MAKE DEW TECHNOLOGIES

41X: Interim Report

Faculty Advisor: Dr. Shohel Mahmud
February 15, 2019

Signature Page

In signing this report and providing my PEO SMP (Student Membership Program) number, I certify that I have been an active member of the team and provided approximately equal contribution to the work. I take shared credit and responsibility for the content of this report. I understand that taking credit for work that is not my own is a form of academic misconduct and will be treated as such.

Name	**Signature**	**SMP Number**
Kineshan Sivanesarajah		058918
Oluwadunsin Agbetuyi		059116
Mohammed Alkhafaji		059036
Ryan Krahn		058833

Date: February 15, 2019

I Executive Summary

Ever since the 20th century, the quality of water around the world has been decreasing due to the unstoppable tide of the industrial revolution and climate change. The need for new ways of extracting safe drinkable water is at an all-time high. In this report, the concept of condensing humid air will be utilized to create a solution that harvests water from the air in humid regions in less developed countries. The solution discussed in this report will be directed to the city Douala, in Cameroon. Douala was selected because it located in a very humid region in Africa with an average relative humidity of 80% throughout the year.

The theory behind the solution is to create a means of harvesting dew from the atmosphere and store it in a sanitized container. The proposed solution will mimic the environment's cooling effect by matching its system temperature to the dew point temperature for a given relative humidity, causing condensation of water vapor within the system.

To ensure that the solution is well suited for Douala, it will need to be a low-energy functioning solution since the city's electricity source can be unreliable. This will be achieved by integrating low energy consuming sensors and Peltier tiles, as well as repurposing cooled waste air. Alternatively, in future designs, the solution can be powered by other means of sustainable energy, such as solar and wind to ensure continuous operation and reliability.

II Acknowledgements

We would like to thank Professor Shohel Mahmud Ph.D., P.Eng. who provided us with the design process for calculations, feedback on mechanical design and insight on heat sinks designs and applications. [1]

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1.0. Proposal Statement

1.1. Problem Definition

It is public knowledge that the planet earth we live on has an abundant supply of water, to be precise, 71% of the earth's surface is covered by large expanses of water. However, 96.5% of earth's water exist as saltwater in vast oceans, leaving our ever-rising global population with about 3.5% and majority of this amount exists as glaciers [2].

Freshwater is water that contains little to no soluble salts, it is the water we drink. Surface water such as rivers and lakes are scarce in arid regions. To compensate for this scarcity, groundwater such as aquifers are exploited. Groundwater replenishes slowly, and if they are continuously exploited, they are at risk of drying out [3].

Water is essential in our everyday lives; besides drinking and cooking it is crucial for the socioeconomic growth of any nation because of its major impact on agriculture and sustaining life. Water makes up a significant percentage of all living organisms, it can be found in every cell of living organisms. Animals and humans have inhabited lands that had water and plants flourished in water-rich environments. With the beginning of the industrial era, and the expansion of the global markets, factories all over the globe are producing merchandise with the mindset that earth has an unlimited amount of water, and contaminated waste water can be dumped into the oceans with no consequences. This mindset has led us to the water scarcity crises.

A general definition of water scarcity is the lack of access to adequate quantities of water for human and environmental use [4]. A common misunderstanding is that water scarcity is only experienced in the poor, undeveloped countries. Developed countries can also experience water scarcity to a degree. The 'Falkenmark Indicator', also known as 'Water stress index' [5], is a way of measuring water scarcity for a nation. The Falkenmark Indicator evaluates water scarcity as the total amount of renewable freshwater available to a person each year. If the amount is less than 1700 m^3 per person per year, the country is experiencing water stress, if the amount is below 1000 m^3, then the country is experiencing water scarcity, and if the amount is below 500m^3, then the country has absolute water scarcity.

In summary, water is a basic human right and an infinite resource on earth, but despite this, its availability to people greatly varies across countries, coupling this with the rising population, pollution and climate change and how they affect the quality and quantity of the resource, it bears the question "is water still an infinite resource?"

1.2. Literature Review

It is reported that approximately 1.29 x10^{16} liters of water in the form of vapor is available in the atmosphere at any moment in the day, and this amount is recycled about 40 times a year by the hydrological cycle [6]. This report aims to propose a set of concepts and designs for a device, using dew harvesting from the atmosphere as the next water reservoir.

South Asian and Sub-Saharan African countries are the main regions of the world that have been plagued by water scarcity, causing water-borne diseases such as malaria, typhoid and overall lower health and mortality rate. The figure below shows regions on the map coloured in yellow and red to have a lower than standard coverage of improved drinking water sources.

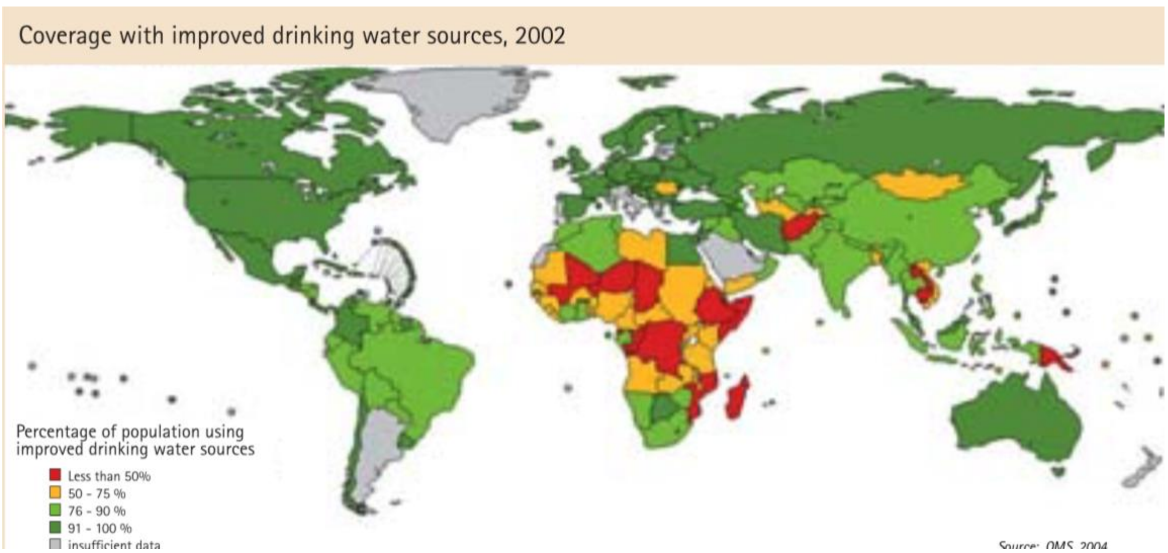

Figure 1: Coverage showing affected African and south Asian countries [3]

Figure 1 shows that south Asia and majority of central, west and east Africa have approximately 50% of their water sources contaminated with harmful microbes. Using this information, our team decided to select one of the affected countries and design our prototype to meet its needs. For this reason, the city of Douala, Cameroon has been selected to be our area of focus. The prototypes discussed later in this report will be adaptable to the region's climate, meet stringent water quality standards and provide at least the minimum daily water requirements for one person.

Douala is located in the western region of Cameroon, beside the Wouri river and the Douala wildlife reserve [7] (Figure 2). The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) reports Douala's coldest and hottest dry bulb temperatures to be 22°C and 34°C respectively. They also claim the temperature will only exceed these extremes 0.4% of the year or approximately 2 days. ASHRAE also reports Douala's relative humidity to be about 15.8 g water/ kg dry air (Figure 36 in the Appendix). The high temperatures and relative humidity in the atmosphere make Douala a good candidate for our dew harvesting proposal.

Figure 2: Map showing location of Douala, Cameroon [7]

Douala is the largest city in Cameroon and the economic capital, its population is approximately 2 million with only 650,000 households that are connected to a water supply. Current water production by SNEC is reported to be 110,000m^3 /day meanwhile the current demand is 250,000m^3/day (Cameroon Tribune, 2007) in addition to this supply there are about 30,000 traditional wells, taps and boreholes that are not monitored for contaminants and have been reported to cause 70% of waterborne diseases [8].

The solution being proposed in this document will either be used as a supplemental or main drinking water supply for households and individuals. Currently, less than half of the population in Douala receive water services from main providers such as La Camerounaise de Eaux (CDE) and Societe Nationale des Eaux du Cameroun (SNEC). Added to that, those who receive a water supply from these service providers have reported major unreliability and insufficiency in services. According to a study that analyzed the impact of the privatization of water supply in Households in Douala [9], 17 out of the 50 houses sampled did not have access to water for 26-30 days and when water was available, 32 of the 50 houses only had access for 2 hours a day. The study concluded that 34.5% of the households have insufficient water supply, 40.5% had just enough water supply, and only 25 Percent of the household had enough water supply. Figure 3 below is the table showing the data collected from the people in Bonaberi Ward of Douala city.

Volume of Water Supplied to Households after Privatisation in Bonaberi Ward of Douala City, Cameroon

Status	Household	Percentage	People in Household	Volume (in Litres)	Percentage
Sufficient	8	16	43	2097	24.75
Manageable	14	28	86	3435	40.55
Insufficient	28	56	169	2938	34.68
Total	50	100	298	8470	100

Figure 3: Percentage of people in Bonaberi Ward receiving sufficient water [9]

Nationwide in Cameroon access to safe drinking water has decreased from 58% to 48% between 1990 to 2015. In populous cities such as Douala, there has been a more drastic decline in access to drinking water from 45% to 22% between 1990 and 2015. below compares the percentage of households in Cameroon with access to safe drinking water [10]. It can be observed from Figure 4 that Douala has the lowest percentage of a population with access to safe drinking water despite the fact it has the highest population in Cameroon.

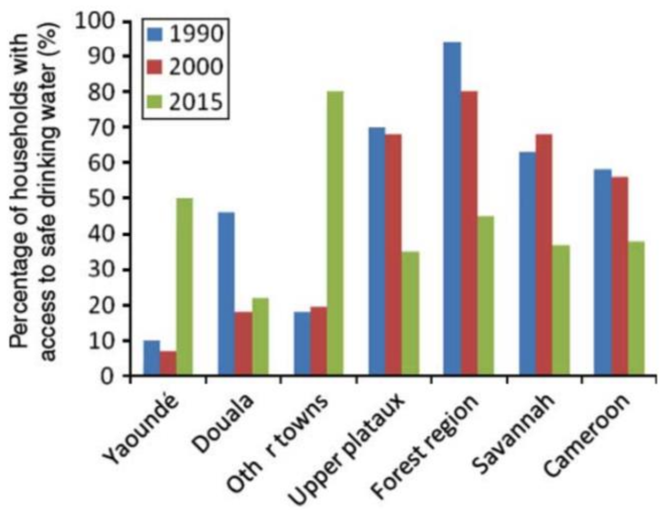

Figure 4: Comparing access to safe drinking water between regions in Cameroon [10]

Records show the world bank has given Cameron "US$ 45 million for the rehabilitation of production facilities and improved access to drinking water in Douala and Yaoundé" [11]. Yet, the Douala citizens still struggle with access to safe drinkable water as a report suggests, "I have not been able to carry water from my tap for more than a month", " My children has developed stomach problems ever since we started using well water" [12].

In conclusion, several reports and studies have been made within Cameroon claiming low access to safe drinking water, Douala is the most populated city, providing most of the jobs and contributing to the economic growth of Cameroon. Despite Douala's significance its nation it maintains the highest percentage of a population without access to safe drinking water. Make

Dew Technologies has been set on course to correct this problem in order to significantly increase the quality of life of Doualans.

Since the enclosure described below will be constructed from Aluminum, investigation went into finding alternative methods of causing this turbulence, without increasing the rate of airflow through system or changing the material being used in the design. One article suggested the use of ribs along the channel to create this turbulence and improve the mixing of temperature streams in the system. This simulation was meant to be "representative of phenomenology found in the internal cooling channels within turbine blades" but should permit similar results in the flow over the ribs. Eddies are formed before and after air passes over the ribs, which circulates air for a certain distance $L_{turbulent}$ before returning to a fully formed velocity profile. [13]

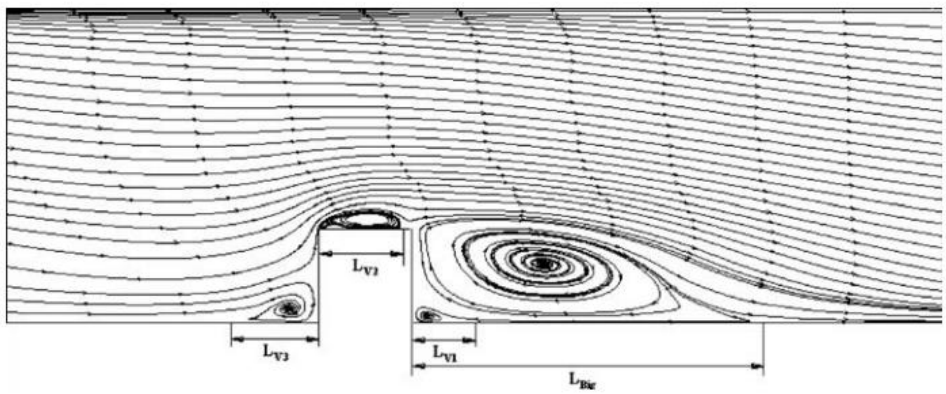

Figure 5: Flow Structures in the Symmetry plane [13]

For large amounts of convective heat transfer to occur on any fluid flow problem, the ideal flow should be turbulent, which causes a greater diffusion of the two Temperature streams. As can be seen in Figure 6 below, with a rougher pipe the Nusselt number increases in the turbulent range, which also increases the heat transfer with Reynolds numbers in that range [14].

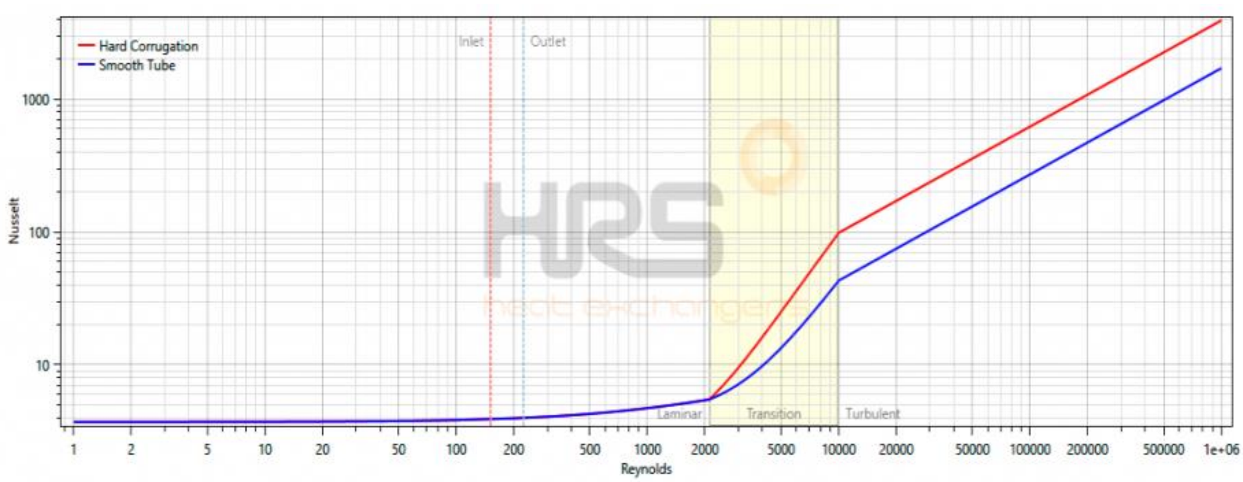

Figure 6:Nusselt number for increasing Reynolds number in rough and smooth pipes [14]

1.2.1. Existing Water Scarcity Solutions

There are few readily available water harvesting techniques being used in the developing world already. The most efficient of these techniques are the fog harvesters [15]. This technique is relativity simple and easy to implement but has many constrains. The Fog harvesting technique is composed of a flat rectangular nets made out of nylon or stainless steel that are supported by poles to make it stand vertically as seen in **Error! Reference source not found.** below. As the fog and clouds pass through the net, the water particles from the humid air get attracted to the surface of the net and eventually form water droplets. the system is completely passive because once large droplets of water are formed, the force of gravity pulls the water droplets down the net and into a gutter that leads to the storage containers. This harvesting technique however is not compatible in many regions of the developing world, this is because the performance depends on the frequency of fog occurrence and the fog water content, making it an unreliable process because it is weather dependent. These harvesting nets are also usually installed on at high altitudes, between 400m to 1200m, depending on the region [15]. In addition, the nets are in open air so they need to be installed away from where people reside to decrease the possibility of contamination. The cost of these nets vary significantly from one region to another, cost depends on material, shipping, and labor costs. In Northern Chili, the cost of installing these nets is $\$90/m^2$, and the output averages to $3.0l/m^2$ of mesh/day [15]The maintenance cost for such a system is estimated at $\$600/year$ [15]

The estimated cost of water production for this setup is @1.41/1000 with a capability of producing 2.5L/m2/day [15]

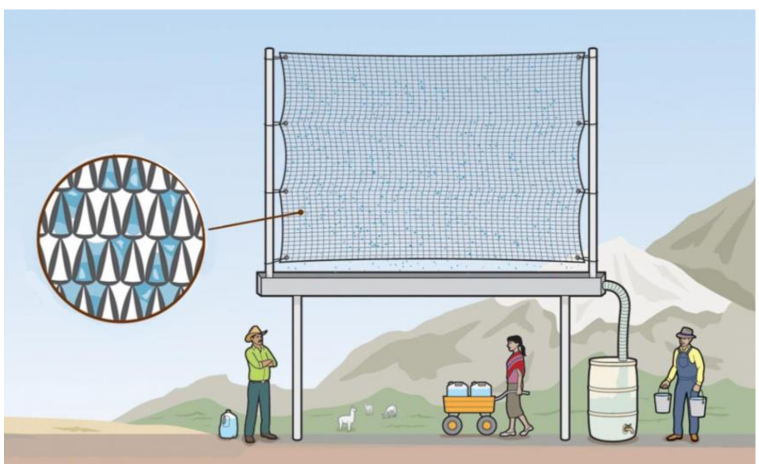

Figure 7:Water Fog Harvesting Net [16]

Another existing design alternative is an Atmospheric water generation from the company WaterGen. The company claims that their product is able to produce 15L/day at 26.7 degrees Celsius and 60% RH. The unit is able to storage up to 15 liters before it needs to be emptied. The product is targeted for both residential and commercial users. However, it is gaining the popularity in armies because the unit can run of batteries. The disadvantage of this device is its high electrical consumption and the price of the unit. The units cost between $18000-40000 USD, depending on the model and requires an annual service to maintain high performance. The electrical consumption of this unit is estimated at 0.33KWh/L. The cost of electricity in Cameroon is around €0.63/kWh [1], hence the average cost of producing 1 liter of drinkable water is €0.21/kWh, or $1.13/L.

