Design and Manufacturing of a
Pre heater assembly for diesel Engine

By

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FINAL YEAR PROJECT REPORT

We hereby recommend that the dissertation prepared under our supervision by: {Muhammad Hamza Alvi (NUST20102042), Muhammad Imad (NUST201200593) and Muhammad Hasin (NUST201200586)} Titled: {Design and Manufacturing of a Pre heater assembly } be accepted in partial fulfillment of the requirements for the award of Bachelors of Engineering in Mechanical Engineering degree with (C+ grade)

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Declaration

I/We certify that this research work titled "*Design and manufacturing of pre heater assembly for a diesel engine"* is my own work. The work has not been presented elsewhere for assessment. The material that has been used from other sources it has been properly acknowledged / referred.

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Copyright Statement

First of all thanks to ALLAH Almighty who choose us to be worthy to study in this prestigious Institute. Dedicated to our parents, friends, and most of all to our batch ME-04 who made this thing possible for us.

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Abstract

Abstract:

The pre-heater we designed for this engine is a shell and tube counter flow heat exchanger. For ten minute preheating, it requires 225 KW of heat energy, so we have to design a heat exchanger for a capacity of more than 225 kW to compensate losses during heat transfer in heat exchanger, pipes, joints and Engine body. It takes input in the form of exhaust gases which are produced at 600 Celsius by combustion of fuel (petrol, diesel) in a burner. These exhaust gases enter in heat exchanger shell at high temperature and high velocity and exchange heat with water entering in the tubes at ambient temperature. The greater the temperature and velocity of input gases, greater will be the heat transfer coefficient and hence greater will be the heat transfer. The water exits tubes at more than 100 Celsius and enters the water galleries of engine. It heat Engine block, run through cylinder head etc, where it heats engine block at 60 Celsius from ambient temperature in ten minutes. So, now the vehicle containing this engine is ready to perform its operation right after ten minutes. Which saves huge amount of time. The Heat exchange requirements are calculated. The heat exchange is designed in accordance with required parameters and size. To reduce volume and increase surface area for better heat exchange, tubes are finned.

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Chapter 1

Introduction

1.1. Background:

Diesel Engines are compression Ignition engines. They do not require a spark plug to initiate ignition. The fuel is mixed and self-ignited in compressed air at high temperature. So, they require air and engine to be at certain temperature already. In cold days, especially, if engine is at a low temperature. Self-ignition does problem. So, the engine require to be preheated to start ignition and smooth running of engine. Ships, Tanks, Trucks and large off road vehicles are run by such large diesel engines that must require some preheating.

Wartsila WL20 engine is a 900 KW diesel engine used to run a ship. It is a big Diesel engine. So, it does problem while starting, like other diesel engines especially if ambient temperature is low, it requires preheating. It is preheated by some other heat sources, such as exhaust from other petrol or diesel engines. There are several engines on a ship. The engine consumes about 2Kw of heat energy to keep it maintain from at 60 degrees Celsius, which is required to initiate auto ignition in this engine. Similarly other preheating systems are also installed in some other similar engines, they preheat engines in a certain amount of time.

The purpose of project was to design a system which preheat engine in short time. We compromise between size and time and developed a pre heater of small size which can be fixed easily inside whole system and it preheats engine within ten minutes. We can calculate required amount of heat to be provided to the engine to heat it at particular temperature in particular amount of time.

1.2. Aims and Objectives

The purpose of project was to design a system which preheat engine in short time. We compromise between size and time and developed a pre heater of small size which can be fixed easily inside whole system and it preheats engine within ten minutes. We can calculate required amount of heat to be provided to the engine to heat it at particular temperature in particular amount of time.

Chapter 2
Literature Review

1. Engine specifications:

2.1. General specifications:

Table 2.1 Engine specifications

Name	W4L20	
Weight	7.2	Tonnes
Dimensions	2510 x 1483 x 2075	Mm
Cylinder bore	200	Mm
Cylinder stroke	280	Mm
Piston displacement	8.8	l/cyc
Number of valves	2 inlet , 2 exhaust	
Cylinder configuration	In line	
Speed	900-1000	Rpm
Mean piston speed	8.4-9.3	m/s
Main engine Power	800	Kw

2.2. Mean effective pressure (P_e) :

Mean effective pressureof Engine can be calculated as:

$P_e = (P \times c \times 1.2 \times 10^9) / (L \times n \times Pi \times D^2)$

Where,

P_e = Mean effective pressure [bar]

P = Output /cylinder [kW]

c = Operating cycle

D = cylinder diameter [mm]

L = Length of stroke [mm]

n = Engine speed [rev. /min]

So, $P_e = (200 \times 4 \times 1.2 \times 10^9) / (280 \times 1000 \times 3.142 \times 200^2)$

P_e = 27.2 bar

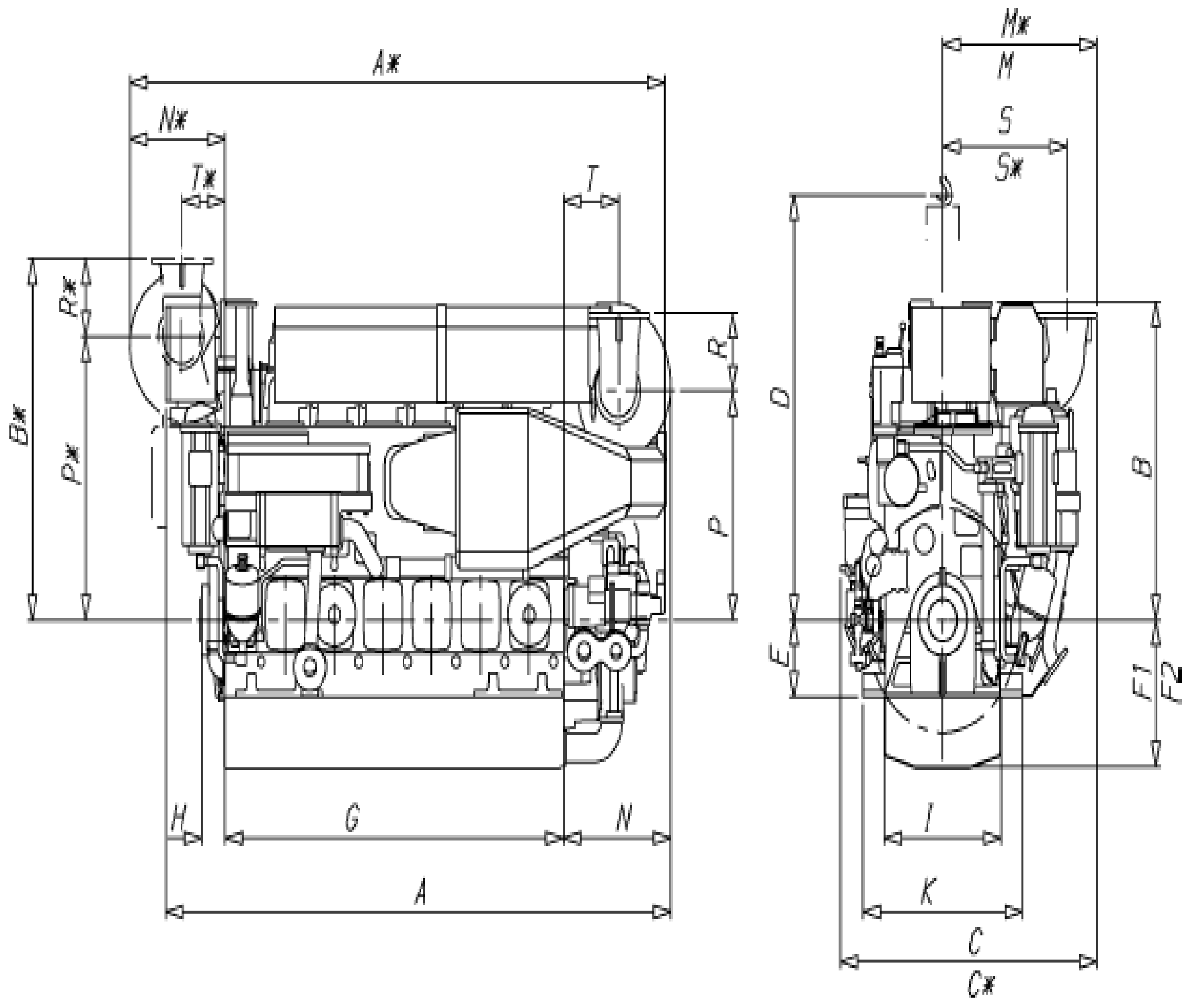


Fig 1.4.1 Main engines (3V92E0068c)

Engine	A*	A	B*	B	C*	C	D	E	F1	F2	G	H	I	K
W 4L20		2510		1348		1483	1800	325	725	725	1480	155	718	980
W 6L20	3292	3108	1528	1348	1580	1579	1800	325	624	824	2080	155	718	980
W 8L20	4011	3783	1614	1465	1756	1713	1800	325	624	824	2680	155	718	980
W 9L20	4299	4076	1614	1449	1756	1713	1800	325	624	824	2980	155	718	980

Fig.2.1 Engine dimensions

2.3. Reference conditions for Engine:

Table 2.2

Total barometric pressure	100	kpa
Air temperature	25	C’
Relative humidity	30	%
Charge air coolant temperature	25	C`

Now we come upto the required auxiliary of engine i.e the preheater. The preheater is designed specifically for this engine. The practical approaches, such as material selection, fitting, size constraints are kept in mind.

Chapter 3
Methodology

3. Burner:

Source of heating for preheating water:

3.1. Electric heating:

There are several methods for heating water by some electrical means.
For a 225 KW heater. We need to give very high voltage and current
Using Joule's Law.
P = V I = 220,000 W
This requires very high voltage of current. So, much power is not feasible to be given inside a vehicle through batteries. So, electrical heating concept is not feasible.

3.2. combustion heating:

The Best option for heating water is through a burner. This burner takes fresh air, fuel input (diesel, petrol), uses a spark and produces gases at 600 Celsius. A fan is installed which blows these exhaust gases with high speed. These gases then enter heat exchanger through input port.

4. Material Selection:

The heat exchanger have to work at high temperature, high gas momentum and even bit high pressure. At these conditions, it has to transfer heat at large magnitude in such short time. So material should be carefully selected. The material selection should have following characteristics:

- It should have very high **thermal conductivity**.
- High **melting point**.
- **Corrosion resisting**, because of high temperature water and gas.
- **Light weight** because of a lot of material is clamped or welded at just two ends.
- A lot of material is consumed, so cost is an important factor. So, Material should be **cheap.**

Table 4.1 Material Selection

#	Material	Melting Point	Thermal Conductivity	Density	Corrosion Resistance	Price
		C`	W/m.K	Kg/m^3		PKR/kg
1	Aluminum	660	237	2.71	Yes	600
2	Copper	1085	410	8.94	Yes	1500
3	Stainless Steel	1450	50	7.85	Yes	350
4	Mild Steel	1450	50	7.85	No	100

On these basis, we reject aluminum although it is very good conductor of heat and it it light weight. But it is highly vulnerable to melting due to its low melting point and high temperature operating conditions prevailing in heat exchanger. Copper being expensive will make heat exchanger cost very high. Because a lot of material is involved in this Heat exchanger.

The best option for heat exchanger is therefore stainless steel. Most of the heat exchanger made for industrial purposes are made of stainless steel, due to its moderate heat conduction, high strength, corrosion resistance and moderate cost.

5. Heat Exchanger calculation:

5.1. Preheating requirement:

There is a formula for calculation of preheating requirement for every engine. This formula depends upon the power of engine, amount of lube oil in engine, High temperature water in engine, the water galleries of engine and most importantly the size of engine. So according to this empirical formula for this engine, the preheating requirement for different temperature ranges and different preheating timings will be as follows.

Formula for calculating required preheating power

$$P = \frac{(T_1 - T_0)(m_{eng} \times 0.14 + V_{LO} \times 0.48 + V_{FW} \times 1.16)}{t} + k_{eng} \times n_{cyl}$$

where:

- P = Preheater output [kW]
- T_1 = Preheating temperature = 60...70 °C
- T_0 = Ambient temperature [°C]
- m_{eng} = Engine weight [ton]
- V_{LO} = Lubricating oil volume [m³] (wet sump engines only)
- V_{FW} = HT water volume [m³]
- t = Preheating time [h]
- k_{eng} = Engine specific coefficient = 0.5 kW
- n_{cyl} = Number of cylinders

Power requirement:

$$P = \frac{(60 - 30) \times (7.2 \times 0.14 + 0.27 \times 0.48 + 0.09 \times 1.16)}{10/60} + 0.5 \times 4$$

P = 225 KW

This table shows the preheating power required to preheat engine in a certain time.

Initial temperature is 30 Celsius, final temperature is 60 Celsius.

Table 5.1 Preheating requirement

	Time	Power
	Minutes	KW
1	5	450
2	10	225
3	20	114
4	30	76
5	45	52
6	60	40

We have designed our Preheater for ten minutes preheating time.

5.2. Calculations:

There are several methods for calculation of heat exchanger. The NTU (Number of Transfer Units) method and LMTD (Log Mean Temperature Difference) methods are two most common. We will choose LMTD method to design our heat exchanger.

Using LMTD method.

