INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

Solar Desalination using Humidification-Dehumidification Cycle

by

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Contents

Physical Constants

Specific area of evaporator	a		m^2m^{-3}
External area of condenser casing	A_c		m^2
Heat transfer area of condenser	A_{cond}		m^2
External area of evaporator casing	A_E		m^2
Heat capacity of brine	C_p	=	2.4359 kJ kg^{-1} $^{\circ}C^{-1}$
Heat capacity of air	C_{p_a}	=	1.009 kJ kg^{-1} $^{\circ}C^{-1}$
Heat capacity of liquid water	C_{p_w}	=	4.193 kJ kg^{-1} $^{\circ}C^{-1}$
Rate of condensed water	D		kg s^{-1}
Correction factor for heat and mass transfer	e		*nondimensional*
Humidity factor	f		*nondimensional*
Dry air mass flow rate	G		kg s^{-1}
Brine mass flow rate	L		kg s^{-1}
Enthalpy of saturated air at temperature T_i	H_i		J kg^{-1} dry air
Mass transfer coefficient	L		kg s^{-1} m^{-2}
Molecular mass of air	M_a	=	0.028966 kg mol^{-1}
Molecular mass of water	M_w	=	0.018016 kg mol^{-1}
Atmospheric Pressure	P		Pa
Vapor Pressure of water at temperature point i	P_i^0		Pa
Ambient Temperature	T_{amb}		$^{\circ}C$
Average surface temperature of condenser	T_{avgC}		$^{\circ}C$
Average surface temperature of evaporator	T_{avgE}		$^{\circ}C$
Temperature at point i	T_i		$^{\circ}C$
Distillate temperature	T_d		$^{\circ}C$
Overall coefficient of heat transfer at condenser	U_{cond}		J s^{-1} m^{-2} $^{\circ}C^{-1}$
Overall heat loss coefficient of condenser	U_{LC}		J s^{-1} m^{-2} $^{\circ}C^{-1}$
Overall heat loss coefficient of evaporator	U_{LE}		J s^{-1} m^{-2} $^{\circ}C^{-1}$
Evaporator volume	A_c		m^3
Humidity of saturated air at temperature T_i	W_i		kg water/kg dry air
Enthalpy of vaporization of water	ΔH_{vap}	=	2332.20 kJ kg^{-1}

Chapter 1

Introduction

Sufficient quantity and reliability of fresh water is a fundamental need for humanity and other living beings. In many parts of the world, people suffer from lack of freshwater and they live mostly in arid, remote areas and islands. On the other hand, in most of these regions, abundance of solar energy is available with the large amount of sea or underground saline water. For those reasons, by many researchers, solar water desalination systems are proposed as an economical and environmentally friendly solution to supply small settlements in these locations with enough drinking water. Standard desalination processes such as MSF (Multi Stage Flash), ME (Multi Effect), VC (Vapor Compression) and RO (Reverse Osmosis) are suitable for large and medium capacity fresh water production. But most remote arid areas need low capacity desalination systems. The HDH desalination process (humidification and dehumidification) will be a suitable choice for fresh water production when the demand is decentralized. HDH is a low temperature process where total required thermal energy can be obtained from solar energy. Capacity of HDH units is between conventional methods and solar stills.

The process based on Humidification–Dehumidification of air principle mimics the natural water cycle. HDH technique has been subject of many studies in recent years due to the possibility of low-temperature energy use (geothermal, solar, waste energy), its simplicity, its low installation and operation costs. Moreover HDH desalination systems work at atmospheric pressure; hence they do not need additional large mechanical energy. These kinds of systems do not require sophisticated high-technologies so that they can be easily implemented in developing countries. Therefore, their design, construction and operation are easy. The system is modular; so that the capacity can be increased system additional solar collectors and additional HDH cycles

Chapter 2

Model Description

In the air humidification–dehumidification cycle, the air acts as a carrying fluid, i.e. evaporation occurs without boiling of water. Air is humidified in one of the system's regions and moved to a cooler region where water condenses, releasing its enthalpy of condensa- tion. In the condenser this heat can be recovered and used to preheat the seawater or brackish water. One of the most commonly studied HDH cycles is the Closed air, Open water(CAOW) cycle, in which air is in continuous circulation and is being reused whereas the water circulation from the inlet is disjoint from outlet. A schematic diagram of such a system is shown as follows:

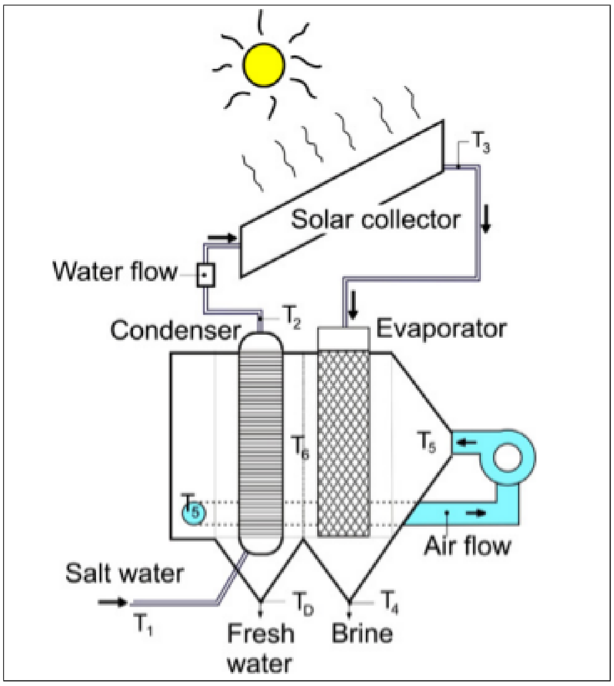


FIGURE 2.1: Schematic diagram of a CAOW cycle

Air flows in a closed loop from condenser to evaporator and vice versa. Water flows in an open circuit as follows : seawater enters the system as a cooling fluid at temperature T_1 and flows through the condenser, thus cooling the humid air and absorbing its change of enthalpy. It leaves the condenser at temperature T_2 having recovered the heat of evaporation of water. Then this water is further heated up in the solar collector/Electric heater leaving it at the temperature of T_3 . The hot water is then distributed on a large area substratum at the evaporator to set conditions for evaporation in air. The heat of evaporation is taken from its own sensible heat, thus cooling it to temperature T_4 . Brine leaves the system at this temperature. Distilled water leaves the system at T_D, a temperature slightly higher than T_1 .

The air loop is as follows: air flowing through the evaporator is humidified and heated by hot water up to T_6 at the exit of this element. At this point the air has the highest absolute water content in the loop. It would be desirable that the air were saturated. However, theoretical considerations regarding the brief retention time in the evaporator make safer to suppose that it is not fully saturated. In the next stage, air is cooled in the condenser region to T_5 , a temperature considerably lower than T_6 . This makes air to reach saturation, forcing the water to condense.

Chapter 3

Detailed part wise description

3.1 Condenser/Dehumidifier

The condenser is the most vital component of this setup in the sense that it has scope both in the material selection and design orientation. These parameters shall be varied throughout the study of this setup.
Brine or brackish water enters the condenser at a temperature T_1 which is usually the ambient temperature $(27°C - 30°C)$ at a mass flow rate of L which may vary between 0.02 kg s^{-1} – 0.2 kg s^{-1}. A unit of the condenser is show below :

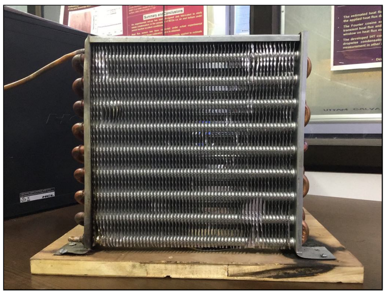


FIGURE 3.1: A condenser unit

A pair of such units may be placed together in an incline to facilitate the droplet condensation and collection of water. Such a setup can be visualized as shown in Figure 3.2:

A series of such inclines make up the entire Dehumidifier which is enclosed in a galvanized steel enclosure of dimension $0.3m \times 0.4m \times 0.3m$. The enclosure has inlet for brine inflow, brine outflow, air inflow at temperature T_6, at a mass flow rate of $G = 0.04$kg s^{-1} which then passes through several units of condenser, and exits as temperature T_5. Heat exchange through the condenser, cools down the air and results in droplet formation on the fins of the condenser which are collected at the bottom. The parameters influencing the droplet formation are: U_{cond}, U_{LC},

FIGURE 3.2: Two condenser units placed at an incline

L and G. The controllable parameters in the design of the condenser are: U_{cond}, U_{LC}, fin separation and fin area. The detailed calculations of these parameters are shown in **Appendix**.

3.2 Humidifier

The humidifier consists of various parts (humidifier casing, swing door and packing material). The humidifier casing is made of galvanized steel with rectangular cross sectional area $0.3m \times 0.3m \times 0.4m$. A built in swing door can be added to the humidifier to facilitate the changeability of humidifier's packing material, which is fixed inside it. Packing material being used is also a parameter of study which has high influence on the production of pure water. Options available are wood, cellulose paper having honeycomb structure (shown in figure) or other materials which provide high surface area for evaporation.

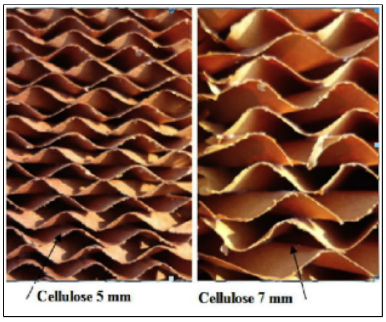


FIGURE 3.3: Cellulose papers of different sizes

Air enters the humidification chamber at a temperature of T_5, is heated along with an increase in temperature and exits the humidification chamber at temperature of T_6. To increase the humidification of the air, the packing material should provide a high surface area per unit volume which is represented by the parameter a. For the materials mentioned this normally lies in the range of $100m^2m^{-3}$ to $400m^2m^{-3}$. The total volume occupied by the packing material is V which is normally half of the humidification chamber volume and the mass transfer coefficient for the humidifier is represented by K. Brine which is heated from the solar heater, is sprayed onto the packing material at temperature of T_3 at a mass flow rate of L and leaves the humidification chamber at a temperature of T_4. The air mass flow rate is represented by G which is set to a value of 0.04kg s^{-1} by an air fan. From the study done in [1], a relation between the above mentioned parameters is obtained as follows:

$$\frac{KaV}{L} = 0.53 - 0.22 \log \left(\frac{L}{G} \right)$$

and

$$\frac{KaV}{L} = 0.52(L/G)^{-0.16}$$

The variation of KaV which represents the total mass flow rate (kg s^{-1}) as a function of the brine flow rate G is obtained and is shown below:

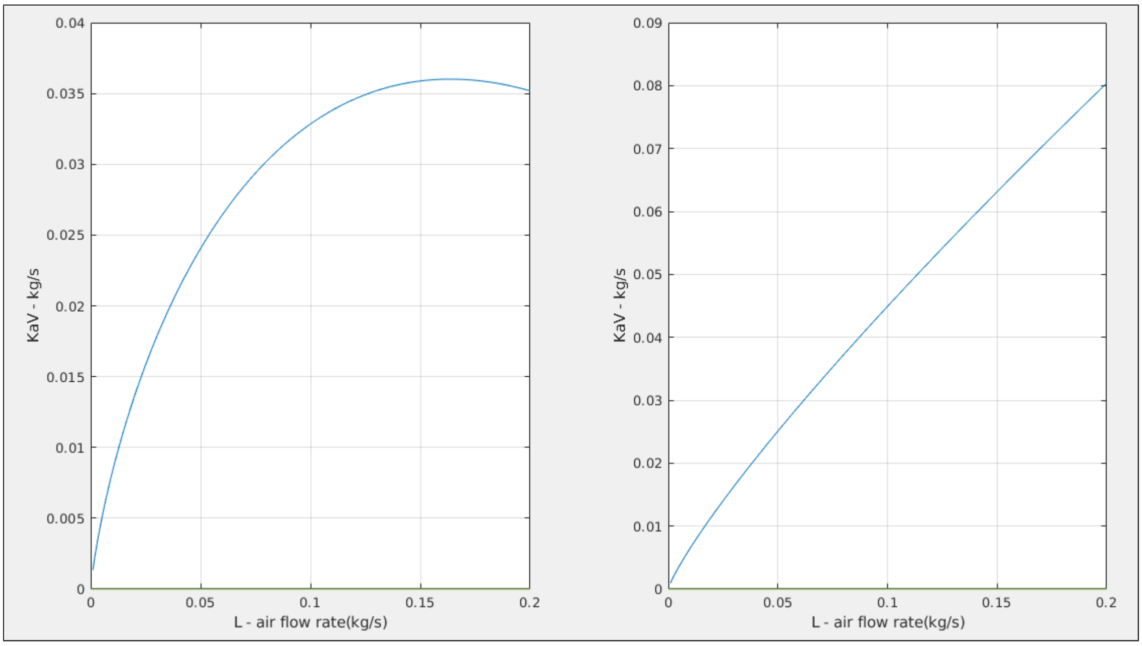


FIGURE 3.4: Variation of Kav vs L

3.3 Blower/ Fan

The air is circulated either by natural or forced circulation. The effect of three types of forced circulating air (up, down and up-down) on the unit performance was done by [2], and it was shown the placing the blower/fan at the bottom has the most effect on the production. For this a commonly available fan of 85 cfm = 0.04 $m^3 s^{-1}$ can be used. The dimension of the fan available is $12cm \times 12cm \times 2.5cm$ and the power requirement is in the range of $15W - 20W$.

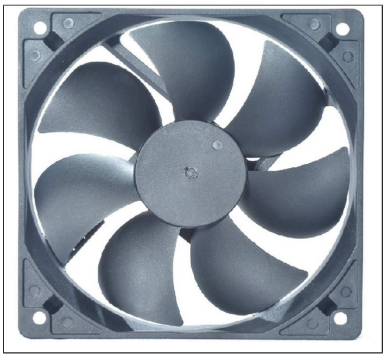


FIGURE 3.5: Air fan (85cfm)

3.4 Piping

PVC piping of small diameter ($3cm - 4cm$) can be used for carrying brine from the condenser to the solar heater and from the solar heater to the sprayer at the humidifier inlet. PVC provides low frictional losses and thus is useful in reducing pumping power.
For air flow, PVC piping of larger diameter so as to allow the air fan to fit in can be used.