1.2.2. Water Standards

In Cameroon packaged drinking water is held to the guideline values outlined by the World Health Organization (WHO). These standards specify that drinking water must be safe and not hazardous to one's health by being free from micro-organisms, chemical substances and radiological hazards. Table 16 in Appendix depicts the acceptable standards for nitrate, fluoride, lead, sulphate and other heavy metals. A study from 2014 analyzing the quality of packaged drinking water marketed in Douala shows that water of local production is unable to meet some of these standards and therefore a need for a sustainable means of producing clean drinking water.

1.3. Project Scope

The primary objective of this project is to completely design a device that can harvest moisture from the atmosphere and provide potable drinking water for an individual to meet their daily requirement. This will be done by cooling the humid surrounding air until its temperature is below the dew point. The scope will include design, conception, engineering analysis, technical calculations, cost analysis, modeling, and completion of a working prototype.

Theoretically the atmospheric water will be contaminant free however, due to fan inlets and exits within the enclosure there is a concern of contaminants building up on the condensing surfaces. This buildup could potentially lead to the contamination of the drinking water despite implementation of filters. The development of a full filtration system which abides by the WHO drinking water quality guidelines is beyond the scope of this project. This project will lay the groundwork for future iterations of the design and the addition of improvements such as a filtration system. The **constraints and criteria** will be used in developing the design to ensure it achieves the goal set out for this project. The chosen design will be completed over the course of **12 weeks** which have been broken down as a timeline outlining major milestones and is presented within the proposal.

1.4. Constraints and Criteria

Table 1: Design Constraints

Constraints	Comments
The system must produce at least 2.5L of drinking water within 24 hours each day for the lifespan of the system.	According to the World Health Organization, for survival humans need at least 2.5 - 3L per day of water [17]. Depending on the selected system, the actual constraint on water produced may increase to match at least the minimum water requirements.
The cost of the system should not exceed $1825.9, yearly maintenance should not exceed $400, and the system should not exceed an operation cost of $1.41/2.5L produced.	This project is targeted towards a problem in developing countries. The main objective is to design a system that is able to sustainably meet the drinking water needs of one person at a more affordable price. The price of cap of $1825.9 CAD was determined through analyzing the capital and operating cost of similar devices that are currently available in the market such as fog condensers and atmospheric water generators and reducing it by various amounts. Further information on pricing is given in the literature review section.
The system must be able to work in at least 65% relative humidity	Our system must be able to function in a minimum 65% relative humidity because during the colder months in Douala, there are days where the humidity will drop to the 60% range [18]. There also isn’t much rainfall during this season, so functioning at a minimum of 65% is necessary.

Table 2: Design Criteria

Criteria	Comments
Low System Cost	Since the system design is aimed for an application in a developing country, the system cost must be attainable for a community or family with low income, or feasible for a NGO to distribute to these groups.
Reduced Environmental Impact	This project should be designed to have a very low environmental impact and manufactured from durable and repairable components so that there is little to no waste created by this device over its lifetime.
Low Energy Input	Ideally the final system should be able to function with a minimal amount of energy in comparison to systems fulfilling similar roles.
Renewable/Sustainable Energy Usage	Energy that is used in this design should be obtained from renewable sources, such as solar photovoltaic (PV). This renewable source shouldn't consume more than twice the footprint area of the system itself.
Withstand Environmental Factors	Since this system will primarily be located in outdoor environments, it should be able to withstand and function properly in a variety of climatic conditions such as severe rain and wind storms.
Ratio of Area consumed to Volume of water produced	The size of the design relative to the water produced will be a major selling point for consumers. For marketability the design should be as small as possible, so it can be used in a wide variety of urban or rural settings while also being able to produce a sufficient amount of water.
Modular Design Features	The design's ability for future additions or expansion will also be used to determine its feasibility. Ideally the design should be modular such that if the consumer wanted to produce more water, they could do so by simply adding another attachment/unit.
Ability to adjust to ambient conditions	The purpose of this design is to harness atmospheric moisture to produce drinking water and to successfully be able to do so the design must be able to work within a range of ambient conditions. The designs' ability to adapt to varying climates will be an important criterion for determining the final design.

2.0. Conceptual Design & Evaluation

2.1. Design Process

2.1.1. Information Gathering

Beginning the design process, a preliminary literature review was conducted to assess the other current designs that were able to fulfill the problem statement for the project. Through analyses of these designs, it was determined that the currently available options were not economically viable solutions for a developing country. In addition to cost, we examined each design's short comings and considered them as possible areas of improvement that our design could address such as water collection inconsistencies and inability to function below certain altitudes.

Further literature review was conducted to determine the background information needed to solve this problem such as; climatic data, average household income and drinking water requirements outlined in relevant codes and regulations for the average person in Douala. Once enough information on the problem had been gathered, a divide and conquer approach was used by assigning each member of the group the task to come up with a conceptual design solution.

One team member was assigned the role of devil's advocate and they critiqued each design, outlining the merits and disadvantages of each design to determine their feasibility. Once each design's viability was validated, a decision matrix was created to determine the optimal solution. This matrix was solely based on the predetermined criteria from the problem identification stage of the project to avoid bias by omitting or choosing criterion in favor of one design. The results from the decision matrices proved the Peltier wall idea was the preferred design solution. The team then met with the faculty advisor to gain feedback on the feasible designs and discuss other possible options. The conclusion at this stage was implementing the Peltier tile wall design would be the optimal solution.

2.1.2. Design of Mechanical Attributes

In order to begin the design of the mechanical components, a lab experiment was conducted using a simple setup that included a Peltier tile with fins on the cold side and a heatsink to remove heat production. This was done to determine if the desired temperature differential was achievable at the fin tip. It was observed that initially the temperature differential was achieved, however with time the temperature differential decreased significantly, and the desired cooling was not achieved. This occurrence in the final design would result in a temperature above the dew point which would lead to the inability to produce and harvest the required condensation. Analyzing present the lab conditions, it was found that this reduced temperature differential could be attributed to two major causes; the first being a lack of heat dissipation on the hot side of the tile, and the second being a need for a closed system with forced convection as oppose to open system with natural and forced convection. Based on the result from these tests the current working design was changed to a closed system with forced convection and a combination of heat sinks and repurposed cooled air to address the issues that were identified during the lab test.

To increase surface area for the collecting of water, one considered improvement involved changing the cooling surface to a corrugated design, seen in Figure 8 below to increase the amount of surface area and thus increase the amount of heat removal. Upon further discussion, it was decided to that this approach had many disadvantages since it would be difficult to validate the model through calculation and the design would largely rely on trial and error methods to determine the heat removal from the design. Additionally, the corrugated design would be increasingly difficult and costly to manufacture as it would require a custom die for forming the corrugations.

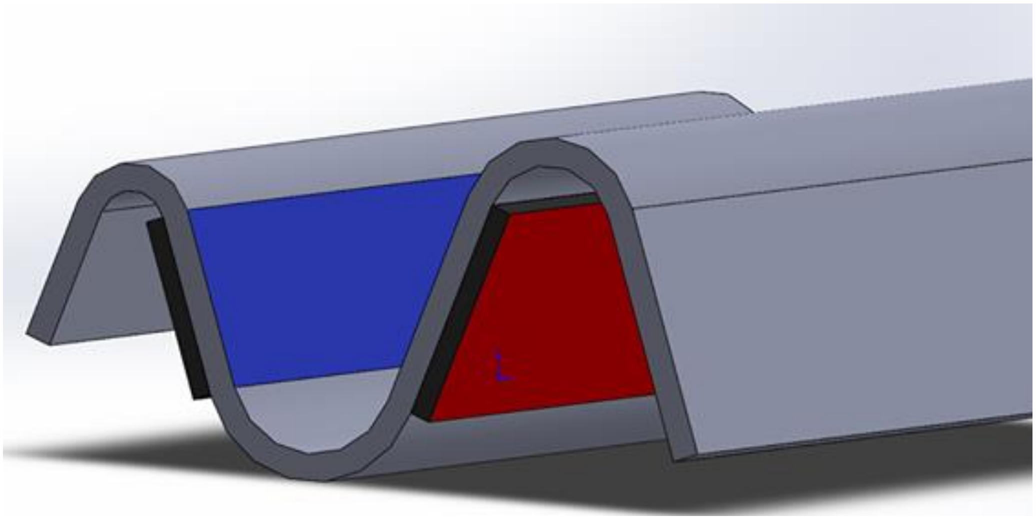

Figure 8: Example of corrugated mounting design for water collection

After the corrugated concept, a new conceptual design was created that implemented the Peltier wall enclosure concept and sketches and calculations were developed to validate the design. Figure 9 below is the initial sketch of the proposed preliminary design, that will be further described in the remainder of this report. The design incorporates flat plates to create a funnel enclosure in the shape of a hexagon or octagon and Peltier tiles will be attached to the walls, allowing a cooling effect inside the enclosure and a heating effect outside the enclosure (exterior shell). Humid air will be pulled into the enclosure using an inlet fan, where condensation will occur within the inner chamber. Fans on the external shell will draw in make-up air to cool the exterior shell of the enclosure and air from the condensation chamber will be mixed with this air to improve heat dissipation in the exterior shell.

After passing through the design review with the group, this conceptual design was then chosen as the final design that would be pursued. The next stage involved further calculations to validate and prove the design's functionality using water phase change and heat transfer principles. Figure 10 below shows a flow chart of the design process.

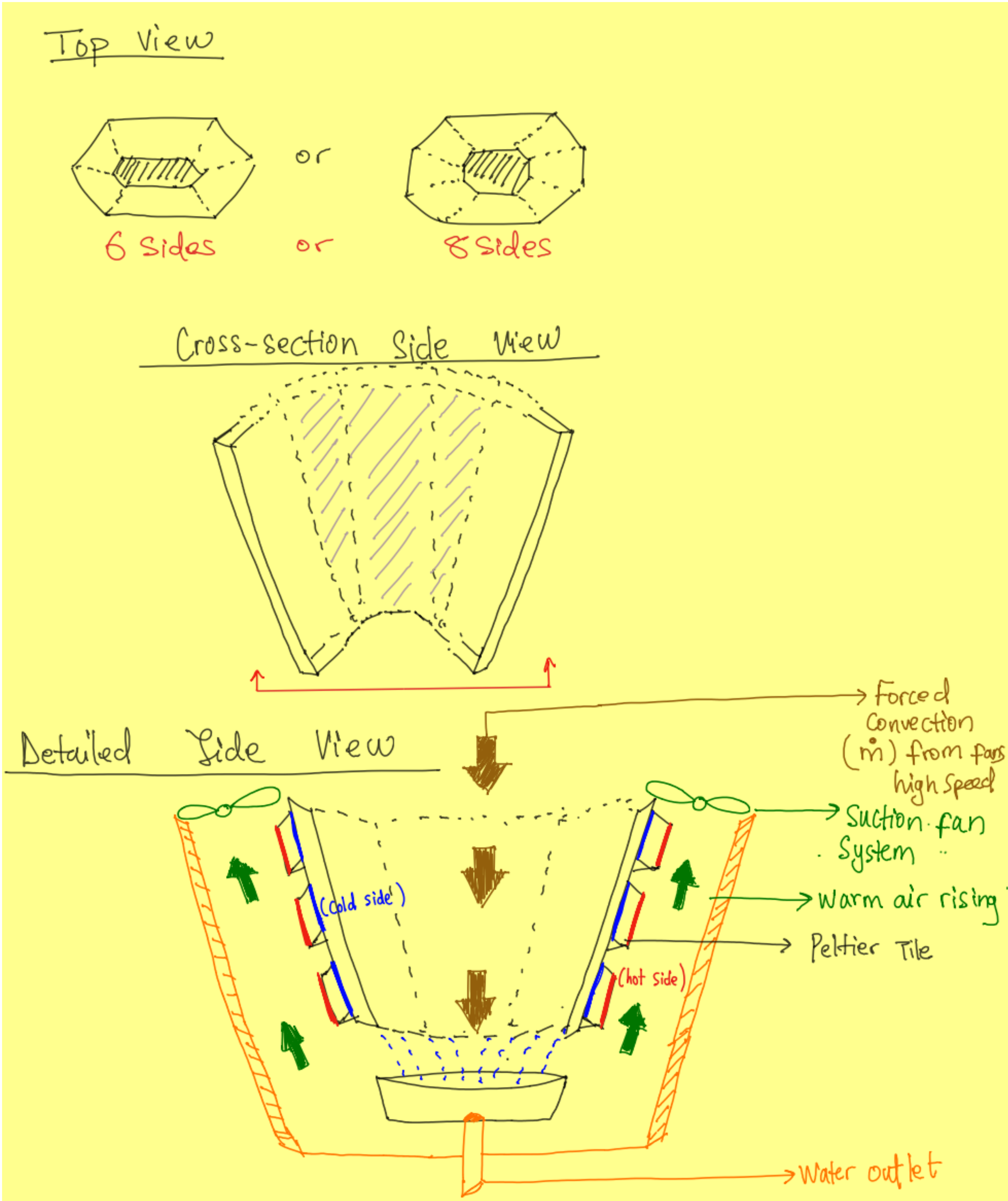

Figure 9: Initial sketch of Closed Funnel Design

Design Calculation Flow Chart

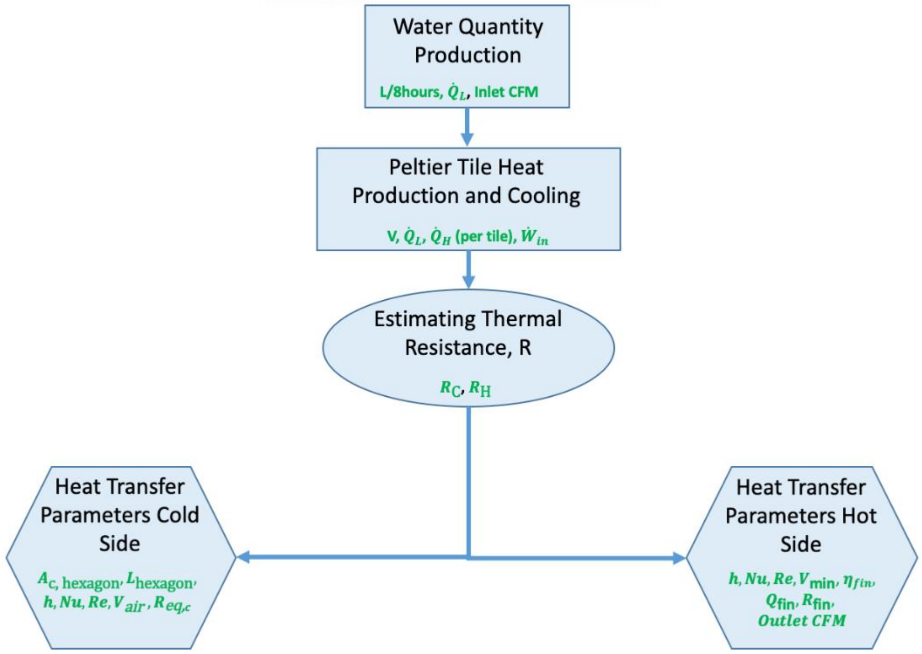

Figure 10: Flow Chart of Completed Heat Transfer Calculations

2.1.3. Water Quantity & Condensation

Stage one of the analytical design was determining the quantity of water that can be obtained for a variation of temperature and humidity conditions occurring in Douala, Cameroon. This is a necessary step because it indicates if it will be possible to acquire the minimum amount of daily water required by an individual to sustain life. For the purpose of this analysis, it was assumed that the user will have access to electricity for at least 8 hours each day, hence the quantity of water collected is in liters per 8-hour intervals. It is important to note that this 8 hour interval should be adjusted to talk place during the peak temperatures of the day. This step was also necessary in estimating the volume flow rate (CFM) of the inlet fan needed and calculating the required amount of cooling energy needed by the whole system to achieve a given amount of water, $\dot{Q}_L$. See *Calculating Quantity of Water Produced* for sample calculations.

					CFM 25				Assumption									W/ Peltier 32.08
	INLET TEMP (°C)	Specific Humidity (kg/kg da)	SPECIFIC VOLUME (m^3/kg)	SPECIFIC ENTHALPY (kJ/kg)	Inlet VOLUME FLOWRATE (m^3/min)	Dew point (°C)	INLET RELATIVE HUMIDITY (%)	OUTLET TEMP Tc (°C)	OUTLET RELATIVE HUMIDITY (%)	Specific Humidity (kg/kg da)	SPECIFIC ENTHALPY (kJ/kg)	Enthalpy of saturated water (kJ/kg)	Dry air flow rate (kg/min)	Dew collection rate (kg/min = L/min)	Total heat removal rate required (kJ/min)	Total heat removal rate required (W)	Dew in 8hour period (L)	Number of Peltier Tiles
	T1	w1	v1	h1	V	Tdp	ϕ1	T2	ϕ2	w2	h2	hw	ma	mw	QL total	QL Total	Water	#
best	34	2.45E-02	0.91	97	0.7075	29.4	72	19.4	100	0.01475	57.5	83.915	0.777472527	0.007580357	30.07405917	501.23	3.638571429	16
worst	21	1.58E-02	0.855	61.8	0.7075	21	98	11	100	8.30E-03	32	46.2	0.82748538	0.00620614	24.46046784	407.67	2.978947368	13

Figure 11: Heat Transfer Calculations - Sheet 1

Figure 11 above is an example of the results determined through use of the design calculator, it compares the design's performance in the absolute worst and best environmental conditions in Douala. The best and worst case being 34°C, 24.5g/kg dry air and 21°C, 15.8 g/kg dry air respectively. Putting the calculations from the first row into words, during the best environmental conditions, with an inlet volume flow rate of 25CFM, 3.6L of water can be collected in 8 hours which requires 500W of cooling and 16 Peltier tiles, assuming each Peltier functions at 4.5A (assuming ideal conditions).

2.1.4. Estimating Optimal Number of Peltier Tiles

The next design stage was estimating the number of Peltier tiles needed to ensure the design is always equipped to provide enough cooling to condense dew. This is a necessary step because the system being built will alternate the working current between at 3.0A and 4.5A. Using a higher current for each Peltier tile results in less Peltier tiles needed, however the higher the current, the higher the amount of heat produced and the system can only dissipate a limited amount of heat. To resolve this issue, the relationship between number of Peltier tiles and water produced was examined for both 3.0A and 4.5A in the worst and best environmental conditions and the graphs below were created.