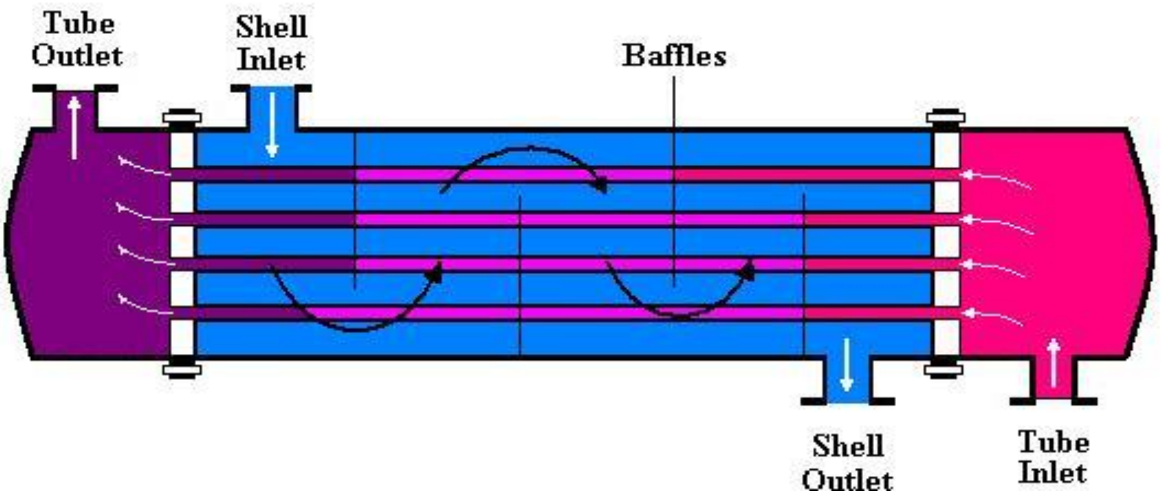


Fig. 5.1 Shell & Tube heat exchanger

Gas Input and output:

Table 5.2 Exhaust Gas from a burner.

Gas In	T_i	600	C
Gas Out	T_e	200	C
Gas avg. temperature	T_{avg}	(600+200)/2 = 400	C
Avg. density @ 400	p_{avg}	0.5	Kg/m^3
Specific heat	$C_{p\ avg}$	1000	J/Kg
Kinematic Viscosity	n	0.00006	m^2/s
Thermal conductivity	k	0.045	W/m.K
Prandtl Number @ 400	P	0.68	
Prandtl Number @100	P_s	0.697	

Fan characteristics:

Air fan is used to blow air into shell.

Fan blade radius = r =12 cm = 0.12 m

Rotational speed = w =2000 rpm = 2000 x 2 x Pi / 3600 = 209.5 rad/s

Fan Tip speed =v = r x w =0.12 x 209.5 =**25 m/s**

Inlet diameter = d =25 cm =0.25 m

Inlet Area = A = Pi x d^2 / 4 = 0.0491 m^2

Mass flow rate = m = p x A x v = 0.5 x 0.049 x 25 = **0.617 kg/s**

Water In and Out:

Table 5.3

Water In	T_i	30	C
Water out	T_e	100	C
Water avg. temperature	T_{avg}	65	C
Avg. density @65	p_{avg}	4200	Kg/m^3
Specific Heat @ 65	$C_{p\ avg}$	1000	J/Kg
Kinematic Viscosity @65	n	4.3 e -7	m^2/s

Water Pump:

Pump flow rate = 3 m^3/hr = **0.8333 kg/s**

Now starting Calculations using LMTD:

<u>Total Capacity of heat transfer for air and water.</u>

$Q_{air} = m \times C_p \times dT$

$= 0.617 \times 1000 \times (600 – 200)$

$= 246{,}800$ W = **246 KW**

$Q_{water} = m \times C_p \times dT$

$= 0.833 \times 4200 \times (100 – 30)$

$= 245{,}000$ W = **245 KW**

<u>Log mean Temperatures for counter flow:</u>

$dt_1 = T_{air\ in} - T_{water\ out}$

$= 600 - 100 = 500$ C

$dt_2 = T_{air\ out} - T_{water\ in}$

$= 200 – 30 = 170$ C

$dt_{log\ mean} = (dt_1 – dt_2) / \ln (dt_1 / dt_2)$

$= (500 – 170) / \ln (500 / 170)$

= **307` C**

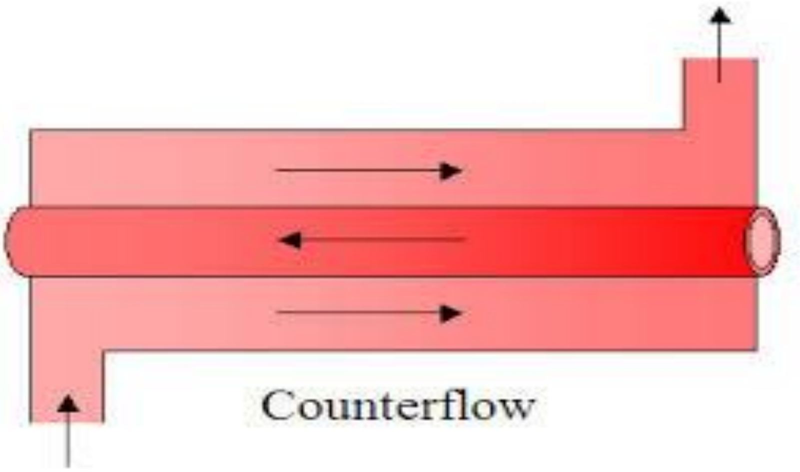


Fig. 5.2

Tube Design & dimensions:

Table 5.4 (All distances are in meters)

# of Tubes		25
Internal diameter	d_{in}	0.019
Thickness	t	0.0012
Outer diameter	D	0.0214
Parallel distance	S_L	0.08
Perpendicular distance	S_T	0.08

Using In Line configuration

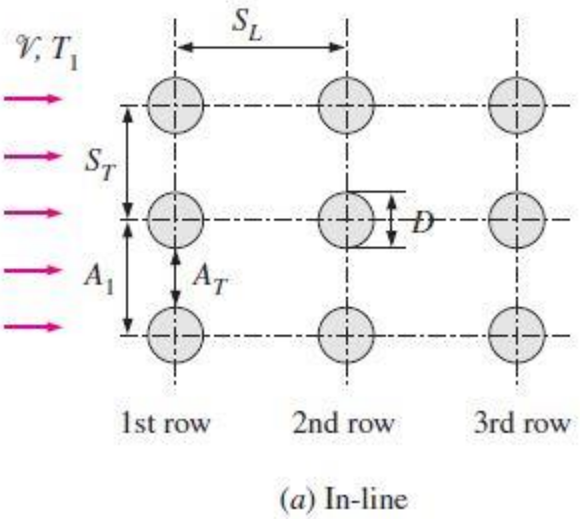
V_{max} = $V \times S_T / (S_T - D)$

25 x 0.08 / (0.08 – 0.021) = **33 m/s**

Fig. 5.3

Inline tube

configuration



Reynolds Number:

Re = V_{max} x D / n = 33 x 0.021 / 0.00006 = **12,239**

Nusselt Number:

Table 5.5 Nusselt number correlation

Nusselt number correlations for cross flow over tube banks for N > 16 and 0.7 < Pr < 500 (from Zukauskas, Ref. 15, 1987)*

Arrangement	Range of Re_D	Correlation
In-line	0–100	$Nu_D = 0.9\ Re_D^{0.4} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	100–1000	$Nu_D = 0.52\ Re_D^{0.5} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$1000–2 \times 10^5$	$Nu_D = 0.27\ Re_D^{0.63} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$2 \times 10^5–2 \times 10^6$	$Nu_D = 0.033\ Re_D^{0.8} Pr^{0.4} (Pr/Pr_s)^{0.25}$
Staggered	0–500	$Nu_D = 1.04\ Re_D^{0.4} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	500–1000	$Nu_D = 0.71\ Re_D^{0.5} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$1000–2 \times 10^5$	$Nu_D = 0.35 (S_T/S_L)^{0.2}\ Re_D^{0.6} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$2 \times 10^5–2 \times 10^6$	$Nu_D = 0.031 (S_T/S_L)^{0.2}\ Re_D^{0.8} Pr^{0.36} (Pr/Pr_s)^{0.25}$

*All properties except Pr_s are to be evaluated at the arithmetic mean of the inlet and outlet temperatures of the fluid (Pr_s is to be evaluated at T_s).

Table 5.6 Correction Factors

Correction factor F to be used in $Nu_{D,N_L} = F Nu_D$ for $N_L < 16$ and $Re_D > 1000$ (from Zukauskas, Ref 15, 1987).

N_L	1	2	3	4	5	7	10	13
In-line	0.70	0.80	0.86	0.90	0.93	0.96	0.98	0.99
Staggered	0.64	0.76	0.84	0.89	0.93	0.96	0.98	0.99

So we use formula #3 of in-line arrangement

Nu_D = 0.93 x 0.27 x $12239^{0.63}$ x $0.68^{0.4}$ x $(0.68 / 0.697)^{0.25}$

= **87**

<u>Heat Transfer co-efficient</u> can be evaluated as

$h = Nu \times D / k$ k = thermal conductivity of gas to that meduim

$h = 87 \times 0.021 / 0.044$

h = **184.7 W / m^2 . C`**

Now we will calculate **<u>area</u>** required for heat transfer:

$Q = h \times A \times dt_{lm}$

$A = Q / (h \times dt_{lm})$

$= 240{,}000 / (184.7 \times 307) = 4.3\ m^2$

So total 4.3 m^2 area is required for carrying out this amount of heat transfer.

Area required per tube = 4.3 / 25 = 0.172 $m^{2.}$

So , evaluating this we calculate each tube's length:

$L = A / 3.142 \times D = 2.6$ m

A 2.6 meter tube will make heat exchanger very long and will induce Pressure drop.

So, we set our heat exchanger length to be 0.7 meters. And then calculate remaining area.

Area of each tube = Pi x D x L

$= 3.142 \times 0.0213 \times 0.7$

$A = 0.0462\ m^2$

So Area deficit = 0.172 – 0.0462 = 0.1258 m^2

To achieve required heat transfer with more surface area and less

5.3. Fin attachment

There are two ways to increase the rate of heat transfer

- To increase the convection heat transfer coefficient h
- To increase the surface area A_s

Increasing h may require the installation of a pump or fan or replacing the existing one with a larger one .The alternative is to increase the surface area by attaching to the surface extended surfaces called fins made of highly conductive materials

This extra area per tube is induced by Using Fins. The feasibility of fins is calculated as:

Long rectangular fins are selected, their dimensions are as follows:

Table 5.7 Fin dimensions

Length	0.64 m
Height	1 inch = 0.0254 m
Width	13 SWG = 0.0025 m
Surface Area	= 2 x0.64 x 0.0254 = **0.0326**
Number	= 0.1258 / 0.0326 = 3.5

5.4. Fins Efficiency:

Fins must be checked for their efficiency whether they are helping heat transfer. If fins are too crowded or not properly designed they can even hinder heat transfer.

{ k = fin's material conductivity}

{ k = 50 for S.S }

$L_c = L + t/2$ = 0.0264

$m = (2h / kt)^{-1/2}$

$= (2 \times 188 / 50 \times 0.002)^{-1/2}$

= 61.3

Fin efficiency = $\tanh(mL_c) / mL_c$

= **57.5 %**

So, Fins are 57.5 % efficient, it means we almost require double the number of fins for same heat transfer. So, **7** fins per tubes are fixed.

Fin effectiveness:

E = heat transfer with fin / heat transfer without fin

Area without fin = 0.0462 x 25 = 1.155 m^2

Area of fins = 0.0326 x 7 x 25 = 5.7 m^2

Heat transfer = $h.A.dt_{lm}$

From unfinned surface = 188 x 1.155 x 308 = 66 kW

From fins = 0.57 x 188 x 5.7 x 308 = 188 kW

Total heat transfer = 245 kW

From fins = 245 – 66 kW = 180 kW

Fin effectiveness = $\grave{Q}_{fin} / \grave{Q}_{no\ fin}$ = 180 / 66 = **2.8**

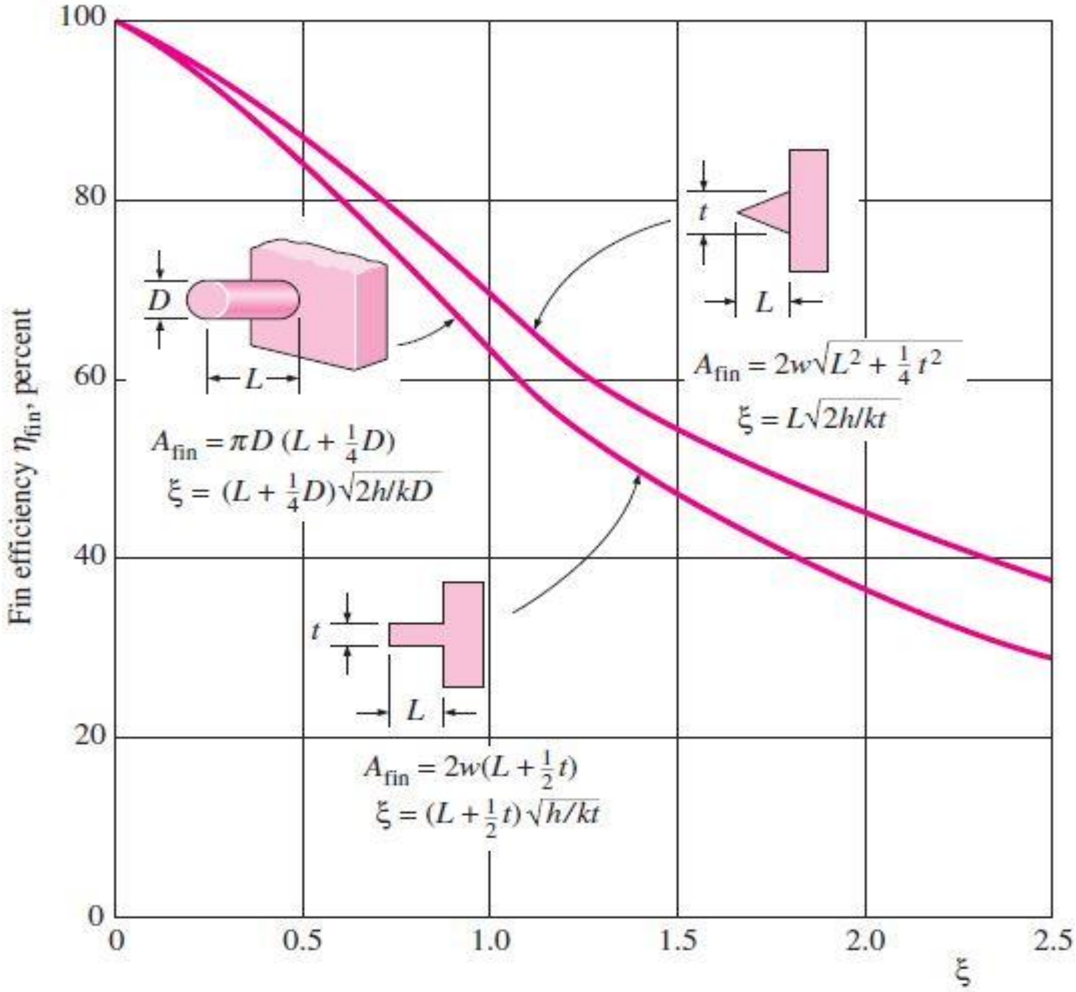


Fig. 5.4 Fin efficiency

6. Drawings

All dimensions are in mm

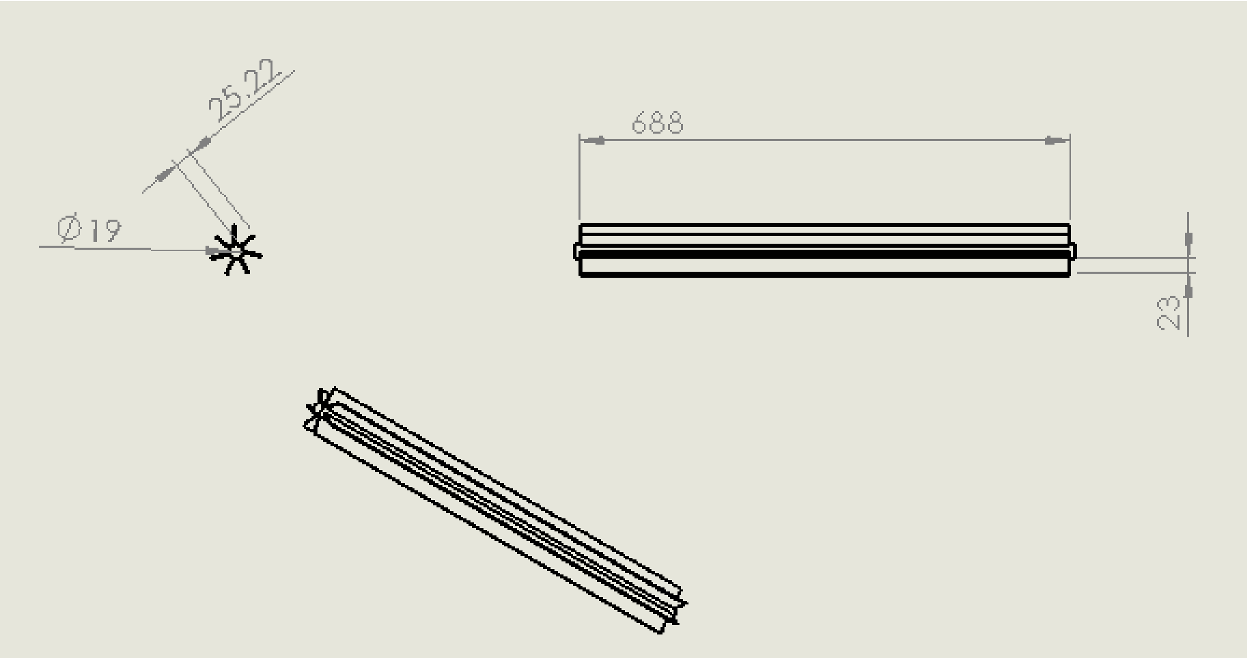


Fig 6.1 fins drawings

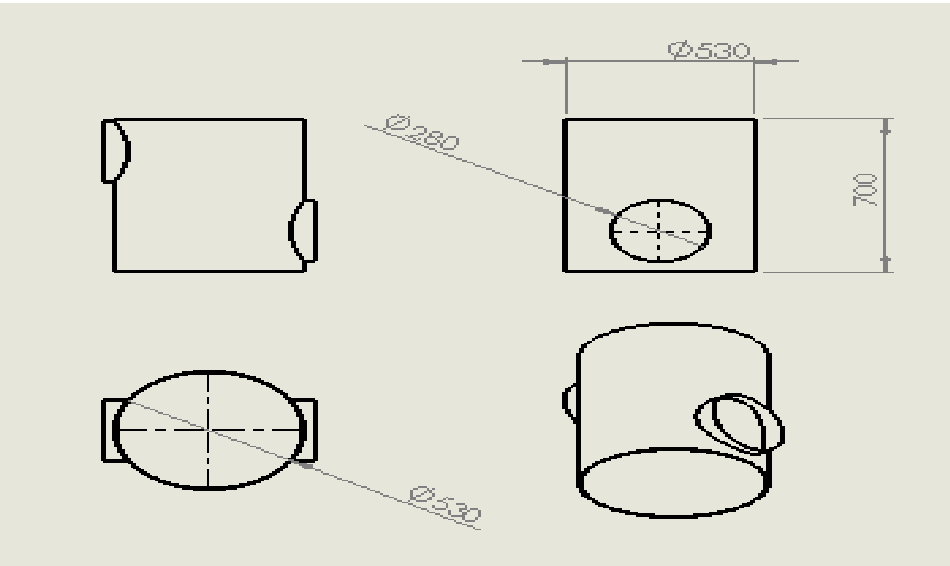


Fig. 6.2 shell drawings

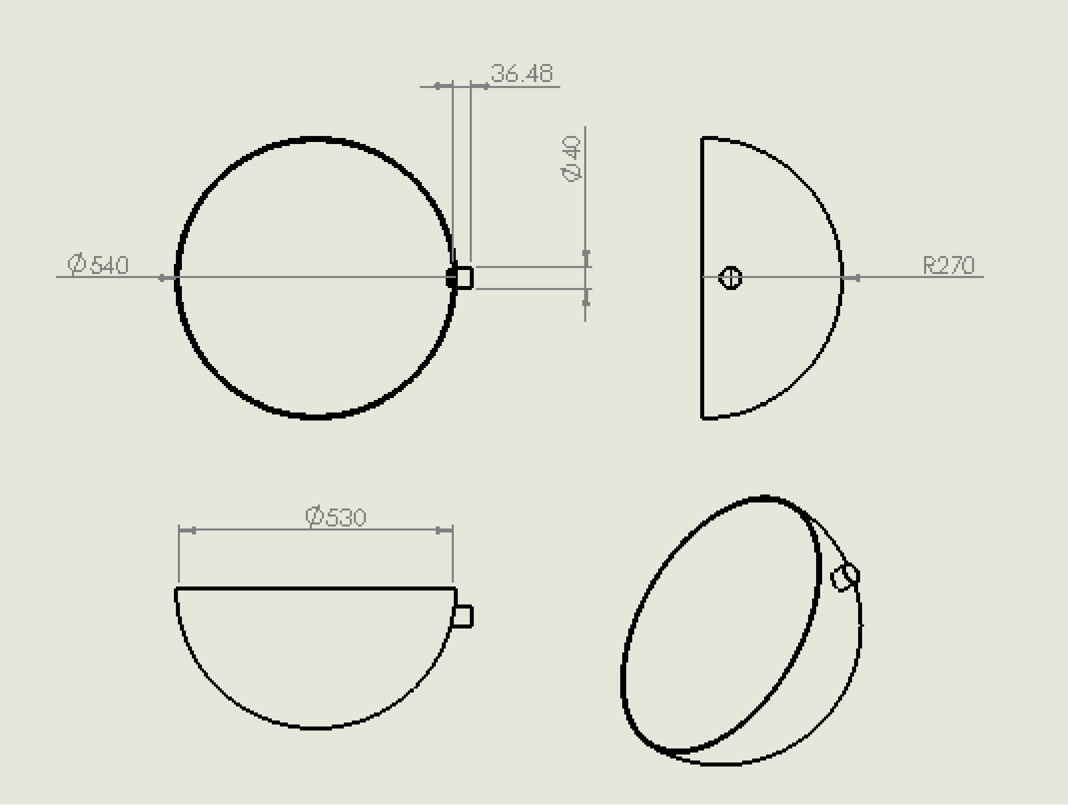


Fig 6.3 Hemispherical lids drawings

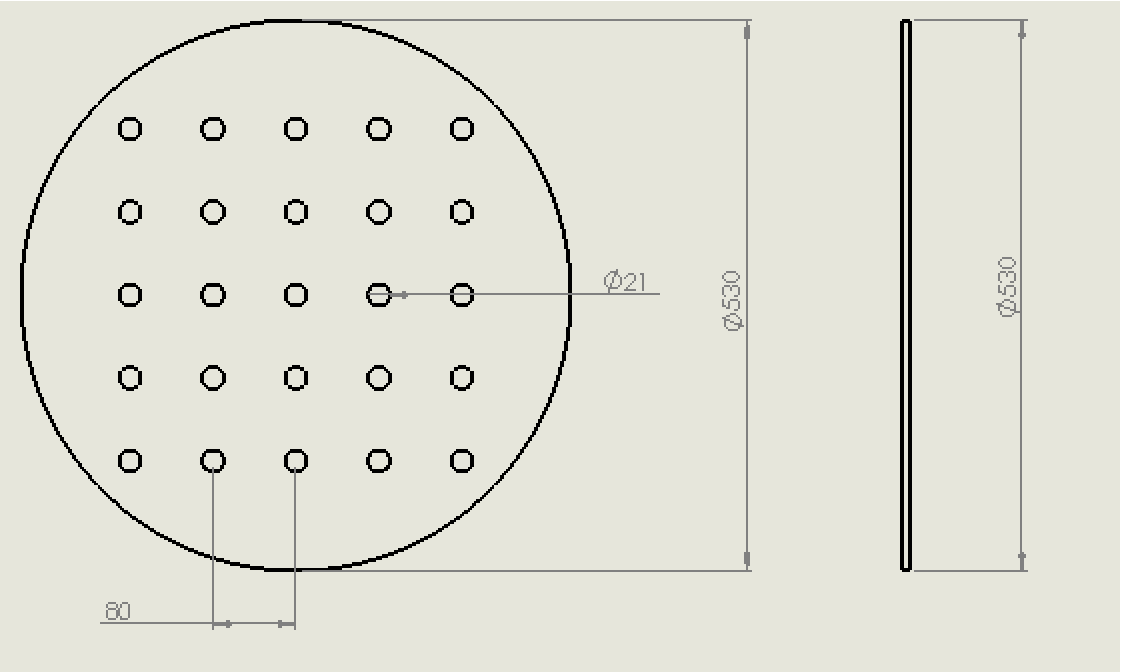


Fig 6.4 Plates drawings

7. CAD Models:

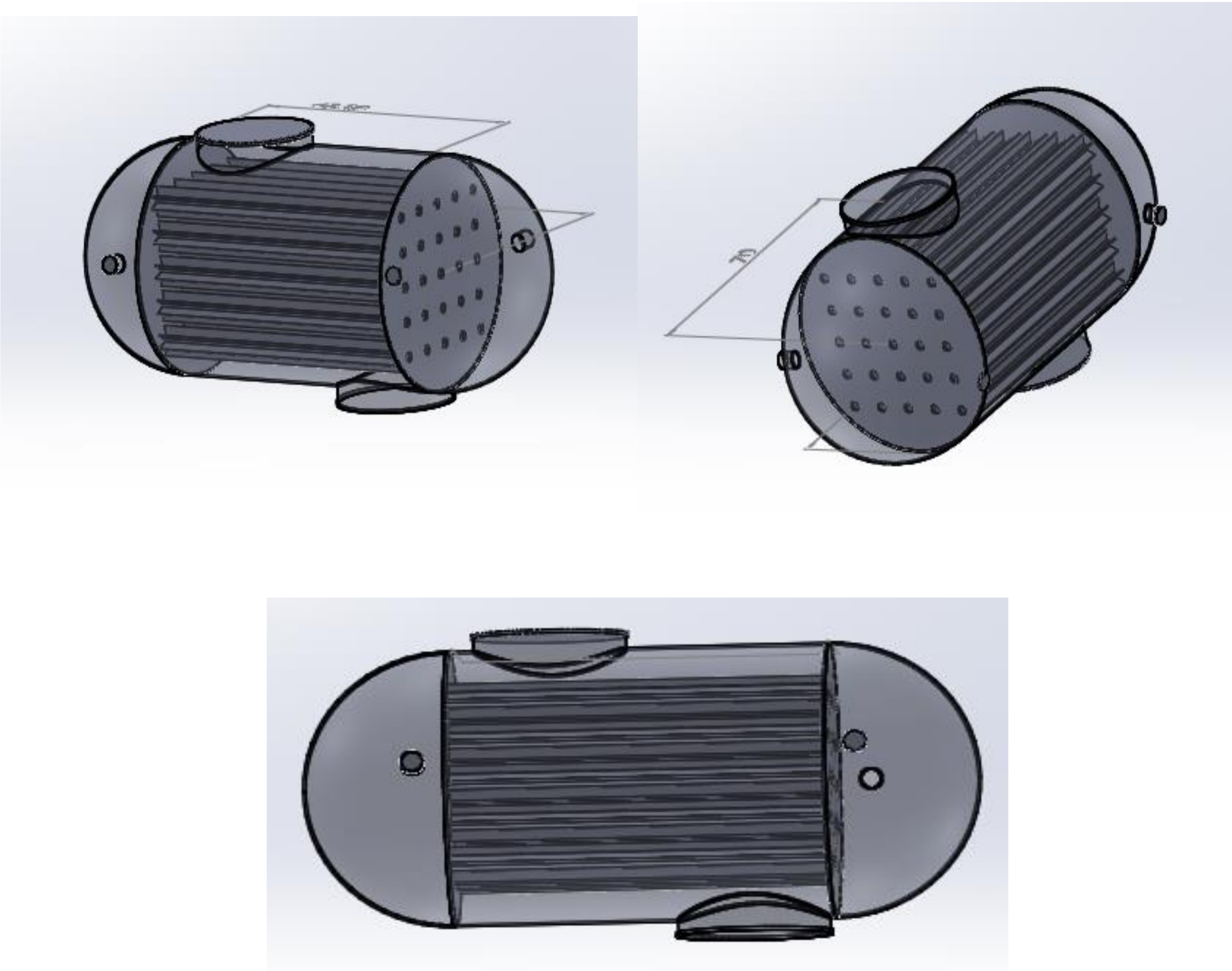


Fig. 7.1 Shell views

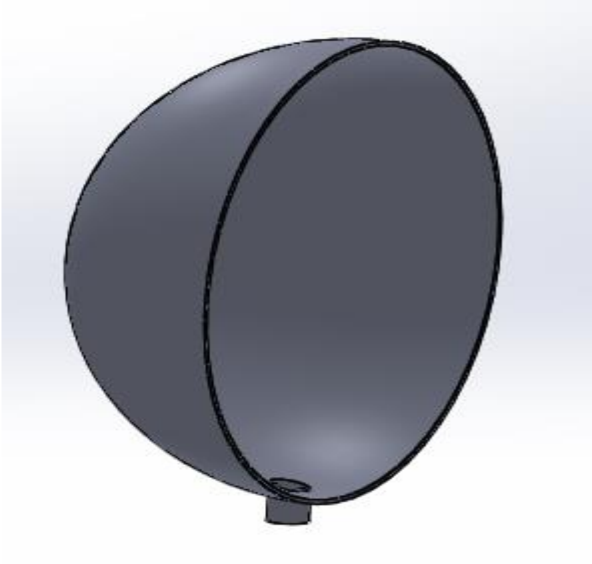