3.5 Pump

FIGURE 3.6: Pump (40W)

Pump is required to maintain the mass flow rate of brine, L across the setup. The head losses in the system are from:

- Frictional loss through the condenser
- Frictional loss through the PVC piping
- Gravitational head loss
- Pressure head loss due to sprayer

Of these, the frictional loss is a function of the mass flow rate of brine L. Use of the Darcy Weisbach equation suggests that, keeping the friction coefficient a constant, the power is a cubical function of the mass flow rate L.

A detailed calculation of Pumping power is shown in **Appendix**. This suggest that a pump of $40W - 120W$ will be required. A suitable centrifugal pump can be used with guide vanes to control the mass flow rates.

3.6 Solar heater/ Electric heater

Considering solar irradiation to be a constant euqal to $1.4kWm^{-2}$, a Solar collection of dimension $2m \times 1m$, having an efficiency of $50\% - 60\%$, can easily produce a power of $1.5kW$. This Solar collector is used to heat the brine coming out from the dehumidifier section at a temperature of T_2 to a higher temperature of T_3 which is then fed to the Humidifier section.
As solar irradiation is dependent on the time of the year and the time of the day, for simulation purposes, an equivalent Electrical heater can be used.
The pump, air fan and other electronics roughly add up to 150-170 W. This can be battery driven or a Solar PV cell of similar dimension as the solar collection can be used making the entire setup self sufficient.

3.7 Other components

Other components to be used in the setup are:

- Sprayer at humidifier inlet (30 psi)
- Thermocouples to measure the six temperatures.
- Humidity sensors to measure the humidity at point 5 and 6.
- Pressure sensors for flow analysis.

Chapter 4

Setup Design and Construction

4.1 Setup Assembly

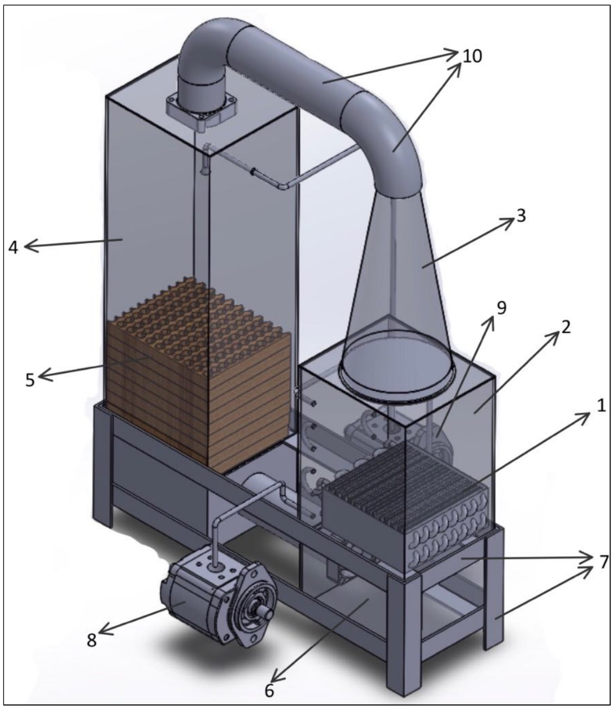


FIGURE 4.1: Setup Assembly

The components of the assembly are as follows:

1. Condenser
2. Dehumidifier casing
3. Diffuser

4. Humidifier casing
5. Cellulose pads
6. Collector
7. Frame and Legs
8. Pumps
9. Electric heater
10. Pipes and bends

The size limiting parameters of the assembly were the available condensers in the market. This limits the cross section of the assembly. The height of the assembly is restricted by the size of the diffuser, the calculation of which has been detailed in the calculation chapter. The components that will be manufactured by us are detailed in sections that follow. The Drawings of each of these components are attached in **Appendix A**

4.2 Dehumidifier Casing

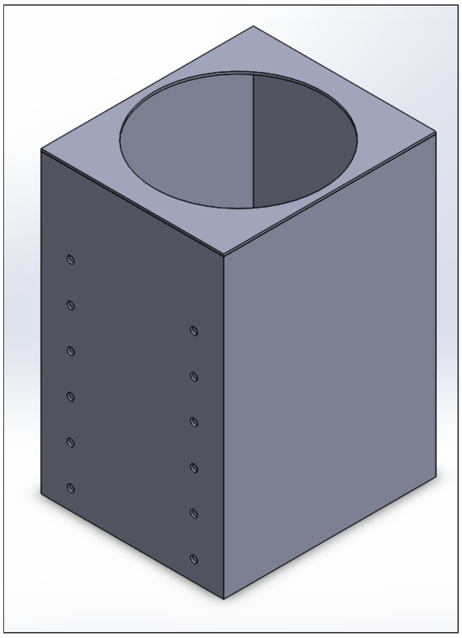


FIGURE 4.2: Dehumidifier Casing

The above figure shows an isometric view of the Dehumidifier casing. This casing houses the condenser units which may be placed in stacks one above the other. the inlets and outlets of condenser units are joined through the holes provided on the side of the dehumidifier casing. the large hole at the top serves as inlet of hot air coming from the humidifier. The large hole serves to spread the air across the condenser so as to maximize pure water production. Material to used for the casing is Mild Steel sheet of thickness of $3mm$. Processes that will be used for making the casing are Welding, Cutting, Machining and Turning.

4.3 Diffuser

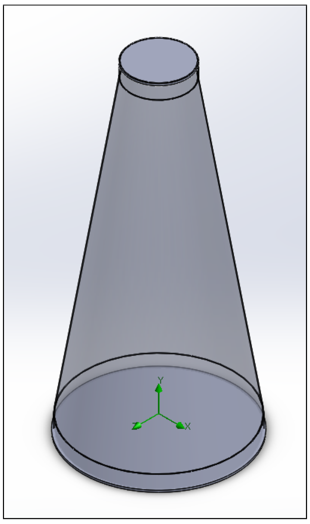


FIGURE 4.3: Diffuser

The above figure shows an isometric view of the diffuser. This is provided for the expansion of air into a larger cross sectional area after it passes through the humidifier casing through the pipes and bends. The larger diameter is to ensure that air passes throughout the cross sectional area of the condenser units. The height of the condenser is calculated using CFD simulations. This has been outlined in the section on Calculations. The material to be used for the diffuser is GI sheet of $0.3mm$. The process that would be employed here are Cutting, Development, Marking, Bending, Soldering and Riveting.

4.4 Humidifier Casing

The figure shown below represents the isometric view of the Humidifier casing. This casing houses the stack of cellulose padding on which hot water coming from the electric heater through a pump is sprayed through the small hole shown on the side wall. The hole present at the top wall is the hot air outlet that passes through the cellulose padding absorbing moisture. The circulation of hot air is ensured with a fan placed at the lower surface of the opening. The material to be used for the casing is Mild Steel sheet of $3mm$. The process that would be employed for the casing are similar to that used for the Dehumidifier casing. The bend carrying the hot air from the top of the casing will be joined through a flange made of persplex.

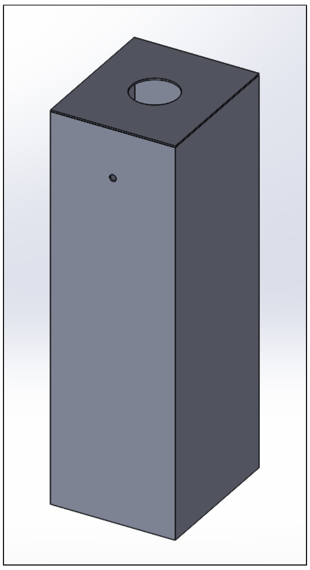


Figure 4.4: Humidifier Casing

4.5 Collector

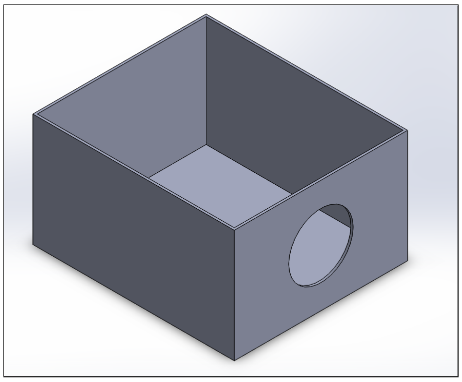


Figure 4.5: Collector

The figure above shows the isometric view of collector. This unit shall be symmetrically placed below the humidifier casing and dehumidifier casing. The collector below the dehumidfier casing is to ensure the capture of pure water which shall be directed out of the setup through a hole made at the bottom of the collector. The collector below the humidifier casing is to ensure the capture of impure water passing through the cellulose padding which shall be directed out of the setup through a hole made at the bottom of the collector. The larger hole made on the side of the collector is to circulate air between the the dehumidifier and the humidifier casings via a fan attached on the inner wall of the collector. The material to be used is Mild Steel sheet of $3mm$. The processes that will be used are similar to that of Dehumidifier casing.

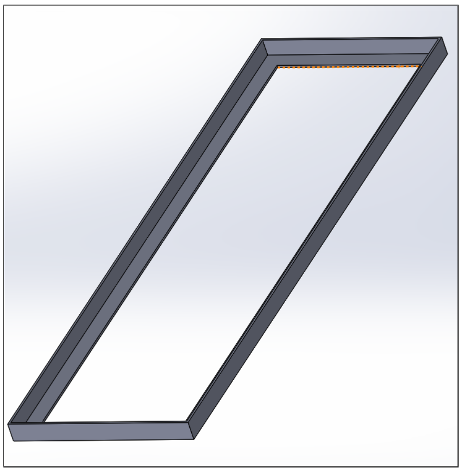


FIGURE 4.6: Frame

4.6 Frame and legs

The figure above shows the isometric view of the frame to used in the setup. This is made of bracket of cross section $45mm - 45mm - 3mm$. This frame support the collector units on the bottom level and another similar frame supports the humidifier and dehumidifier casing on the upper level. The horizontal faces of the frame is also used to house the condensers and the cellulose paddings. Legs made of the same bracket are to be used to raise the entire setup above ground level to ensure efficient usage of piping for the outlets of the collectors. The material to be used for this is Mild Steel. The process to be used are Welding and Cutting.

Chapter 5

Calculations

5.1 Mathematical Model

The general mathematical mode is based on the conservation of mass and energy for each of the three components of the desalination system namely : humidifier, condenser and a solar collector. The air is saturated at point 5, thus enthalpy of humid air is calculated with the following equation, which assumes $T = 0°C$ and zero humidity as a reference point for $H = 0$:

$$H_i = Cp_a T_i + (Cp_w T_i + \Delta H_{vap}) W_i$$

where Cp_a and Cp_w are the heat capacities of air and water respectively, and ΔH_{vap} is the heat of vaporization of water. The absolute humidity of saturated air, W_i is computed from the expression :

$$W_i = \frac{P_i^0 M_w}{(P - P_i^0) M_a}$$

According to the paper, the vapor pressure of water, P_i^0 is calculated as a function of temperature with the following equation : $P = P_0 e^{A+B/T+ClnT+DT}$ where T is given in kelvin and $P_0 = 7.384kPa$, $A = 67.35$, $B = -7218.15K$, $C = -7.9939$ and $D = 0.00052333K^{-1}$ But this expression did not led to correct results(after verification from the internet). So we used *Teten's equation* :

$$P = 0.61078 \exp(\frac{17.27T}{T + 237.3})$$

where T is in $°C$ and P is in kPa.

Air is not saturated at point 6. Thus we can consider a humidity factor f here : the enthalpy at point 6 is therefore $fH6$, where f ranges from 0 to 1, and is defined as the ratio of actual humidity to saturation humidity (both being at same temperature).

Assuming steady state, energy balance for the condenser can be written as :

$$\dot{Q}_{entering} - \dot{Q}_{leaving} - \dot{Q}_{loss} = 0$$

where $\dot{Q}$ represents heat rate. Heat enters condenser through humid air with mass flowrate G, and with an enthalpy $fH6$. Also heat enters through the seawater at flowrate L at temperature T_1. Thus

$$\dot{Q}_{entering} = GfH6 + LC_pT_1$$

Heat leaves the condenser through air and water flows. Assuming that the air which is leaving is saturated at temperature T_5, thus its heat content is GH_5. Water leaves at temperature T_2, which is than T_1 due to heat exchanging process :

$$\dot{Q}_{loss} = U_{LC}A_C(T_{avgC} - T_{amb})$$

Then, substituting and rearranging terms we get :

$$G(fH_6 - H_5) + LC_p(T_1 - T_2) - U_{LC}A_C(T_{avgC} - T_{amb}) = 0$$

For condenser, according to [2] :

$$LC_p(T_2 - T_1) = eU_{cond}A_{cond}\left[\frac{(T_6 - T_2) - (T_5 - T_1)}{ln\frac{T_6 - T_2}{T_5 - T_1}}\right]$$

where e is the correction factor for cross-flow heat transfer, U_{cond} is the overall heat transfer coefficient in the condenser, A_{cond} is the heat transfer area. Similarly, the heat balance in the evaporator on rearranging terms is :

$$(H_5 - fH_6) + LC_p(T_3 - T_4) - U_{LE}A_E(T_{avgE} - T_{amb}) = 0$$

where T_{avgE} is the average temperature in the evaporator, U_{LE} is the overall heat transfer coefficient and A_{cond} is the external area of condenser. The mass transfer rate in the evaporator is written as

$$G(fH_6 - H_5) = eKaV\left[\frac{(H3 - fH_6) - (H_4 - H_5)}{ln\frac{H_3 - fH_6}{H_4 - H_5}}\right]$$

where K is the mass transfer coefficient, a is the evaporator substratum specific area, V is the evaporator volume. Thus, aV is the evaporator mass transfer area.

Heat input is added between condenser output and evaporator input, *i.e.* between points 2 and 3. It can be simply modelled as :

$$\dot{Q} = LC_p(T_3 - T_2)$$

Here $\dot{Q}$ is the solar input, which is assumed to be constant for simplicity. The flow rate for the distillate is obtained from mass balance, considering the change of water content between the air entering and leaving the condenser, which is expressed by :

$$D = G(fW_6 - W_5)$$

5.2 Model solution

There are some assumptions which are considered to reduce the number of unknowns and to simplify calculations :

- Condenser and evaporator operate between the same temperatures, T_5 and T_6, thus the average temperatures can be expressed as $T_{avg} = T_{avgC} = T_{avgE} = \frac{1}{2}(T_5 + T_6)$
- The external areas of condenser and evaporator are similar, so $A_C \approx A_E$ and we can use A to represent external area of both parts.
- If the shape and thermal insulation of both elements are similar, then we can assume the global heat transfer coefficients are also similar : $U_{LC} \approx U_{LE}$ and use a single U_L

Now the variables are :

- Flow rates : L and G.
- Properties : Cp_w, Cp_a, H_{vap}, P.
- External geometry : A.
- Internal geometry : A_{cond}, a, V.
- Heat and mass transfer : $Q, U_L, U_{cond}, K, f, e, T_1, T_{amb}$.