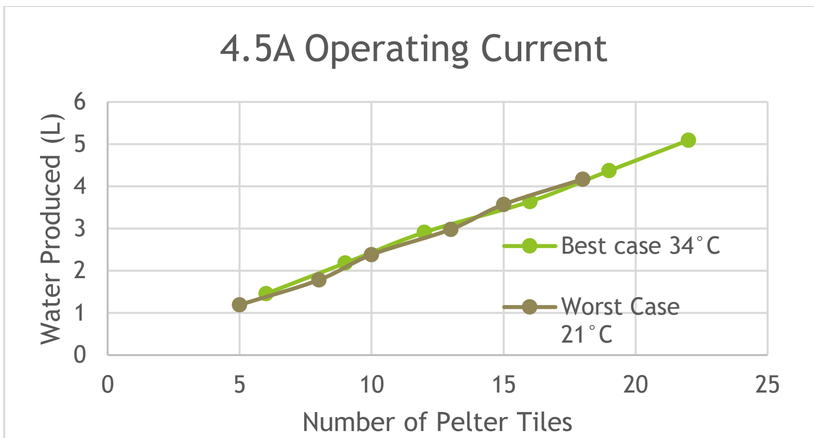

Figure 12: Water Produced by Peltier tiles operating at 4.5A

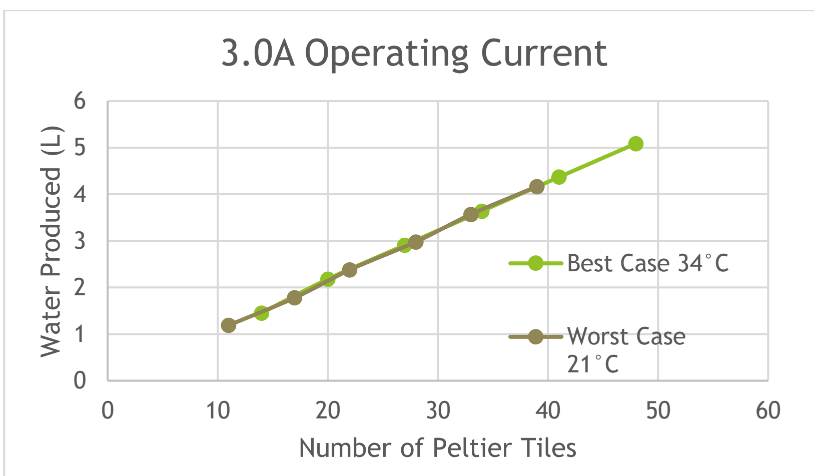

Figure 13: Water Produced by Peltier tiles operating at 3.0A

The purpose of Figure 12 and Figure 13 is to identify the number of Peltier tiles that will result in the same amount of water collection in both the best and worst cases. On the graphs this can be identified as the point where the worst case line intersects with the best case line and then exceed the best case line. The number of Peltier tiles where this occurred using the

4.5A and 3.0A are 14 and 30 respectively. The average of the numbers is 22, which is the optimal number of tiles required for this design. To minimize costs and improve reliability, 24 Peltier tiles have been chosen for this design.

2.1.5. Peltier Tile Cooling Effect and Heat Production

The following stage is conducting heat transfer calculations to assess the required amounts of heating and cooling necessary for a functional design. After the number of Peltier tiles required for the design was established to be 24. The information from the Peltier tile datasheet was incorporated into the calculator to reduce errors and make the process more efficient, see *Thermoelectric Cooler Datasheet*. Figure 14 below shows how the calculator estimates voltage, cooling $\dot{Q}_L$ per tile, power input and heat created $\dot{Q}_H$ per tile. These variables are a function of temperature difference on each tile ($T_H - T_c$) and the current provided. In this example the hot and cold side have temperatures of 50°C and 20°C respectively. The Voltage and Cooling effect can be gotten directly from the graphs provided in the datasheet while other variables are calculated below using equations:

$$\dot{W}_{in} = Current\ (I) * Voltage(V)$$

$$\dot{Q}_H = \dot{W}_{in} + \dot{Q}_L$$

Tc (°C)	TH (°C)					
20	50					
	Delta T Th - Tc (°C)	Current (I)	Voltage (V)	Heat Removal on Cold side (W)	Power Input per Tile (W)	Heat created on hot side (W)
	ΔT	Amps	V	QC/QL	Win	QH
	30	6.0	16.5	37.7	99.0	136.7
		4.5	12.5	32.1	56.3	88.3
		3.0	8.5	14.7	25.5	40.2
		1.5	5.0	1.3	7.5	8.8

Figure 14: Peltier Tile Power Calculator

The calculator predicts that with a required temperature difference of 30°C and using 3.0A for each Peltier tile, the required voltage, power input, cooling power and heat produces will be 8.5V, 25.5W, 14.7W and 40.2W respectively. It is important to note that when using 3.0A and 4.5A the heat produced is approximately 2.7 times larger than the cooling produced which is quite low; however, when using 6.0A, heat is 3.6 times larger and with 1.5A, heat is 6.7 times larger than cooling. This means working with 1.5A and 6.0A amps will be unacceptable for this design because they produce larger amounts of heat to provide the same cooling effect and this design's heat dissipation system is limited.

2.1.6. Estimating Thermal Resistance

The thermal resistance value, R represents the system's resistance to heat flow through a material or medium. It is necessary that it is calculated so that parameters can be selected for efficient conduction and forced convection, which will contribute to the success of the design. A high R value means the material or medium has a high thermal resistance and is able to retain heat or shows insulating properties. For this design, the R value of the cooling and heating side of the system is determined. Hence, the system must achieve an R value greater or equal to that of the estimated R value on the cold side, R_C as well as achieve an R value less or equal to the estimated R value on the hot side, R_H . This means the cold side will resist the flow of heat into the system and the hot side will not resist the flow of heat, allowing for an easier heat removal system. Figure 15 below shows the estimated R values on the cold and hot sides of the system, depending on the current being used. The equations used to calculate the R values and sample calculations can be found in *Sample Calculations Estimating R values*.

Delta T Th - Tc (°C)	Air Temp Surrounding (°C)	Air Temp Outlet Cold (°C)	Air Temp Outlet hot (°C)	Current (I)	Resistance on cold side (°C/W)	Resistance on hot side (°C/W)
ΔT	**T∞**	**Tc**	**Th**	**Amps**	**Rc**	**Rh**
30	34.0	19.4	50.0	6.0	0.39	0.12
				4.5	0.46	0.18
				3.0	0.99	0.40
				1.5	11.12	1.82

Figure 15: System Resistivity Calculations

Following the previous examples, using 3.0A with a temperature difference of 30°C the R values of the cold and hot side are $0.99\ °C/W$ and $0.40\ °C/W$ respectively.

2.1.7. Designing Heat Transfer Parameters on the Cooling side.

The heat transfer parameters being designed must allow for an equivalent R value on the cold of the system to be equal or greater than the R_c value of $0.99\ °C/W$. The factors that contribute the thermal resistance on the cold side is the thermal conduction from the cold side of the Peltier tile through the aluminum plate where dew condenses and the forced convection happening within the vessel as a result of the inlet fan. The occurring thermal conduction and convection contribute R values and the sum of these R values must be equal or larger than $0.99\ °C/W$.

The thermal conduction present will have a very negligible R value because it is dependent on the aluminum plate thickness, thermal conductivity of aluminum and the cross-sectional area in contact with the Peltier tile. The thermal conduction R value can be easily found, using the equation for $R_{Conduction}$ below and the thermal convection will have a significant R value which depends on the cold side's Nusselt's Number, *Nu*, heat transfer coefficient, *h* and Reynolds number, *Re*.

$$R_{Conduction} = \frac{L}{KA_C} <<< R_{Convection} = \frac{1}{hA_C}$$

These heat transfer parameters can now be designed to meet the R value constraint using a series of equations that can be seen in *Sample Calculations for Cooling Parameters*. Figure 16 shows the calculator's results for h, Nu and Re. This example shows that the Reynolds number is laminar and using this value the required inlet air velocity can be calculated, V_{air}. The inlet fan flow rate CFM (volume flow rate) which has been designed in a previous section can now be used with the velocity to design the inlet cross sectional area required.

Plate Resistance/ Peltier (°C/W)	Plate Resistance/ # peltiers (°C/W)	Cold air Resistance/ Peltier (°C/W)	Cold air Resistance/ # Peltiers (°C/W)	Heat Transfer Coeff Cold air (W/m^2K)	Nusselt Number	Reynolds Number Laminar	Reynolds Number Turbulent	Velocity of Air (m/s)	Vessel Cross Sectional Area (m^2)	Hexagonal Vessel side Length (m)	Octagonal Vessel side Length (m)
Rplate	**Rplate**	**Rair,c**	**Rair,c**	**h, c**	**Nu**	**Re,l**	**Re,t**	**v, air**	**Ac**	**HexAc**	**HexAc**
9.19E-05	3.83E-06	1.02	4.26E-02	7.98	156.85	6.89E+04	3.90E+04	21.77	0.103	0.199	0.146

Check if 0: 0

Insert Selected Re values: 6.89E+04

Figure 16: Calculations of system parameters

The Results show an h, Nu, Re, V values of $7.98\, W/m^2K$, $156.85, 6.89x10^4, 21.77m/s$ respectively. The cross-sectional area required is estimated to be $0.103\, m^2$, therefore if the design is Hexagonal in shape, each side will have a length of 0.199m while an octagonal shaped design will have side lengths of 0.146m.

						Vessel Height (CM)	
						50.0	
Cooling Vs. Heating	Air film Temp (°C)	Thermal Conductivity of Vessel (W/mK)	Thermal conductivity of Air (W/mK)	Kinematic Viscosity (m^2/s)	Prandtl Number	Vessel height (m)	Plate Cross Section area (m^2)
	Tf	**Kal**	**K air**	**υ**	**Pr**	**h**	**Ac**
C	26.7	222	0.025436	1.58E-04	0.72904	0.50	0.12

	Plate Parameters	
	mm	m
L	350.0	3.50E-01
W	350.0	3.50E-01
t	2.5	2.50E-03

Figure 17: Estimate of Aluminum plate Sizing

Figure 17 above shows how the aluminum plate sizes have been estimated. Each Peltier tile will be in contact with an area of aluminum, which the dew is expected to condense on. The calculator can estimate the area of aluminum plate required by each Peltier tile to achieve a reasonable inlet velocity and cross-sectional area. Each Peltier tile is 40mm x 40mm if the aluminum plate was also designed to be these dimensions, the Reynolds number will be turbulent with a value of $8.91x10^6$ and the velocity at the inlet would be $2817.1m/s$ with a cross sectional inlet area of $13.29\, m^2$. These design parameters are not acceptable because the size criteria agreed upon by the team was to keep as compact in size as possible for portability and safe daily use. Therefore, the calculations need to be modified to reduce the inlet air speed to a more acceptable and realistic value.

2.1.8. Designing Heat Transfer Parameters for Heat Dissipation

Heat dissipation is an essential feature of this design, without proper heat dissipation the cooling effect from the tiles may not be sufficient to cool the air and allow the formation of dew. A variety of methods can be used to dissipate the heat produced by 24 Peltier tiles, however for this design the heat transfer medium facilitating heat dissipation is air. As previously discussed, the heat dissipation will comprise of make-up air from the exterior and cold air from the cooling system after the water condensation cycle is complete. The temperature of the cold air exiting the cooling system will depend on the initial air temperature as it enters the system and how long it is being cooled for and because of this, the properties of the cold air may vary. In order to ensure the system will be able to remove as much heat from the system as needed, the heat dissipation calculations will solely be based on the make-up air. Assuming the make-up air can achieve the required heat dissipation, then additional cooling will only improve the system functionality.

The two factors contributing to heat dissipation using make-up air is the thermal resistance contributed by the heat sink and forced convection. It was determined earlier that the R_H for this design is $0.40\ °C/W$, now the goal is to achieve an equivalent thermal resistance less than or equal to that, using heat sinks and outlet fans. Figure 18 below shows the result of the equivalent R value of the system and the heat dissipation acquired from the design calculator.

Heat Transfer Coeff (W/m^2K)	Heat Transfer factor	Fin Efficiency (%)	Area of fins (m^2)	Area no fins (m^2)	Heat Lost w/ fins (W)	Heat Lost w/ no fins (W)	Total heat lost/1 heat sink (W)	Fin Resistance (1 fin) (°C/W)
h	**m**	**ηfin**	**Afin**	**Ano,fin**	**Qfin**	**Qno,fin**	**Qfin, total**	**Rfin**
21.429	16.607	0.68	0.14	0.004	61.73	2.53	64.26	0.48
							% Heat Dissipated	**R Value Ratio**
							161.46%	0.84
							>= 80%	>=1

Figure 18: System thermal resistances

The estimated R value of the system is calculated to be $0.48\ °C/W$ using an outlet volume flow rate of 210 CFM and heat sink dimensions that will be discussed below. This value does not meet the requirement of $0.40\ °C/W$ or less, which means the heat sink properties can be improved or volume flow rate should be increased to further reduce this value by 15%. An important factor that will also be considered is the percentage of heat being dissipated, which Is calculated to be 160%. This means 1.6 times of the heat produced by each Peltier tile is completely dissipated. Because of the high dissipation rate, it may be worthwhile considering operating at an R value ratio of 0.84.

The heat transfer coefficient, *h* and heat sink efficiency, η_{fin} are dependent on the velocity of air between each fin, V_{min} and corresponding Reynolds and Nusselts's numbers *Re, Nu* respectively. See *Sample Calculations for Heating Parameters*

			MATERIAL	CFM		PROPERTIES AT TF		FIN CALCULATIONS (1 FIN)			
			Aluminium	*210*							
Heating	Air film Temp (°C)	Air Temperature in Shell (°C)	Thermal Conductivity of Fin (W/mK)	Outlet VOLUME FLOWRATE (m^3/min)	Thermal conductivity of Air (W/mK)	Kinematic Viscosity (m^2/s)	Prandtl Number	Temp Diff (fin base to ambient) (°C)	Corrected Length (m)	Perimeter (m)	Cross Sectional Area (m^2)
	Tf	**Tc**	**Kal**	**V**	**K air**	**υ**	**Pr**	**(Tb-T∞)**	**Lc**	**p**	**Ac**
H	42	19.4	222	0.099	0.026768	1.72E-05	0.72494	30.60	7.54E-02	0.151	5.25E-05

	Fin Parameters	
	mm	m
L	75.0	7.50E-02
W	69.0	6.90E-02
S	2.0	2.00E-03
H	37.0	3.70E-02
t	0.7	7.00E-04
n	25	

Figure 19: Heat Sink Calculations

Figure 19 above represents the heat sink calculations and how they have been derived. Using the heat sink dimensions shown, which are length (L), width (W), spacing (S), height (H), thickness (t) and number of fins (n) with a volume flow rate of 210 CFM the R value was calculated. This heat sink dimensions can be easily purchased or manufactured, increasing fin height, decreasing thickness or spacing of fins will result in a required lower R value but heat sinks of that dimensions are difficult to procure and expensive. In addition to this, increasing volume flow rate from 210 to 340 CFM will result in the required R value as well, however, to supply that amount will require more outlet fans and input power which may not be useful considering the high heat dissipation rate currently being achieved.

2.2. Required Resources

Required Resources for this project include both analytical software and tools for manufacturing our final prototype. The needed tools can be broken down into three categories: Informational, Analytical and Manufacturing.

Table 3: Required Project Resources

Informational	Analytical	Manufacturing
The Internet	SolidWorks	CNC Mill
Faculty Advisor	Ansys Workbench and Fluent	SOE Manufacturing Shop
Library	MATLAB	Mastercam

The resources required for the completion of the project are both digital and physical. First, Professors, the library, and the Internet will be utilized to gather data for the research phase of the design process. Professors are also needed for guidance as well as support when an obstacle occurs. For the designing phase, SolidWorks and Ansys will both be required to design a model and then simulate the stresses as well as the heat transfer properties experienced by the model. Alongside the modeling software, MATLAB will be required to design the electrical and control circuit of the machine. Lastly, the machine shop inside Thornbrough will be required to fabricate a physical model of the project.

2.3. Conceptual Design Alternatives

2.3.1. Biomimicry Concept

Biomimicry is a method to resolve material and generate design problems by ambulating species that developed over a long period of time [19]. Essentially, the idea is that, through evolution nature has developed highly efficient systems and processes that can be give rise to solutions to many of the waste, resource efficiency and management problems that we now grapple with today.

The fundamental Biomimicry laws [20] can be utilized to achieve a sustainable and efficient water condenser. First, the product will have to draw its power from sunlight. This can be achieved by using solar panels to create electricity and storing it in a battery. Second, a humidity sensor will measure the humidity level and trigger the cooling mechanism to cool the surface just enough to let water. Lastly, the surface on which the cooling effect is taking place will have to be built with enhanced functionality, whether to collect the water easier, prevent water from staying on the cooling surface, or simply to maximize efficiency by decreasing the energy needed to create the dew. Figure 20 below shows our proposed solution using biomimicry in the form of a leaf.

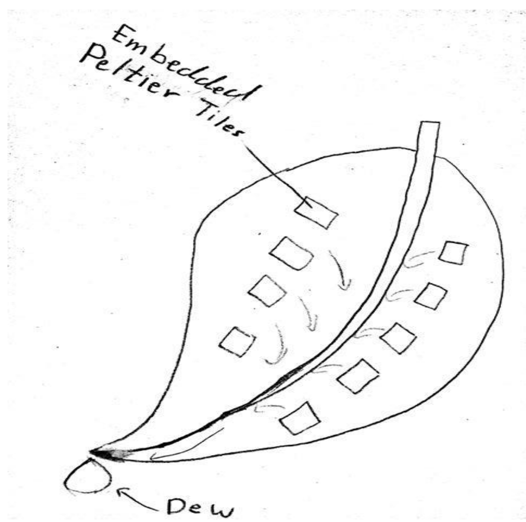

Figure 20: Peltier tiles used on a leaf shape design

The Biomimicry concept above is to create a cooling surface that resembles the shape of a leaf because the leaves have a large surface area and low volume, therefore making it easy to have multiple leaves close to each other while also providing enough space for the heat caused by the Peltier tiles to dissipate. Leaves are also shaped such that water dew can slide into the central vane of the leaf and then drip at the tip of the leaf.

2.3.2. Dehumidification Concept (Desiccant Wheel)

Another design we have utilizes the wasted dehumidification water in air conditioning units (AHUs) for buildings and the Peltier effect. In humid countries the air enters an AHU at a high relative humidity, typically greater than 70%, however for human comfort purposes supplying air at this percentage of relative humidity is uncomfortable and as a result air must be dehumidified before it is supplied to a space. This is typically done through use of a desiccant dehumidification wheel, which is made of a silica gel material that has pours that are excellent at trapping moisture. In a typical AHU there are two air streams one that is exhausted out of the building after gaining moisture and the other is supplied to a space after losing moisture and reducing its temperature.

