exchangers i.e. with fin heat exchanger and without fin heat exchanger. An experimental rig was built for this object which provides hot water with different water circulating rate. A centrifugal blower was installed as a major part of the experimental rig to provide atmospheric air with different velocities. Result indicates that the effectiveness increases with increasing of air velocity.**

***Index Terms*—Fin tube heat exchanger, Effectiveness, Air velocity.**

I. INTRODUCTION

Heat transfer is a discipline of thermal engineering that concerns the generation, use, conversion, and exchange of thermal energy and heat between physical systems. Heat transfer is classified into various mechanisms, such as thermal conduction, thermal convection, thermal radiation, and transfer of energy by phase changes.

The modes of heat transfer are as under [1]

1 Conduction heat transfer
2. Convection heat transfer
3. Radiation heat transfer

A "Heat Exchanger " is process equipment designed for the effective transfer of heat energy between two fluids; a hot fluid and a coolant.the purpose may be either to remove heat from a fluid or to add heat to a fluid. .Notable examples are:

(i) Boilers, super heater and a condenser of a power plant.
(ii) Automobiles radiators and oil coolers of heat engine.
(iii) Evaporator of an ice plant.
(iv) Condensers and evaporators in refrigeration units.
(v) Water and air heaters or coolers. [2]

1.1 FINS. (EXTENDED SURFACES)

When the available surface is found inadequate to transfer the required quantity of heat with the available temperature drop and convective heat transfer coefficient, extended furfaces or fins are used. This practice, invariably, is found necessary in heat transfer between a surface and gas as the convective heat transfer coefficient is rather low in these situations.

The finned surfaces are widely used in:

1. Economiser for steam power plant;
2. Convectors for steam and hot-water heating systems;
3. Radiators of automobiles;
4. Air-cooled engine cylinder heads;
5. Cooling coils and condenser coils in refrigerators and air conditioners;
6. Small capacity compressors;
7. Electric motor bodies;
8. Transformers and electronic equipments etc. [3]

1.2 TYPES OF FIN.

Different fin configurations are illustrated in Figure 1.1. A straight fin is any extended surface that is attached to a plane wall. It may be of uniform cross-sectional area, or its cross-sectional area may vary with the distance x from the wall. An annular fin is one that is circumferentially attached to a cylinder, and its cross section varies with radius from the wall of the cylinder. The foregoing fin types have rectangular cross sections, whose area may be expressed as a product of the fin thickness t and the width w for straight fins or the circumference $2\pi r$ for annular fins. In contrast a pin fin or spine is an extended surface of circular cross section. Pin fins may also be of uniform or no uniform cross section. In any application, selection of a particular fin configuration may depend on space, weight, manufacturing, and cost considerations, as well as on the extent to which the fins reduce the surface convection coefficient and increase the pressure drop associated with flow over the fins. [4]

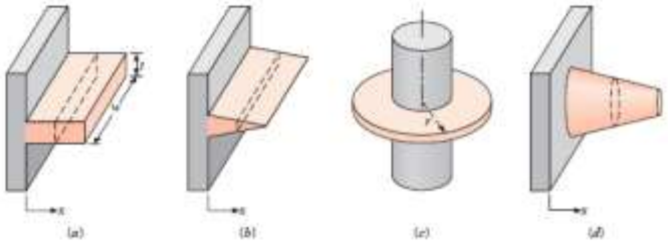

Figure 1.1. (a) Straight fin of uniform cross section. (b) Straight fin of Non uniform cross section. (c) Annular fin. (d) Pin fin

1.3 Fin Performance

1. Fin Effectiveness 2. Fin efficiency

1.3.1 Fin Effectiveness

Fins are used to increase the heat transfer from a surface by increasing the effective surface area. However, the fin itself represents a conduction resistance to heat transfer from the original surface. For this reason, there is no assurance that the heat transfer rate will be increased through the use of fins. An assessment of this matter may be made by evaluating the fin effectiveness ε_f. It is defined as the ratio of the fin heat transfer rate to the heat transfer rate that would exist without the fin Therefore

$$\varepsilon_f = \frac{q_{fin}}{q_{without\ fin}} \quad (1.1)$$

Where $A_{c,b}$ is the fin cross-sectional area at the base. In any rational design the value of $\sum_f$ should be as large as possible, and in general, the use of fins may rarely be justified unless $\varepsilon_f \geq 2$

1.3.2 Fin efficiency

Another measure of fin thermal performance is provided by the fin efficiency ηf .The maximum driving potential for convection is the temperature difference between the base (x = 0) and the fluid, $\Theta_b = T_b - T_\infty$. Hence the maximum rate at which a fin could dissipate energy is the rate that would exist if the entire fin surface were at the base temperature. However, since any fin is characterized by a finite conduction resistance, a temperature gradient must exist along the fin and the preceding condition is an idealization. A logical definition of fin efficiency is therefore,

$$\eta_f = \frac{q_f}{q_{max}} \quad (1.2)$$

Where A_f is the surface area of the fin. [5]

2. EXPERIMENTAL SETUP

The schematic diagram of experimental setup as shown in fig.2.1. Consists of two basic circuits and instrumentations i.e.