Fig. 7.2

Lid m

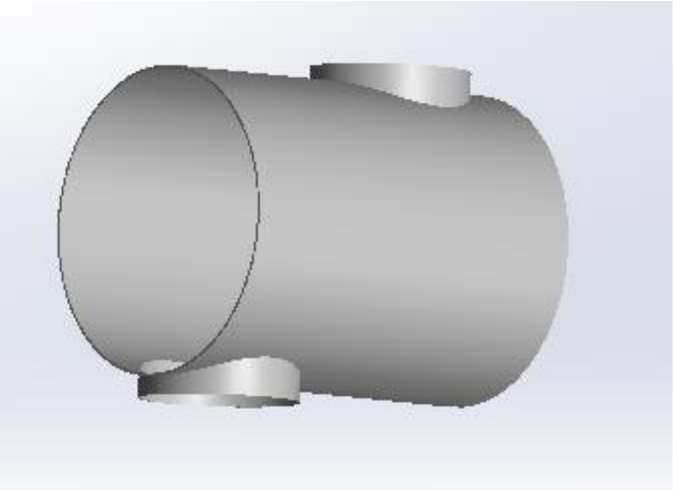


Fig. 7.3 shell model

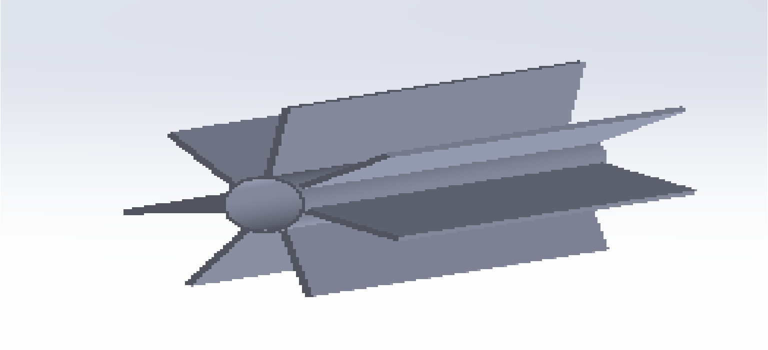


Fig 7.4 fins model

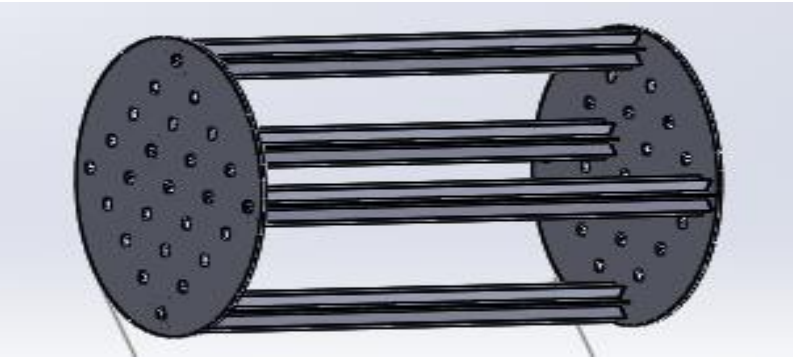


Fig. 7.5 Fin-plate sub assembly

8. Simulations:

Simulation were run using Solid Works.

Simulation #1 was run on original model according to design condition.

The flow was assumed fully developed. Because fully developed flow performs better heat exchange. The external surface of shell is assumed to be insulated. In practical situations this can be done by

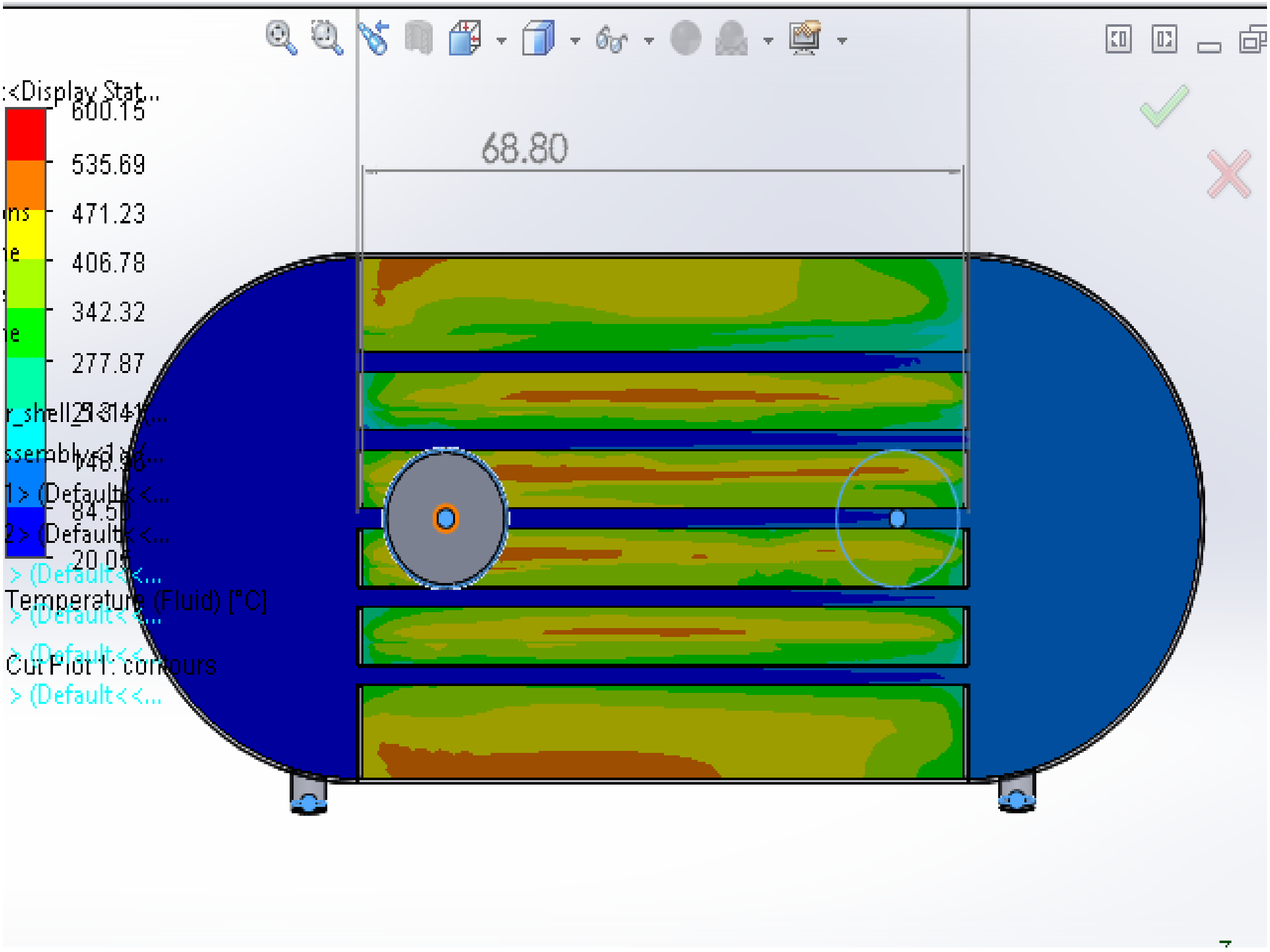


Fig. 8.1 temperature simulation

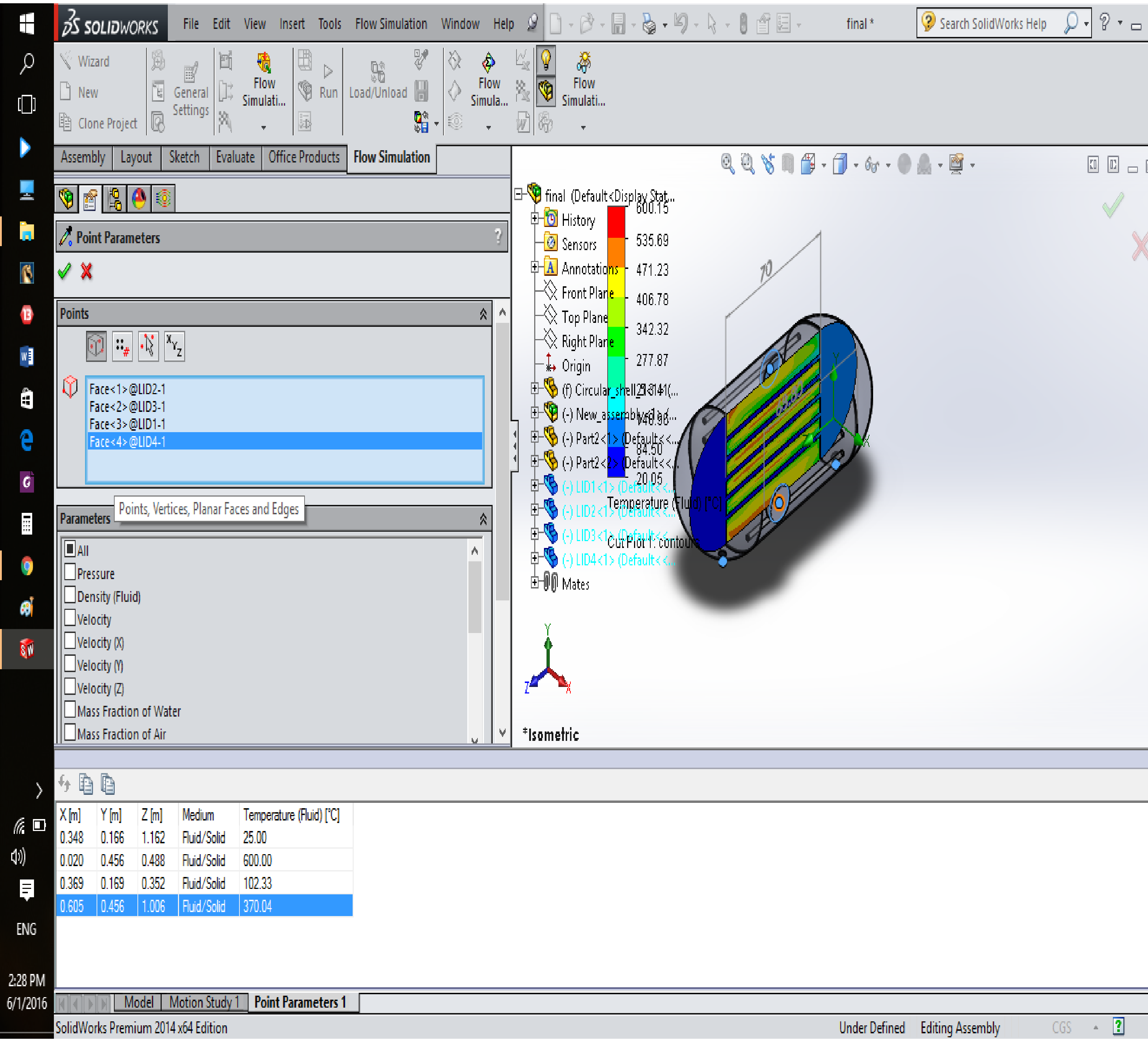


Fig. 8.2 Temperature simulation results

The results obtained were according to the design. The water is exiting at 102 `C. which was ~100 `C in design. The difference is shown because, safety factors were involved in design calculations.