Our design will primarily target the exhaust air stream to cool the air below the dew point and harvest the water for drinking purposes. We plan to do this by the following process; In stage 1 hot humid ambient air is pulled into the unit by a fan, stage 2, the air is heated using Peltier tiles to increase its moisture holding capacity. Stage 3 the air passes through the desiccant dehumidification wheel to gain moisture maintaining its temperature and increasing its relative humidity as show. Stage 4 the air is cooled to below its dew point by another set of Peltier tiles to harvest the dew as water.

The second air stream (supply air) begins with ambient air entering at stage 5 which is then dehumidified by the desiccant wheel. This air is then cooled in stage 6 by the cooling side of the Peltier tiles from stage 2 producing cool supply air at the desired relative humidity at stage 7.

The primary advantage of this design is its ability to utilize otherwise wasted energy put into humidifying the exhaust air. As a result, the energy to further cool the air once this high temperature and high relative humidity is reached is relatively low in comparison to cooling the ambient air. The main disadvantages of this design is the capital cost and limitability for use. In developing countries, the capital cost of equipment is always a concern and as a result, air handling units which are rather pricey are often times replaced by split units which don't work as well but are significantly less expensive. Another major disadvantage of the air handling unit design is the fact that the unit will require a significant amount of space in comparison to the other designs as well as contribute noticeable amounts of noise to the environment it is placed in. With this in mind it would be difficult to place these units outside of an urban building setting. However, these issues can be addressed through varying the equipment used in building the design and warrants further research.

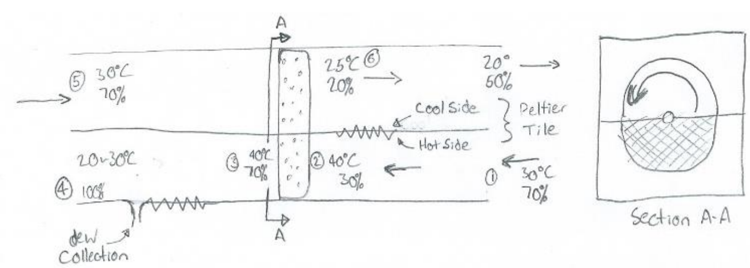

Figure 21: Desiccant wheel for dehumidification

2.3.3. Peltier Wall Concept

The Peltier-Seebeck effect creates a temperature difference at a junction of two dissimilar conductive materials when a current is run across them. Inversely, if there is a temperature difference across the thermoelectric circuit, a current is generated in the electric circuit. These thermoelectric coolers (TECs) are manufactured in a very compact form and are often referred to as Peltier tiles, due to his discovery of the effect. The tiles have a thermal efficiency between 10-15%, which is less efficient than refrigeration units using compressors, but has benefits depending on the application.

The design shown in Figure 22 uses TEC tiles in a single layer to extract heat from an aluminum surface, cooling the air to its dew point and collecting the resulting water for use. Copper, or another similar thermally conductive material would also be embedded in the aluminum sheet to ensure that the coolest temperature occurs at the peaks of the machined plate. The surface pattern is meant to mimic natural shapes and provide channels for the water to flow down the wall and into a collection trough.

This collected water will be filtered before storage to remove any dust or particulate matter that might be in the air, since the plate will be a semi-open system to the environment. A control system will also be embedded in the plate to monitor temperature and relative humidity and then regulate how much power is being drawn by the TEC tiles and when to turn them off or when to extract more heat and create more dew.

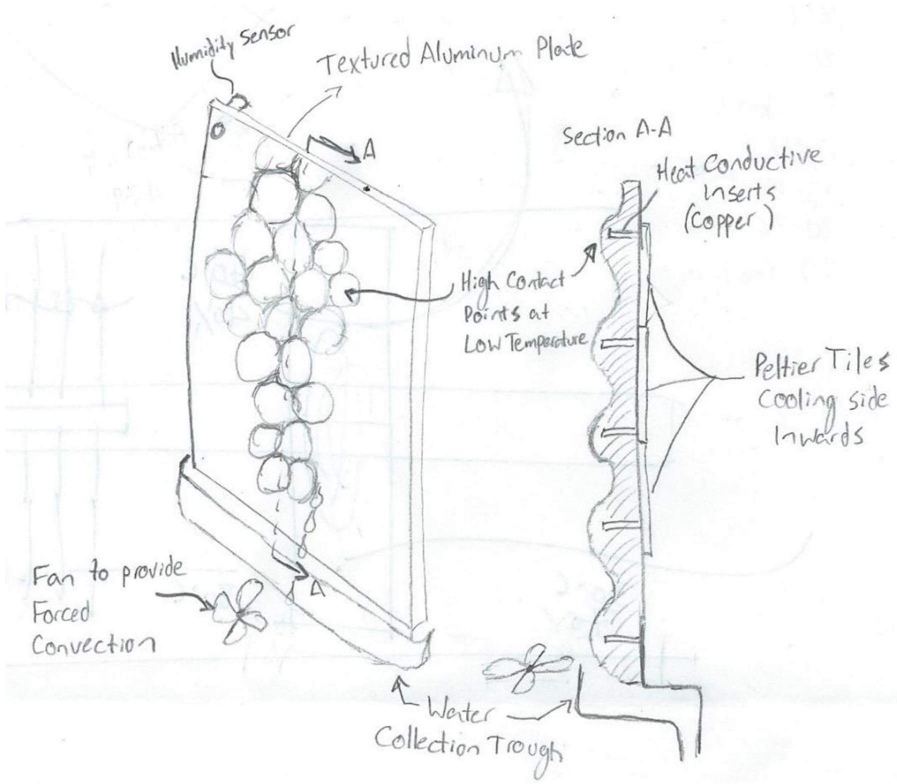

Figure 22: Peltier Water Collection Wall

2.4. Design Evaluation

2.4.1. Design Evaluation Methodology

To Evaluate between the conceptual designs and determine a final design solution a weighted decision matrix has been used. This qualitative tool was useful for evaluating the multiple conceptual design alternatives based on the predetermined criteria from earlier sections. To begin the evaluation, process the design alternatives were compared relative to a reference alternative, the biomimicry concept, using a Likert 5-point scale for each criterion. It is important to note that when comparing the relative rating of the designs only one criterion was considered at a time and all other criterions were ignored. For the purpose of this matrix the Likert scale was –2, significantly worse, -1 marginally worse, 0 about the same, 1 marginally better, and significantly better.

To determine the weightings of each criterion a pairwise comparison was then conducted. First a square matrix was established, using the criteria listed vertically and horizontally as shown in Figure 23. In this matrix each cell corresponded to a comparison between 2 criterions and the more important criterion is listed, or if they are of equal importance both are listed. Based on this analysis the diagonal and bottom half of the matrix would be irrelevant as they would either be a comparison of a criterion to itself or a redundant comparison.

		A	B	C	D	E	F	G	H
Ability to adjust to ambient conditions	A	-	A	C	A	A	F	AG	A
Modular Design features	B	-	-	C	D	E	F	G	H
Ratio of Volume of Water Produced to footprint area	C	-	-	-	C	C	CF	C	C
Renewable/sustainable energy usage	D	-	-	-	-	D	F	G	H
Reduced environmental impact	E	-	-	-	-	-	F	G	H
low system cost	F	-	-	-	-	-	-	F	F
low energy input	G	-	-	-	-	-	-	-	GH
ability to withstand enviromental factors	H	-	-	-	-	-	-	-	-

Figure 23: Decision Matrix

Another comparison table is created to rank the criteria being used above, the ranking in this table was equivalent to the number of cells containing each criterion. This gave us the relative importance of each criterion, see Figure 24. The rankings were multiplied by a factor x until the sum of the weights equaled approximately 100, in this case this value was 96.1. The remainder of the weights, 3.9 were distributed to the lowest ranked criterion to ensure each criterion had a weight greater than 0. Each design was compared to the other before the relative weights were determined to remove bias in ratings and achieve consistency. These weights were then inserted into the decision matrix and multiplied by the rating to determine the score for each criterion. The sum of the scores were then determined for each design and they were then ranked in order of highest score.

	Identifying letter	Calculated Ranking	Calculated percentage weights	Rounded percentage weights	Remaining weight distribution using X=3.1
Ability to adjust to ambient conditions	A	5	15.5	15.5	
Modular Design features	B	0	0	3	3
Ratio of Volume of Water Produced to footprint area	C	7	21.7	22	
Renewable/sustainable energy usage	D	2	6.2	6	
Reduced environmental impact	E	1	3.1	4	1
Low system cost	F	7	21.7	22	
Low energy input	G	5	15.5	15.5	
Ability to withstand enviromental factors	H	4	12.4	12	
Total			96.1	100	4

Figure 24: Design Ranking

The final design option 4 had the highest rated score and was chosen as the final design. To ensure that this design was the correct choice a sensitivity analysis was conducted. In this analysis the leading two criterion were varied and as shown in Figure 25, Figure 26 and Figure 27, the lowest rating of the chosen design exceeds that of the highest rating of the other designs. This implies that our chosen final design is not sensitive to changes in 44% of the weighted criteria.

Two large components of this design process are the comparison between criterions and the comparison of designs based on criterions. For the pairwise matrix in **Error! Reference source not found.** the comparison between criterions was rather simple as the team decided through consensus which criterion of the two being compared provided better immediate value to the user and was then given the cell. The comparison between alternative designs was a bit more difficult, the thought process has been summarized below.

The purpose of this design is to harness atmospheric moisture to produce drinking water and to successfully be able to do so the design must be able to work within a range of ambient conditions. For this criterion the reference biomimicry concept had some capability of achieving this as did Options 2 and 3 through use of sensors and a closed feedback loop and due to their equivalence in achieving this all three of these options were given a rating of 0. It is important to note that these three options were open systems and as discussed in earlier sections open systems are more difficult to adjust to as there is no fixed controlled volume. Option 4 being a closed system had a better ability to adjust to ambient conditions and was given a rating of 2.

The modular design features criterion is a measure of a design's ability for future additions or expansion. Ideally the design should be modular such that if the consumer wanted to produce more water, they could do so by simply adding another attachment or component. The reference design and option 4 have no capability of adding expansion components to increase the dew capacity. Instead these designs required a total replacement of a larger scale version of the design to achieve this. As a result, these designs were given ratings of zero indicating their equivalence in inability to add modular improvements. Option 3 could incorporate modular expansions to an extent as most air handling units are constructed in segments and was given a rating of 1. Option 2 had significantly more room for expansion as the entire purpose of the design was to have connectable expansion units that could be stacked vertically and as a result was given a rating of 2.

The size of the design relative to the water produced will be a major selling point for consumers. For marketability the design should be as small as possible, so it can be used in a wide variety of urban or rural settings while also being able to produce enough water. For comparison based on this criterion each design was evaluated based on surface area required to produce 2L of water. The reference Option footprint was slightly larger than Option 4 and as a result Option 4 was given a rating of 1. Option 3 required an air handling unit and was given a rating of –2 due to its requirement of a large footprint. Option 2 was given a rating of 2 as its footprint was limited to the width of the plate, stand and the convection inducing fan which was considerably smaller than that of the reference Option.

Ideally energy that is used in this design should be obtained from renewable sources, such as solar photovoltaic (PV). For the renewable/sustainable energy usage criterion Options 2,3 and 4 did not use a renewable energy source and the reference Option did and as a result Options 2,3 and 4 were given ratings of –2.

The solution should be designed to have a very low environmental impact by a manufacturing the solution from durable and repairable components so that there is little to no waste created by this device over its lifetime. The reference Option along with Options 2 and 3 required roughly the same amount of material and major components and as a result these options were given an equivalent rating of 0. Option 3 however was given a rating of -2 as the number of components and materials associated with the air handling unit design was significantly greater than that of the other three options.

Every design should aim to minimize cost this is especially true for this solution as it is aimed for application in a developing country, the system cost must be attainable for a community or family with low income, or feasible for an NGO to distribute to these groups. Of the four options the reference biomimicry option has the largest cost due to its utilization of renewable energy sources. Options 2 and 4 were given a rating of 1 as they require similar major components to that of the reference option however these options do not require a renewable energy source and as a result, they were less expensive. Options 3 was given a rating of 2 since it had the lowest cost as this design would utilize modifications to an existing air handling system.

Ideally the final system should be able to function with minimal amounts of energy. The amount of energy required for each design would be largely related to the number of tiles required to remove the desired moisture from a common controlled volume. Since each design would be removing the same amount of moisture, they would also require the same number of tiles and energy consumption. As a result, each design was given an equivalent rating of 0.

The final design should be able to withstand and function properly in varying environmental conditions such as rain storms or windy climates. For rating based on this criterion open systems would be less likely to withstand harsher climates and were given a lesser rating. As a result, Options 3 and 4 were given a rating of 2 since both designs were closed systems contained within an enclosure. Options 2 alongside the reference option were both open systems however, Option 2 was given a rating of –1 because of integrity. Due to Option 2’s vertical design it was more susceptible to large gusts of wind.

	14.5	17	19.5	22	24.5	27	29.5
14.5	0.255	0.305	0.355	0.405	0.455	0.505	0.555
17	0.28	0.33	0.38	0.43	0.48	0.53	0.58
19.5	0.305	0.355	0.405	0.455	0.505	0.555	0.605
22	0.33	0.38	0.43	0.48	0.53	0.58	0.63
24.5	0.355	0.405	0.455	0.505	0.555	0.605	0.655
27	0.38	0.43	0.48	0.53	0.58	0.63	0.68
29.5	0.405	0.455	0.505	0.555	0.605	0.655	0.705

Figure 25: option 2 sensitivity to leading 2 criterion

	14.5	17	19.5	22	24.5	27	29.5
14.5	0.38	0.33	0.28	0.23	0.18	0.13	0.08
17	0.43	0.38	0.33	0.28	0.23	0.18	0.13
19.5	0.48	0.43	0.38	0.33	0.28	0.23	0.18
22	0.53	0.48	0.43	0.38	0.33	0.28	0.23
24.5	0.58	0.53	0.48	0.43	0.38	0.33	0.28
27	0.63	0.58	0.53	0.48	0.43	0.38	0.33
29.5	0.68	0.63	0.58	0.53	0.48	0.43	0.38

Figure 26: option 3 sensitivity to leading 2 criterion

	14.5	17	19.5	22	24.5	27	29.5
14.5	0.72	0.745	0.77	0.795	0.82	0.845	0.87
17	0.745	0.77	0.795	0.82	0.845	0.87	0.895
19.5	0.77	0.795	0.82	0.845	0.87	0.895	0.92
22	0.795	0.82	0.845	0.87	0.895	0.92	0.945
24.5	0.82	0.845	0.87	0.895	0.92	0.945	0.97
27	0.845	0.87	0.895	0.92	0.945	0.97	0.995
29.5	0.87	0.895	0.92	0.945	0.97	0.995	1.02

Figure 27:option 4 sensitivity to leading 2 criterion

2.5. Proposed Preliminary Design & Defence

Through completion of the weighted decision matrix above, it was determined that the funnel design was the best solution for Douala's problem. The final design features a funnel like enclosure which utilizes a closed system comprised of Peltier tiles and forced internal convection induced by various fans. In this design air will be pulled from above by an intake fan into a tapered channel which will be lined with Peltier tiles to provide cooling for the air. The air will circulate through this channel until the desired moisture is removed and will then be directed to the exhaust channels. To achieve this moisture removal, the Peltier tiles will be set to a point below the dew point temperature of the incoming air. The exhaust channel will feature the repurposed cooled air from the tapered cooling channel mixed with makeup air to provide additional cooling for the hot side of the Peltier tiles. The air in this channel will be ejected horizontally with enough throw by the exhaust fans to avoid short cycling. The hot side of the tiles will also have fins attached to increase the surface area and thus the rate of heat transfer between the tiles and the exhaust air.

A key criterion for the final solution was to maximize the surface area available per square foot of footprint area. Peltier tiles were used in this design even though they are 10-15% less efficient than a traditional refrigerant system due to their compact size. This size advantage allowed for the minimization of the required footprint area in comparison to the compressors needed in traditional refrigeration systems. In addition, Peltier tiles also have the advantage of silent operation and instantaneous temperature change as refrigerant systems require the use of a compressor and time to change the temperature of the working fluid.

The primary advantage of this design over the other alternatives is that it is a closed system. As mentioned earlier in the approach section through testing it was determined that open systems were less capable of providing the necessary cooling below the dew point temperature. This deficiency in cooling was due to a lack of movement or stagnation of air and the inability to handle large quantities of air. The biomimicry and Peltier wall options are examples of designs utilizing open systems. This funnel design addresses both issues through retaining the air from which moisture is being removed within a closed system, the cooling channel, using fans. Since the volume flowrate can be varied with the intake fan speed and the size of the control volume is fixed, the time within the cooling channel can be varied as well. Through use of sensors at the inlet and exit alongside a closed feedback loop this design ensures that the exiting air reaches the dew point temperature, thus the intended amount of moisture removal is achieved.

This design also has the advantage of repurposing cooled air for increased heat dissipation on the heat rejection side. All the alternative designs the team developed merely eject the cooled air back into the environment. The final funnel design however, does not and uses a combination of ambient air and the cooled air to increase the rate of heat dissipation. Through this process the final design utilizes the otherwise wasted cooled air to reduce this load and the total energy consumption which is a clear advantage over the other designs.

Another advantage of this design would be its ease of manufacturing. Since a majority of this design will be comprised of off the shelf components and sheet metal it can be easily mechanically fastened and assembled. From an environmental perspective this is a clear benefit as the use of sheet metals will lower the overall weight of the product thus reducing the amount of fuel burned per unit shipped. This would also have economic implications as well since use of fastened sheet metal over welding will also aid in reducing the size and weight which are important factors when determining shipping costs.

Although this design has many advantages over the alternative it has a few disadvantages as well. The main disadvantages of this design are the increased energy consumption for cooling, non-modular design and difficulty to clean. As mentioned above the use of Peltier tiles over typical refrigerant comes at an inefficiency cost which is a concern as one of the criterions for this design is also low energy input. This energy cost can be mitigated but not eliminated by use of a higher end tile, but this will result in increased costs and thus a cost benefit analysis is required to determine the course of action. This design also has the inability of incorporating modular additions, however it does possess the ability to vary the fan speed to in return vary the amount of water removed to a certain extent. These areas of concern warrant further analysis and are possible areas of improvements for later iterations however, it is the team's belief that this option is the best solution.