Given these values, the unknowns in the model are the internal temperatures, T_2, T_3, T_4, T_5 and T_6. Thus, we have five variables and we need five independent equations to solve the model. To solve this system of highly nonlinear equations, we used MatLab command *fsolve* which solved the equations leaving a residual in the order of 10^{-4} for about every equation. It works by minimizing the sum of squares of residuals of every equation.

5.2.1 MatLab code

Following figure is the snapshot of the MatLab code which was used to solve the equations.

```
clear;
clc;

calc_temp;
function calc_temp

T = @(T2,T3,T4,T5,T6) [T2 T3 T4 T5 T6]

% Constants
L=0.02;
G=0.040;
Cpw=4.193*10^3;
Cpa=1.009*10^3;
Hvap = 2332.20*10^3;
%Hvap = 0;
P = 101.325*10^3;
Acond = 3.5;
Tamb = 30.5+273;
T1 = 27.9+273;
Cp = 2.4359*10^3;
% Cp = 3.12*10^3;
V = 0.0143;
a = 100;
% A = 2*(0.305*0.335*2+0.305*0.335);
Qdot = 1120;
Uloss = 7.04;
Ulc = Uloss;
Ule = Uloss;
Ac = 2*(0.3*0.3*2+0.3*0.4);
Ae = 2*(0.3*0.3*2+0.3*0.4);
e = 0.82;
Ucond = 47.90;
K = 0.0015;
f = 0.82;


Ma = 0.028966;
Mw = 0.018016;

% Constants obtained through inverse problem
% e = 0.82;
```

Figure 5.1: Code-1

```
% Ucond = 46.78;
% K = 0.0014;
% Uloss = 10.7174;
% Ulc = Uloss;
% Ule = Uloss;
% f = 0.8158;


%-----------------------

H3 = @(T) (Cp*T(2));
H4 = @(T) (Cp*T(3));
H5 = @(T) (Cpa*T(4) + (Cpw*T(4)+Hvap)*610.78*exp((17.27*(T(4)-273))/(T(4)-273+237.3))*(Mw/Ma))/(P-(610.78*exp((17.27*(T(4)-273))/(T(4)-273+237.3))));
H6 = @(T) (Cpa*T(5) + (Cpw*T(5)+Hvap)*610.78*exp((17.27*(T(5)-273))/(T(5)-273+237.3))*(Mw/Ma))/(P-(610.78*exp((17.27*(T(5)-273))/(T(5)-273+237.3))));

% Solution
opts = optimoptions('fsolve', 'TolFun', 1E-20, 'TolX', 1E-20,'MaxFunctionEvaluations',1000);
T0 = [30,32,40,35,36]+273.15*[1,1,1,1,1];
Temp = fsolve(@CalcTemps, T0,opts);



function fun = CalcTemps(T)

    fun(1) = G*(f*H6(T)-H5(T))+L*Cp*(T1-T(1))-Ulc*Ac*(((T(4)+T(5))/2)-Tamb);
    fun(2) = L*Cp*(T(1)-T1) - e*Ucond*Acond*(T(5)-T(1)-T(4)+T1)/(log((T(5)-T(1))/(T(4)-T1)));
    fun(3) = G*(H5(T)-f*H6(T))+L*Cp*(T(2)-T(3))-Ule*Ae*(((T(4)+T(5))/2)-Tamb);
    fun(4) = G*(f*H6(T)-H5(T))-e*K*a*V*((H3(T)-f*H6(T)-H4(T)+H5(T))/log((H3(T)-f*H6(T))/(H4(T)-H5(T))));
    fun(5) = -Qdot+L*Cp*(T(2)-T(1));


W5r = 610.78*exp((17.27*(T(4)-273))/(T(4)-273+237.3))*(Mw/Ma)/(P-(610.78*exp((17.27*(T(4)-273))/(T(4)-273+237.3))));
W6r = 610.78*exp((17.27*(T(5)-273))/(T(5)-273+237.3))*(Mw/Ma)/(P-(610.78*exp((17.27*(T(5)-273))/(T(5)-273+237.3))));


D_ = G*(f*W6r-W5r)*3600;

fprintf('Production rate is: %i\n', D_);
fprintf('fun1 is: %i\n ',fun(1));
fprintf('fun2 is: %i\n ',fun(2));
fprintf('fun3 is: %i\n ',fun(3));
```

FIGURE 5.2: Code-2

```
fprintf('fun4 is: %i\n ',fun(4));
fprintf('fun5 is: %i\n ',fun(5));


end

Temp - [1,1,1,1,1]*273
end
```

FIGURE 5.3: Code-3

```
 Production rate is: 9.307176e-01
fun1 is: 2.751967e-03
 fun2 is: -7.408847e-04
 fun3 is: -2.772249e-03
 fun4 is: 2.821839e-03
 fun5 is: -1.364242e-12

Equation solved, inaccuracy possible.

The vector of function values is near zero, as measured by the selected value
of the function tolerance. However, the last step was ineffective.



ans =

   47.1616   70.1510   48.0986   43.9583   49.2298
```

FIGURE 5.4: Results for $G = 0.02 kg/hr$

5.3 Verification with referred papers

	Reference paper (℃)	Our Result (℃)
T2	47.4	47.17
T3	68.9	70.16
T4	46.5	48.12
T5	43.4	43.96
T6	49.7	49.23

FIGURE 5.5: Table of comparison

5.4 Our Expectations

```
 Production rate is: 1.019209e+00
fun1 is: 1.714062e-03
 fun2 is: -3.051665e-04
 fun3 is: -1.733633e-03
 fun4 is: 1.762195e-03
 fun5 is: 3.410605e-13

Equation solved, inaccuracy possible.

The vector of function values is near zero, as measured by the selected value
of the function tolerance. However, the last step was ineffective.




ans =

   30.0000   32.0526   29.8503   34.8708   37.9342
```

FIGURE 5.6: Results for $G = 0.2kg/hr$

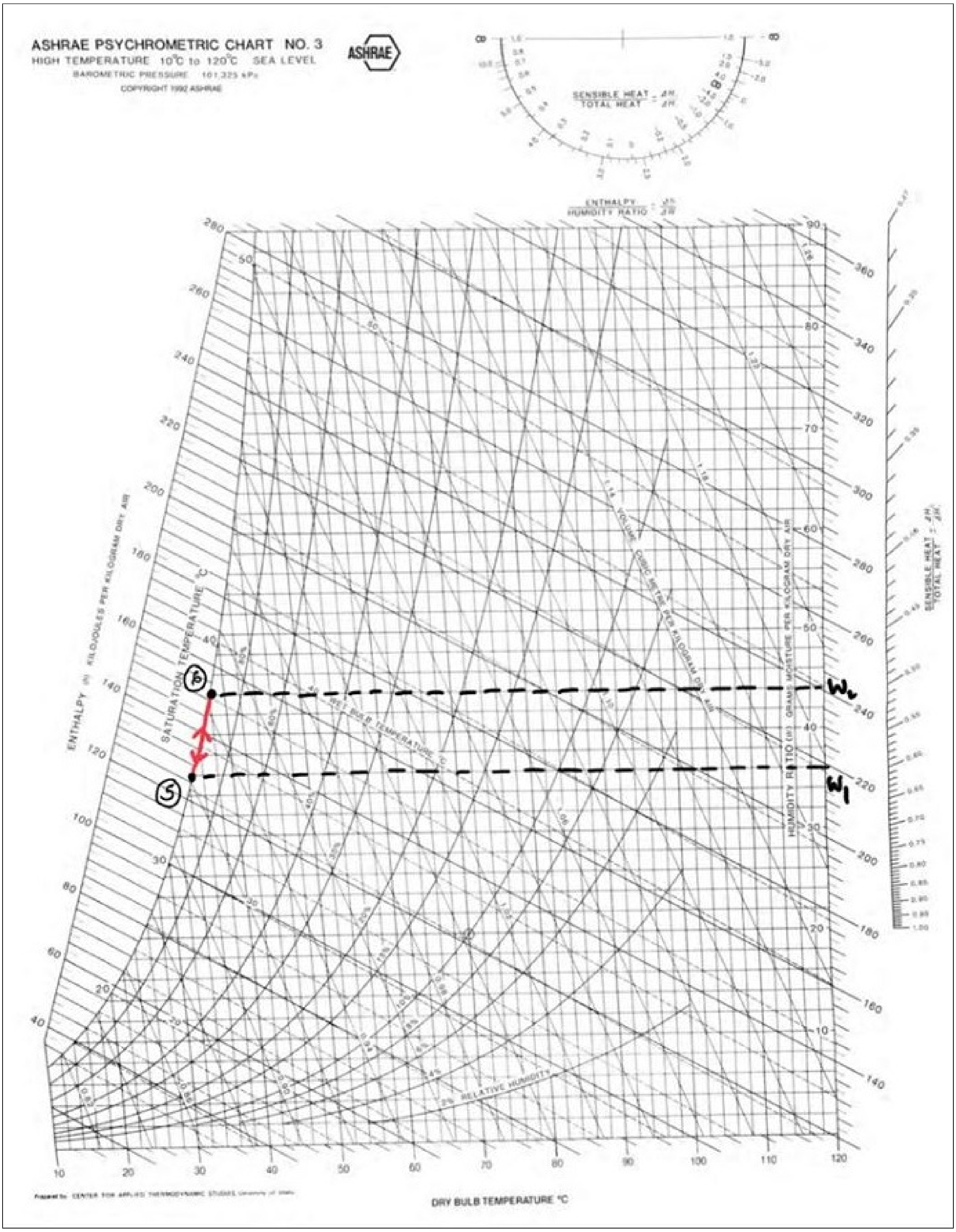


Figure 5.7: Psychrometric Chart demonstrating the process

5.5 Cost analysis

The Estimated upper bounds on the cost of the components involved are as follows:

- Condenser units (4-6 condenser, each of Rs. 1500). Total = Rs. 9000
- Cellulose pads (each unit costs around Rs.300). Total cost = Rs.900
- Pump, Average power of 80W, costs around Rs.1000
- Blower, 85psi, costs around Rs.300
- PVC piping and galvanized steel roughly add up to Rs.1000

Therefore, total cost for the setup adds up to Rs. 12200

5.6 Calculations for pumps and condensers

5.6.1 Calculating overall heat transfer coefficient

Formula for overall heat transfer coefficient:

$$\frac{1}{U} = \frac{1}{h_{air}} + \frac{1}{h_{water}} + \frac{dr}{K_{cu}}$$

$h_{air} = 200W/m^2K$, $K_{cu} = 385W/mK$, $h_{water} = 500W/m^2K$
Substituting these values in the above equation :

$$\frac{1}{U_{cond}} = \frac{1}{200} + \frac{1}{500} + \frac{15 \times 10^{-4}}{385}$$

$$U_{cond} = 142W/m^2K$$

5.6.2 Calculating Fin pitch using Bond number:

Minimum distance between 2 Fins such that water bubble doesn't coalesce is given by the expression :

$$Bo = \frac{gL^2(\rho_L - \rho_G)}{\sigma}$$

where Bo = Bond number, σ = Inter-facial surface tension, ρ_L = density of liquid, ρ_G = density of the surrounding gas.

For our case: $\sigma = 72.8 \times 10^{-3}N/m$, $\rho_L = 997kg/m^3$, $\rho_a = 1.3kg/m^3$

Taking Bond number to be 1.5

$$1.5 = \frac{9.8 \times (997 - 1.3) \times D^2}{72.8 \times 10^{-3}}$$

$$D = 3.34mm$$

This is the minimum distance between fins to be placed such that bubble doesn't get coalesce and gets condensed. Considering factor of safety to be two, we should have a minimum distance between fins to be around 7 to 8mm.

5.6.3 Specification of One heat exchanger:

- Dimensions of heat exchanger : 22.5 cm X 23 cm X 4 cm
- Number of fins = 59
- Copper pipe inner diameter= 0.5 cm, Outer diameter = 1 cm.
- Total Length of Copper pipe = $17 \times 23 = 391cm$
- Area of 1 fin = 78.655 cm^2
- Total Area of Fins in 1 unit = 0.4640 m^2
- Area of pipe= 0.1201 m^2
- Total Area of 1 unit of condenser= 0.5841 m^2

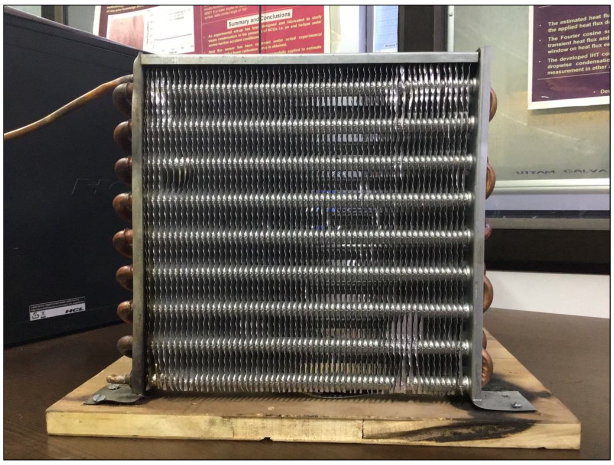


FIGURE 5.8: A condenser unit

Area of 1 fin :

$$A_{fin} = 4 \times 23 - \frac{17 \times \pi \times 1^2}{4} = 78.655cm^2$$

Area of 59 fins :

$$A_{total} = 59 \times 78.655 = 0.4640m^2$$

Area of pipe :

$$A_{pipe} = \pi DL \times 17 = 17 \times 3.14 \times 1 \times 22.5 = 0.1201m^2$$

Thus, the total surface area of fin :

$$A_s = A_{pipe} + A_{fin} = 0.5841m^2$$

5.6.4 Sample calculation of fin efficiency:

Now, considering these dimensions of fins and C_u tubes :

- d_o = diameter of Copper tube =9.52 mm.
- D_o = outer diameter of fin = 381 mm.
- L = length of Copper tube = 30 m.
- t = thickness of fin = 1 mm.
- h = length of fin = 14.29 mm.
- U = heat transfer coefficient = 142 W/m^2
- α_i = Thermal conductivity of Cu tube = 385 W/mK