1. Air circuit,
2. Water circuit and
3. Instrumentation

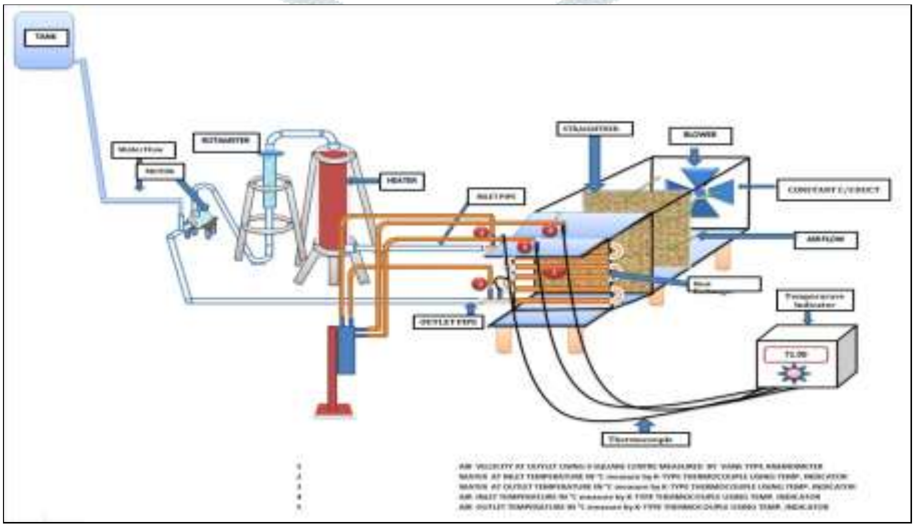

Figure 2.1 Schematic diagram of test apparatus

2.1 Air circuit

Air circuit consists of following major units:

1. Centrifugal cum axial blower,
2. Flow control arrangement,
3 Flow straightners,
4. Flow convergent section,
5. Heat exchanger unit
6. Exit duct

2.1.1 Centrifugal cum axial blower

Air is allowed to enter axially at the suction of centrifugal cum axial blower, which is basically a modified version of a centrifugal air blower with conventional centrifugal entry of air at the suction of the blower and with a modified exit at the rim of the blower so that at the exit of the blower, flow streams are in the axial directions.

The advantage of this unit is its compactness and vibration free operations. This unit is ideally suited for low pressure drop application.

2.1.2 Flow control arrangement

Air entering to the centrifugal entry at the blower is varied with the use of a butterfly valve arrangement provided at the suction of the blower. This arrangement provides the tool for varying flow condition at the suction of the blower. Arrangement contains a circular obstruction with a locking device, which allows locking of the butterfly damper at a given position.

2.1.3 Flow straightners

In order to predict the thermal performance of the given crossed flow finned tube heat exchanger, one important condition is that the flow at the face of the test section must be directed perpendicular to the test section and the same should be in uniform floe condition. This condition is very essential and the same achieved by providing enough number of flow straightners at the different strategic locations in the flow passages. Two sets of such straightners are located at the air duct one at the entry and another at the exit of the flow duct provided at the sections. The flow straightners are made out of a closely spaced flow straightner strips, bank of such strips are provided at the horizontal-vertical-horizontal thin strips. Bundle of such strips slashes the whirl component of the air flow and this can creates almost streamline flow at the entry of the test section.

2.1.4 Flow convergent section

In order to direct the flow of the air to the test section, a flow convergent section of sufficient length is provided at the end of the air duct. Providing separate flow convergent section will give interchangeability and flexibility in the event of testing several test sections of different frontal cross section area. Convergent angle of the flow convergent section should be such that the flow should not unduly create concentration effect at any point other than the centre of the section or to the frontal area of the unit.

2.1.5 Heat exchanger unit

Heat exchanger unit consist of a cross flow arrangement, which consist of a number of horizontal flat tubes (10 tube pass) having provision for fluid to flow through these tubes.

Two types of heat exchanger set-up used: (1) With fin heat exchanger and (2) Without fin heat exchanger

Tube length is approximately 500 cm in that & around 900 fin.
Tube diameter: - 14.6 mm
Circular Fin diameter: - 34.3 mm

In a circular fin heat exchanger, outside the tube circular fins are fixed with a much closed pitch (approximate 5mm each). The fins are fixed over the horizontal tubes using special process so as to eliminate all / any air gap. This helps in improving heat transfer performance of the test section. Hot Water enters in the test section through pipe at the upper tube and then flows through all the tubes. The test section gets cooled with the help of the flow of the air across these tubes. Heat transfer process is enhanced by provision of fins.