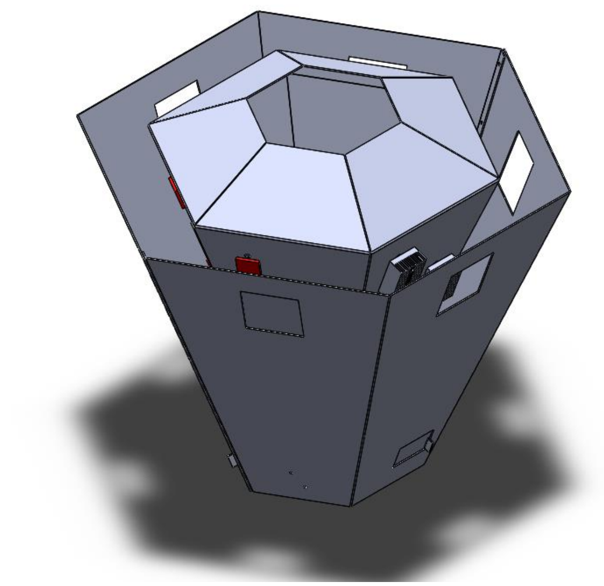

Figure 28: Preliminary CAD of Preliminary Design

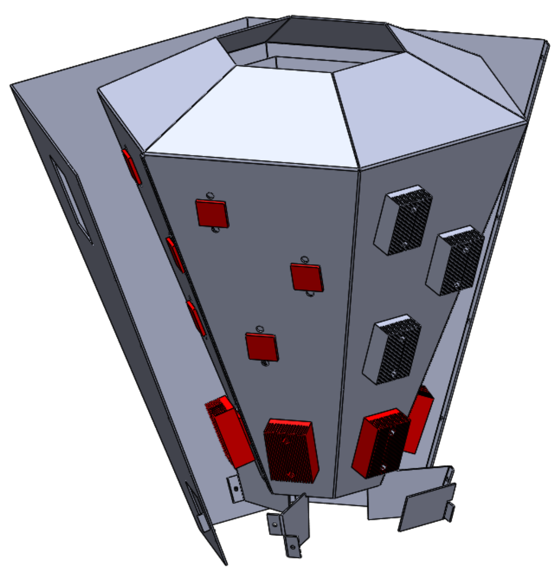

Figure 29: Sectioned View of the Mounting Configuration

2.5.1. User Safety & Maintenance

For any water harvesting device there are several guidelines and regulations as for what constitutes as safe drinking water however, as mentioned in earlier sections for Douala, Cameroon there aren't any drinking water requirements clearly stated. Since harvested dew water is theoretically pure drinking water, the safety aspect of the drinking water was not considered at this stage. During the conceptualization of the final funnel design there were two major areas of concern that were raised and addressed. The first area of concern was the ability of contaminants, small animals, and possibly body parts entering the enclosure through the inlet and exhaust fans. To mitigate this risk bird cages, small mesh guards, were placed on all entry and exit points. These guards would prevent most foreign objects from entering the device to a certain extent, as they would be limited to the size of the mesh. One possibility is that the water harvested from the humid air is capable of damaging the electrical circuit. To protect the circuit from water damage, wires and connections will be covered with electrical tape, but relay switches and the Arduino will need to be kept in a dry environment. The electrical components are also vulnerable to damage from the heat dissipated by the Peltier tiles, therefore, the electrical panel containing the components will need to be kept at a safe distance to reduce possible heat damages.

Since water is condensed inside the cooling funnel and then drips into a water collection contained, it is safe to assume that the water collected is not polarized since there is no connection between the electrical components and the water collected.

3.0. Updated Work Plan and Resources

3.1. Project Resources & Materials

The project resources can be split into four categories, electrical, refrigeration, structural, and programs. Electrical items are used to both power the system and control the algorithm of the system by integrating a processor and sensors. Refrigeration items are components that need to be assembled together to cool the humid air to below the dew point temperature as well as keeping the Peltier tiles cool to maximize the tiles efficiency. Structural resources are material to be used to manufacture the unit and create fastening brackets. Lastly, Digital resources are programs used to model a system electronically and running both structural and heat transfer simulations before the actual build.

Structural material:

- Aluminum sheet metal
- Thermal resistant material
- Screw and nuts

Starting off with the Structural equipment, the 0.16" thick aluminum sheets will be used to create the physical build of the device. Some of the aluminum sheets will be cut into small pieces and bent to create brackets for mounting the funnels, heat dissipation funs, and other smaller units. Screws and nuts are used for fastening purposes wherever needed. The thermal resistant sheets will be placed on the metal sheet that has the heat dissipation fins attached to it, this is to prevent heat from conducting through the metal sheet and increasing the temperature of the chilling funnels.

Table 4 below shows the options considered for the thermal resistant material. The Foxnovo tape was selected as the heat resistant material of choice because of the quality, cost, and the application purposes as the DEI tape is mainly used to block radiating heat.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
010408 DEI	1.5" Heat reflective tape (30' roll)	• Withstands up to 2000°F	• Cost: $39 CAD • Has adhesive material for mounting • More radiation resistant than conduction resistant. • medium quality
B00N1QZY4M Foxnovo	5cm high temperature heat resistant tape (33m long)	• Resists up to 280°C	• Cost: $17 CAD • Has adhesive material for mounting • High consumer rating • Crocking and tear resistant • High quality

Table 4: summary of the thermal resistant materials considered

Refrigeration resources:

- Peltier Tiles
- Thermal Paste
- Heat sink fins
- Heat sink fans
- Air inlet fans

For refrigeration, the Peltier tiles are the main components as they are responsible for transferring heat from the cool funnels to the outside. The other components such as fins and fans are complementary to prevent the Peltier tiles from overheating. The thermal paste is placed on both sides of the Peltier tile to fill any microscopic air gaps to increase the heat transfer efficiency.

Table 5 below shows the Peltier Tiles considered for the design. The Peltier tile of choice was the YIKESHU Peltier tiles, this is because a wattage Peltier tile requires a bigger heat sink to dissipate the heat. From the team's calculations, it was decided that the tiles should operate at 40W, which can be achieved with the YIKESHU tiles. The YIKESHU Peltier tiles are also significantly cheaper.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
TEC1-12706 YIKESHU	60W Peltier tiles	• 4.3-4.6 A at 12 volts	• Cost: $33 CAD for 10 tiles
TEC1-12709 Ecowsera	138W Peltier tiles	• 9A at 20V	• Cost: $33 CAD per tile

Table 5 : summary of Peltier tiles considered to cool the chilling funnels

Table 6 below show the thermal pastes considered for the design. Due to all the thermal pastes having similar price per volume, the paste with the highest heat transfer properties was selected. Therefore, the thermal paste selected was the MX-4 ARCTIC

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
MX-2 ARCTIC	Thermal compound paste	• Conductivity: 5.7W/mK	• Cost: $2.38/ gram
MX-4 ARCTIC	Thermal compound paste	• Conductivity: 8.5W/mK	• Cost: $2.19/ gram

Table 6: Summary of thermal pasted considerations

Table 7 below shows the air inlet fans considered for this design. The fan of choice was the 800RPM thermaltake fan because it has a higher CFM value at marginal difference in power consumption. The fan speed can also be controlled because it has an analog input.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
CL-F016-PL20BU-A Thermaltake	800 RPM air inlet fan	• CFM: 129.6 • Power: 2.76W • 200X200mm	• Cost: $22 CAD • Noise: 28.2 dBA • Analog input to control speed
1500 PWM Noctua	1500 RPM air inlet fan	• CFM: 100 • Power: 2.4W • 140x140mm	• Cost: $18.95 CAD • 29.7 dBA • Analog input to control speed

Table 7: summary of the air inlet fans considered.

Table 8 below shows the heat sink fins considered for this design. The fin of choice was the Sodial fin because it has long fins and base dimension. Out of the three fins considered, it has the largest surface area which is needed for maximum heat dissipation.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
082639 SODIAL®	Aluminum heat sink	• 150x69x37	• Cost: $15.39 CAD • Long fins • Best base and fin length combination
M7170904017 Yibuy	Aluminum heat sink	• 69x69x36	• Cost: $12.84 CAD • ideal length to width proportion • Long fins
CA-LDRK002 SenMod	Aluminum heat Sink	• 90x90x15	• Cost $20.98 CAD • Ideal length to width proportions • Really short fins

B07 Acogeder	Aluminum heat Sink	• 300x140x20	• Cost $28 CAD • Large base area • Short fins

Table 8 : Summary of the heat sink fins considered

Table 9 below shows the heat exhaust fans considered for this design. The fan of choice was the one made by Rosewill, this is because it has the CFM. High CFM is important to keep the heat sink fins cool and maintain high performing Peltier tiles.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
RFA-120-K Rosewill	Heat exhaust fan	• CFM: 74.78 • Power: 4.5	• Cost: $20 CAD
14BK3-3 Uphere	Heat exhaust fan	• CFM : 48.9 • Power:4.2W	• Cost: $18 CAD for 3 fans
B00NTUJTAK ARCTIC	Heat exhaust fan	• 53 CFM • Power: 4W	• Cost $27 CAD for 5 fans

Table 9: Summary of heat exhaust fans considered

Table 10: Electrical Components Required

Electrical resources:	
Power supply	Voltage regulator
12 and 18 gauge wires	Thermocouples
Relay switch	Humidity sensor
Infrared sensor	Arduino

As for the Electrical components, it can be split furthermore into two categories, powering the system and controlling it. For the power aspect, a 600W power source will power the entire system. A 12-Gauge wire will be used to transfer any high amperage loads, and an 18-Gauge wire will be used for the lower amperage loads. A voltage regulator is required to step down the voltage from 12V to 5V, which is required for the Arduino. For the control components, the Arduino will be used as the main controller of the circuit, jumper wires are used to connect the sensors and relay switches. Relay switches that are controlled by the Arduino will be used to power the Peltier tiles and exhaust fans when needed. The three sensors to be used are; thermocouples to measure the temperature of the air entering and leaving of the entire system, humidity sensors to measure the humidity of the air entering and leaving the chilling funnel, and an infrared sensor to monitor the surface temperature of the heat dissipation fins. **Error! Reference source not found.** in the appendix shows the electrical circuit diagram of the system.

There were not many options on the market for 720W power supply unit. The two optioned that were sensible for the design were to buy a desktop power supply and modify it to output the 12V required for many of the electrical components, or to purchase a power supply unit that outputs the 12V required. While the desktop power supply is significantly cheaper, the team decided to go with buying the normal power supply in case any unanticipated issues arise from using the modified desktop power supply.

Table 11 below show the voltage regulators considered for the design. It was decided by the team to use the Dork voltage regulator because it has a heat sink, inputs and outputs can be monitored on the LCD screen, and adjustments to the output can be made easily.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
LM2596 Lysignal	Voltage regulator	• Input: 3V-35V • Output 1.5V-35V • 3A	• Cost: $7 • No heat sink
180081 Dork	Voltage regulator	• Input: 6V-55V • Output 10V-50V • 5A	• Cost: $39 CAD • Physical adjustment knobs • LCD screen • Has heat sink

Table 11: Summary of voltage regulators considered

Table 12 below shows the relay switches considered for the design. The relay switch of choice was the unit made by Yeeco as it is able to control higher currents, does not create system noise, and the output can be controlled by the Arduino.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
101-70-102 Sainsmart	8 channel relay module	• H/L control. Can't edjust output as needed.	• Cost: $16CAD • Noisy. Might affect the reading on the sensors close to it
Y13000199 Yeeco	2 channel relay module	• Imax: 30A at 12 volts	• Cost: $13 CAD • Relay module requires 12V

Table 12: summary of relay switches considered.

Table 13 below shows the Infrared thermometers considered for the design. Even though both sensors are manufactured by the same company, the supplier KEYSTUDIO was preferred because the connection pins were included and soldered to the sensor. KEYSTUDIO also supplies the program library with the purchase.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
Mlx90614 KEYESTUDIO	Infrared thermometer	• Temperature range: -40 to 125 °C • Accuracy: ± 0.5°C	• Cost: $20CAD • Sensor library available online. • Pins soldered
Mlx906ESF-BCC FidgetGear	Infrared thermometer	• Temperature range: -40 to 125 °C • Accuracy: ± 0.5°C	• Cost: $19.5 CAD • Pins require soldering

Table 13: Summary of infrared thermometers considered

Table 14 below shows the temperature sensors considered for the design. It was decided to use the XCSOURCE thermocouples because both sensors have the same specification, but the one from XCSOOURCE is water resistant and better quality.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
KY-013 Global	Analog temperature sensor	• Temperature range: -55 to 125 °C • Accuracy: ± 0.5°C	• Cost: $3.6CAD • Cheap quality
DS18B20 XCSOURCE (R)	Temperature thermocouples	• Temperature range: -40 to 125 °C • Accuracy: ± 0.5°C	• Cost: $16 CAD for 5 sensors • High quality • Water proof

Table 14: Summary of temperature sensors considered

Table 15 below shows the sensors considered for the design. Both sensors are made by the same manufacturer, but one sensor is more accurate. Therefore, it was decided to use the DHT22 sensors because it has a greater measuring range and higher accuracy.

Model and manufacturer	Description	Specifications	Benefits/ Pitfalls
DHT11 Gikfun	Humidity and temperature sensor	• Temperature range: 0 to 50 °C ± 2°C • Humidity range: 20-95% RH ± 5%	• Cost: $12 CAD • Low accuracy • Sensor calibrated • Code library available
DHT22 Gikfun	Humidity and temperature sensor	• Temperature range: -40 to 80 °C ± 0.5°C • Humidity range: 0-99% RH ± 2%	• Cost: $13 • Sensor calibrated • Code library available

Table 15: Summary of Humidity sensors considered

3.2. Budget Updates

Since the initial project proposal was submitted in November of 2018, the budget has been updated to reflect a better understanding of the scope and goals of the project. The estimated current cost for the completion of the project is $32,205.00. The initial proposal budgeted $20,822 for the completion of the project, but severely underestimated the amount of computational work and testing that would be undertaken to optimize and design the system. This new budget is found in *Project Budget*

In this updated estimate, currently $18,066.55 in time and materials has been consumed. Analyzing time spent on completed deliverables, the initial proposal was predicted to take $3000 to complete and 30 hours but ended up consuming more time than anticipated at 32 hours, which raised the cost to $3200. The Proposal Summary memo by comparison came in under budget at $1050, which was $950 under the predicted cost for the deliverable. Design calculations and computational studies are currently ongoing, but there is still 13 hours of time left that is allotted to this task out of the 45 that were estimated.

Purchasing and sourcing of materials is also anticipated to come in under budget as final purchasing is being completed by February 25th. This deadline has been imposed in our schedule to allow ample time for manufacturing of the prototype design. For this task, 15 hours were allotted and only 7.5 have been used.

The budget for Manufacturing, Development and Testing was set at 50 hours and currently 24 hours have been used. The current time consumed accounts for all time spent developing CAD models and manufacturing drawings as well as the testing of Peltier modules to enhance the heat dissipation when integrated into a complete Prototype.

The cost of the completion of the Interim report was set at 40 hours and $4000 to compile all gathered data and complete additional research to support the development of the project. At the completion of this report, 36 hours were used at a total cost of $3600.

Tasks that will be starting later in the semester include the Final Design Report and Marketing Development, which includes a poster for the Design tradeshow. These two tasks were allotted a total of 50 hours and $5000 to complete. It is estimated that given the estimation from other tasks, these will be completed on time and under budget. This trend of coming in under budget for most tasks is a good development for the project but shows that the number of hours being quoted can still be refined based on the expertise of the group.

3.3. Schedule Updates

The weekly deliverables for this project are broken down in the Gantt chart in Appendix. Major upcoming deliverables are the manufacturing of the Final Prototype and finalizing additions that need to be made to the design to improve heat transfer. Simulation of the system will continue for several weeks still and completed using ANSYS Fluent. Manufacturing and prototype testing are currently on schedule and manufacturing will conclude by March 8th. Progress has been good on the design and given that the project is currently under budget, al targets are being met.

3.3.1. Anticipated Challenges

There are several anticipated challenges that will likely be encountered in the construction of the final prototype. The first concern is maintaining the cooling system at the desired temperature, which will be difficult because the Peltier tiles will be mounted using directly to an aluminum plate. The tile will be held there under pressure as it will be sandwiched between the plate and an aluminum fin which will amplify the heat transfer. Some losses will occur in this setup due to the holes drilled for the mounting hardware, which will create small gaps between the cooling compartment and heat rejection side of the device. There will also be heat loss to the surroundings if the sides of the tile are not insulated. If this loss proves to greatly impact the design and results, a press-sensitive adhesive insulating material will be used to fill the gaps.

Another anticipated challenge is heat dissipation. A solution is currently in progress to address this by optimizing the air flow rate that is used to dissipate the heat. Once investigative cases have been completed, the fan speed will be controlled, or more powerful fans might be purchased that can provide the required airflow.

3.3.2. Unanticipated Issues During Testing

To decrease the possibilities of unanticipated cooling issues occurring in the final prototype, several tests were conducted to evaluate the design theories and calculations before committing to the final build.

The first test completed measured both the surface and fin tip temperature. The setup for this experiment consisted of a finned surface for cooling, one Peltier tile and one CPU fan for heat dissipation that was composed of a fan and a heat sink. Thermal paste was used on both sides of the Peltier tiles to increase contact and heat transfer. Let it be noted that at this stage of testing, the proposed design consisted of having finned surfaces open to the environment and not within a closed funnel system. The Peltier tile was running at 11.6V and 3 Amps, with a desired set point of 18°C.

The first issue encountered in the test was the CPU fan's inability to exhaust the heat being generated on the hot side of the Peltier tile. As the test proceeded, the finned surface on the cooling side started to warm up, after the initial cooling that occurred. This is because the CPU fan was not able to dissipate the heat, which caused the Peltier tile to overheat and effect the cooling capability of the tile. The results of the test were not conclusive as there are two possible reasons for the finned surface not cooling to the 18 °C setpoint. The first possibility was that the finned surface was in open environment and there was no insulation to separate the hot side from the cold side and that heat was constantly being absorbed from the surrounding. The second possibility is that the CPU Cooler was not strong enough to prevent the Peltier tile from overheating.

The second experiment conducted used ice water to dissipate heat from the heat sink rather than using a CPU fan. Additionally, the cooling finned surface was replaced with a flat plate, similar to how the prototype design will be assembled. At this stage in the design process, the proposed design was changed from an open system with a finned surface for collecting water, to a flat surface design within a closed channel. Two aluminum plates with different thicknesses were tested; Sample A was 64 x 51 x5mm and sample B was 80 x 70x 2.5mm (L x W x t). The results from the second test were more conclusive, as seen in Figure 30. The temperature of sample B reached 17.6°C at the edges after 6 minutes of cooling. Sample A, which was a thicker plate, cooled to -2.5°C at the center and 9°C at the edge of the plate. From the temperature measured at the various points around the plates, it is possible to verify that the theoretical calculations of efficient conduction are accurate.