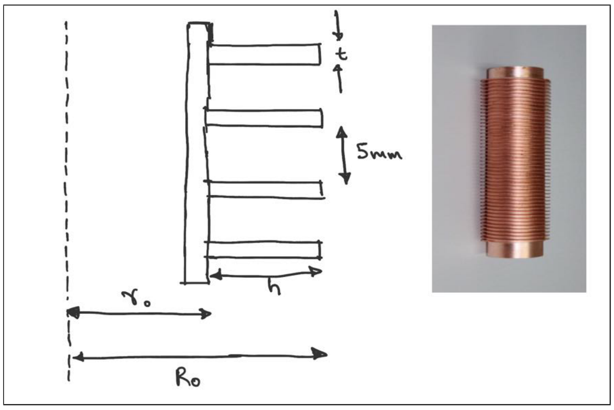


FIGURE 5.9: Fin design

For annular fin design, efficiency is calculated by solving the Bessel equation:

$$\eta_r = \frac{tanh(mh\psi)}{mh\psi}$$

$$\psi = 1 + 0.35ln(1 + \frac{h}{r_o}) = 1 + 0.35ln(1 + \frac{14.29}{4.76}) = 1.48$$

Calculation of value of m:

$$m = \sqrt{\frac{2U}{\alpha_i t}}$$

$$m = \sqrt{\frac{2 \times 142}{385 \times 10^{-3}}} = 27.16$$

Now calculating final efficiency

$$\eta_r = \frac{tanh(mh\psi)}{mh\psi}$$

$$\eta_r = \frac{tanh(27.16 \times 14.29 \times 10^{-3} \times 1.48)}{27.16 \times 14.29 \times 10^{-3} \times 1.48} = 90.28\%$$

5.6.5 Pumping power requirement calculations:

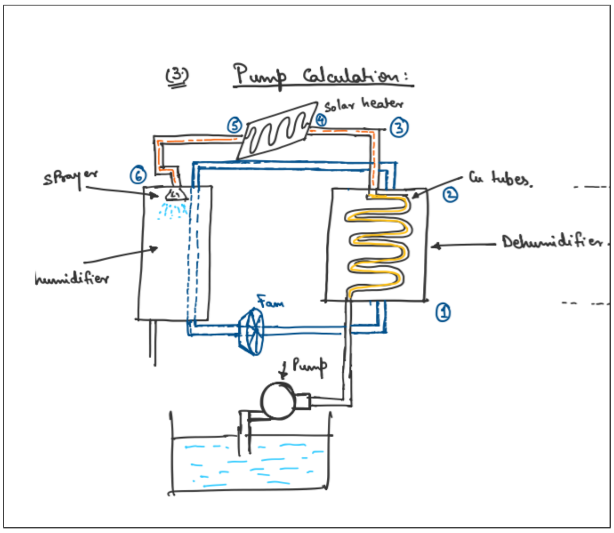


FIGURE 5.10: Overall system

Applying Bernoulli's equation between 1 and 6: Pressure difference between Water sprinkler/sprayer= 30 psi = 2 bar.

$$h_p = \frac{\Delta p}{\rho g} + \Delta z + h_f$$

where $\Delta z = 0.7m, \Delta p = 10^5 Pascal$, Flow rate of water=$10^{-4}m^3/s$.

Calculation of head loss h_f for C_u tubing

$$h_f = \frac{fLV^2}{2gD}$$

Calculation of velocity:

$$V = \frac{4 \times 10^{-4}}{\pi \times 0.25 \times 10^{-4}} = 5.09m/s$$

Calculating Reynolds number for the flow:

$$R_e = \frac{\rho V D}{\mu}$$

$$R_e = \frac{10^3 \times 5.09 \times 0.5}{8.90 \times 10^{-2}} = 28610$$

$\mu_{water} = 8.9 \times 10^{-4} Pa.s$, Relative roughness of C_u $pipe = \frac{e}{D} = 10^{-4}$

From Moody's chart f = 0.02.

Frictional head loss through *Copper pipe* for six dehumidifiers:

$$h_f = \frac{fLV^2}{2gD} = \frac{0.02 \times 6 \times 0.391 \times 5.09^2}{2 \times 9.8 \times 0.5 \times 10^{-2}} = 12.414m$$

Applying Bernoulli's equation across points 6 and 1.

$$h_p = \frac{\Delta P}{\rho g} + \Delta Z + h_f$$

$$h_p = \frac{10^5}{1000 \times 9.8} + 0.7 + 12.414 = 23.314m$$

Total power requirement to pump across 1 to 2:

$$P_{pump} = \rho g h_p \times Q = 10^{-4} \times 10^3 \times 9.8 \times 23.314 = 22.8W$$

Total power requirement:

Assuming other electronics will consume 10 W (this includes sensors like Thermocouple, Humidity sensor, Pressure sensor etc.) Blower has power rating of 28W. So,

$$P_{total} = 10 + 28 + 22.8 = 60.84W$$

5.7 CFD Simulation of Diffuser

Since we are using a fan of $12cm \times 12cm$, and we have to blow it through a circular cross-section of diameter $8in$, including a diffuser to increase the area of flow is necessary. Flow simulation facility provided in SolidWorks was used to simulate the flow to check whether flow separation was happening in the diffuser or not.

Diffuser has inlet diameter of $3in$, and outer diameter of $8.5in$. We had two options, either to have the hot air outlet on the side wall of Humidifier casing or at the top. The two cases resulted in diffuser of length $6in$ and $15in$ respectively. Following are the images of Flow simulation done on SolidWorks.

5.7.1 Boundary Conditions

We are using a fan of 85cfm capacity which is about $0.04m^3/s$. So inlet boundary condition has volume flow rate $0.04m^3/s$. The outlet has been given a static pressure which equals atmospheric pressure. The walls are having surface roughness of $30\mu m$.

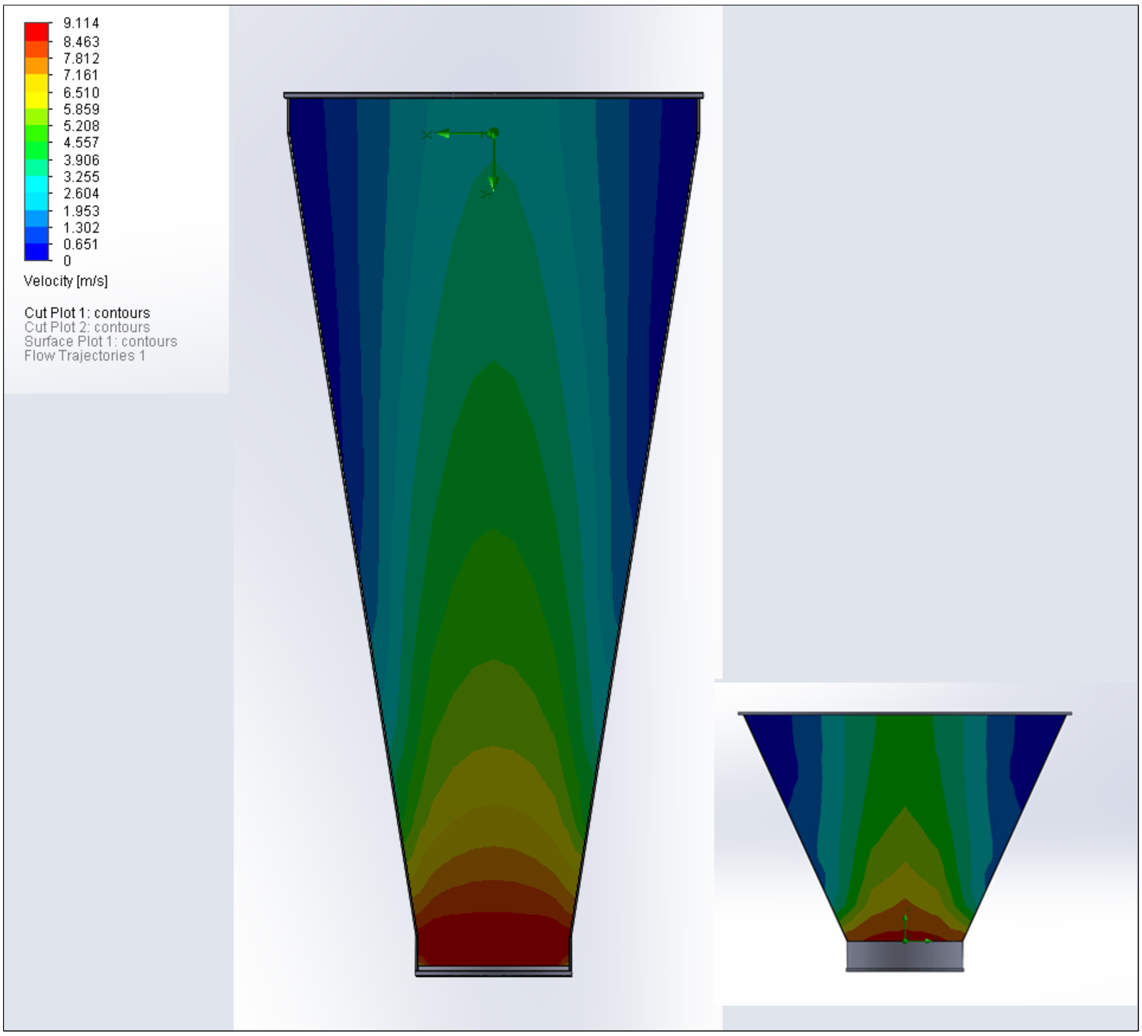


FIGURE 5.11: Velocity contour plot

5.7.2 $k - \epsilon$ Turbulence model

For simulation, we have chosen $k - \epsilon$ model [5]. Turbulent energy $k = \frac{3}{2}(UI)^2$. Turbulent intensity at the core of the pipe for fully developed flow is given by $I = 0.16Re_{d_h}^{-\frac{1}{s}}$, where Re_{d_h} is the Reynolds number for a pipe with hydraulic diameter d_h. U is the mean flow velocity. The turbulent dissipation rate ϵ is calculated using the expression $\epsilon = C_\mu^{\frac{3}{4}} \frac{k^{\frac{3}{2}}}{l}$, where C_u is turbulent model constant and is usually equal to 0.09. l is the turbulence length scale and describes the size of large energy-containing eddies in a turbulent flow. For a fully developed pipe $l = 0.07d_h$. The final results are : $k = 32.85J/kg$ and $\epsilon = 5737W/kg$.

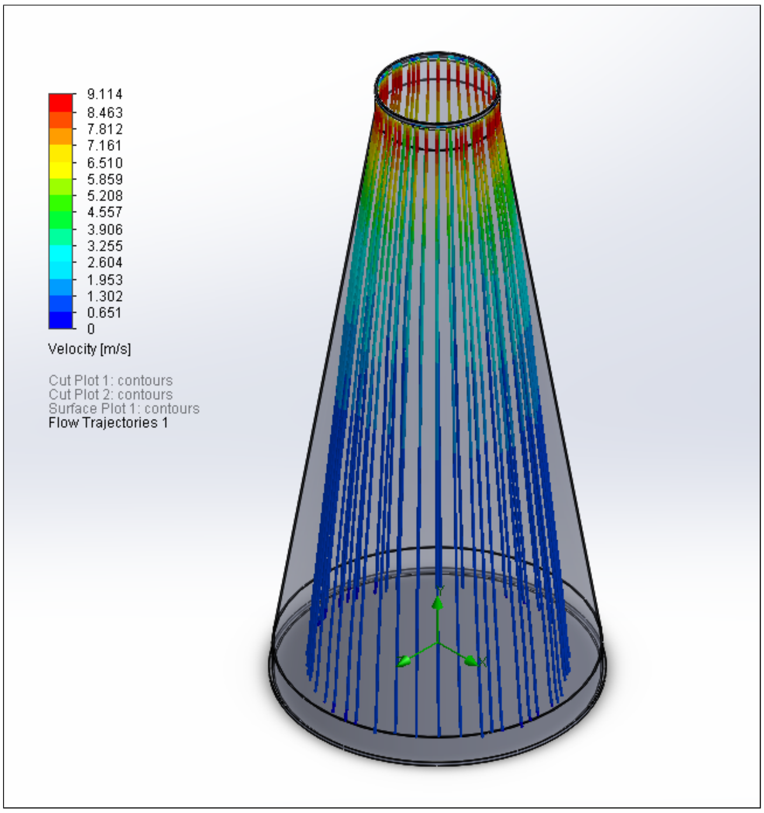


FIGURE 5.12: Flow trajectory plot

From the velocity contours, we can observe that shorter diffuser does not spread the air throughout the cross-section. Hence, we decided to go with the longer diffuser.

Appendix A

Handwritten Calculations

The next few pages contain handwritten calculations for condenser parameters and also the pump calculations.

Formulae for overall heat transfer Coefficient :

$$\boxed{\frac{1}{U_{cond}} = \frac{1}{h_{air}} + \frac{1}{h_{water}} + \frac{dr}{K_{cu}}}$$

- $h_{air} = 200\ W/m^2 K$.
- $K_{cu} = 385\ W m^{-1} K^{-1}$
- $h_{water} = 500\ W/m^2 K$

$$\therefore \quad \frac{1}{U_{cond}} = \frac{1}{200} + \frac{1}{500} + \frac{15 \times 10^{-4}}{385}$$

$$\Rightarrow \quad \boxed{U_{cond} = 142\ W/m^2 K}$$

Btp-3

Monday, 9 September 2019 10:28 AM

Calculation of Fin pitch using Bond number

- Min. distance b/w 2 fins such that water bubble doesn't collaesce.

$$\boxed{\text{Bond number} = \frac{g L^2 \times (\rho_L - \rho_G)}{\sigma}}$$

" σ " = Interfacial surface tension.
" ρ_L " = density of liquid.
" ρ_G " = density of surrounding gas.

Now, in our Case :

- $\sigma = 72.8 \times 10^{-3}\ N/m$
- $\rho_L = 997\ Kg/m^3$
- $\rho_a = 1.3\ Kg/m^3$

Taking Bond number to be 1.5

$$1.5 = \frac{9.8 \times (997 - 1.3) D^2}{72.8 \times 10^{-3}}$$

$$\boxed{D = 3.34 \text{ mm}}$$

This is the minimum distance b/w fins to be placed such that bubble doesn't get collasce & gets condensed.

- So on safer side we will place fins at a distance of 5mm gap.

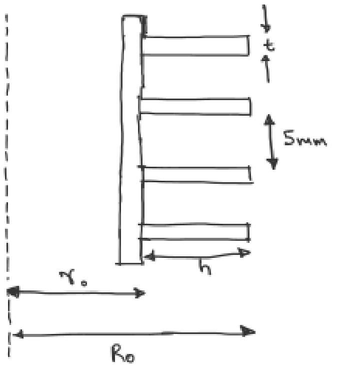
Calculation of fin efficiency.