2.1.6 Exit duct.

Air at exit of the test section is hot and the same is channelled through an exit duct made to measure the hot air parameters like flow and temperature. Proper sealing is ensured at the exit of the test section, to avoid any escape of the air emerging out from the test section unit. Exit air duct is also contains provision for the fixing of flow meter and special arrangement to measure temperature of air.

2.2 Water circuit

Water circuit consists of following major units

1. Overhead Tank,
2. Electric Motor with Pump,
3. Heater,
4. Flow meter and
5. Test Section

2.2.1 Overhead Tank

Near to experiment setup a small overhead tank is placed at certain height, which is used as a feeding tank at the time of charging or priming the unit by Water. Tank is also used as a source of supply at the time of charging the unit and the same is connected at the lower most point so that the air can be removed from the top.

2.2.2 Electric Motor with Pump

Electric Motor with pump is provided at the outlet of the cold flow from the test section. Provision of the motor and the pump at the outlet of the test section is provided from the point of view of the handling the coldest flow from the test section, which will impart the smooth operation of the pump and longer operation life.

2.2.3 Heater With Variac

Electric Heater is used to act as a heat source to the circulating fluid i.e. water. The heater is fully insulated in order to prevent any conduction as well as convection loss of heat transfer to the surrounding, so that temperature of the circulating fluid may not dropped. Electric Heater is basically a counter flow heat exchanger in which the cold working fluid is admitted from the top and after getting heated due to the hot inside surface by means of electric resistance heater of uniform pitch, circulated in the close loop of the experimental set-up. Electric supply is provided with the help of variac for different voltages (i.e.80 volts, 100 volts, 120volts, 140volts).

2.2.4 Flow meter.

An obstruction type flow meter or Rotameter is used for the measuring the flow of the fluid in the circuit. Flow control valves are provided at the suction and discharge so as to regulate the flow of the working fluid as desired. Rotameter is situated in the circuit in such a way that it encounters only the cold working fluid. This will allow the use of less costly rotameter material and smoother operation life.

2.2.5 Test Section.

Test Section consists of two type of heat exchanger set-up.

1. With fin heat exchanger and
2. Without fin heat exchanger.

Figure 2.2 With fin heat exchanger

Figure 2.3 Without fin heat exchanger.

The general specification of heat exchangers is shown in table 2.1

Table 2.1 General specification of finned tube heat exchanger

Title	Value
Fin material	Aluminum
Tube material	M.S
Tube internal dia.	13 mm
Tube outer dia.	14.6 mm
Fin outer dia.	34.3 mm
Fin thickness	1 mm
Fin space	4 mm
Fin pitch	180/meter
No of fins	900
No of tube pass	10
No of fins for one tube pass	90
Total Length of tube	500 cm

Air flows outside the tubes and the working fluid flows inside the tubes. Water enters in the test section through a pipe at the top of the test section and then flows through the circular tube passages and coming out from the bottom row of the tube and again passed to the motor.

2.3 Instrumentation Scheme

In order to estimate the thermal performance of the cross flow heat exchanger, suitable instrumentation are employed in the experimental set-up for the estimating various parameters are given as under:

1. Temperature measurement,
2. Flow measurement,
3. Pressure measurement

2.3.1 Temperature measurement.

In order to estimate the absolute temperature of the cold air flowing in to the test section, a K-type thermocouple is provided in the air circuit at the entry to the test section. For exit temperature of air from the test section, another K-type thermocouple is provided. The temperature of air at entry and exit of the test section is indicated on temperature indicator.

Water inlet and outlet temperature to the test section is measured with the help of K-type thermocouple. The General specification of Thermocouple is as shown in table 3.2.

Table 2.2 General specification of Thermocouple

Temperature Sensor	**K Type**
Sensor	Open element type 3 meter cable simplex
Make	Sensewell instruments pvt.ltd.
Range	. 0 to 100 0C.

The surface temperature of bare tube and finned tube are measured at nine different locations with the help of Infrared Thermometer. . The General specification of Infrared Thermometer is as shown in table 2.3.

Table 2.3. General specification of Infrared Thermometer

Display	3-1/2 digit (1999 count) LCD with back lightning.
Ranges	-50 0c to 450 0c
Response time	Less than 1 second.
Operating temperature	0 0c to 50 0c.
Over range indication	LCD will show “1”.
Emissivity	0.95 fixed value.
Spectral response	6-14 μm

2.3.2 Flow measurement

In order to measure the flow of the air, a vane type anemometer is provided to measure the velocity of the air at the exit of the duct. While measurement of flow of water is carried out by rotameter. The General specification of Rotameter is as shown in table 2.4.