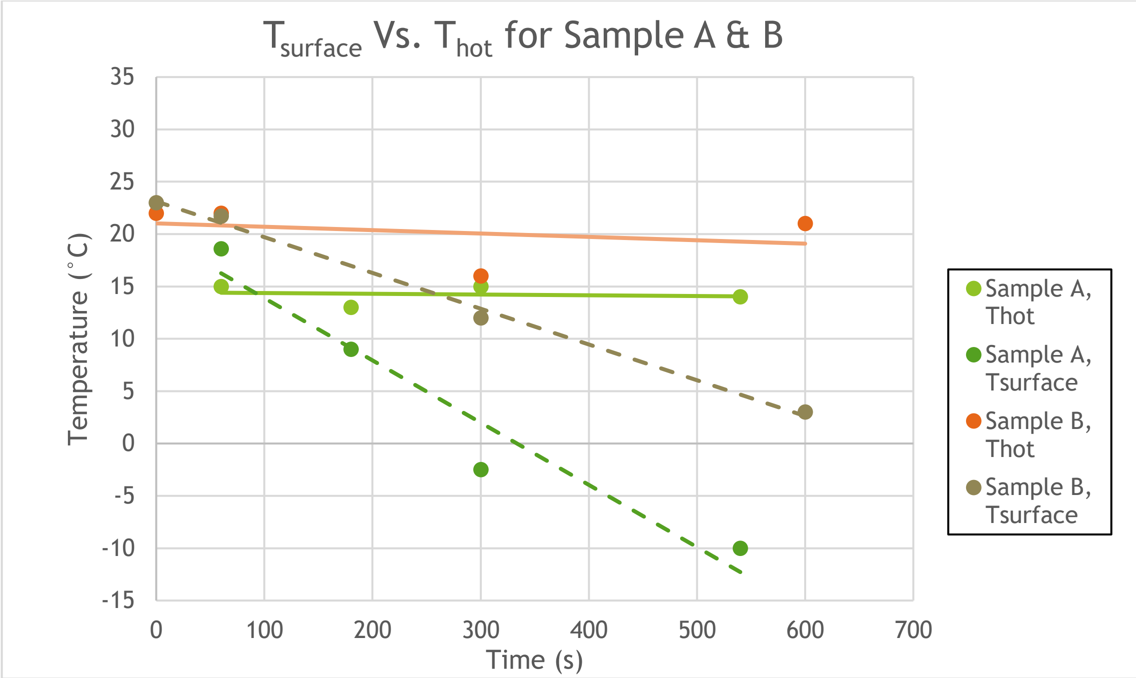

Figure 30: Temperature Profiles after 6 minutes operation

From Figure 30, it can be observed that the slope of sample A is steeper than sample B when measuring temperature at the plate and heat sink. Hence it is concluded that using a thinner plate will result in rapid cooling, however, the thicker plate provides sufficient cooling to meet the project requirements. Between the undertaking of these two experiments, the initial design was changed dramatically and greater understanding of Peltier tile spacing was achieved. With this knowledge, the possibility of future unanticipated challenges is reduced for this design.

One conclusion that was drawn from the experimental results, was that for a better cooling distribution over a larger surface area, it is advantageous to use several Peltier tiles operating at lower power rather than one Peltier tile operating at twice the power. If one Peltier tile is operating at 40W, vs. two tiles operating at 20-30W, it will take a higher airflow and finned surface area to cool down one tile that is creating a higher heating load vs two tiles, which require smaller fins and a slower air flow rate to dissipate the heat.

4.0. Conclusions & Recommendations

4.1. Conclusions

After the completion of calculations, and through verification from experimental tests, it was determined that the proper aspects of the design were being focused on and that challenges with heat dissipation and proper cooling of inlet air would be properly handled within the system design. By changing from the initial concept design, which was an open system with a combination of natural and forced convection, to a closed system which is being modelled as forced internal convection, the amount of control over the yields of water are greatly improved. A closed system also allows greater control of heat dissipation, since the airflow can be more directed within a channel than it can be on an open surface.

One development that came late in the process of the design was simulating the type of flow that was desired to cause mixing and heat transfer within a channel. This came about after it was requested that CFD simulation be completed for a simplified model of the system to determine at what length in the channel the majority of the air was cooled to within 2 degrees of the setpoint temperature. As discussed in the literature review (1.2) above, a fully-developed laminar flow formed in the duct and produced a boundary layer which maintained the setpoint temperature, but it didn't mix with any of the other air in the duct and impact the temperature. Through investigation, it was determined that turbulent flow is optimal for heat transfer as it disrupts the boundary layer and cools down a greater amount of the air flowing through the channel. Investigation will be continuing to determine the optimal spacing or ribbing within the channel to improve mixing and ensure that the proper analytical model is being used for simulation purposes.

This project is currently progressing on schedule to a prototype completion date of March 29th. Given that testing has already occurred, and some difficulties have already been identified, there should be fewer unexpected issues with the manufacturing and testing of the design than there could have been if one of the original concept designs was used.

4.2. Recommendations

For future iterations of this design, several improvements are recommended, including the inclusion of solar panels to provide power to the design and potential uses for the heat being dissipated by the tiles. In the current prototype, all electrical power will be drawn from a central grid, which reduces design costs if the infrastructure already exists. For bringing this design to rural villages without a steady source of electricity, it would be essential to have a sustainable power source, like solar, that could assist in providing drinking water to a small family or community. For using the waste heat, it has been proposed that it could be channeled into a drying chamber to expedite the drying of food products, which can improve the longevity of produce. This is outside the scope of the current research but would be essential to a complete system being deployed in the developing world.

A future area of improvement the team could purse is a method of increasing the amount of heat transfer within the cooling channel. This can be done through the addition of surface area within the cooling channel. In Figure 31 below this is achieved through incorporating angled plates within the cooling channel which would increase the amount of turbulent flow. As mentioned in earlier sections turbulent flow is better for heat transfer due to the mixing of air. In addition, the incorporation of these channels will increase the time the air spends within the cooling channel which will allow for the reduction of power consumption. Since the air is within the system for longer period of time the tiles do not need to work as hard to immediately cool the air and instead can do so in a gradual process by slowly removing heat.

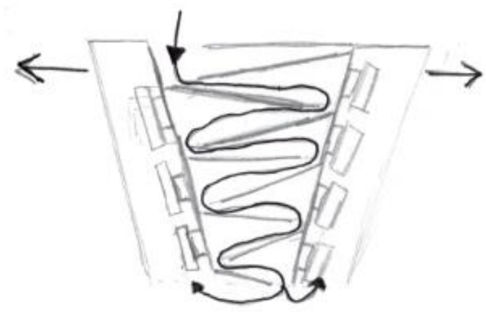

Figure 31: Spiral Improvement for Funnel

Another Improvement that can be made to the condensation system within the enclosure is adding thin hexagonal shells as shown in Figure 32 below. This will allow the humid air introduced into the enclosure to have contact with more surface area, which will improve the design's ability to condense dew. Humid air will be brought into the first central shell, when reaching the bottom, be passed into the next shell and when reaching the top of the 2nd shell be passed to the 3rd shell. The hexagonal shells will be thin aluminum sheets so they can be easily cooled by the Peltier tiles to dew point before the humid air enters the system and because of its incline, dew can be easily collected at the bottom

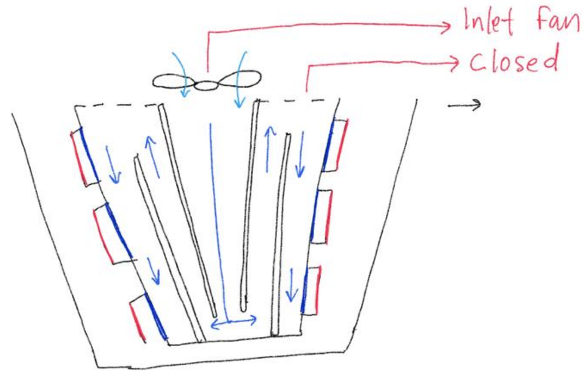

Figure 32: Potential Labyrinth design for Funnel

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I Appendix

A. Project Gantt Chart

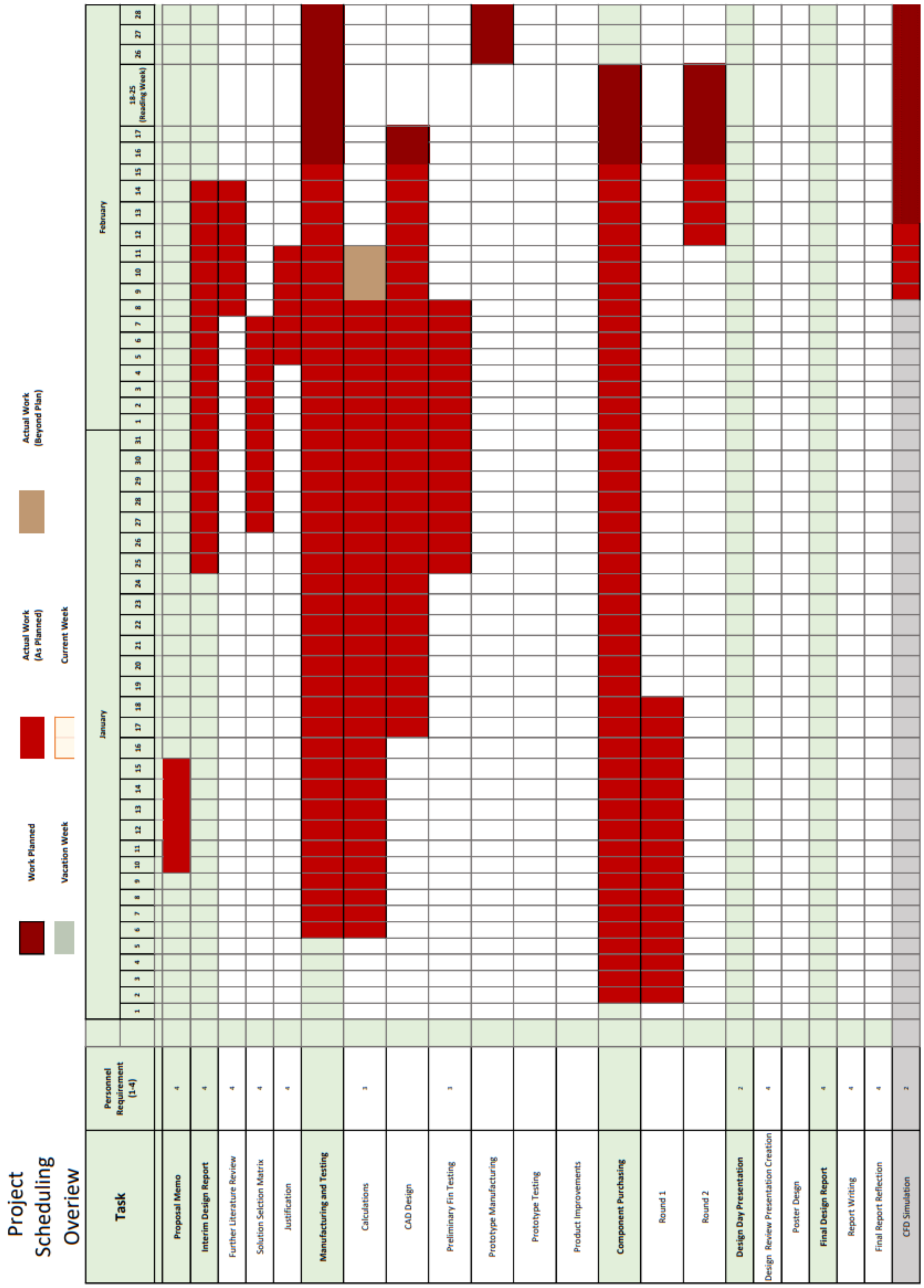


Figure 33: Preliminary Project Gantt chart (1/2)

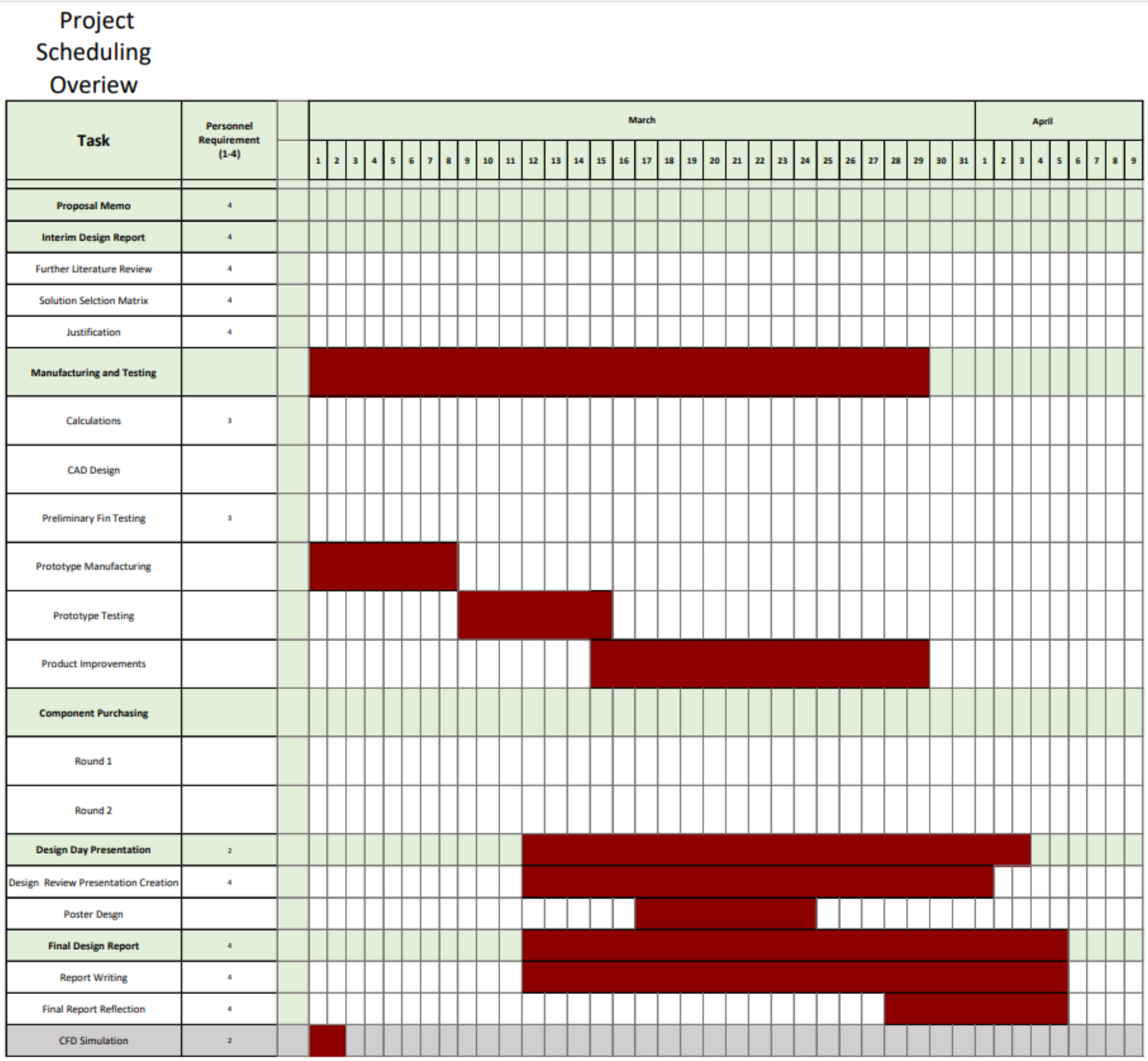


Figure 34: Preliminary Project Gantt chart (2/2)

B. WHO Guidelines for chemicals in drinking water that impact health

Table 16: WHO Chemical guideline values for drinking water

Chemical	Guideline value		Remarks
	mg/l	µg/l	
Acrylamide	0.0005[a]	0.5[a]	
Alachlor	0.02[a]	20[a]	
Aldicarb	0.01	10	Applies to aldicarb sulfoxide and aldicarb sulfone
Aldrin and dieldrin	0.000 03	0.03	For combined aldrin plus dieldrin
Antimony	0.02	20	
Arsenic	0.01 (A, T)	10 (A, T)	
Atrazine and its chloro-s-triazine metabolites	0.1	100	
Barium	1.3	1 300	
Benzene	0.01[a]	10[a]	
Benzo[*a*]pyrene	0.0007[a]	0.7[a]	
Boron	2.4	2 400	
Bromate	0.01[a] (A, T)	10[a] (A, T)	
Bromodichloromethane	0.06[a]	60[a]	
Bromoform	0.1	100	
Cadmium	0.003	3	
Carbofuran	0.007	7	
Carbon tetrachloride	0.004	4	
Chlorate	0.7 (D)	700 (D)	
Chlordane	0.0002	0.2	
Chlorine	5 (C)	5 000 (C)	For effective disinfection, there should be a residual concentration of free chlorine of ≥ 0.5 mg/l after at least 30 min contact time at pH < 8.0. A chlorine residual should be maintained throughout the distribution system. At the point of delivery, the minimum residual concentration of free chlorine should be 0.2 mg/l.
Chlorite	0.7 (D)	700 (D)	
Chloroform	0.3	300	
Chlorotoluron	0.03	30	
Chlorpyrifos	0.03	30	
Chromium	0.05 (P)	50 (P)	For total chromium
Copper	2	2 000	Staining of laundry and sanitary ware may occur below guideline value
Cyanazine	0.0006	0.6	