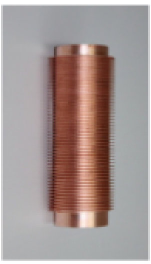
Now, considering the dimensions of fins & Cu tubes:

- d_o = diameter of Cu tube = 9.52 mm
- D_o = outer diameter of fin = 38.1mm.
- l = length of Cu tube = 30m.
- t = thickness of fin = 1mm.
- h = 14.29 mm

r_o = 4.76 mm (radius of Cu tube)
R_o = 19.05 mm. (radius of Annular fins)
U = heat transfer coefficient = 142 W/m²K
α_i = Thermal conductivity of Cu tube = 385 $W m^{-1} K^{-1}$
h = 14.29 mm

So our annular fin looks like this:





For a Annular fin design, efficiency is calculated by solving the bessel equation:

$$\boxed{\eta_R = \frac{\tanh(mh\psi)}{mh\psi}}$$

$$\boxed{\psi = 1 + 0.35 \ln\left(1 + \frac{h}{r_o}\right)}$$

Calculating: $\psi = 1 + 0.35 \times \ln\left(1 + \frac{14.29}{4.76}\right)$

$$\boxed{\psi = 1.48}$$

$$\boxed{m = \sqrt{\frac{2U}{\alpha_i \times t}} = \sqrt{\frac{2 \times 142}{385 \times 10^{-3}}} = 27.16}$$

$$\Rightarrow \quad \eta_R = \frac{\tanh(mh\psi)}{mh\psi}$$

$$\eta_R = \frac{\tanh(27.16 \times 14.29 \times 10^{-3} \times 1.48)}{27.16 \times 14.29 \times 10^{-3} \times 1.48}$$

$$\boxed{\eta_R = \frac{0.5185}{0.5744} = 90.28\,\%}$$

→ Specifications of Heat Exchanger:

- Dimensions of heat exchanger : 22.5 cm × 23 cm × 4 cm
- Number of fins = 59.
- Cu pipe inner diameter = 0.5 cm.
  Outer diameter = 1 cm.
  Total length of Cu pipe = 17 × 23
  = 391 cm.

→ Calculation of Area.

$$\text{Area of 1 fin} = 4 \times 23 - \frac{17 \times \pi \times 1^2}{4}$$

$$\text{Area} = 92 - 13345$$

$$\Rightarrow \boxed{\text{Area}_{1\,\text{fin}} = 78.655\ \text{cm}^2}$$

$$\Rightarrow \text{Area of 59 fins} = 59 \times 78.655$$

$$\boxed{A_{\text{Total}}|_{\text{fin}} = 0.4640\ \text{m}^2}$$

$$\Rightarrow \text{Area of pipe} = (\pi d l) \times 17$$

$$= 17 \times 3.14 \times 1 \times 22.5$$

$$\boxed{A_{\text{pipe}} = 0.1201\ \text{m}^2}$$

$$\boxed{\begin{aligned}\text{Total Area} &= A_{\text{pipe}} + A_{\text{fin}} \\ &= 0.5841\ \text{m}^2\end{aligned}}$$

[+] Power Calculations:

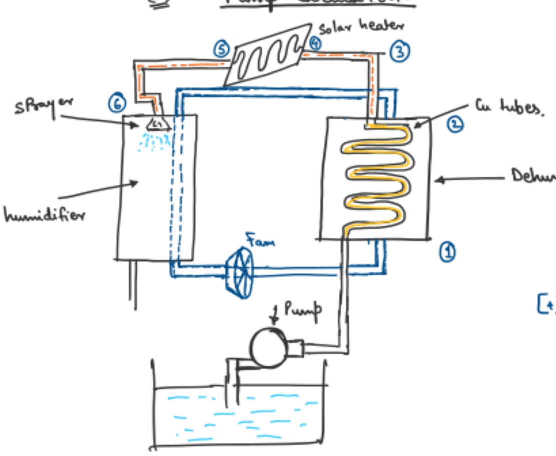
(1) Water sprinkler / sprayer:

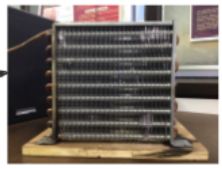
- 30 psi ≈ 2 bar.

(2) Blower / Fan:

- 28 Watts.

(3) Pump Calculation:





[+] Pumping Power required:

Bernoullie eqn: across point ⑥ & point ①

$$\boxed{h_p = \frac{\Delta P}{\rho g} + \Delta Z + h_f}$$

$$\boxed{\text{Flow rate of water inside Cu tube} = 1\times10^{-4}\ m^3/s}$$

$\Delta Z = 0.7\ m$

$\Delta P \approx 10^5$ Pascal (2 bar - 1 bar)

$$\boxed{\text{"}h_f\text{" for Cu tubing} : \frac{fLV^2}{2gD}}$$

- $V = \dfrac{1\times10^{-4}\ m^3/s}{\frac{\pi \times (0.5\times10^{-2})^2}{4}} = 5.09\ m/s$

- $\mu_{water} = 8.90\times10^{-4}\ Pa$ ⇒ $Re = \dfrac{\rho V D}{\mu} = \dfrac{10^3 \times 5.09 \times 0.5\times10^{-2}}{8.90\times10^{-4}}$

$Re = 28610$

Now, Relative roughness for Cu pipe $= \frac{\varepsilon}{D} = 10^{-4}$

⇒ from Moody's chart $f = 0.02$

⇒ frictional head loss through Cu tube

$$\boxed{h_f = \frac{fL'V^2}{2gD} = \frac{0.02 \times 6 \times 0.391 \times (5.09)^2}{2 \times 9.8 \times 0.5\times10^{-2}} = 12.414}$$

For 6 dehumidifiers:

$L' = 6 \times L =$

⇒ $h_p = \dfrac{\Delta P}{\rho g} + \Delta Z + h_f = \underbrace{\dfrac{10^5}{1000\times9.8}}_{10.20} + 0.7 + 12.414$

⇒ $\boxed{h_p = 23.314}$

⇒ Total power req. to pump across points ① & ②:

Power $= \rho g h_p Q$

$= 1\times10^{-4} \times 10^3 \times 9.8 \times 23.314$

$\boxed{P = 22.8\ W}$

[+] Total Power requirement

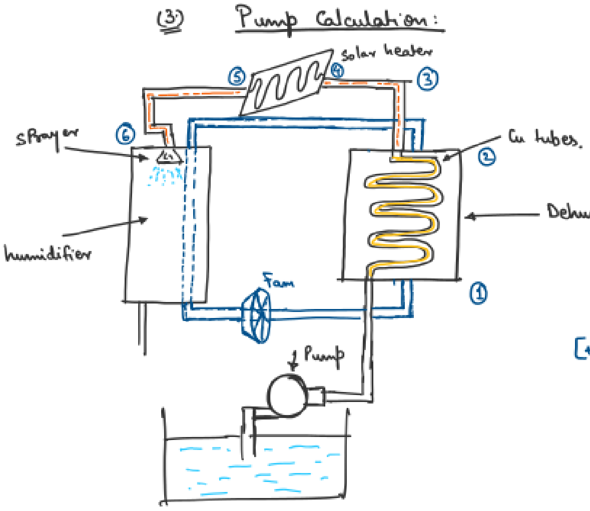
- Electronics ≃ 10 W (Thermocouple, Humidity Sensor, Pressure Sensor)

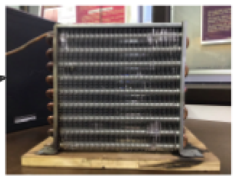
- Blower / fan ≃ 28 W

- Total Power = 10 + 28 + 22.8 W

$$\boxed{P_{Total} = 60.84\ W}$$

(3) Pump Calculation:





[+] Pumping Power required: Bernoullie eqn across point ⑥ & point ①

$$\boxed{h_p = \frac{\Delta P}{\rho g} + \Delta Z + h_f}$$

$$\boxed{\text{Flow rate of water inside Cu tube} = 2\times10^{-4}\ m^3/s}$$

$\Delta Z = 0.7\ m$

$\Delta P \approx 10^5$ Pascal (2 bar - 1 bar)

$$\boxed{\text{"}h_f\text{" for Cu tubing} : \frac{fLV^2}{2gD}}$$

h_f → frictional head loss

- $V = \dfrac{2\times10^{-4}\ m^3/s}{\dfrac{\pi * (0.5\times10^{-2})^2}{4}} = 10.185\ m/s$

- $\mu_{water} = 8.90\times10^{-4}$ Pa ⇒

$$\boxed{Re = \frac{\rho V D}{\mu} = \frac{10^3 \times 10.185 \times 0.5\times10^{-2}}{8.90\times10^{-4}} \qquad Re = 57220}$$

Now, Relative roughness for Cu pipe $= \dfrac{\varepsilon}{D} = 10^{-4}$

⇛ from Moody's chart $f = 0.02$

⇛ frictional head loss through Cu tube

$$\boxed{h_f = \frac{fLV^2}{2gD} = \frac{0.02 \times 0.391 \times (10.185)^2}{2 \times 9.8 \times 0.5 \times 10^{-2}} = 8.277\,m.}$$

⇛

$$h_p = \frac{\Delta P}{\rho g} + \Delta Z + h_f = \frac{10^5}{1000 \times 9.8} + 0.3 + 8.277$$

10.20

⇛ $\boxed{h_p = 18.781\,m}$

⇛ Total power req. to pump across points ① & ② :

$$\text{Power} = \rho g h_p Q$$

$$= 2 \times 10^{-4} \times 10^3 \times 9.8 \times 18.781$$

$$\boxed{P = 36.810\ W}$$

[+] Total Power requirement

- Electronics ≃ 10 W (Thermo couple, Humidity Sensor, Pressure Sensor)
- Blower / fan ≃ 28 W

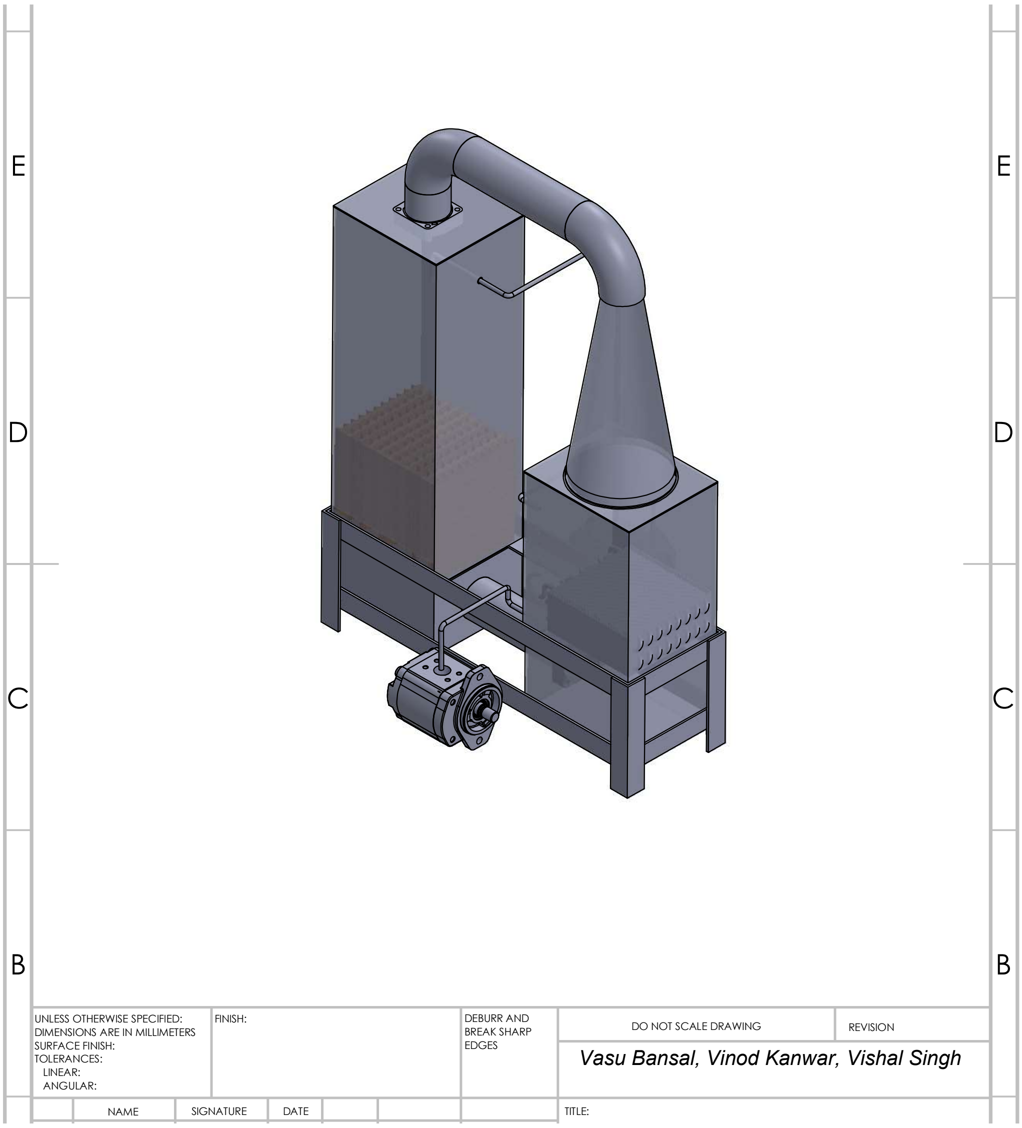
- Total Power = 10 + 28 + 36.810 W

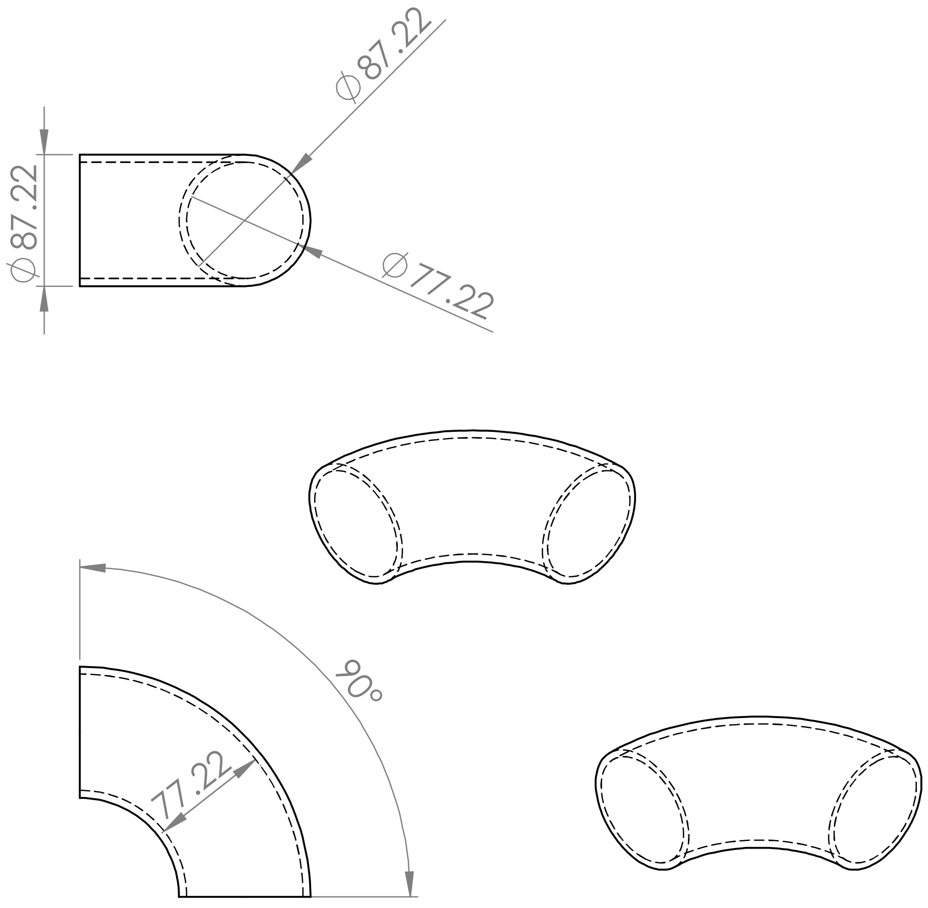
$$\boxed{P_{Total} = 74.810\ W}$$

Appendix B

Drawings

The next few pages contain drawings for the parts which are going to be manufactured.