Table 2.4. General specification of Rota meter

Manufacturing company	**Instronics**
Range	0 to 1000 lph
Quantity	1

2.3.3 Pressure measurement.

A U-Tube manometer is used for measuring the pressure difference of water in mm of Hg at inlet and outlet of the test section. The General specification of U-Tube manometer is as shown in table 2.5.

Table 2.5 General specification of U-Tube manometer

Type	U-tube manometer
Range	0 to 250 mm of Hg

A Water column manometer is used for measuring the pressure difference of Air at before and after test section. The General specification of Water column manometer is as shown in table 2.6.

Table 2.6. General specification of Water column manometer

Type	Water column manometer
Range	o-50 mm of water
Quantity	1

3. EXPERIMENTAL PROCEDURE

In order to measure the thermal performance of the fin tube heat exchanger following are the steps carried out for the experimentations:

1. Ensure the overhead tank full of the working fluid which is water in our case.
2. Connect the supply of the overhead tank at the bottom most connection for admitting the working fluid.
3. Provide the air vent valve open to discharge the air from the system.
4. Open both the valves of working fluid as well as of the air vent valve.
5. Open the valve and allow the working fluid to admit the system until the working fluid start emerging out from the air vent valve.
6. Close working fluid inlet and air vent valve.
7. Run the centrifugal pump & check for the removal of the air bubble settle in the corners of the system, which can be observed from the rotameter.
8. Switch off the pump, if the presence of air bubble is found in the circulating fluid.
9. Repeat the procedure shown in the steps 2 to 7 until circulating fluid show no sign of air bubble.
10. Remove the air from the system without stopping the centrifugal pump and fill the working fluid by opening the inlet simultaneously.
11. Continue the process until the system is completely filled with water without any air bubble.
12. Run the pump for continuous circulation of working fluid.
13. Set the required working fluid flow by adjusting the valve and set the flow rate of as observed from the scale of the rot meter.
14. Start the air blower so that air flow is established in the wind tunnel.
15. Adjust the required air flow by moving the damper.
16. Set the required quantity of air flow by observing anemometer reading and that is 4m/s in first case.
17. Set the variac to zero voltage &then set it to the required voltage and that is 80 volts in first case.
18. Check the temperature of water at inlet & outlet of the test section by means of K-type thermocouple.
19. Wait until these temperatures becomes constant, this is essential to ensure the steady state condition.
20. Note the temperature of water at inlet and outlet of the test section.
21. Note the temperature of air at before and after the test section by K- type thermocouple.
22. Note air flow rate at various nine locations of the front section of the setup from anemometer display, and takes average of these nine readings.
23. Take surface temperature of the tube at nine various locations of the front section of the setup and takes average of these nine readings.
24. Take pressure difference of the working fluid inlet and outlet of the test section by U-tube manometer.
25. Take the pressure difference of the air before and after test section by water column manometer.
26. Change water flow rate to next level (i.e. 180 litre/hour to 500 litre/hour in the step of 80 litre/hour) and note down the observations.
27. Change the air flow rate to next level (i.e. 4 m/s to 12 m/s in the step of 2 m/s.) and note down the observations.
28. Now change the voltage giving to the heater (i.e. 80 volts, 100 volts, 120 volts, 140 volts and 160 volts) and repeat the above procedure.

4. OBSERVATIONS AND CALCULATION.

As mentioned before one internal working fluids, i.e. the water and one external working fluid i.e. the air were used in current study. In this dissertation the subscripts a, w, and f represent the air, water, and fin surface of the test specimen respectively.

Assumptions were made that the used fluids are incompressible Newtonian fluids and their properties are independent of pressure but the functions of temperature only. The liquids were assumed to be uniformly distributed through all the tubes in the test section.

In the experiments the measurements and data were taken for the fundamental and independent parameters. These parameters were as follows.

4.1 Liquid side measured parameters (water)

1. Mass flow rate.
2. Inlet or entrance temperature
3. Exit or outlet temperature
4. Pressure drop of water

4.1.1 Mass flow of Water:

The mass flow of water is governed by rotameter which is passing through the test section through tubes. The mass flow of water varies from 200 LPH to 500 LPH in steps of 80 LPH.

4.1.2 Water inlet temperature:

The water is passed through the tubes in the test section. The temperature of water at inlet to the test section is measured with the help of K-type thermocouple in ºc. As water is passing through the tubes, the temperature of water falls down due to heat transfer process.

4.1.3 Water outlet temperature:

Temperature of water at outlet to the test section is measured with the help of K-type thermocouple in ºc. Water is again re-circulated to the test section with the help of circulating pump through the rotameter.

4.1.4 Pressure drop of water;

As water is passed through the test section, the pressure of water falls down. The pressure drop is measured with the help of U-tube manometer in mm of Hg.