2,4-D[b]	0.03	30	Applies to free acid
2,4-DB[c]	0.09	90	
DDT[d] and metabolites	0.001	1	
Dibromoacetonitrile	0.07	70	
Dibromochloromethane	0.1	100	
1,2-Dibromo-3-chloropropane	0.001[a]	1[a]	
1,2-Dibromoethane	0.0004[a] (P)	0.4[a] (P)	
Dichloroacetate	0.05[a] (D)	50[a] (D)	
Dichloroacetonitrile	0.02 (P)	20 (P)	
1,2-Dichlorobenzene	1 (C)	1 000 (C)	
1,4-Dichlorobenzene	0.3 (C)	300 (C)	
1,2-Dichloroethane	0.03a	30[a]	
1,2-Dichloroethene	0.05	50	
Dichloromethane	0.02	20	
1,2-Dichloropropane	0.04 (P)	40 (P)	
1,3-Dichloropropene	0.02[a]	20[a]	
Dichlorprop	0.1	100	
Di(2-ethylhexyl)phthalate	0.008	8	
Dimethoate	0.006	6	
1,4-Dioxane	0.05[a]	50[a]	Derived using tolerable daily intake approach as well as linearized multistage modelling
Edetic acid	0.6	600	Applies to the free acid
Endrin	0.0006	0.6	
Epichlorohydrin	0.0004 (P)	0.4 (P)	
Ethylbenzene	0.3 (C)	300 (C)	
Fenoprop	0.009	9	
Fluoride	1.5	1 500	Volume of water consumed and intake from other sources should be considered when setting national standards
Hexachlorobutadiene	0.0006	0.6	
Hydroxyatrazine	0.2	200	Atrazine metabolite
Isoproturon	0.009	9	
Lead	0.01 (A, T)	10 (A, T)	
Lindane	0.002	2	
Mecoprop	0.01	10	
Mercury	0.006	6	For inorganic mercury
Methoxychlor	0.02	20	
Metolachlor	0.01	10	
Microcystin-LR	0.001 (P)	1 (P)	For total microcystin-LR (free plus cell- bound)
Molinate	0.006	6	

Monochloramine	3	3 000	
Monochloroacetate	0.02	20	
Nickel	0.07	70	
Nitrate (as NO_3^-)	50	50 000	Based on short-term effects, but protective for long-term effects
Nitrilotriacetic acid	0.2	200	
Nitrite (as NO_2^-)	3	3 000	Based on short-term effects, but protective for long-term effects
***N*-Nitrosodimethylamine**	0.0001	0.1	
Pendimethalin	0.02	20	
Pentachlorophenol	0.009[a] (P)	9[a] (P)	
Perchlorate	0.07	70	
Selenium	0.04 (P)	40 (P)	
Simazine	0.002	2	
Sodium	50	50 000	As sodium dichloroisocyanurate
dichloroisocyanurate	40	40 000	As cyanuric acid
Styrene	0.02 (C)	20 (C)	
2,4,5-T[e]	0.009	9	
Terbuthylazine	0.007	7	
Tetrachloroethene	0.04	40	
Toluene	0.7 (C)	700 (C)	
Trichloroacetate	0.2	200	
Trichloroethene	0.02 (P)	20 (P)	
2,4,6-Trichlorophenol	0.2[a] (C)	200[a] (C)	
Trifuralin	0.02	20	
Trihalomethanes			The sum of the ratio of the concentration of each to its respective guideline value should not exceed 1
Uranium	0.03 (P)	30 (P)	Only chemical aspects of uranium addressed
Vinyl chloride	0.0003[a]	0.3[a]	
Xylenes	0.5 (C)	500 (C)	

C. Project Budget

Project Costs

Date: 02-16-19

Invoice #: 1

To: Shohel Mahmud, Ph.D., P.Eng.
School of Engineering
University of Guelph
Guelph, ON N1G 2W1

Salesperson	Job	Payment Terms	Due Date
Ryan Krahn	Engineering Designer	Due upon receipt	4-30-19

Hours	Description	Unit Price (CAD)	Estimated Total	Current Total
30.00	Initial Design Proposal and Research	$ 100.00	$ 3,000.00	$ 3,200.00
20.00	Proposal Summary Memo	100.00	2,000.00	$ 1,050.00
45.00	Design Calculations & Computational Studies	100.00	4,500.00	$ 3,200.00
40.00	Interim Design Report	100.00	4,000.00	$ 3,700.00
15.00	Purchasing & Sourcing	100.00	1,500.00	$ 750.00
50.00	Manufacturing, Development & Testing	100.00	5,000.00	$ 2,400.00
20.00	Marketing Development	100.00	2,000.00	$ -
30.00	Final Design Report	100.00	3,000.00	$ -
20.00	Senior Engineering Consultant	100.00	2,000.00	$ 300.00
		Hourly Subtotal	27,000.00	14,600.00
Materials		Material Costs		
1.00	Materials Necessary to Complete Prototype Design	1,500.00	1,500.00	$ 1,388.10
		Subtotal	$ 28,500.00	$ 15,988.10
		HST	0.13	0.13
		Total	$ 32,205.00	$ 18,066.55

Quotation prepared by: Ryan Krahn

This is a quotation on the goods named, subject to the conditions noted below:
(Describe any conditions pertaining to these prices and any additional terms of the agreement.
You may want to include contingencies that will affect the quotation.)

To accept this quotation, sign here and return: ______________________________

Figure 35: Project Budget

D. Design Calculations

DOUALA, Cameroon WMO#: **649100**

Lat: **4.006N** Long: **9.719E** Elev: **10** StdP: **101.2** Time Zone: **1.00 (AFC)** Period: **92-14** WBAN: **99999**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB 99.6%	Heating DB 99%	Humidification DP/MCDB and HR 99.6% DP	99.6% HR	99.6% MCDB	99% DP	99% HR	99% MCDB	Coldest month WS/MCDB 0.4% WS	0.4% MCDB	1% WS	1% MCDB	MCWS/PCWD to 99.6% DB MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	8	22.0	22.5	21.1	15.8	24.9	21.8	16.5	24.5	5.8	27.5	5.3	27.5	0.8	80	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB 0.4% DB	0.4% MCWB	1% DB	1% MCWB	2% DB	2% MCWB	Evaporation WB/MCDB 0.4% WB	0.4% MCDB	1% WB	1% MCDB	2% WB	2% MCDB	MCWS/PCWD to 0.4% DB MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	2	7.8	33.9	27.1	33.1	27.0	32.8	26.9	28.9	31.2	28.2	30.8	27.7	30.6	3.1	220	(2)

	Dehumidification DP/MCDB and HR 0.4% DP	0.4% HR	0.4% MCDB	1% DP	1% HR	1% MCDB	2% DP	2% HR	2% MCDB	Enthalpy/MCDB 0.4% Enth	0.4% MCDB	1% Enth	1% MCDB	2% Enth	2% MCDB	Extreme Max WB	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	28.2	24.5	29.9	27.9	24.0	29.7	27.1	22.8	29.1	94.4	31.5	91.1	30.6	88.5	31.6	33.0	(3)

Extreme Annual Design Conditions

	Extreme Annual WS 1%	2.5%	5%		Extreme Annual Temperature Mean Min	Mean Max	Standard Deviation Min	Standard Deviation Max	n-Year Return Period Values of Extreme Temperature n=5 years Min	n=5 years Max	n=10 years Min	n=10 years Max	n=20 years Min	n=20 years Max	n=50 years Min	n=50 years Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	5.3	4.7	4.1	DB	20.5	35.0	2.2	0.5	18.9	35.4	17.6	35.7	16.4	35.9	14.8	36.3	(4)
(5)				WB	20.2	29.8	2.0	1.5	18.7	30.8	17.5	31.7	16.4	32.5	14.9	33.6	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	27.3	28.4	28.7	28.5	28.0	27.8	26.9	25.8	25.5	26.1	26.7	27.5	28.1	(6)
(7)			DBStd	1.47	0.81	0.99	1.13	1.22	1.15	1.00	0.96	0.90	1.01	0.97	1.12	0.76	(7)
(8)			HDD10.0	0	0	0	0	0	0	0	0	0	0	0	0	0	(8)
(9)			HDD18.3	0	0	0	0	0	0	0	0	0	0	0	0	0	(9)
(10)			CDD10.0	6322	571	524	573	542	553	506	488	480	483	517	524	561	(10)
(11)			CDD18.3	3281	312	291	314	292	294	256	230	222	233	259	274	303	(11)
(12)			CDH23.3	30054	3250	3218	3431	3000	2950	2210	1484	1240	1616	2065	2537	3053	(12)
(13)			CDH26.7	9738	1234	1285	1375	1159	1055	558	204	127	301	539	811	1091	(13)
(14)	Wind		WSAvg	1.2	1.1	1.3	1.3	1.3	1.2	1.2	1.3	1.4	1.3	1.2	1.1	1.0	(14)
(15)	Precipitation		PrecAvg	3851	36	64	168	230	272	429	695	755	626	410	134	35	(15)
(16)			PrecMax	4930	90	216	310	423	412	790	980	1240	1026	604	223	120	(16)
(17)			PrecMin	3233	1	5	47	124	168	223	330	498	347	196	40	6	(17)
(18)			PrecStd	497	26	51	78	69	68	147	165	248	203	112	57	31	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		33.9	34.2	34.2	34.2	33.9	32.2	30.4	30.0	31.2	32.1	33.0	33.2	(19)
(20)			MCWB		26.5	27.0	27.4	27.6	27.5	26.6	25.8	25.8	26.1	26.7	27.1	26.6	(20)
(21)		2%	DB		33.1	33.2	33.5	33.2	32.9	31.2	29.4	29.0	30.2	31.1	32.1	32.7	(21)
(22)			MCWB		26.6	26.9	27.2	27.4	27.2	26.5	25.4	25.4	25.9	26.5	26.8	26.5	(22)
(23)		5%	DB		32.2	32.9	32.9	32.7	32.1	30.5	28.8	28.0	29.3	30.4	31.3	32.0	(23)
(24)			MCWB		26.4	26.8	27.0	27.2	26.8	26.2	25.2	25.1	25.5	26.2	26.7	26.4	(24)
(25)		10%	DB		31.8	32.1	32.0	31.8	31.2	29.8	27.9	27.1	28.6	29.8	30.8	31.2	(25)
(26)			MCWB		26.4	26.6	26.8	26.9	26.6	25.9	25.0	24.9	25.3	26.0	26.6	26.4	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		28.8	29.2	29.4	30.2	29.0	27.2	26.4	27.0	26.7	28.1	28.5	28.7	(27)
(28)			MCDB		31.0	31.4	31.4	31.7	31.6	30.6	29.0	27.9	29.4	29.6	30.4	30.8	(28)
(29)		2%	WB		28.0	28.4	28.3	28.6	27.9	26.7	25.9	26.0	26.2	27.1	27.6	27.5	(29)
(30)			MCDB		30.7	30.9	31.1	31.2	31.3	29.9	28.4	27.8	28.9	29.7	30.0	30.4	(30)
(31)		5%	WB		27.3	27.7	27.7	27.7	27.3	26.4	25.6	25.5	25.9	26.6	27.1	26.9	(31)
(32)			MCDB		30.4	30.5	30.8	30.7	30.6	29.4	27.9	27.2	28.4	29.2	30.2	30.4	(32)
(33)		10%	WB		26.8	27.1	27.2	27.2	26.8	26.1	25.2	25.1	25.6	26.1	26.6	26.6	(33)
(34)			MCDB		30.1	30.3	30.4	30.3	30.0	28.7	27.2	26.7	27.8	28.7	29.6	29.8	(34)
(35)	Mean Daily Temperature Range		MDBR		7.9	7.8	7.9	7.8	7.4	6.0	4.5	4.1	5.3	6.5	7.1	7.5	(35)
(36)		5% DB	MCDBR		8.4	8.3	8.6	8.8	8.4	7.2	5.8	5.2	6.5	7.6	7.9	8.1	(36)
(37)			MCWBR		3.0	2.9	3.5	3.8	3.6	3.0	2.5	2.3	2.8	3.6	3.3	2.9	(37)
(38)		5% WB	MCDBR		7.9	8.0	8.1	8.2	7.9	6.5	5.1	4.6	5.8	7.1	7.4	7.6	(38)
(39)			MCWBR		3.3	3.5	3.8	4.3	3.7	3.0	2.5	2.5	2.8	3.8	3.5	3.1	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.803	0.832	0.805	0.570	0.503	0.524	0.522	0.507	0.450	0.467	0.587	0.739	(40)
(41)			taud		1.525	1.495	1.543	2.016	2.213	2.141	2.146	2.193	2.369	2.335	1.950	1.623	(41)
(42)			Ebn,noon		597	595	615	763	796	767	773	802	865	855	747	629	(42)
(43)			Edh,noon		294	308	294	180	143	151	152	148	127	131	190	262	(43)
(44)	All-Sky Solar Radiation		RadAvg		4.92	4.86	4.89	4.81	4.60	3.92	3.50	3.36	3.90	4.27	4.51	4.77	(44)
(45)			RadStd		0.16	0.18	0.22	0.14	0.21	0.28	0.28	0.18	0.18	0.17	0.16	0.13	(45)

Nomenclature: See separate page

Figure 36: ASHRAE Handbook Data for Douala, Cameroon

From ASHRAE (a) Thot = 33.9°C, HR = 24.5 g/kg da (b) Tcold = 21.1°C, HR = 15.8 g/kg

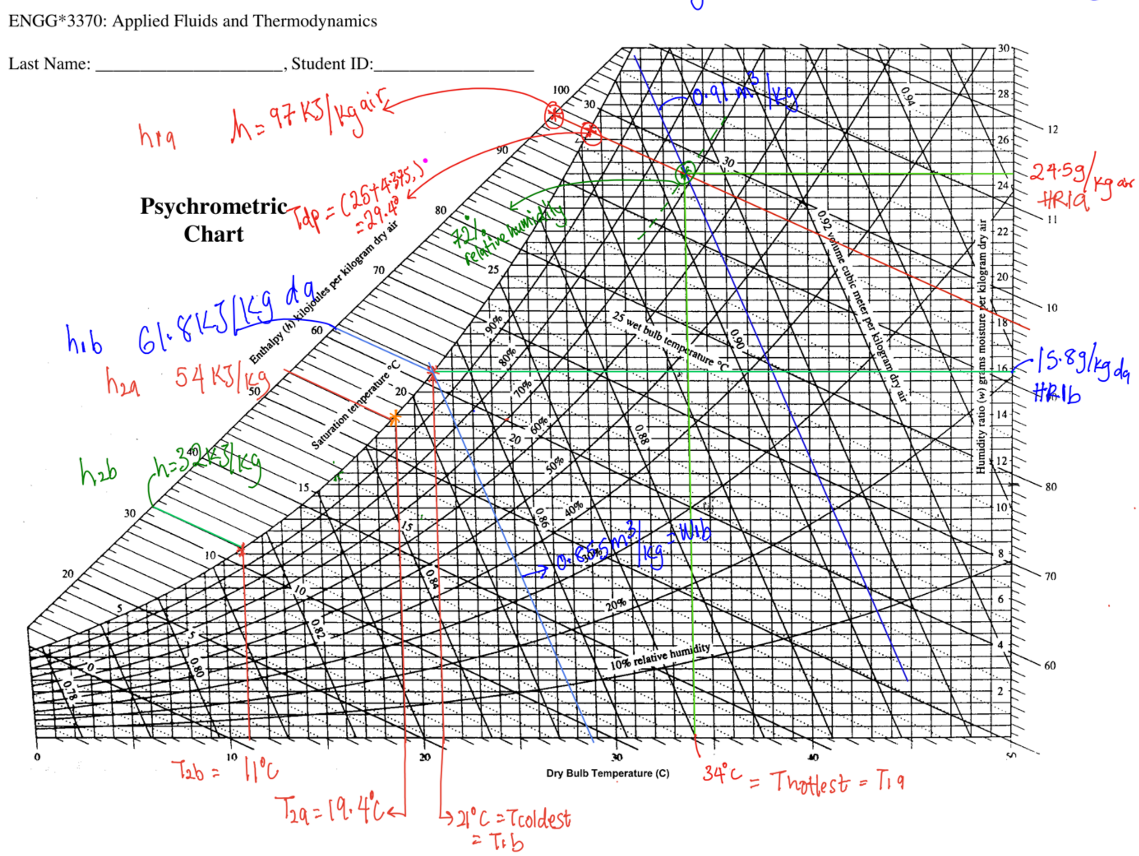


Figure 37: Psychrometric chart used in calculations

+ Overview of Calculations & Steps.

Initial Conditions of Environment:

Extreme Hot & Humid

$T = T_\infty = 34°C$

$\omega = 24.5$ g/kg dry air

DP = 28.2°C

P_a = 1atm ∴

From psychrometric:

Relative Humid. = 72%

Tsat = 29.4°C

Enthalpy = 97 KJ/kg (h_1)

Specific volume = 0.91 m³/kg (v)

Table A-4 (Condensate properties)

Saturated liquid water @ 20°C

h_w = 83.92 KJ/kg

V_{fw} = 0.001 m³/kg

Peltier Tile Cooling:

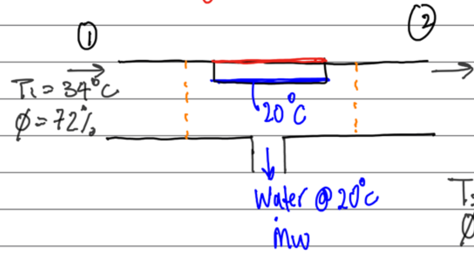
Extreme temp

T_C = −14°C

T_H = 50°C

Assume $\boxed{T_C \text{ Set to } 20°C}$ → Program for ΔT

System design



Assume:
T_2 air is same Temp as T_C
& ϕ_2 is fully saturated

$T_2 = 20°C$, $\phi = 100\%$ $\Big\}$ $h_2 = 57.5$ KJ/kg, $\omega = 14.75$ g/kg

Sample Calculations

Dry air balance: $\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a$

Water mass balance: $\dot{m}_{a_1} W_1 = \dot{m}_{a_2} W_2 + \dot{m}_w \quad \therefore \dot{m}_w = \dot{m}_{a_1} W_1 - \dot{m}_{a_2} W_2 = \dot{m}_a (W_1 - W_2)$ $\left[\frac{\cancel{kg}}{s} \times \frac{g}{\cancel{kg}}\right] = \left[\frac{g}{s}\right]$

Energy balance: $\dot{m}_{a_1} h_1 - \underset{\text{cooling}}{\dot{Q}_{in}} = \dot{m}_{a_2} h_2 + \dot{m}_w h_w \quad \therefore \dot{Q}_{in} = \dot{m}_{a_1} h_1 - \dot{m}_{a_2} h_2 - \dot{m}_w h_w$

[Cooling load] Q_L $\quad \dot{Q}_{in} = \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w$

Quantity of Water Produced.

Fan volume flow rate = 30 cfm
30 cfm = 0.849 m³/min

$\dot{m}_{a_1} = \frac{\dot{V}}{v}$

$\dot{m}_w = \dot{m}_a (W_1 - W_2)$

$\boxed{\dot{m}_{a_1}} = \frac{\dot{V}_1}{v} = \frac{0.849 \text{ m}^3/\text{min}}{0.91 \text{ m}^3/\text{kg}} = 0.933 \frac{\cancel{m^3}}{min} \times \frac{kg}{\cancel{m^3}} = \left[\frac{kg}{min}\right]$

*KIN USED 1 m³/min this is very large

$\boxed{\dot{m}_w} = 0.933 \frac{\cancel{kg}}{min} (24.5 - 14.75) \frac{g}{\cancel{kg}} = 0.1556\ (9.75) = 9.096 \text{ g/min}$

$\boxed{\dot{m}_w} = 9.096 \frac{\cancel{g}}{min} \times \frac{1 \text{ kg}}{1000 \cancel{g}} = 9.096 \times 10^{-3} \text{ kg/min}$

① * Note: Use Excel to calculate $\dot{V}$ (volume flow rate) required to achieve $\dot{m}_w$ required.