E
D
C
B
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND
BREAK SHARP
EDGES
DO NOT SCALE DRAWING
REVISION
Vasu Bansal, Vinod Kanwar, Vishal Singh
NAME
SIGNATURE
DATE
TITLE:



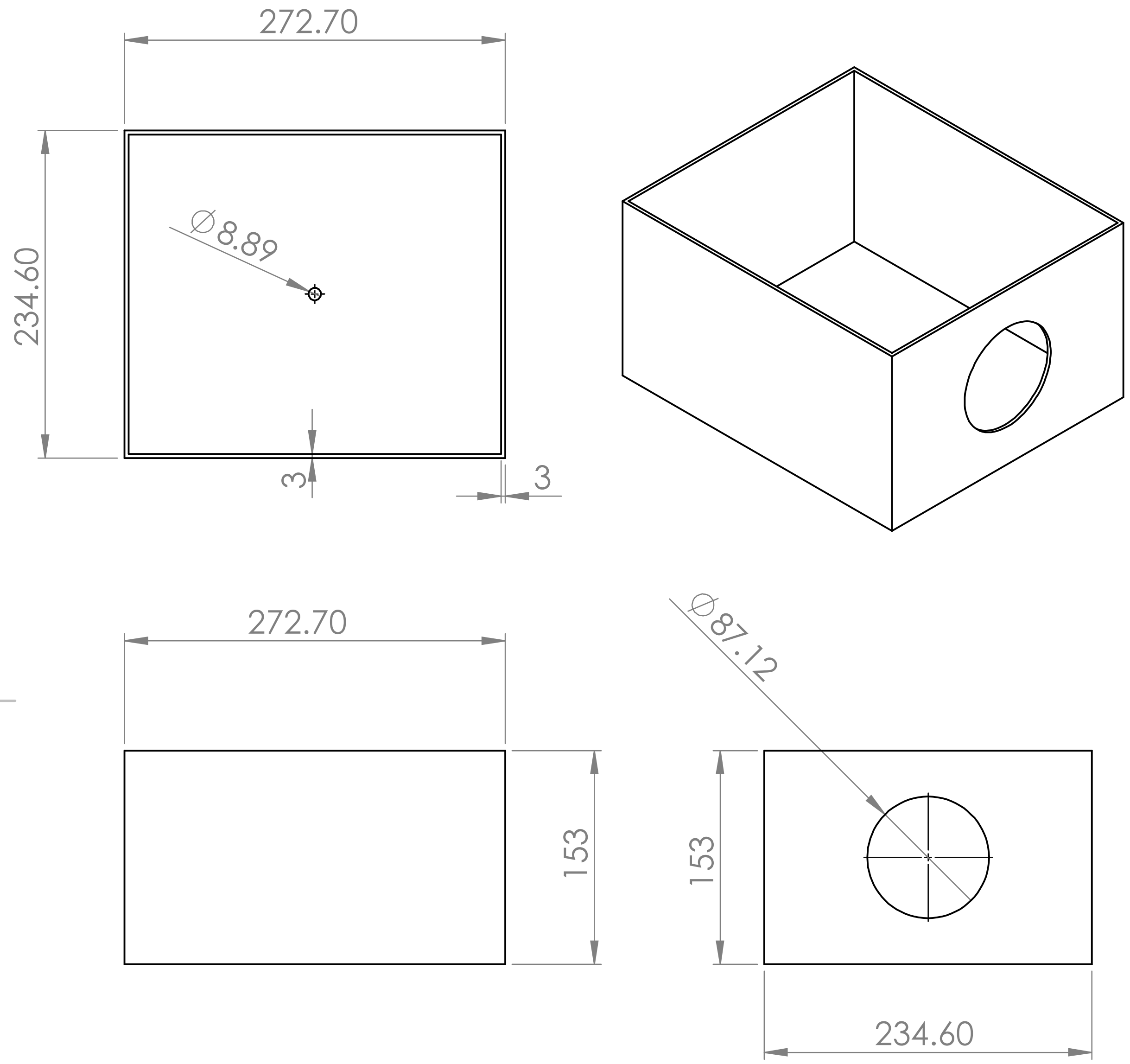
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING *1 unit*	REVISION

	NAME	SIGNATURE	DATE		
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A				MATERIAL: *PVC pipe*	

TITLE: *Vasu Bansal, Vinod Kanwar, Vishal Singh*

DWG NO. **Bend**

A4



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING
			REVISION
			2 units

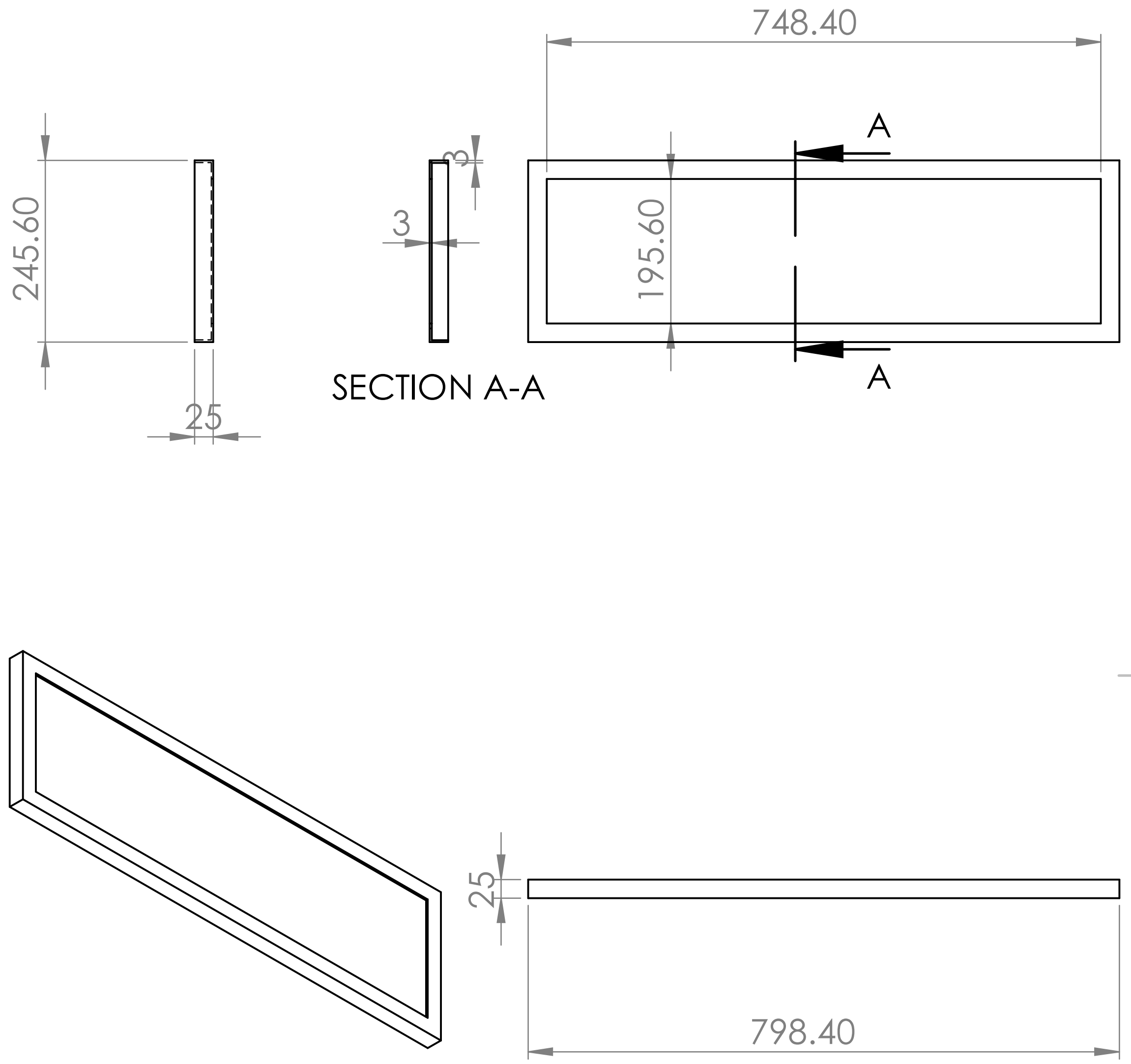
	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL: *MS Sheet*

TITLE: *Vasu Bansal, Vinod Kanwar, Vishal Singh*

DWG NO. Collector

A4



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

2 units

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Vasu Bansal, Vinod Kanwar, Vishal Singh

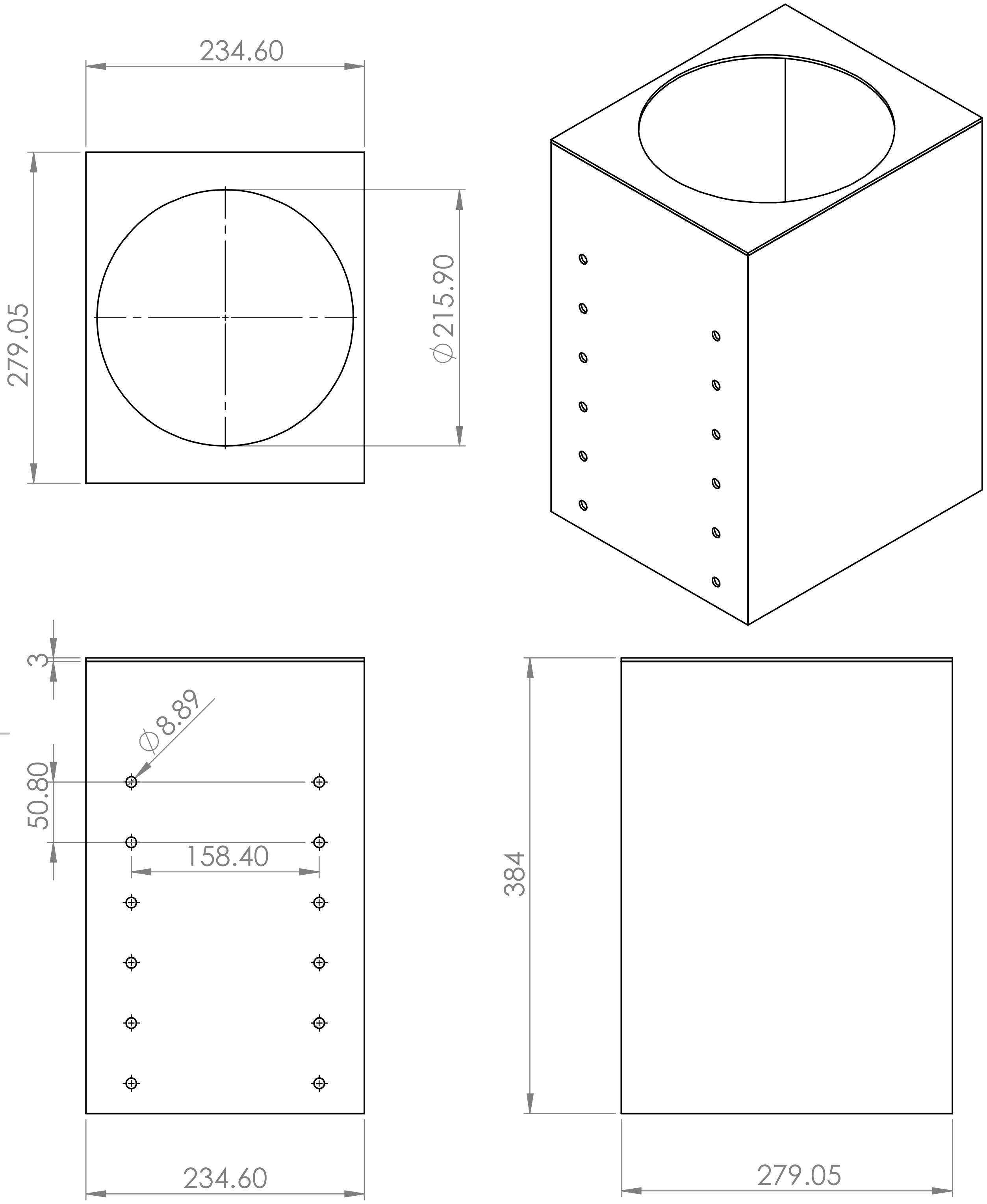
MATERIAL:

MS sheet

DWG NO.