4.2 Air side measured parameters

1. Flow velocity (Va), through the wind tunnel
2. Inlet or entrance temperatures (Ta, i)
3. Exit or outlet temperatures (Ta,o)

4. Differential pressure across test section

4.2.1 Flow velocity (Va), through the wind tunnel.

The velocity of air through the test section was measured with the help of vane type Anemometer at the exit of the wind tunnel. The air velocity varied from 3 m/s, 4 m/s, 5 m/s and 6 m/s.

4.2.2 Air inlet temperature:

The temperature of air at inlet to the test section is measured with the help of K-type thermocouple in ºc.

4.2.3 Air outlet temperature:

The temperature of air at outlet to the test section is measured with the help of K-type thermocouple in ºc.

4.2.4 Pressure drop of Air:

As air is passed through the test section, the pressure of air falls down. The pressure drop is measured with the help of water column manometer in mm of water.

4.3 Evaluations of thermo physical properties of working fluids:

Unless otherwise stated, usually the thermo physical properties of the working fluids in current study were evaluated at bulk flow temperature between the inlet and exit of the flow path. Considering a linear variation of temperature between the inlet and exit of both fluids, the bulk temperatures were deduced as follows,

Liquid side bulk flow temperature

$$T_{b,w} = \frac{T_{w,i}+T_{w,o}}{2} \quad (4.1)$$

And

Air side bulk flow temperature.

$$T_{b,a} = \frac{T_{a,i}+T_{a,o}}{2} \quad (4.2)$$

Unless otherwise stated, all the thermo physical properties of all the working fluids.

Were evaluated at their respective bulk temperature defined by Equation 4.1 and 4.2 above. The properties of water and air mixture for each data point were derived from the fluid property calculator. [6]

Where Ta,i, Ta,o, Tw,i and Tw,o are the consolidated mean inlet and exit temperatures of the flowing air and liquid.

SR NO	V	va, air (m/s)	mw LPH	Ta,i	Ta,o	ΔTa (air)	Tw, i	Tw,o	ΔT (water)	WATER PRESSURE DROP.(MM of HG)	AIR PRESSURE DROP. (MM OF WATER)
1	80	3	180	33	34	1	52	46	6	24	0.5
2	80	3	260	33	34	1	52	45	7	40	0.5
3	80	3	320	34	35	1	51	45	6	62	0.5
4	80	3	420	34	35	1	51	45	6	81	0.5
5	80	3	500	34	35	1	50	45	5	121	0.5
6	80	4	180	34	35	1	50	44	6	26	0.7
7	80	4	260	34	35	1	50	45	5	41	0.7
8	80	4	320	34	35	1	50	44	6	65	0.7
9	80	4	420	34	35	1	49	43	6	83	0.7
10	80	4	500	34	35	1	48	43	5	124	0.7
11	80	5	180	34	35	1	50	42	8	28	0.9
12	80	5	260	34	35	1	49	44	5	42	0.9
13	80	5	320	34	35	1	49	44	5	66	0.9
14	80	5	420	34	35	1	48	43	5	86	0.9
15	80	5	500	34	35	1	47	43	4	127	0.9
16	80	6	180	34	35	1	49	40	9	29	1.2
17	80	6	260	34	35	1	48	44	4	43	1.2
18	80	6	320	34	35	1	47	44	3	65	1.2
19	80	6	420	34	35	1	46	43	3	86	1.2
20	80	6	500	34	35	1	45	43	2	132	1.2

Table 4.1 Observation table for 80 voltage with fin

Table.4.6 Observation table for 80 voltage without fin

SR NO	v	va, air (m/s)	mw LPH	Ta,i	Ta,o	ΔTa (air)	Tw,i	Tw,o	ΔT (water)	WATER PRESSURE (MM of HG)	AIR PRESSURE (MM OF WATER)
1	80	3	180	34	35	1	53	48	5	24	0.5
2	80	3	260	34	35	1	52	48	4	40	0.5
3	80	3	320	34	35	1	51	48	3	62	0.5
4	80	3	420	34	35	1	51	47	4	81	0.5
5	80	3	500	34	35	1	50	47	3	121	0.5
6	80	4	180	34	35	1	51	47	4	26	0.7
7	80	4	260	34	35	1	50	48	2	41	0.7
8	80	4	320	34	35	1	50	48	2	65	0.7
9	80	4	420	34	35	1	49	47	2	83	0.7
10	80	4	500	34	35	1	48	47	1	124	0.7
11	80	5	180	34	35	1	50	47	3	28	0.9
12	80	5	260	34	35	1	49	47	2	42	0.9
13	80	5	320	34	35	1	49	47	2	66	0.9
14	80	5	420	34	35	1	48	47	1	86	0.9
15	80	5	500	34	35	1	47	46	1	127	0.9
16	80	6	180	34	35	1	50	47	3	29	1.2
17	80	6	260	35	36	1	48	45	3	43	1.2
18	80	6	320	35	36	1	47	45	2	65	1.2
19	80	6	420	35	36	1	46	45	1	86	1.2
20	80	6	500	35	36	1	45	44	1	132	1.2

4.6 HEAT TRANSFER PERFORMANCE PARAMETERS

The heat transfer performance of any heat exchange device is associated to some key parameters such as the overall heat transfer coefficient, thermal resistance, and number of transfer units (NTU), effectiveness, and fin efficiency, individual fluid side heat transfer coefficients, etc., which are described and deduced in this section.