② $\dot{V}$ = volume of tunnel / air change rate of tunnel.

$\dot{Q}_{in} = \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w = 0.933 \left[\frac{\cancel{kg}}{min}\right] (97 - 57.5) \frac{KJ}{\cancel{kg}} - 9.096 \times 10^{-3} \frac{\cancel{kg}}{min} (83.92) \frac{KJ}{\cancel{kg}}$

$\dot{Q}_{in} = 36.09$ KJ/min $\quad \therefore \dot{Q}_{in} = 36.09 \frac{\cancel{KJ}}{\cancel{min}} \times \frac{1 \cancel{min}}{60 sec} \times \frac{1000 J}{\cancel{KJ}}$

$\dot{Q}_{in} = 601.5\ J/s$ or W $= \dot{Q}_L$ (from Total number of Peltiers)

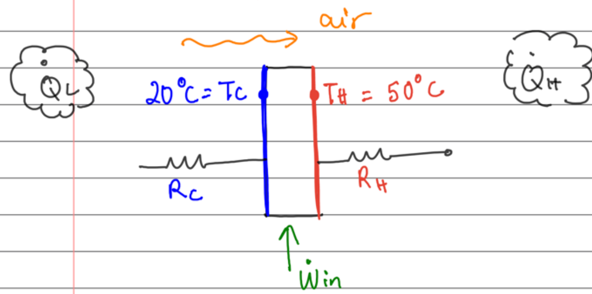
The Conclusion to this section is:

To condense air flow at a rate of 1CFM from 34°C inlet temp to Tc temp. 20°C will require 295 W and will produce 0.00446 L/min ⇒ {2.14L in 8hrs}

Figure 38: Calculating Quantity of Water Produced

SECTION 2

HEAT & MASS ANALYSIS



Assumption from Section 1
air Temp = 34°C
∴ $T_\infty = 34°C$

Cooling Side

$$\boxed{\dot{Q}_{conv} = hA_s(T_s - T_\infty) = \frac{(T_s - T_\infty)}{R}, \quad R = \frac{1}{hA}}$$

$$\dot{Q}_L = \frac{T_\infty - T_{sc}}{R_c} = \frac{34-20}{R_c} \quad \therefore \quad R_c = \frac{34-20}{\dot{Q}_L} = \frac{14}{32.1}$$

$$R_c = 0.4361\ °C/W$$

we want Ts to be as low as possible

$$\therefore \ (-ve)\ \overset{const}{\dot{Q}_L} R_c + \overset{const}{T_\infty} = T_s$$

∴ Rc has to be as Large as possible

∴ $R_c \geq 0.4361\ °C/W$

Heating Side

$$\dot{Q}_H = \frac{T_{sH} - T_\infty}{R_H} = \frac{50-34}{R_H} \quad \therefore \quad R_H = \frac{50-34}{\dot{Q}_H}$$

$$R_H = \frac{50-34}{88.3} = 0.1812\ °C/W$$

For T_{SH} to be as low as possible $\therefore \overset{const}{\dot{Q}_H} R_H = T_{SH} - \overset{const}{T_\infty}$

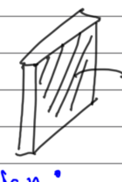
$\dot{Q}_H R_H + T_\infty = T_{SH} \quad \therefore \quad R_H \leq 0.1812\ {}^{o}C/w$

Figure 39: Sample Calculations Estimating R values

Cooling Side & Relation to Convection.

$R_C = 0.4361\ {}^{o}C/w$

$\therefore R_C = R_{eq} = R_{air_C} + R_{plate}$



* Aluminum

$A_c = 40mm \times 40mm$

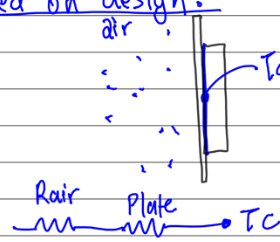
$= 1.6\times10^{-3}\ m^2$

K= 222 W/mK

$R_{plate} = \dfrac{L}{KA_c} \quad — ①$

$R_{air_C} = \dfrac{1}{hA_c} \quad — ②$

Based on design:



① Assume L ⇒ thickness of plate ⇒ $\boxed{1\times10^{-2}\ m}$

$R_{plate} = \dfrac{1\times10^{-2}}{222(1.6\times10^{-3})} = 0.02815$

② $R_{air_C} = \dfrac{1}{h_C A_C} \qquad R_c = R_{plate} + R_{air_C}$

$\therefore R_{air_C} = R_c - R_{plate} = 0.4361 - 0.02815 = 0.40795$

$\therefore R_{air_C} = 0.40795 = \dfrac{1}{hA_c} \qquad \therefore h = \dfrac{1}{0.40795\, A_c}$

$h = \dfrac{1}{0.40795(1.6\times10^{-3})} = 1532.05\ W/m^2K$

L ⇒ lenght Air is contact with.

$\boxed{Nu = \dfrac{hL}{K} = \dfrac{1532.05(0.25)}{0.025436} = 15057.89}$

* Assume laminar Air flow

$\therefore \quad Nu = 0.664 (Re_L)^{0.5} (Pr)^{1/3} \Rightarrow \boxed{Re_L = \left[\frac{Nu}{0.664 (Pr)^{1/3}}\right]^2}$

① $Re_L = 6.349 \times 10^8$ * greater than 5×10^5 ∴ not Laminar

② $\boxed{Re_t = \left[\frac{Nu}{0.037 (Pr)^{1/3}}\right]^{1/0.8}} = 1.17 \times 10^7$ * greater than 1×10^7 ∴ not Turbulent

➡ Design parameters have failed tests, must go back to re-design.

Velocity of air = $\boxed{\frac{Re\,\nu}{L} = V}$

$\boxed{V = \frac{\dot{V}}{A_c}} \quad \therefore \; A_c = \dot{V} \cdot A_c$

Hexagon A_c = Octagon A_c

$\therefore \quad 2(1+\sqrt{2})a^2 = \frac{3\sqrt{3}\,a^2}{2}$

- Sample Calculation with Hexagon

$V = \frac{0.849 \, m^3/min}{0.0259 \, m^2} = 32.779 \, m/\cancel{min} \times \frac{\cancel{1min}}{60s} = 0.5463 \, m/s$

$Re = \frac{Vh}{\nu} = \frac{0.5463 (0.25)}{1.5804 \times 10^{-4}} = 864.23 < 5 \times 10^5$

∴ Laminar flow

$Nu_L = \frac{hL}{K} = 0.664 Re_L^{0.5} Pr^{1/3} = 0.664 (864.23)^{0.5} (0.72904)^{1/3}$

$Nu_L = 17.568 = \frac{hL}{K} \qquad \therefore \; h = \frac{Nu_L K}{height}$

$= \frac{17.568 (0.025436)}{0.25}$

$= 1.787$

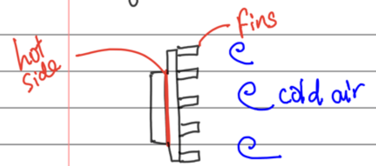
Figure 40: Sample Calculations for Cooling Parameters

HEATING SIDE & CONVECTION

$$T_f = \frac{T_{hs} + T_\infty}{2} = \frac{50+34}{2} = 42°C$$

Air properties at 42°C & 1atm, TABLE A-15
$K = 0.026768\ W/m \cdot K$
$\nu = 1.7212 \times 10^{-5}\ m^2/s$
$Pr = 0.7249$

System model



fixed — changeable

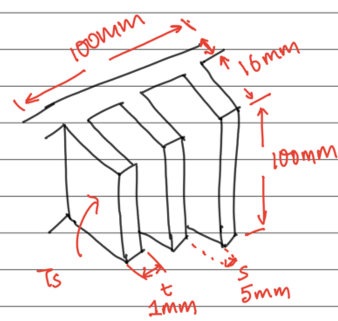
Th — Rfin — Rconv — cold air

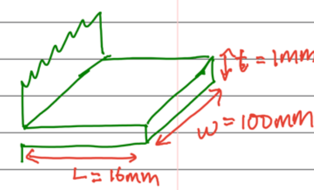
Table 3-3 pg 177
• or fig 3-43 pg 179

Calculating Rfin

$$\beta = \frac{1}{T_f} = \frac{1}{(42+273)} =$$

$$3.175 \times 10^{-3}\ K^{-1}$$





$m = \sqrt{\frac{2h}{Kt}}$ $L_c = L + t/2$ $A_{fin} = 2WL_c$ $\eta_{fin} = \frac{\tanh mL_c}{mL_c}$

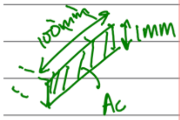
Assuming The hotter the peltier tile is, the more Convection Will be needed

Fin Case: Specified Temperature ($T_{fin,tip} = T_L$)

this assumption means that the Tfin,tip will be Known based on the Tbase (Temp of hot side) this will ensure no matter the Tbase Tfin,tip will be Cool enough for functionality

$\dot{Q}_{fin} = \eta_{fin}\, h A_{fin} (T_b - T_\infty)$
$\dot{Q}_{unfin} = h A_{unfin} (T_b - T_\infty)$

$$\dot{Q}_{total} = \dot{Q}_{fin} + \dot{Q}_{unfin} = \frac{\Delta T}{R}$$



h => based on forced Convection of fan.

$A_c = (1\times10^{-3})(100\times10^{-3}) = 1\times10^{-4}\,m^2$ $\quad K = 222\,W/mK$ (aluminum)

$P = 2(1\times10^{-3} + 100\times10^{-3}) = 0.202\,m$

Assume

$(T_b - T_\infty) \Rightarrow (50-20) = 30$

$m = \sqrt{\dfrac{2h}{Kt}}$, $L_c = L + t/2$, $\eta_{fin} = \dfrac{\tan(mL_c)}{mL_c}$, $A_{fin} = 2wL_c$

+ Calculating h using fin air gap Cross-sectional area

Cross-sectional area for air gap

+ Area 1 gap = $(5\times10^{-3})\times(16\times10^{-3}) = 8\times10^{-5}\,m^2$

+ for whole fin => assume air entering from one side of fin => $(n-1)\times 8\times10^{-5} = 1.2\times10^{-3}$

$\boxed{\text{Assuming Outlet fan at 50CFM}}$

To find the Velocity through fins:

$\dfrac{\dot{V}}{A_c \text{ of air of all fins}} \Rightarrow \dfrac{\dot{V}}{A_{air}(24)} = \dfrac{50(4.719\times10^{-4})}{0.0288}$

$V_{min} = 0.8192\,m/s$ => through all fin gaps at 50CFM & 24 fins

$Re = \dfrac{VL}{\nu}$ Check if Laminar or Turbulent

$Re = \dfrac{(0.8192)(100\times10^{-3})}{(1.72\times10^{-5})} = 4.76\times10^3$ ∴ Laminar

Assume uniform temp. accross fins

∴ $Nu_L = \dfrac{hL}{K} = 0.664\,Re_L^{0.5}\,Pr^{1/3}$ (Laminar)

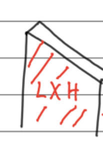
or

$Nu_T = \dfrac{hL}{K} = 0.037\,Re_L^{0.8}\,Pr^{1/3}$ (Turbulent)

Using $Re_L = 4.76\times10^3$, $Nu_L = 41.5$ ∴ $h = \dfrac{Nu\,K}{L} = 11.02\,W/m^2K$

$\eta_{fin} = \dfrac{\tanh(mL_c)}{mL_c} = 0.76$, $m = \sqrt{\dfrac{2h}{Kt}} = 9.962$

A_{fin} (1 fin) $= 2\times(L\times H) + 2(t\times h)$



A_{fin} (1 heat sink) $= n\times A_{fin} = 16\,(A_{fin}) = 0.05\,m^2$

Space not covered by fins

$A_{nofin} = (L\times W) - (n\times t\times L) = 0.008\,m^2$

$\dot{Q}_{fin} = \eta_{fin}\,h\,A_{fin}(T_b - T_\infty) = 12.89$

fins

$$\dot{Q}_{fin} = \eta_{fin} h A_{fin} (T_b - T_\infty) = 12.89$$
$$\dot{Q}_{nofin} = h A_{nofin} (T_b - T_\infty) = 2.78$$
$$\dot{Q}_{TOTAL} = 15.66 \, W$$
$$\therefore \boxed{R = \frac{T_b - T_\infty}{\dot{Q}_{total}} = 1.92 \, °C/W}$$

Figure 41: Sample Calculations for Heating Parameters

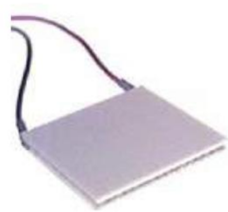
Hebei I.T. (Shanghai) Co., Ltd.

Thermoelectric Cooler

TEC1-12706

Performance Specifications

Hot Side Temperature (° C)	25° C	50° C
Qmax (Watts)	50	57
Delta Tmax (° C)	66	75
Imax (Amps)	6.4	6.4
Vmax (Volts)	14.4	16.4
Module Resistance (Ohms)	1.98	2.30



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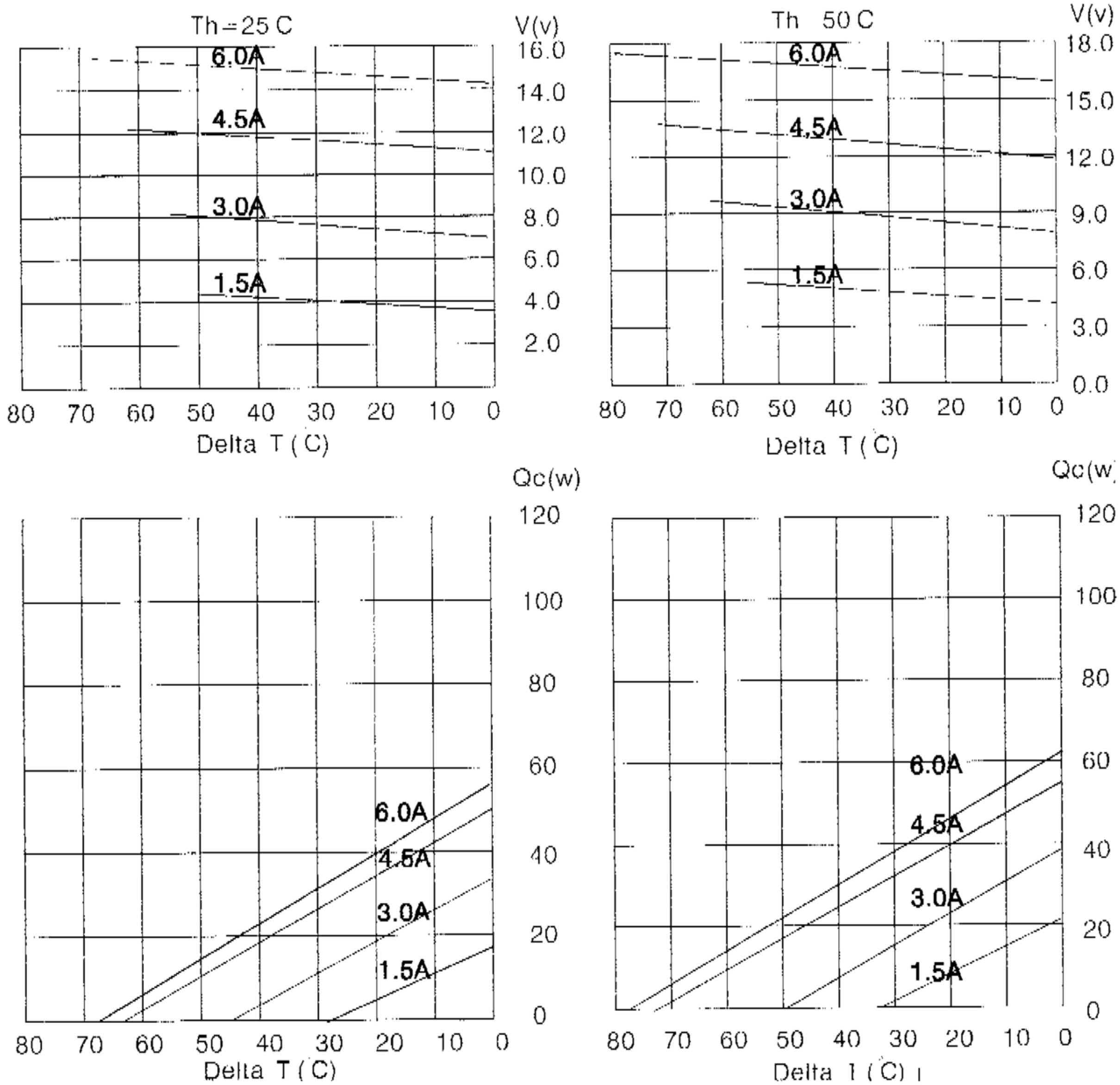
Performance curves:



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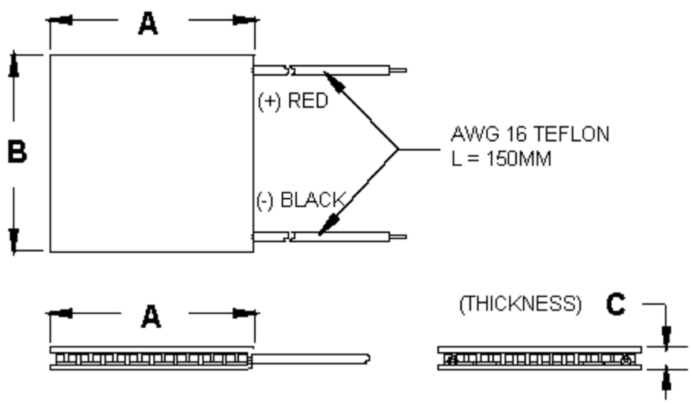

Ceramic Material: Alumina (Al_2O_3)
Solder Construction: 138° C, Bismuth Tin (BiSn)

Size table:

A	B	C			
40	40	3.9			

Operating Tips

- Max. Operating Temperature: 138°C
- Do not exceed Imax or Vmax when operating module.
- Life expectancy: 200,000 hours
- Please consult HB for moisture protection options (seeling).
- Failure rate based on long time testings: 0.2%.

Figure 42: Thermoelectric Cooler Datasheet