Container frame

A4



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

1 unit

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Vasu Bansal, Vinod Kanwar, Vishal Singh

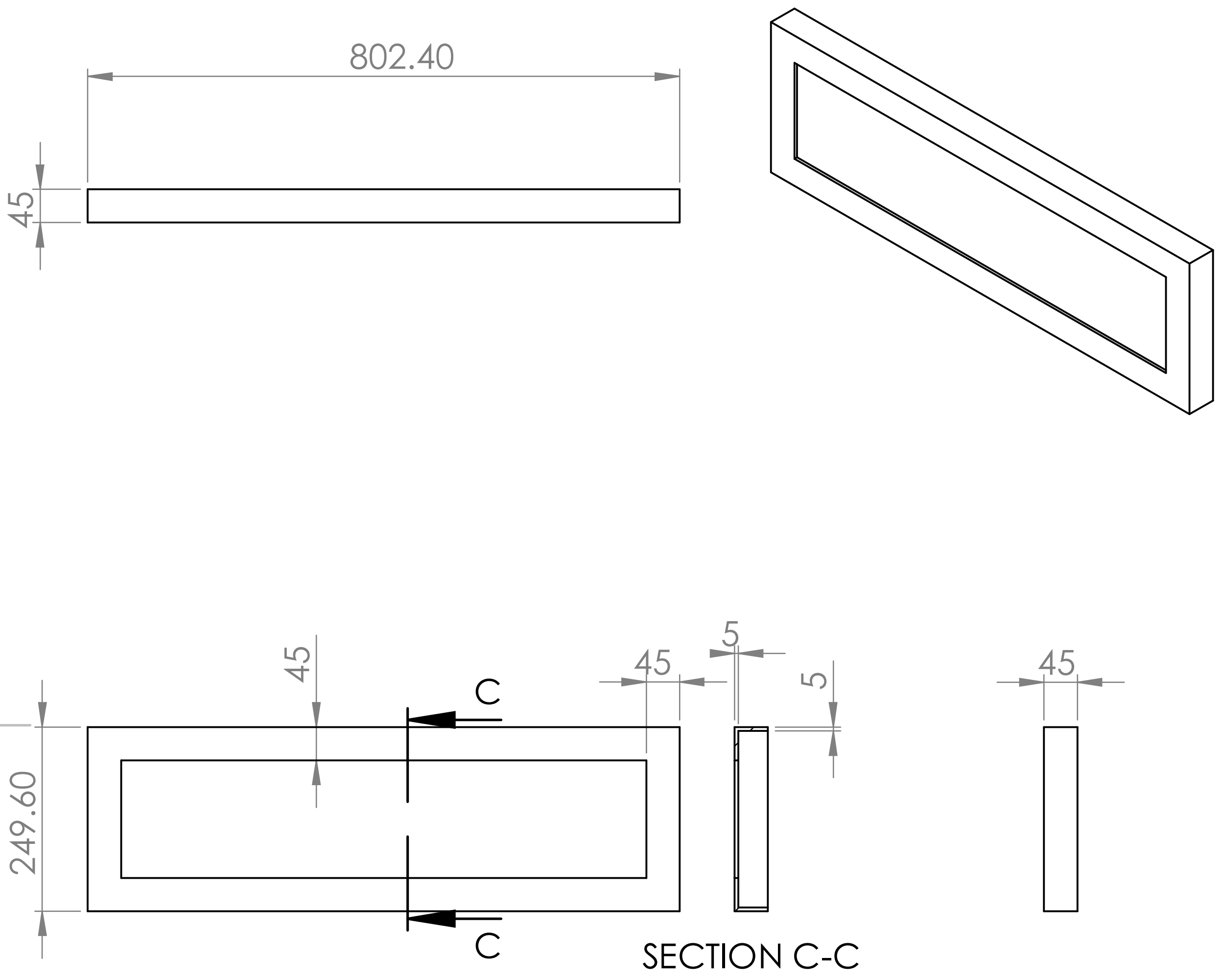
MATERIAL:

MS sheet

DWG NO.

Dehumidifer Casing

A4



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

1 unit

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Vasu Bansal, Vinod Kanwar, Vishal Singh

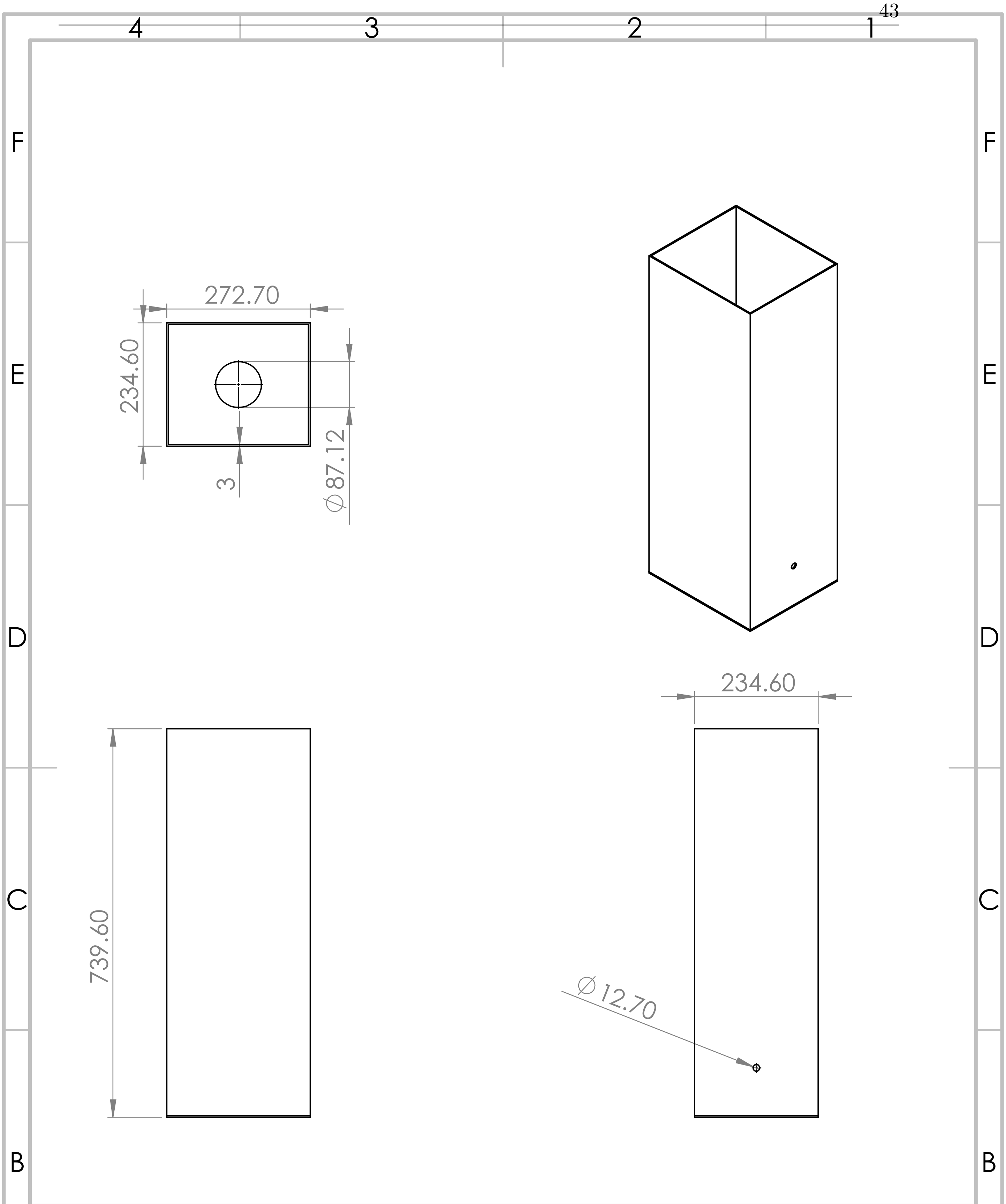
MATERIAL:

MS sheet

DWG NO.

Frame 2

A4



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

1 unit

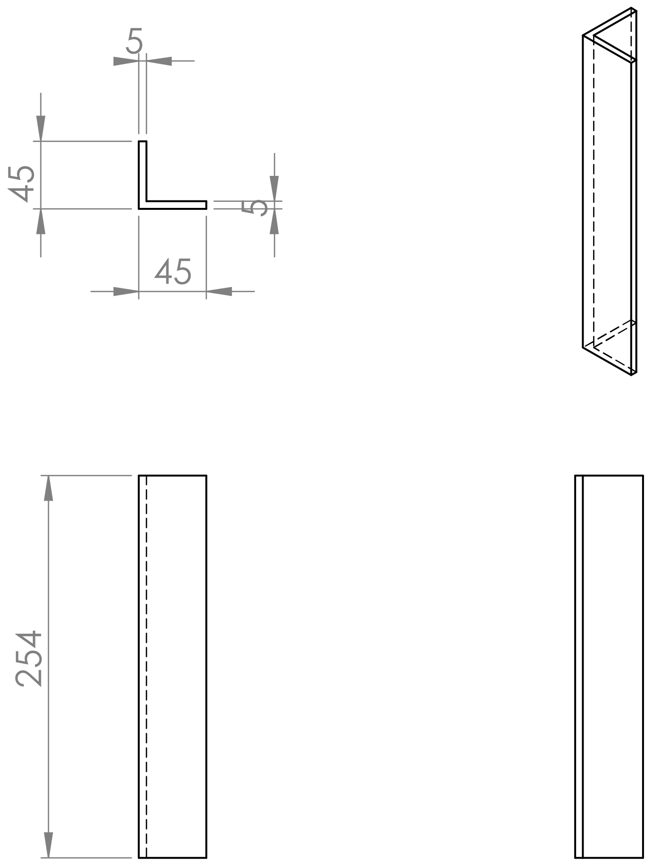
	NAME	SIGNATURE	DATE		
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A					

MATERIAL:
MS sheet

TITLE:
Vasu Bansal, Vinod Kanwar, Vishal Singh

DWG NO.
Humidifier Casing

A4



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

4 units

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Vasu Bansal, Vinod Kanwar, Vishal Singh

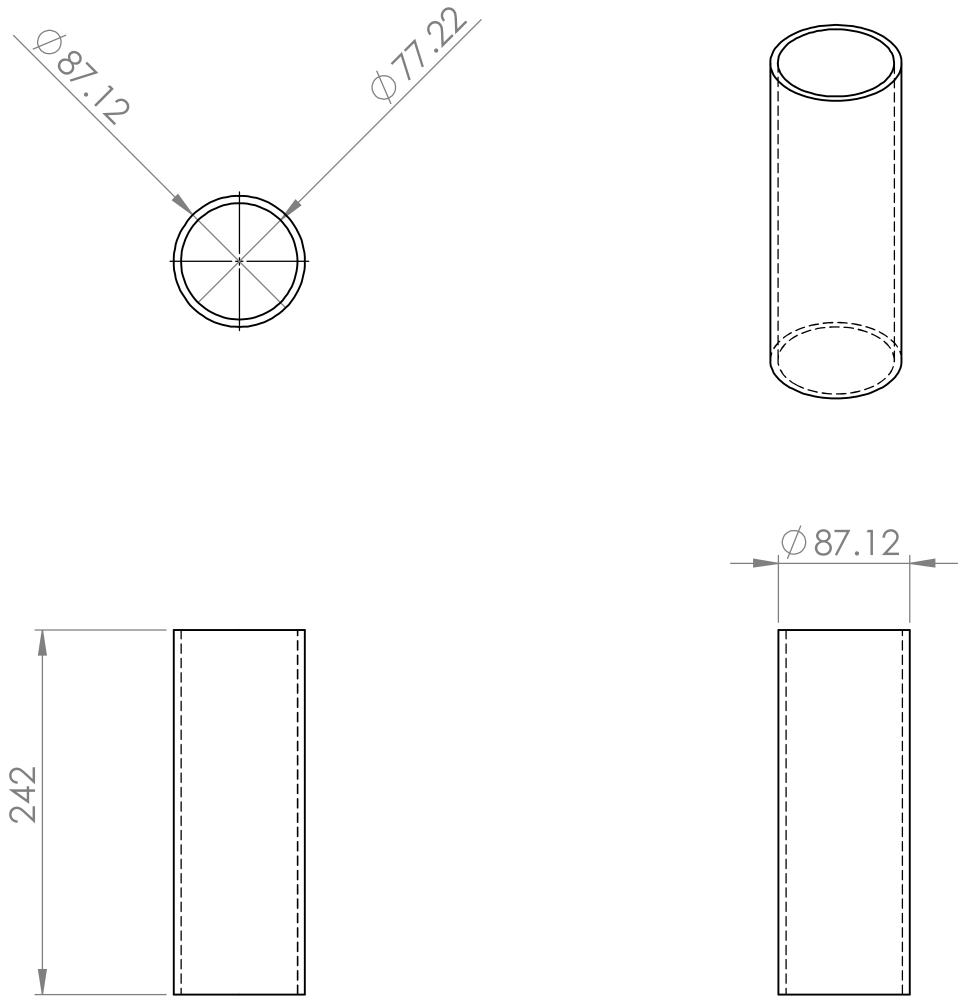
MATERIAL:

MS sheet

DWG NO.