Heat capacity of water= mw* Cp,w
Heat capacity of air = ma * Cp,a
Heat transfer rate of water

$$Q_w = m_{w\ cpw} (T_{w,o} - T_{w,i}) \quad (4.3)$$

Heat transfer rate of water

$$Q_a = m_a\ c_{p,a}\ (T_{a,o} - T_{a,i}) \quad (4.4)$$

Average heat transfer rate:

$$Q_{avg} = 0.5\ (Q_w + Q_a) \quad (4.5)$$

The performance of the heat exchangers is analyzed by the conventional ε-NTU technique and the effectiveness, ε, is defined as:

$$\varepsilon = \frac{Q_{avg}}{(\dot{m}\ C_p)\ \min(T_{w,i} - T_{a,i})} \quad (4.6)$$

The number of transfer unit (NTU) is usually proportional to the heat transfer surface area A. Traditionally the larger the NTU the larger the heat exchanger.

The relationship of the effectiveness, the number of transfer unit (NTU), and the minimum heat capacity flow rate (m*cp)min min, at the air side could be

$$\varepsilon = \frac{1}{C^*}[1 - e^{-(1-e^{-NTU})}] \quad (4.7)$$

$$NTU = \frac{UA}{(\dot{m}\ C_p)\ \min} \quad (4.8)$$

$$C^* = \frac{(\dot{m}\ C_p)\ min}{(\dot{m}\ C_p)\ \max} \quad (4.9)$$

Using Eq. (4.8) and (4.9) the experimental overall heat transfer coefficient, UA_{pra}, could be evaluated.
The tube-side heat transfer coefficient can be calculated by Dittus Boelter - correlation for the turbulent flow through the tubes. [7]

Nu = 0.023*Re0.8*Prn (4.10)

Where n = 0.4 for heating and 0.3 for cooling. These equations have been confirmed experimentally for the range of conditions

$$0.6 \leq Pr \leq 60$$
$$Re \geq 10000$$
$$\frac{L}{D} = 10$$

The air-side heat transfer coefficient can be calculated from Vithayasai et al. [8] correlation suggested for the cross flow heat exchanger,

Nu_a = 10145 x Ln (Rea- 46.081) x Pr0.33 (4.11)

Airside overall surface or fin efficiency (η_0)

The temperature effectiveness or the overall airside fin efficiency (η_0) was first expressed as follows

$$\eta_0 = 1 - \frac{A_f}{A_o}(1-\eta) \quad (4.12)$$

$$A_o = A_f + A_b + \text{Space area} \quad (4.13)$$

A_o Is the total surface area of the finned tube, A_f is surface area of the Fin, A_b is surface area of the bare tube. The fin efficiency η, can be approximated from the Schmidt approximation [9] as follows:

$$\eta = \frac{tanh\,(ml)}{ml} \quad (4.14)$$

Where $m = \sqrt{\frac{2 \times h_0}{k_f \times f_t}}$

The overall heat transfer coefficient can also be estimated from the following overall resistances for the comparison with the experimental data:

$$\frac{1}{UA} = \frac{1}{\eta_0 h_0 A_0} + \frac{\delta}{k_t A_t} + \frac{1}{h_i A_i} \quad (4.15)$$

4.7 Sample calculation for 80 voltage with fin and its validation.

4.7.1. Calculation for Water and Air

Air velocity va = 3 m/s,

Voltage = 80 volts,

Mass flow rate of water m_w = 180 LPH

Ta,i = 33˚C

Ta,o = 34˚C

Tw,i = 52˚C

Tw,o = 46˚C

$$\Delta T_{air} = \text{Ta,o} - \text{Ta,i} = 34-33 = 1^\circ C$$

$$\Delta T_w = \text{Tw,o} - \text{Tw,i} = 52-46 = 6^\circ C$$

$$\Delta T_{air}\ \text{bulk} = \frac{Tair_i + Tair_o}{2} = \frac{67}{2} = 33.5$$

$$\Delta T_w bulk = \frac{T_{w,i}+T_{w,o}}{2} = \frac{98}{2} = 49.5$$

Table .4.11 Thermal Properties of water and air at bulk temp.