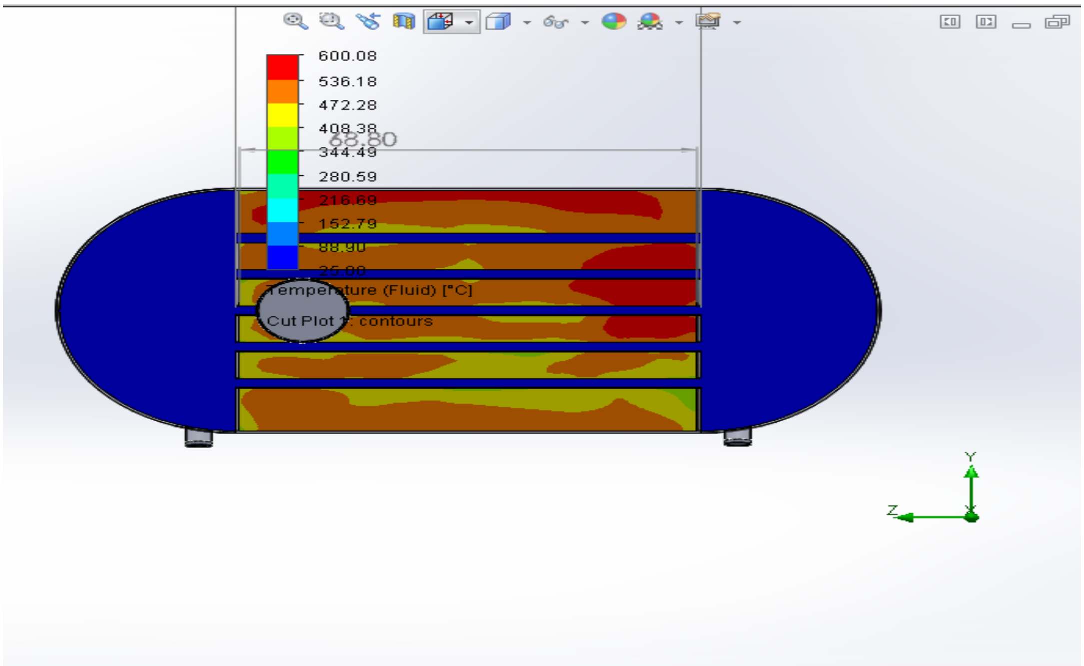
X [m]	Y [m]	Z [m]	Medium	Temperature (Fluid) [°C]
0.348	0.166	1.162	Fluid/Solid	25.00
0.020	0.456	0.488	Fluid/Solid	600.00
0.369	0.169	0.352	Fluid/Solid	102.33
0.605	0.456	1.006	Fluid/Solid	370.04

Table 8.1TemperatureResults

Simulations of Model without fins were also run. In order to check

- Heat transfer enhancement by fins
- Efficiency of fins
- Effectiveness of fins
- Flow restriction by fins

The simulations were run and show that Heat transfer with fins was according to design. So, fins were not choking the flow. And the required criterion is not met without fins. So, fins play a great role in heat transfer.



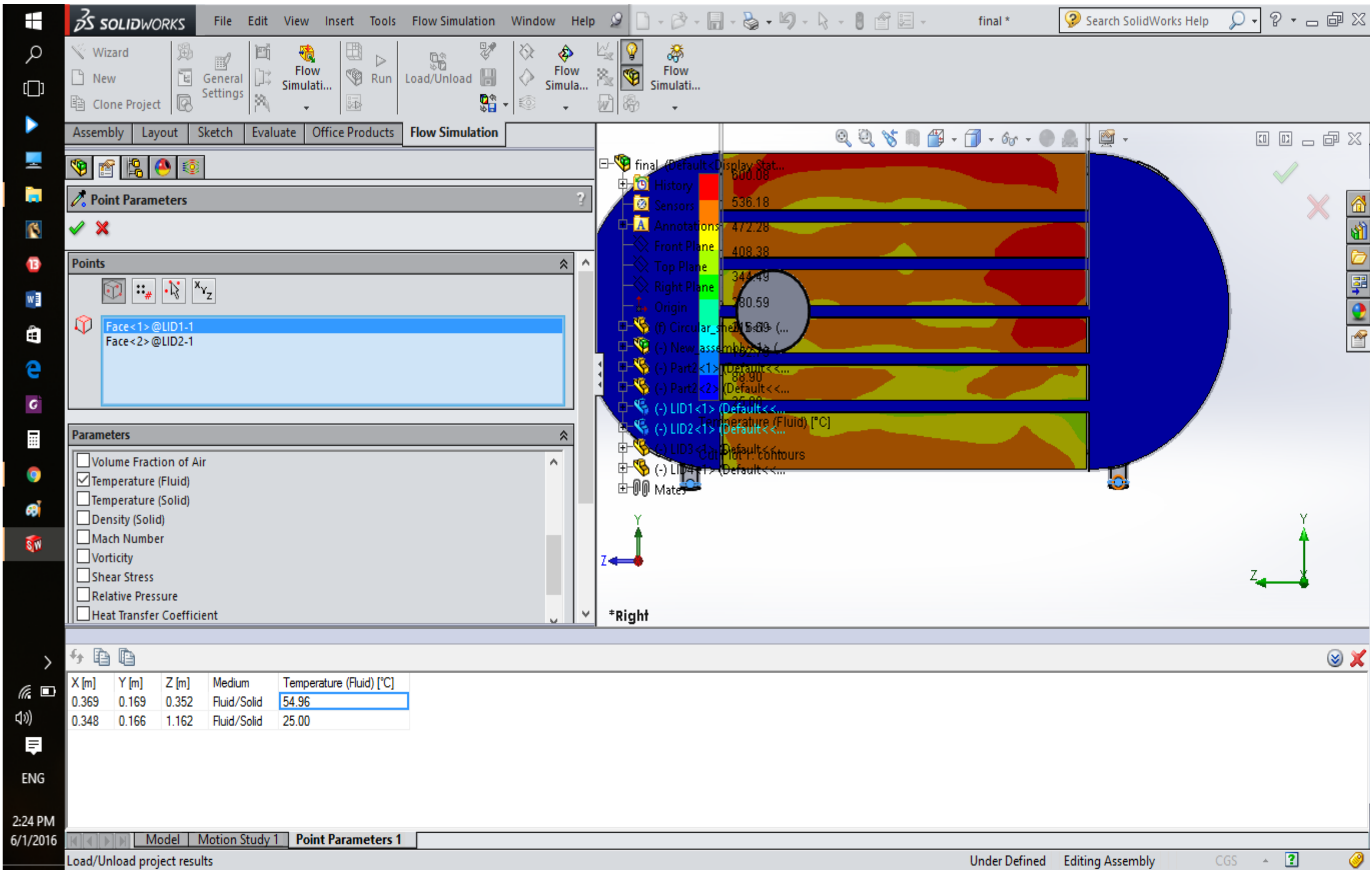
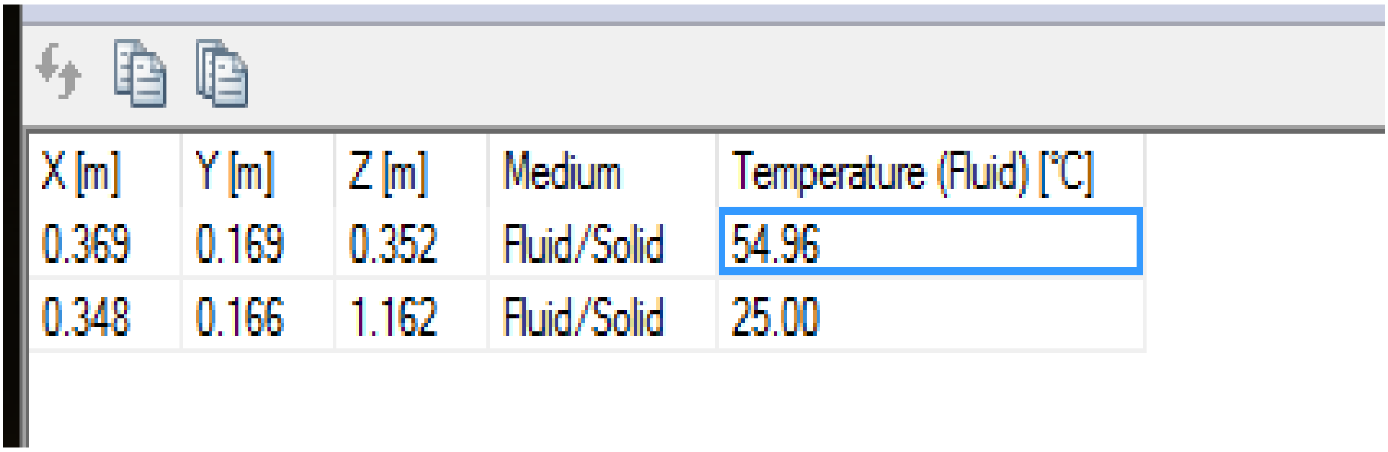


Fig. 8.3Prototype simulation



X [m]	Y [m]	Z [m]	Medium	Temperature (Fluid) [°C]
0.369	0.169	0.352	Fluid/Solid	54.96
0.348	0.166	1.162	Fluid/Solid	25.00

Table 8.2 Prototype results

The results show the exit water temperature to be 55`C which is almost half of the temperature using fin. So, it proves fin effectiveness to be around 2 times.

9. Small scale prototype:

Small scale prototype for manufacturing was designed.

Total Capacity of heat transfer for air and water.

$Q_{air} = m \times C_p \times dT$

$= 0.054 \times 1000 \times (75 - 56)$

$= 1050$ W = **1.05 KW**

$Q_{water} = m \times C_p \times dT$

$= 0.245 \times 4200 \times (26 – 25)$

$= 1030$ W = **1.03 kW**

Gas Input and output:

Table 9.1 Hot air from drier.

Gas In	T_i	75	C
Gas Out	T_e	56	C
Gas avg. temperature	T_{avg}	(75+56)/2 = 67	C
Avg. density @ 67	p_{avg}	0.7	Kg/m^3
Specific heat	$C_{p\ avg}$	1000	J/Kg
Kinematic Viscosity	n	0.000046	m^2/s
Thermal conductivity	k	0.045	W/m.K
Prandtl Number @ 67	P	0.682	
Prandtl Number @30	P_s	0.69	

Water In and Out:

Table 9.2 Prototype water table

Water In	T_i	25	C
Water out	T_e	27	C
Water avg. temperature	T_{avg}	25.5	C
Avg. density @25	p_{avg}	1000	Kg/m^3
Specific Heat @ 25	$C_{p\ avg}$	1004	J/Kg
Kinematic Viscosity @65	n	4.5 e -7	m^2/s

Inputs and outputs for model:

For water supply, a submersible pump is used.

Table 9.3 Submersible pump data

Volumetric Flow rate	1000 liters/hour
Head	1-1.8 meters
Power	
Voltage	220 Volts

For supply of hot air:
A hair drier consisting of filaments and blower fan.
Table 9.4 Drier data

flow rate	0.046 kg/s
Velocity	5.5 m/s
temperature	75 celcius
Area	12.6 in^2
Power	1900 Watts
Voltage	220 V

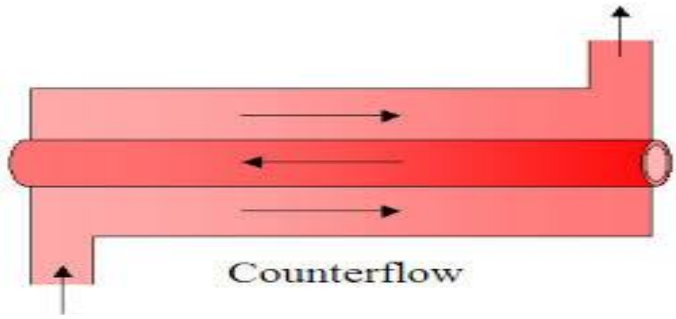
Log mean Temperatures for counter flow:

$dt_1 = T_{air\ in} - T_{water\ out} = 75 - 27 = 52$ C

$dt_2 = T_{air\ out} - T_{water\ in} = 60 - 25 = 35$ C

$dt_{log\ mean} = (dt_1 - dt_2) / \ln(dt_1 / dt_2) = (52 - 35) / \ln(52 / 35)$

= 42` C



Tube Design & dimensions:

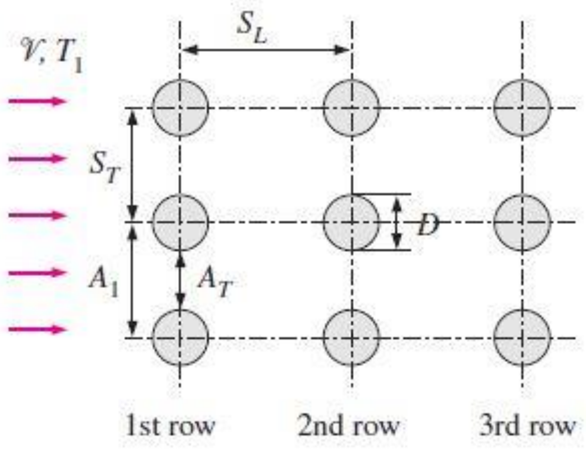
Table 9.5 Prototype dimensions All distances are in meters.