Leg

A4



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

1 unit

	NAME	SIGNATURE	DATE		
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A					

TITLE:

Vasu Bansal, Vinod Kanwar, Vishal Singh

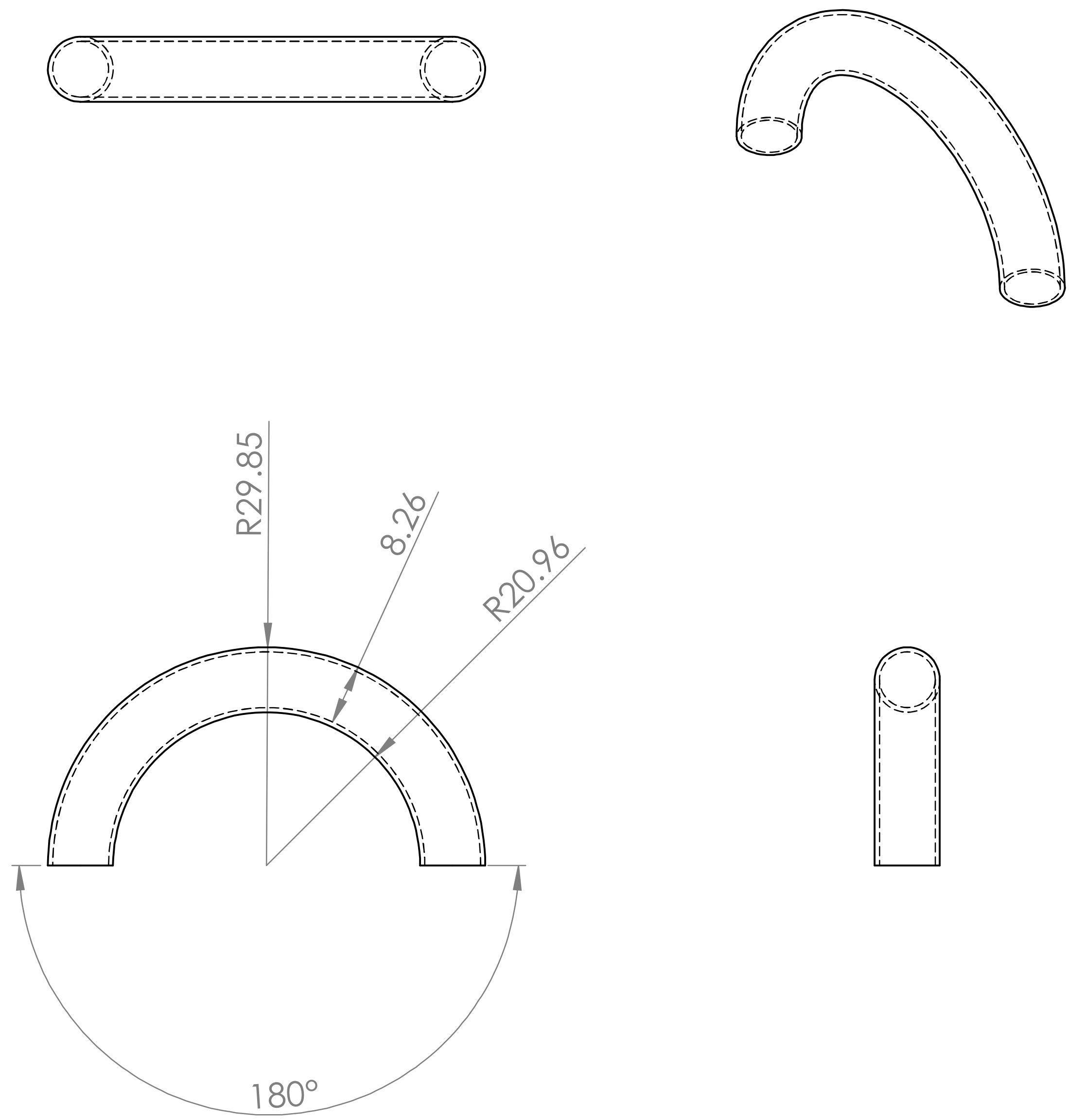
MATERIAL:

PVC pipe

DWG NO.

Pipe1

A4



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

1 unit

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:
PVC pipe

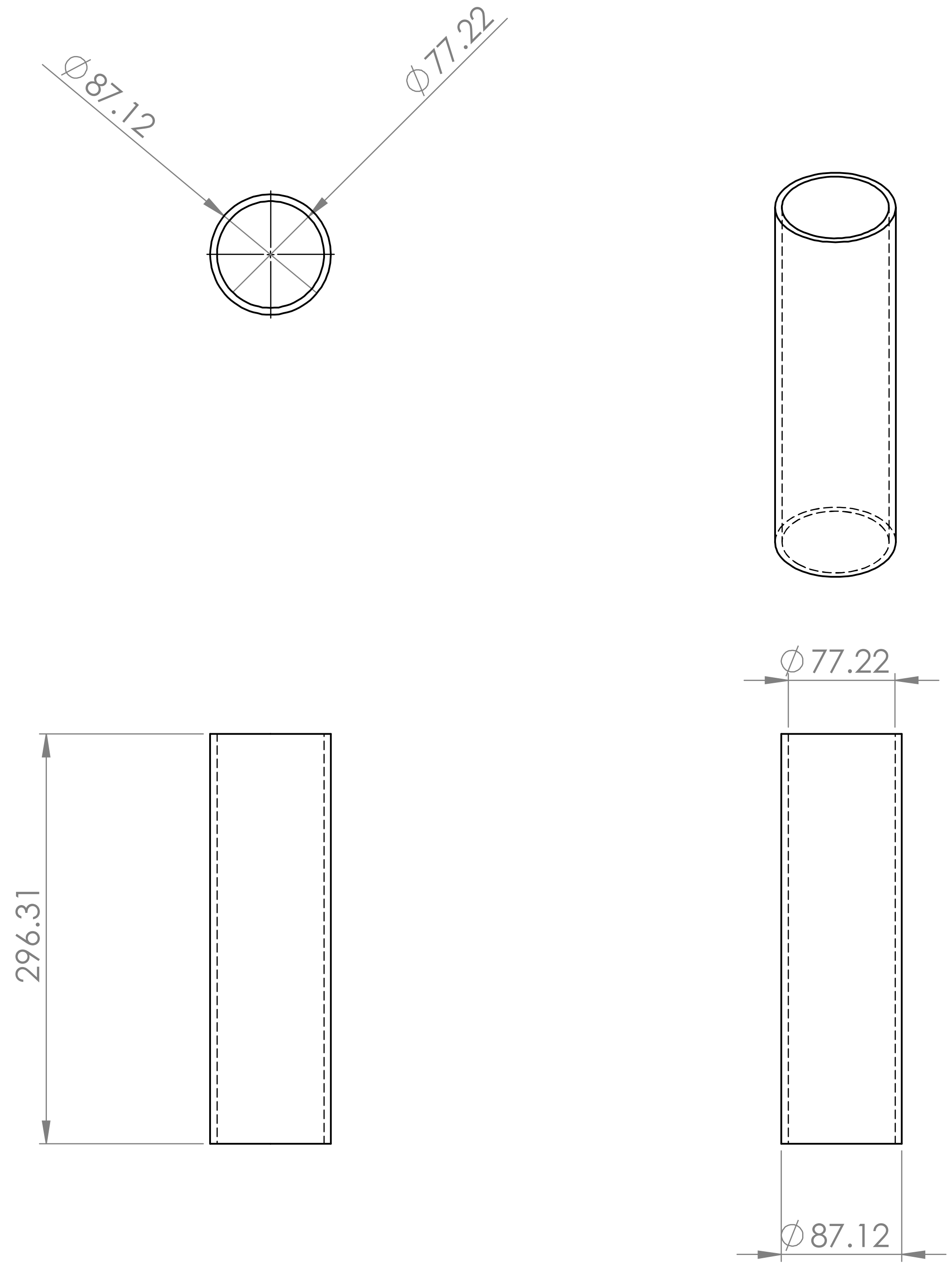
TITLE:

Vasu Bansal, Vinod Kanwar
Vishal Singh

DWG NO.

Tube Bend

A4



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING / REVISION / *1 unit*

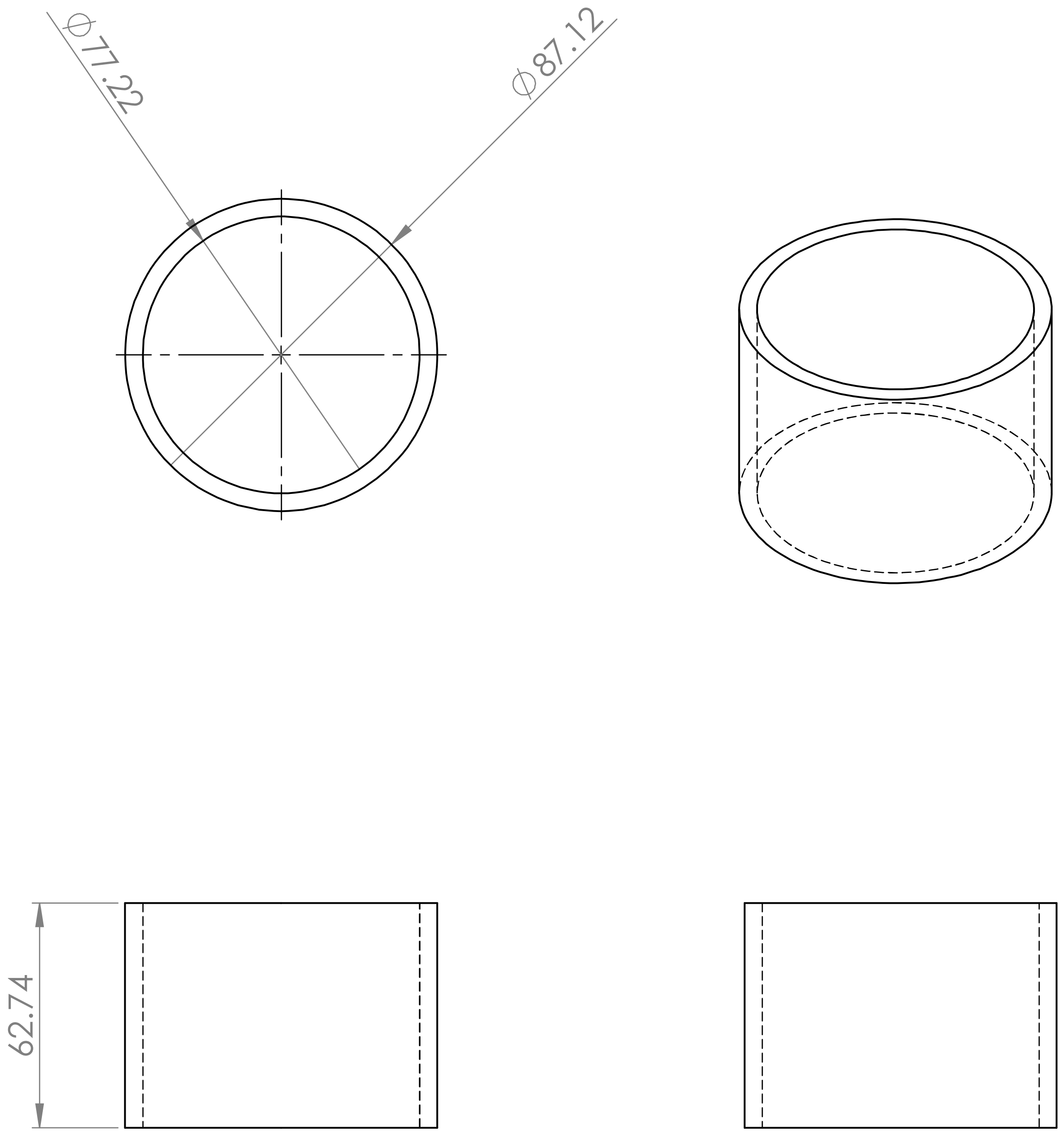
	NAME	SIGNATURE	DATE		
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A					

TITLE: *Vasu Bansal, Vinod Kanwar, Vishal Singh*

MATERIAL: *PVC pipe*

DWG NO. Pipe2

A4



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

1 unit

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Vasu Bansal, Vinod Kanwar, Vishal Singh

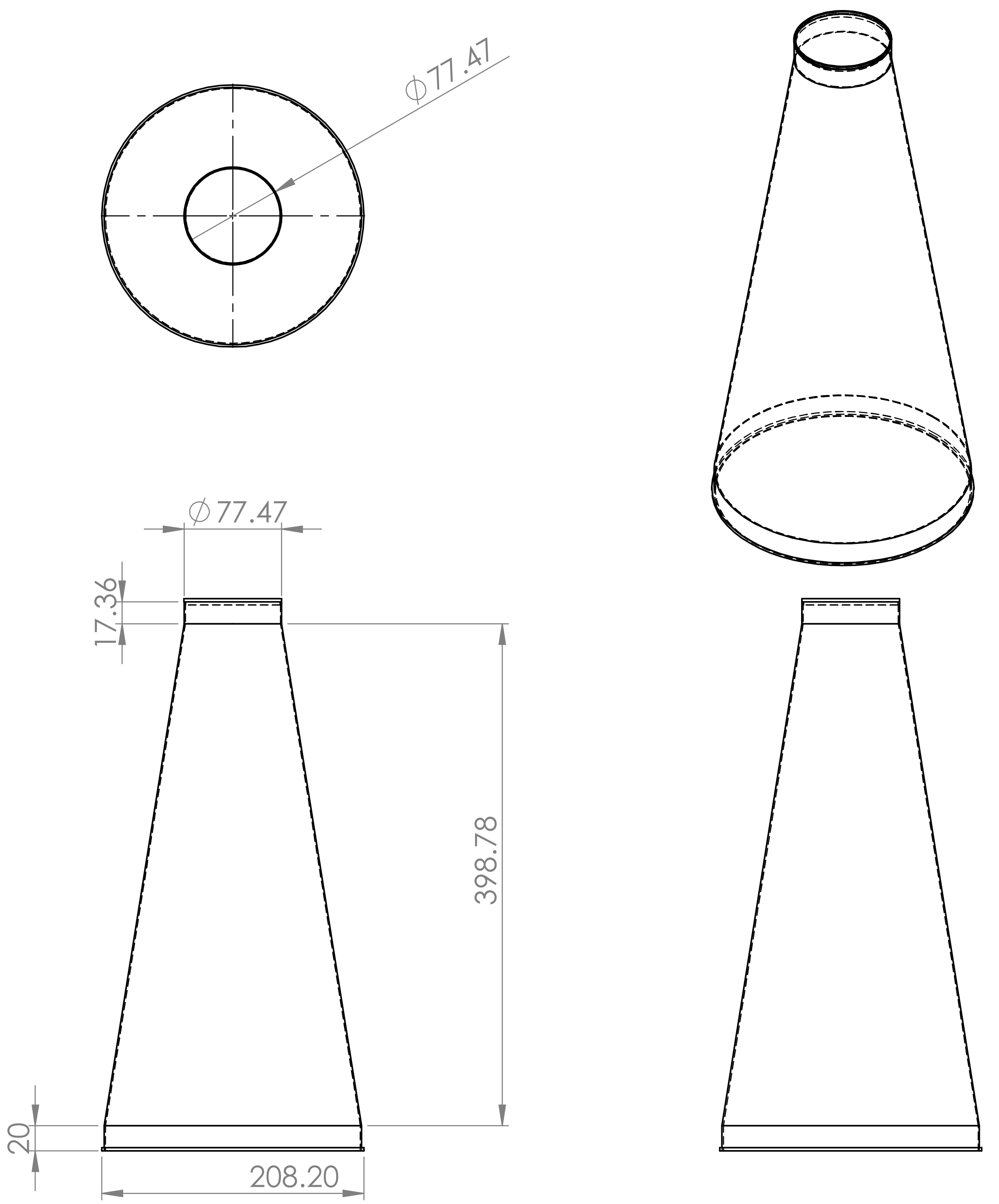
MATERIAL:

PVC pipe

DWG NO.

Pipe3

A4



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH: *GI sheet - 0.3mm*

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE		
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A					

MATERIAL:

TITLE: *Vasu Bansal, Vinod Kanwar, Vishal Singh*

DWG NO. Diffuser2

A4

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