Water	Water	Water	Water		Air	Air	Air	Air
ρ	k	M	Cp		P	k	µ	Cp
988.13	0.6429	0.00054089	4066.1		1.1515	0.0266	0.000018845	1006.6

4.7.1.1 Calculation for Water

$$\text{Mass flow rate of water-180 LPH} = \frac{180 \times 10^{-3} \times \rho}{3600}$$

$$m = 0.04942 \text{ kg/s}$$

$$Q_W = m_W * C_{P,W}(T_{W,O} - T_{W,i})$$

$$Q_W = 1205.35$$

$$V_a = 3 \text{ m/s}$$

$$m_a = \rho * A * V$$

$$m_a = 0.8636$$

$$Q_a = m_a * C_{P,a}(T_{a,O} - T_{a,i})$$

$$Q_a = 869.32$$

$$Q_{avg} = 0.5\,(Q_a + Q_w)$$

$$Q_{avg} = 1037.33$$

Heat capacity of water = mw * Cpw = 200.8918

Heat capacity of air = ma * Cpa = 869.32

$$\varepsilon = \frac{Q_{avg}}{(\dot{m}\ C_p)\min(T_{f,i} - T_{a,i})}$$

$$\varepsilon = 0.2717$$

$$C^* = \frac{(\dot{m}\,C_p)\ min}{(\dot{m}\ C_p)\max}$$

$$\mathbf{C^* = 0.2310}$$

$$\varepsilon = \frac{1}{C^*}[1 - e^{-(1-e^{-NTU})}]$$

$$NTU = -\ln[1+\ln(1-\varepsilon \times C^*)$$

$$NTU = 0.067$$

$$NTU = \frac{UA}{(\dot{m}\ C_p)\min}$$

$$UA = NTU \times (m \times C_p)_{min}$$

$$UA (Pra) = 13.472$$

$$m_w = \rho * A * V_w$$

$$V_w = 0.3766$$

$$Re_w = \frac{\rho \times v \times d}{\mu}$$

$$Re_w = 8946.25$$

$$Pr_w = \frac{\mu \times C_p}{K}$$

$$Pr_w = 3.4209$$

$$Nu_w = 0.023 \times Re^{0.8} \times Pr^{0.3}$$

$$Nu_w = 48.2263$$

$$hi = \frac{Nu * K}{di}$$

$$hi = 2384.97$$

Table.4.12. General Specification for Tube and Fin

Tube	Fin
di = 0.013 m do = 0.0146 m	Di = 0.0146 m Do = 0.0343 m
Tube length = 0.5 m	Thickness = 0.001 m
Thickness = 0.016 m	Space = 0.004 m
Total duct length = 3.27 m	Length = 0.0099 m
dh,wt hydraulic dia. =0.5	

4.7.1.2 Calculation for Air

Mass flow rate of air-3 m/s

$$m_a = \rho \times A \times V$$

$$m_a = 0.8636$$

$$Re_a = \frac{\rho \times v \times L}{\mu}$$

$$Re_a = 91655.61157$$

$$Pr_a = \frac{\mu \times C_p}{K}$$

$$Pr_a = 0.71313$$

$$Nu_a = 10145 \times Ln [Re_a - 46.081] \times Pr.33$$

$$Nu_a = 103.67$$

$$Nu_a = \frac{hL}{K}$$

$$h_a = \frac{Nu_a \times K}{L}$$

$$ho = 5.51527$$

$$m = \sqrt{\frac{2 \times h_o}{k_f \times f_t}}$$

$$m = 7.0017$$

$$\eta = \frac{tanh\,(ml)}{ml}$$

$$\eta = 0.99$$

$$A_o = A_f + A_b$$

$$A_o = Total\ surface\ area\ of\ fin$$

$$A_f = surface\ area\ of\ fin\ tube$$

$$A_b = area\ of\ bare\ tube$$

$$A_f = \frac{\pi}{2} [D^2{}_o - D^2{}_i] + [\pi \times D_o \times \delta]$$

$$A_f = 1.45886$$

$$A_b = [n \times \{ (n-1) \times \pi \times D_o \times s \}] + [n\{ \pi \times D_o \times (0.03 + 0.03)\}]$$

$$A_b = 0.18807$$

Space area =0.99630

$$A_o = A_f + A_b + \text{Space area}$$

Ao =2.64

$$\eta_0 = 1 - \frac{A_f}{A_o}(1-\eta)$$

$$\eta_0 = 0.99$$

$$A_t = \pi * d_{i,f} * l_t * \text{no. of tubes}$$

$$A_t = 0.2293$$

Ai =Π*di*L

$$\frac{1}{UA} = \frac{1}{\eta_0 h_0 A_0} + \frac{\delta}{k_t A_t} + \frac{1}{h_i A_i}$$

UAth = 14.5512

Table.4.13. Comparison between UA_{th} and UA_{pra}.