# of Tubes		9
Internal diameter	d_{in}	0.019
Thickness	t	0.0012
Outer diameter	D	0.0214
Parallel distance	S_L	0.08
Perpendicular distance	S_T	0.08

Using In Line configuration

V_{max} = $V \times S_T / (S_T - D)$

$5 \times 0.08 / (0.08 - 0.021)$ = **7 m/s**



(a) In-line

Nusselt number correlations for cross flow over tube banks for N > 16 and 0.7 < Pr < 500 (from Zukauskas, Ref. 15, 1987)*

Arrangement	Range of Re_D	Correlation
In-line	0–100	$Nu_D = 0.9\, Re_D^{0.4} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	100–1000	$Nu_D = 0.52\, Re_D^{0.5} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$1000–2 \times 10^5$	$Nu_D = 0.27\, Re_D^{0.63} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$2 \times 10^5–2 \times 10^6$	$Nu_D = 0.033\, Re_D^{0.8} Pr^{0.4} (Pr/Pr_s)^{0.25}$
Staggered	0–500	$Nu_D = 1.04\, Re_D^{0.4} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	500–1000	$Nu_D = 0.71\, Re_D^{0.5} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$1000–2 \times 10^5$	$Nu_D = 0.35 (S_T/S_L)^{0.2} Re_D^{0.6} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$2 \times 10^5–2 \times 10^6$	$Nu_D = 0.031 (S_T/S_L)^{0.2} Re_D^{0.8} Pr^{0.36} (Pr/Pr_s)^{0.25}$

*All properties except Pr_s are to be evaluated at the arithmetic mean of the inlet and outlet temperatures of the fluid (Pr_s is to be evaluated at T_s).

Correction factor F to be used in $Nu_{D,N_L} = F Nu_D$ for $N_L < 16$ and $Re_D > 1000$ (from Zukauskas, Ref 15, 1987).

N_L	1	2	3	4	5	7	10	13
In-line	0.70	0.80	0.86	0.90	0.93	0.96	0.98	0.99
Staggered	0.64	0.76	0.84	0.89	0.93	0.96	0.98	0.99

Reynolds Number:

Re = V_{max} x D / n = 7 x 0.021 / 0.000046 = **2800**

Nusselt Number:

Nu_D = 0.86 x 0.27 x $12239^{0.63}$ x $0.68^{0.4}$ x $(0.68/0.697)^{0.25}$

= **30**

Heat Transfer co-efficient can be evaluated as

k = thermal conductivity of gas to that medium

h = Nu x D / k

h = 30 x 0.021 / 0.0385

h = **66 W / m^2 . C`**

Now we will calculate area required for heat transfer:

$Q = h \times A \times dt_{lm}$

$A = Q / (h \times dt_{lm})$

$= 1030 / (66 \times 42)$ $= \mathbf{0.3768\ m^2}$

So total 0.3768 m^2 area is required for carrying out this amount of heat transfer.

Area required per tube = 0.3768 / 9 = 0.042 $m^{2.}$

So, evaluating this we calculate each tube's length:

L = A / 3.142 x D = 0.65 m

So, we set our heat exchanger length to be 0.48 meters. And then calculate remaining area.

Area of each tube = Pi x D x L

= 3.142 x 0.0213 x 0.48

A = 0.029 m^2

So Area deficit = 0.042 – 0.029 = 0.013 m^2

This extra area per tube is induced by Using Fins. The feasibility of fins is calculated as:

Long rectangular fins are selected, their dimensions are as follows:

Table 9.6 Prototype Fins

Length	0.4 m
Height	1 inch = 0.0254 m
Width	13 SWG = 0.0025 m
Surface Area	= 2 x0.4 x 0.0254 = **0.02**
Number	= 0.013 / 0.02 = 1

Fins Efficiency:

Fins must be checked for their efficiency whether they are helping heat transfer. If fins are too crowded or not properly designed they can even hinder heat transfer.

{k = fin's material conductivity}

{k = 50 for S.S}

$L_c = L + t/2$ = 0.0264 m

$m = (2h / kt)^{-1/2}$

$= (2 \times 70 / 50 \times 0.002)^{-1/2}$

= 52

Fin efficiency = Tanh $(mL_c) / mL_c$

= 60 %

So, Fins are 60 % efficient, it means we almost require double the number of fins for same heat transfer. So, **2** fins per tubes are fixed.

Fin effectiveness:

E = heat transfer with fin / heat transfer without fin

Area without fin = 0.029 x 25 = 0.26 m^2

Area of fins = 0.02 x 4 x 9 = 0.72 m^2

Heat transfer = $h.A.dt_{lm}$

From unfinned surface = 70 x 0.26 x 42 = 0.7 kW

From fins = 0.6 x 70 x 0.72 x 42 = 1.2 kW

Fin effectiveness = $\grave{Q}_{fin} / \grave{Q}_{no\ fin}$ = 2

10. Manufacturing of model:

The small scale prototype of heat exchanger is manufactured.

The material use for manufacturing is Mild steel, due to its low cost. The model has to run few times just for operation and its testing and after all it will be placed just as a display model. So, rusting is not an issue for that model. So, it is not a wise decision to build model from expensive stainless steel. Mild steel will keep model's price in limits.

Metal purchased:

Table 10.1 Mild steel sheet data

1	Length	4 ft (~1.2 m)
2	Width	4 ft
3	Thickness	13 SWG (2.5 mm)
4	Weight	30 kg

Table 10.2 Mild Steel Pipe data

1	Length	20 ft (~6 m)
2	Diameter	0.75 inch (21.2 mm)
3	Thickness	18 SWG (1.22 mm)

The model required following manufacturing operations:

- Sheet metal Works
- Turning
- Shaping
- Plasma Cutting
- Welding
- Soldering

The material was bought from City Saddar Road, Rawalpindi. The manufacturing Operations were performed in Manufacturing Resource Center (MRC) at NUST, Gowalmandi and Saddar in Rawalpindi.

Procedure:

1. A sheet of Mild steel is purchased. Usually the Mild Steel Sheets come in size of 8ft x 8ft. The sheet is cut into half by Hydraulic shear.

Fig. 10.1 Shear Machine

2. That sheet is then cut into pieces of dimensions 315 mm x 500 mm by hand shear.
3. That cut piece of metal is rolled to form a cylinder of diameter 315 mm and 500 mm long using a manual sheet roller. That cylinder serves as shell of heat exchanger. Now the rolled piece's open ends are welded using electric arc welding.

Fig. 10.2 Rolled Cylinder

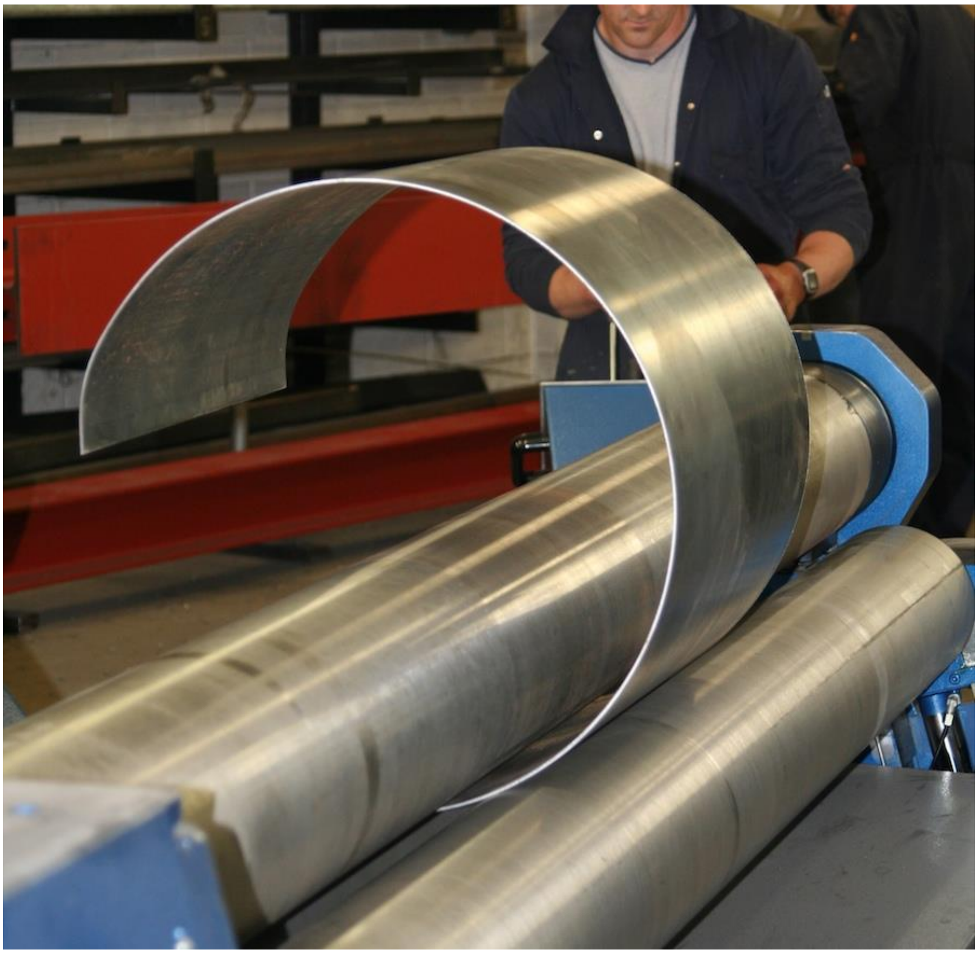


Fig. 10.3 Rolling machine

4. Two holes of ~6 inch (150 mm) diameter are cut using Plasma cutter on shell for air inlet and outlet.

Fig. 10.4 Plasma Cutting

5. From the remaining main sheet two square pieces of 350 mm x 350 mm are cut using Hand shear. These are the plates to be fixed on sides of this cylinder are machined (turning) from same metal sheet into 320 mm diameter plates. 9 holes of diameter 20 mm are drilled in theses plates according to design.

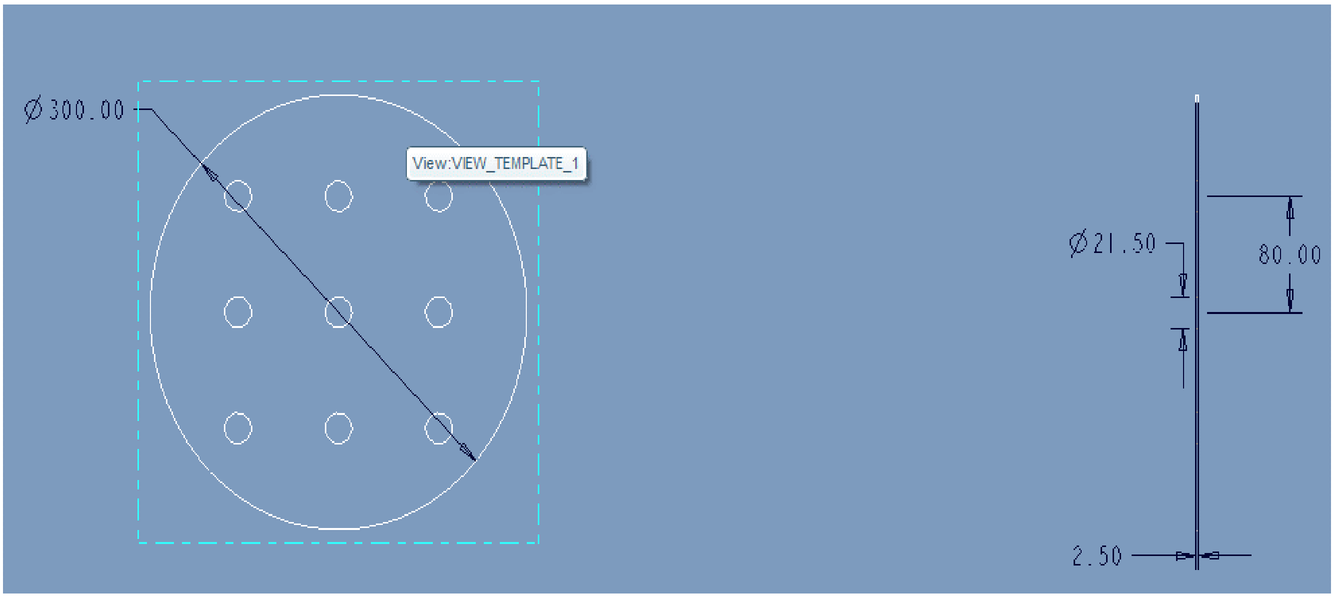


Fig. 10.5 Plate drawings

6. Another 13 SWG Mild steel sheet is cut into thirty six 35 mm x 450 mm rectangular pieces using shear for making fins. The shear machines available were unable to reduce their width further. They either unable to cut or bend the sheet. So machining operation was required to reduce thickness further. These 36 pieces are stacked above each other and point welded. Now this metal pile is machined using a shaper machine. The shaper machine took considerable time to cut metal from both sides. The thickness of each fin was 25 mm after shaping.

Fig. 10.6 Shaper machine

7. The M.S pipe is then cut into 45 mm lengths. Total nine pipes are required. The cut pipes are then finished using grinder.
8. These fins are soldered on these 0.75 inch dia pipes. Four fins on each pipe. The soldering is an expensive operation due to cost of metal, other ingredients and skill required to join. But Welding is not suitable for metal sheets less than 2 mm. The welding produces enough heat to melt the pipe at that spot. So, the quickest and good option was to solder.