Sr no.	Voltage	Mass flow of water(kg/sec)	Air velocity m/s	UA_{th}	UA_{pra}	% Error
1	80	180	3	14.55	13.472	7.40
2	80	180	4	14.91	13.732	7.90
3	80	180	5	15.20	14.243	6.30
4	80	180	6	15.43	14.741	4.45

5. VALIDATIONS OF EXPERIMENTAL DATA WITH OTHER CORRELATION.

A typical comparison between the experimental overall heat transfer coefficient and the prediction of different correlations is shown in Fig.5.1. Results show that at the air velocity increase the value of overall heat transfer coefficient increase at constant mass flow of water 0.049396 and at constant supply voltage of 80 volts.. A good agreement exists between the experimental results and theVithayasai et al. [1] correlation. The absolute average error exists from 4.5 % to 7.9 %.

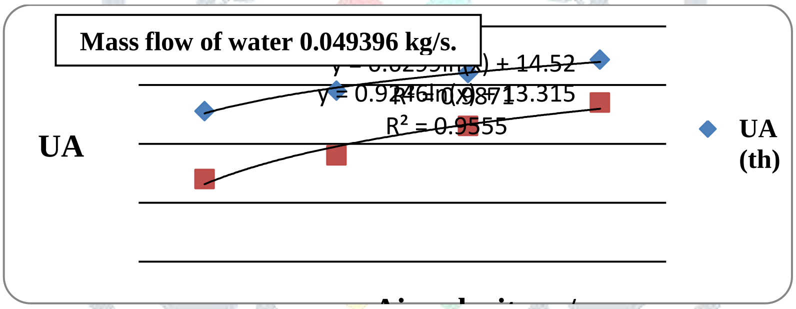

Figure 5.1 Comparison of value of UA_{pra} and UA_{th}

5.1 MAIN EFFECTS FOR EFFECTIVENESS Ɛ

The main effects of each input parameters on Ɛ is shown in fig 5.2. It is clear from the fig. that as mass flow of water increases the value of Ɛ decrease. But as air velocity increases the value of Ɛ goes increase.

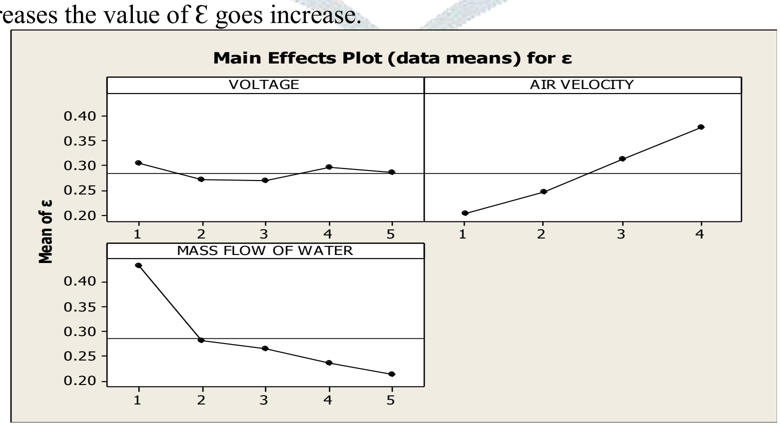

Figure 5.2 Main effects plot for Effectiveness Ɛ

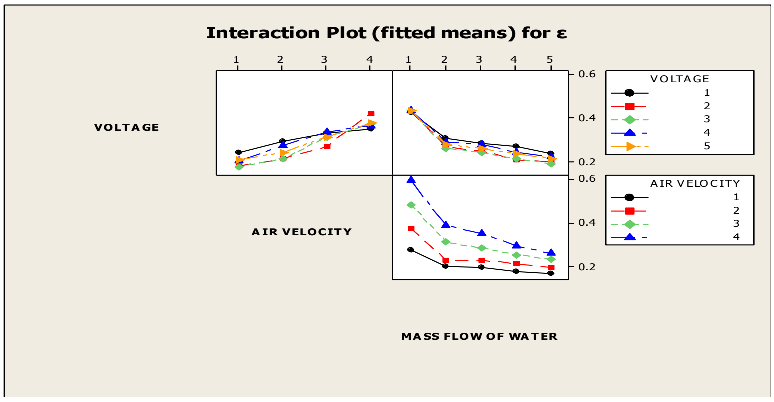

Figure 5.3 Chart for Interaction effect on Effectiveness Ɛ

6. CONCLUSION.

Based on the thermo fluid analysis of a circular fin tube heat exchanger by forced convection presented in the earlier section and with the help of DOE (Design of Experiment) by minitab-14 software, following conclusion can be made.

It is also observed that the value of overall heat transfer coefficient of finned tube heat exchanger was more than that of without fin heat exchanger. The value of average overall heat transfer coefficient of finned tube heat exchanger was 46.90 at 80 voltages and at air velocity 3m/s. While the value of average overall heat transfer coefficient of without fin tube heat exchanger was 32.40 at same condition. And also the highest value of (UA=85.18) for finned tube heat exchanger is higher than that of without fin heat exchanger (UA=56.39).

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