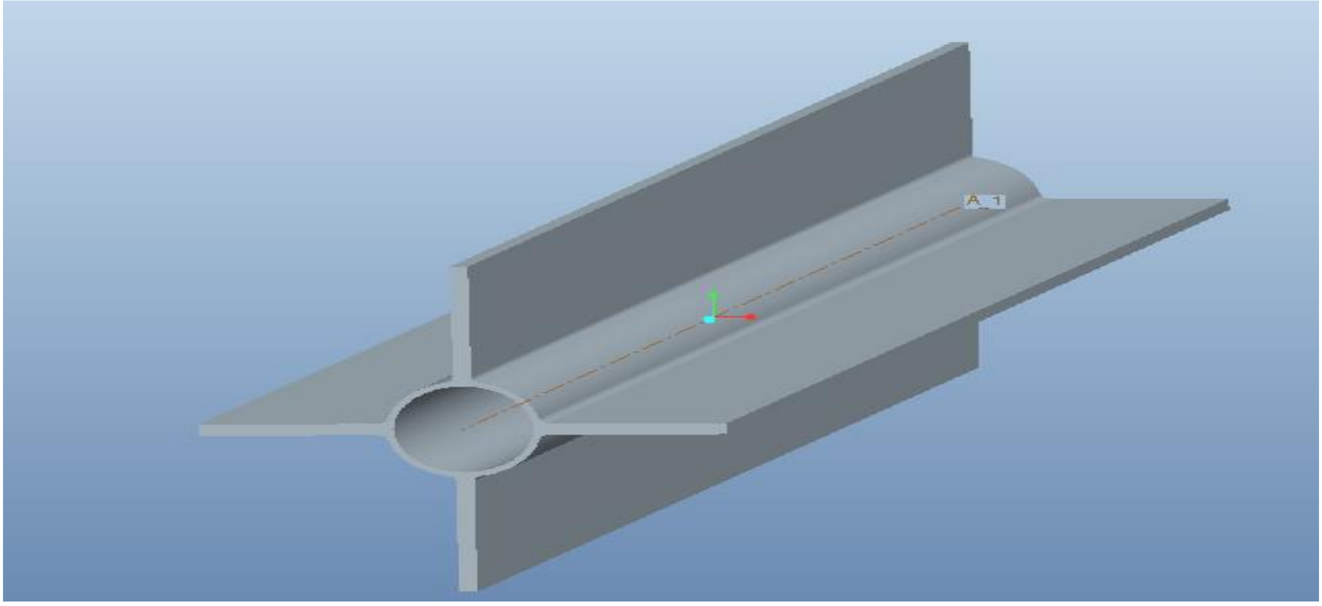


Fig. 10.7 Fin attached on pipe

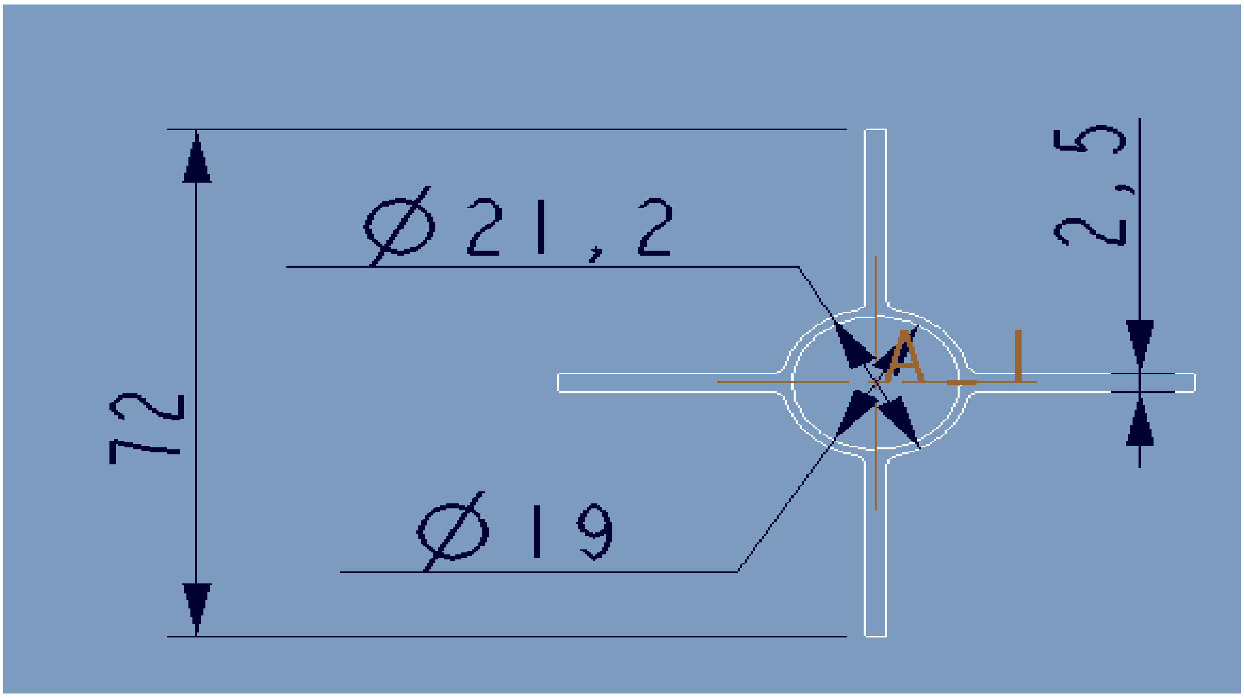


Fig. 10.8a model drawings (fins)

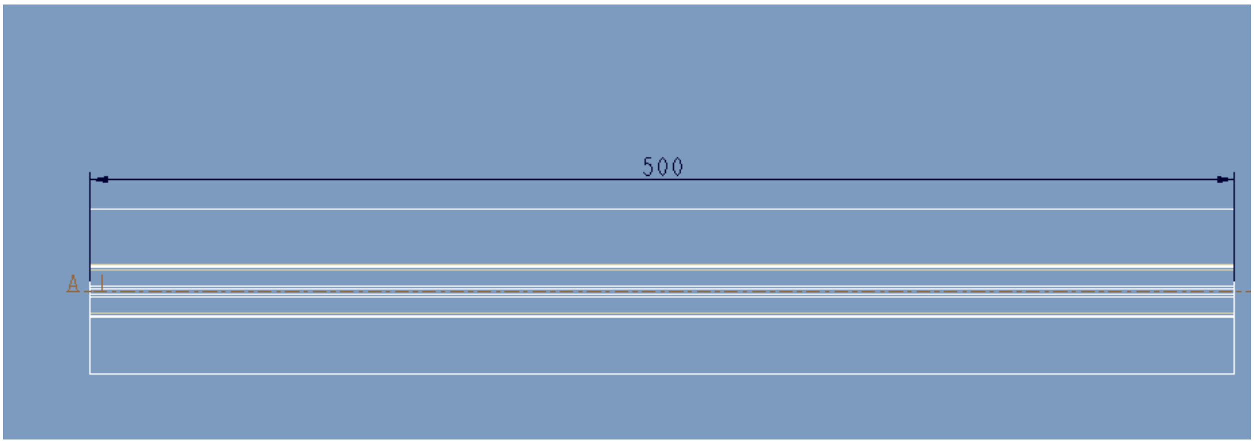


Fig. 10.8b model drawings

9. These 9 pipes are then soldered with those plates. The solder joint was the best option here for many reasons. One reason was that welding joint can melt the metal at that point. Other reason was the leak proof. The solder joint ensures the leakage proof of joints. Now those plates are joined to the main cylinder by soldering.

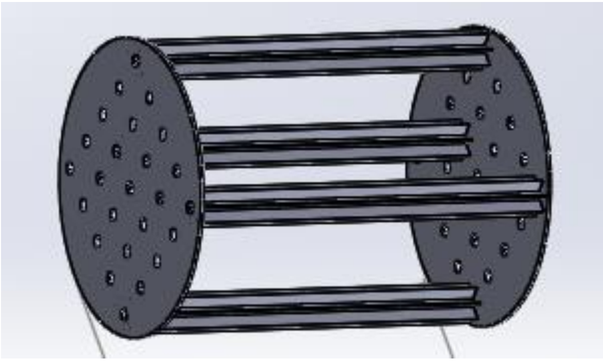


Fig. 10.9
Sub assembly

10. Two hemispherical lids of 310 mm diameter and four inch depth are purchased from black smith. Hole of 25 mm are drilled on each lid for water inlet and outlet. These lids are then joined with main cylinder plates first by spot welding and then sealed with silicone gel. The pressure was not beyond limits so silicone was a good option. The complete welding could melt the solder joints in surroundings and soldering these lids was an unnecessary addition in budget.
11. A stell pipe of thickness 0.5 inches (13 mm), 6 inch (15 cm) outer diameter and 4 inch (10 cm) long is machined and then welded in the hole on shell to make air inlet. Same was done with air outlet.
12. The Whole model is first covered with Lead oxide coating. Then it is silver painted.

11. Electronics:

This manufactured prototype was then made ready for running a test. For measurement of temperature, a temperature sensing circuit was designed.
Electronic items used for temperature sensing circuit.

11.1. Electronic Circuit

- Arduino (Uno)
- Temperature sensor LM35
- Connecting wires
- PC for Arduino interface and 5V supply

Fig. 11.1 LM35

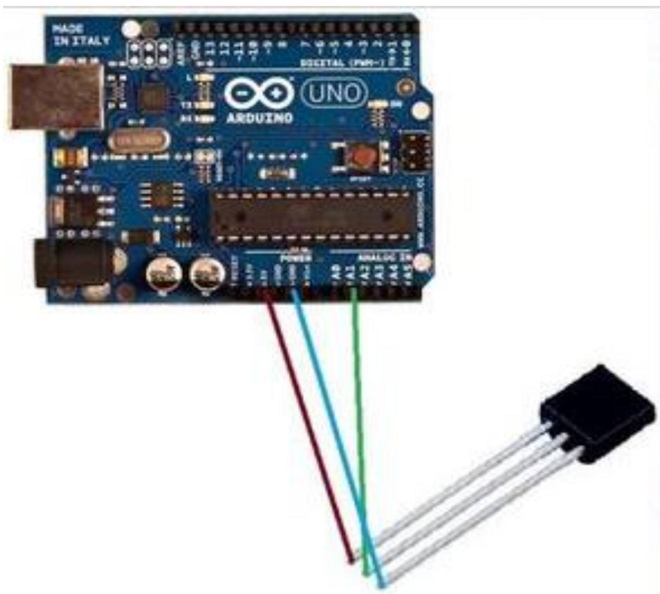


Fig. 11.2 Circuit

11.2. Arduino code:

```
int val;
int tempPin = 1;

void setup()
{
Serial.begin(9600);
}
void loop()
{
val = analogRead(tempPin);
float mv = ( val/1024.0)*5000;
float cel = mv/10;
float farh = (cel*9)/5 + 32;

Serial.print("TEMPRATURE = ");
Serial.print(cel);
Serial.print("*C");
Serial.println();
delay(1000);

Serial.print("TEMPRATURE = ");
Serial.print(farh);
Serial.print("*F");
Serial.println();


*/
}
```

12. Operation of Model:

The model was run by providing air input from drier. Water inlet was attached to pump submerged in bath tub. The water exit line was thrown in another bathtub. The hair drier was switched and water was run through the heat exchanger. The electronic circuit was attached. LM35 sensor's wiring was covered with silicone gel for prevention and sensor was submerged in first cold water inlet and then warm water outlet to check temperature.

The inlet temperature shown was 28.3`C and outlet temperature was 29.9`C. almost change of 1.6 `C which was very close to desired results.

13. Glow plugs:

Along with heating engine block using heat exchanger, there is another strategy that can be employed in parallel. i.e heating the air directly.

The air entering inside a diesel engine can be heated in advance, so after compression it attain temperature that kindle it. This can be accomplished by use of glow plugs. Glow Plugs are metallic rods that are heated with help of electricity. They produce very high temperatures. When incoming air comes in contact with them, it is heated. The point to be noted here is that, air must be heated with in a certain limit. Heating air too much can spoil volumetric efficiency and engine performance suffers.

13.1. Cold start:

The term "cold start" describes all initial processes occurring while the engine and other related systems have not yet equal to optimum operating temperature. The conditions for a quick ignition and complete, environmentally friendly combustion are not favorable if the temperature is lower than required. Certain assisting procedures are employed to help during the cold start and so that starting will not be unacceptably long or even impossible. These compensate for the poorer start conditions while initiating a well-timed and even ignition to ensure stable combustion. The glow plug is one component that assists during cold start. It creates ideal ignition conditions for the injected fuel through electrically generated thermal energy that is brought into the combustion chamber. It is significant as cold start aid for engines with a divided combustion chamber, in order to ensure that these can start even in the frequently occurring temperature range of 10–30 °C. Since the start quality declines significantly at below freezing point, the glow plug is also used as cold start aid for direct-injection diesel engines.

Glow plugs comes in variant forms and types the most common one is Self-regulating pencil type glow plugs

13.2. Requirements of glow plugs:

SHORT HEAT-UP TIME Glow plugs must provide a high temperature within as short a time as possible to assist with ignition – and they must maintain this temperature regardless of the ambient conditions, or even adjust the temperature depending on them.

PRECISE ADAPTATION TO THE COMBUSTION CHAMBER Ideally, the glow rod should be situated precisely at the edge of the mixture vortex - however, it must still project sufficiently deep into the combustion chamber or the antechamber. Only then is it able to introduce the heat accurately. It may not protrude too far into the combustion chamber, otherwise it would disturb the ignition and the preparation of the mixture for an ignitable fuel-air mixture. This will increase exhaust gas emissions. Other than the glow plug SUFFICIENT GLOWING VOLUME , the injection system is of particular significance in the engine cold start. Only a system that has been optimized in terms of its injection point, quantity and mixture composition in conjunction with the correct position and thermal rating of the glow plug will ensure good cold start performance. Even after the engine has been started, the glow plug may not be "blown cold" by the increased air movement in the combustion chamber. Very high air speeds are in particular present in antechamber or turbulence chamber engines at the glow plug tip. In this environment, the plug will only work if it has sufficient reserves; i.e. if sufficient glowing volume is available so that heat can immediately be brought on into the cold-blown zone. The glow plugs developed by BERU fulfill all these requirements in an optimal manner. BERU engineers work closely with the automotive industry especially during the engine development stage. The result: an environmentally-sound diesel quick start in 2-5 seconds (in conjunction with the Instant Start System ISS a maximum of 2 seconds), a reliable start up to –30 °C, a steady engine start-up that is gentle on the engine, with up to 40 % less carbon-particulate emissions in the warm-up phase for post-heating glow plugs

Design and function The BERU glow plug basically comprises the plug body, heating rod with heating and regulating coil, as well as the connecting bolt. The corrosion resistant glow rod is pressed in the housing so that as to be gas-tight. The plug is additionally sealed by a sealing ring or a plastic component at the connector. A battery supplies the electrical energy for the glow plug. It is controlled by an electronic glow time control unit. HEATING AND REGULATING COIL The basic principle of a modern glow plug is the combination of a heating and a regulating coil into a single common resistor element. The heating coil is made of high-temperature resistant material the electrical resistance of which is largely temperature independent. Together with the front part of the glow rod, it forms the heating zone. The regulating coil is attached to the live connecting bolt; its resistance has a large temperature coefficient. The entire coil is firmly packed in a compressed, electrically insulating but highly heat-conductive ceramic powder. During mechanical compaction, the powder is compressed so much that the coil is fitted as if it was cast in cement. This makes it so stable that the thin wires of the heating and regulating coil can permanently resist all vibrations. Even though the individual windings are arranged only a few tenths of a millimeter apart, no winding short circuits can be produced – and certainly no short circuit to the glow tube, which would destroy the plug. With the different materials, lengths and diameters, and different wire thicknesses for the heating and regulating coil, it is possible to change the heat-up times and glow temperatures of the plug in accordance with the respective requirements of the engine. Function During pre-heating, a high current initially flows via the connecting bolt and the regulating coil to the heating coil. The latter heats up quickly, causing the heating zone to glow. Glowing quickly expands – after 2-5 seconds, the heating rod glows up to near the plug body. This additionally increases the temperature of the regulating coil that has already been heated up by the current. Then, the electrical resistance increases and the current is reduced to a point where it cannot cause any damage to the glow rod. Overheating of the glow plug is thus not possible. If the engine is not started, the glow plug will be switched off by the glow time control unit after a certain stand-by time. The resistance of the alloy used on BERU glow plugs increases with the temperature. It is thus possible to design the regulating coil in such manner that it will initially let through a higher current to the heating coil than when it reaches the target temperature. The target temperature is thus reached quicker and is maintained within the permissible range by an increased regulating effect. 6 Connecting bolt Design of a self-regulating, fast-heating pencil type glow plug. Plug body O-ring seal Insulation disc Round nut Gasket Insertion thread Annular gap Glow tube Regulating coil Insulating filling Heating coil Self-regulating pencil type glow plugs

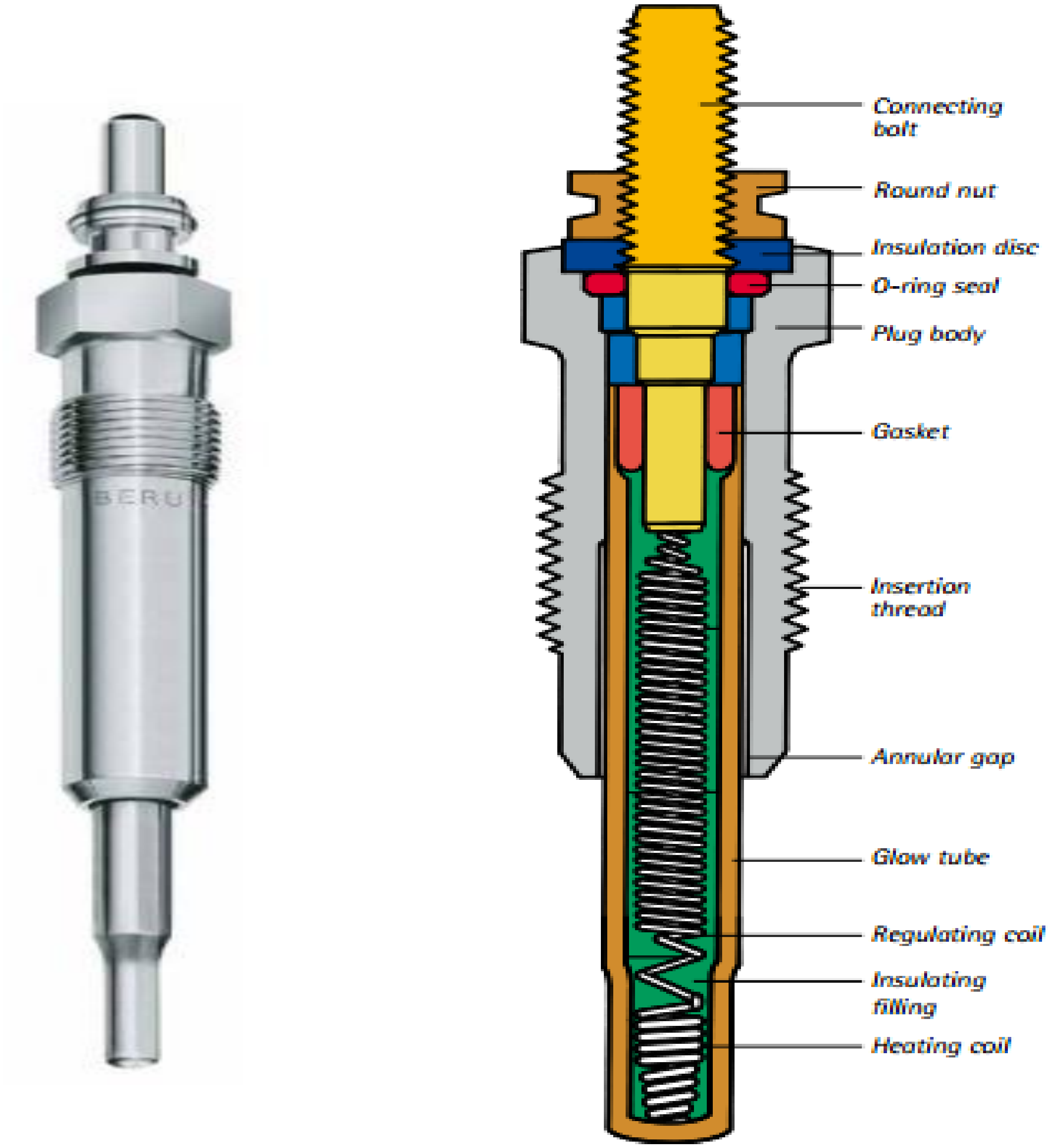
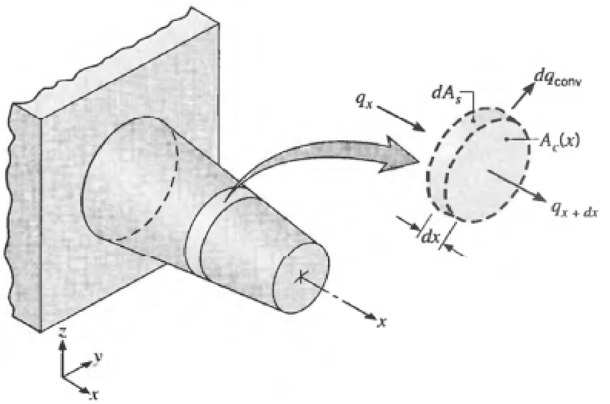


Fig. 13.1 Glow plugs

14. Appendix:

Fin Equation:



$$\dot{Q}_x = -kA_c \frac{dT}{dx}$$

$$\dot{Q}_{x+dx} = \dot{Q}_x + \frac{d\dot{Q}_x}{dx} dx$$

$$d\dot{Q}_{conv} = hdA_s(T - T_\infty)$$

Energy Balance:

$$\dot{Q}_x = \dot{Q}_{x+dx} + d\dot{Q}_{conv} = \dot{Q}_x + \frac{d\dot{Q}_x}{dx} dx + hdA_s(T - T_\infty)$$

$$\frac{d}{dx}\left(A_c \frac{dT}{dx}\right) - \frac{h}{k}\frac{dA_s}{dx}(T - T_\infty) = 0$$

$$\frac{d^2T}{dx^2} + \frac{1}{A_c}\frac{dA_c}{dx}\left(\frac{dT}{dx}\right) - \left(\frac{1}{A_c}\frac{h}{k}\frac{dA_s}{dx}\right)(T - T_\infty) = 0$$

$$\frac{d^2T}{dx^2} + \cancel{\frac{1}{A_c}\frac{dA_c}{dx}}\left(\frac{dT}{dx}\right) - \left(\frac{1}{A_c}\frac{h}{k}\frac{dA_s}{dx}\right)(T - T_\infty) = 0$$

$$\frac{dA_c}{dx} = 0$$

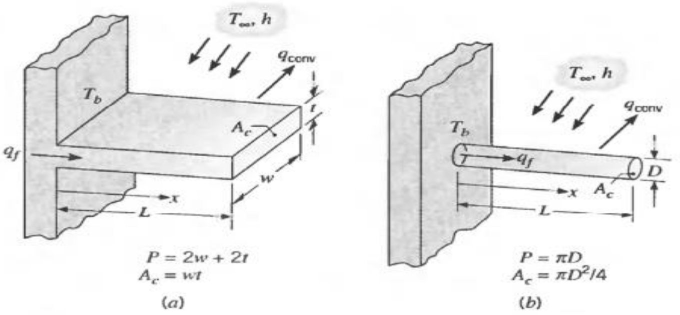
$$A_s = Px \qquad \frac{dA_s}{dx} = P$$

$$\frac{d^2T}{dx^2} - \left(\frac{hP}{kA_c}\right)(T - T_\infty) = 0$$

Excess temperature θ

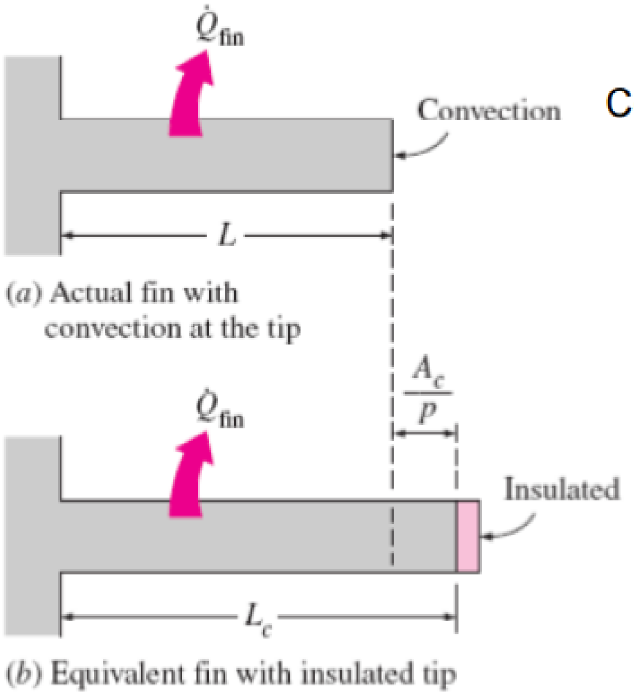
$$\theta(x) \equiv T(x) - T_\infty$$

$$\boxed{\frac{d^2\theta}{dx^2} - m^2\theta = 0}$$



Straight fins of uniform cross section
(a) Rectangular Fin (b) Pin fin

Corrected fin length



Corrected fin length: $$L_c = L + \frac{A_c}{P}$$

Multiplying the relation above by the perimeter gives

$$A_{corrected} = A_{fin\ (lateral)} + A_{tip}$$

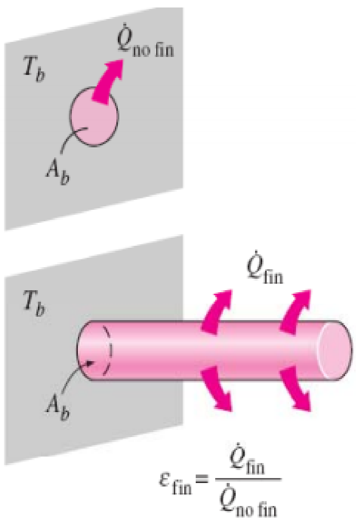
$$L_{c,rectangularfin} = L + \frac{t}{2}$$

$$L_{c,cylindricalfin} = L + \frac{D}{4}$$

Fin Effectiveness

$$\varepsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{nofin}} = \frac{\dot{Q}_{fin}}{hA_b(T_b - T_\infty)} = \frac{\text{Heat transfer rate from the fin of base area } A_b}{\text{Heat transfer rate from the surface of area } A_b}$$

$$\varepsilon_{longfin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{nofin}} = \frac{\sqrt{hPkA_c}\,(T_b - T_\infty)}{hA_b(T_b - T_\infty)} = \sqrt{\frac{kP}{hA_c}}$$



15. References:

- *Heat and mass transfer Transfer by Yunus Cengel*
- *Heat Transfer by Carlson*
- *Thermodynamics- an engineering approach by Yunus cengel*
- *Federal Mogul Motor Parts Manual*
- *Saunders, E. A. (1988). Heat Exchanges: Selection, Design and Construction. New York: Longman Scientific and Technical.*
- *"A new configuration of bend tubes for compound optimization of heat and fluid flow". Energy.*
- *White, F.M. 'Heat and Mass Transfer' © 1988 Addison-Wesley Publishing Co. pp. 602–604*
- *Heat Exchangers Kevin D. Rafferty, Gene Culver Geo-Heat Center, 1996–2001*
- *Sadik Kakaç; Hongtan Liu (2002). Heat Exchangers: Selection, Rating and Thermal Design (2nd ed.). CRC Press. ISBN 0-8493-0902-6.*
- *"FEATool - a Matlab FEM Finite Element Multiphysics Toolbox". Precise Simulation Ltd.*
- COMSOL: official site
- *www.Efunda.com*
- *www.sciencedirect.com*
- *www.youtube.com/ solid works tutorials*