DESIGN AND TESTING OF AN EVAPORATIVE COOLING SYSTEM USING AN ULTRASONIC HUMIDIFIER

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STATEMENT OF PURPOSE

This paper topic was chosen at the behest of the course coordinator, Dr. Raghavan, who wanted work to be performed with a client's (Plastique Frapa Inc.) humidifying technology in relation to part of his research interest area: postharvest storage.

This paper was part of a senior year design project and reports the results of the experiment work that was performed in analyzing the effectiveness of the designed setup. Aside from the topic definition, the project was self-directed in terms of methodology and analysis. Although it should be noted that research associates of the department were instrumental in solving problems that occurred throughout the experimentation.

A co-author for this paper must be recognized for his contributions to the formulation of this paper. Michael Schwalb, worked very closely to assist in setup, conceptualization and other technical aspects.

EXECUTIVE SUMMARY

Postharvest losses throughout the world account for significant reductions in food supply which negatively impacts incomes of farmers, prices of food, and food availability. Many of these losses may be mitigated by providing reduced temperature and increased humidity. This report contains the details of the design of an evaporative cooling system which is intended to perform these functions. The system's design was based upon the humidifying capacity of an ultrasonic humidifier supplied by a Québec company, Plastique Frapa Inc. The evaporative cooling system initially took shape as an external input unit to feed cool, moist air into a control volume such as a cold-storage room. After trials which yielded a very limited temperature reduction (max 2℃) but an acceptable relative humidity increase (to 80%). A second setup with the unit directly inside the control volume yielded slightly greater temperature reduction (max 4℃) and significant relative humidity increase to the point of saturation (100%). This second setup's results were marred by significant condensation within the control volume. A third setup was established with the unit outside of the control volume, however with direct humidity input to the control volume. A similar temperature reduction and relative humidity were obtained as in the second setup however, condensation was limited because the control unit for relative humidity was adjusted to a lower level. The unit was also tested for its effect on produce weight loss. The setup for this test was consistent with the third setup for cooling. The results were compared with a similar experiment conducted by Dadhich et al. (2008), and the values for weight loss from this experiment were found to be considerably higher.

ACKNOWLEDGEMENTS

Throughout the duration of our project, we had quite a bit of help along the way. We relied on the experience and knowledge of a number of people who were able to point us in the right direction regarding a starting point or how to navigate around the many road blocks we encountered.

Our first thanks go to M. Yvan Gariepy who gave much of his time. His expertise in problem solving, instrumentation instruction and general testing design was instrumental in the completion of this project.

We would also like to thank Dr. Sam Sotocinal who lended his opinion to the setup design, guiding around the machine shop and helping troubleshoot when problems arose.

Dr. Raghavan warrants our thanks for providing the project as well as being a guide for project definition.

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INTRODUCTION

The global population in the year 2008 has hit an unprecedented level of 6.5 billion. It continues to rise drastically, and is predicted to hit 8 billion by the year 2025 (United Nations, 2008). In addition, with the economic rise of highly populous countries such as China and India, there is also an unprecedented rise in the overall global standard of living. From an agriculture perspective, these facts translate into an immense increase in the demand for food. Naturally, this entails a subsequent increase in the price of food and agricultural products worldwide. It should also be noted that industries such as the biofuel industry constitute market forces that contribute to the rise in the demand and price of agricultural products. Agricultural engineers are faced with the task of not only meeting the food requirements of the ever increasing global population, but to also maintain relatively low costs for food products. From basic economics principles, stabilizing the cost of a product undergoing an increase in its demand requires either an increase in its supply or a decrease in its cost of production. However, with agricultural products, such a task is not so simple: the amount of land that can be cultivated has reached its practical limit. Furthermore, soaring energy prices continue to push the costs of production from the growing stage to the transportation stage of production higher and higher. The only possible way left to stabilize the cost of food is to establish and implement new and efficient methods for crop production, post-harvest drying and storage, as well as distribution and transportation. It should be noted, that such Although all of these aspects are important when trying to tackle the challenge of meeting stabilizing the cost of food, this design project only explores the post harvest storage aspect of food production. In southeast Asia, postharvest losses range from 10%-37% for rice (International Rice Commission, 2002). Furthermore, in India, post harvest losses are in the range of 25-50%. These losses translate into a significant loss in the overall supply of agricultural products.

To realize the relevance of this project it is important to recognize the reasons for postharvest losses in relation to temperature and humidity. Losses related to temperature and humidity can include spoilage due to disease, over-ripening, negative physiological and compositional effects, loss of mass (produce water mass), and aesthetic appeal. The goals of postharvest cooling are to counteract these effects by the following mechanisms:

-slow and inhibit water loss

-suppress enzymatic degradation and respiration

-slow and inhibit the growth rate and activity of pathogens

-reduce the production of ethylene or minimize a commodity's reaction to ethylene

(Narayanasamy, 2006)

Some background on each aspect is necessary in understanding why it is important to analyze and correct this problem. The first problem of water loss is related to food structure, texture and appearance changes. Water loss after harvest is mainly dependent on two things: 1. the water vapour pressure deficit that exists between produce and its immediate environment, and 2. the surface transfer resistance to water vapour movement (determined by shape and internal resistances as well as surface resistances) which occurs now that the plant is no longer transpiring through its stomata. Transpiration resistance can be increased as a boundary layer of water vapour-saturated air is fashioned because of zero air movement, but the interrelation of factors can be seen in this case because to remove respiratory heat there must be air convection, so a balance must be found. Water loss affects commodities' quality and taste in that water increases the turgor pressure of fruits and vegetables which lends itself to a perceived "freshness" and "crispness" upon consumption. (Wills et al., 2007)

Very important to the economic side of production is that that fruits and vegetables are largely sold on the basis of weight, loss of water is indicative of a loss in revenue which farmers can ill afford.

Suppressing respiration is integral in food preservation; respiration refers to the breakdown of sugars and other compounds within the cell which releases carbon dioxide, water and heat to the surroundings (linking respiration to water loss). It is important to note that the rate of metabolic activity in cells increases with temperature; thus the importance of minimizing produce temperature (and environment temperature) is clear. Produce is typically harvested at ideal times (i.e. ripeness) depending on whether or not it is considered climacteric or non-climacteric, but remains a living entity; thus, cells perform the normal functions of aging (DeEll et al., 2003). Eventually a stage of senescence is reached whereby the cell continues respiring, but loses the ability to divide and becomes a dying entity. During the stage of senescence, cell-wall structure and composition changes and subsequently produce may become more palatable to a point, but these changes make the produce more susceptible to damage from external forces.

Also involved in ripening and eventual produce edibility and/or degradation is the chemical ethylene which is a hormone that may be either produced naturally by the commodity, or applied as a catalyst to hasten the ripening process. There is a large variability between commodities in terms of the actual production rate of ethylene and the commodities' response to ethylene. The final effect of pre-cooling treatments that are implemented is to improve the limitation of pathogens and disease found in vegetables. Postharvest diseases causing spoilage in perishable commodities can be significant as shown by Table 1.1 in Narayanasamy (2006). Pathogens and disease tend to flourish in high temperature and high humidity environments and produce may become affected at any point in the postharvest pathway to the consumer; in terms of pre-cooling, removing initial field heat is an important first step to limiting the influence of diseases.

Moreover, there is a tradeoff to be made with respect to humidity control and it is certainly beneficial to have a high humidity environment for fruits and vegetables with regard to water loss, but leaves the produce more susceptible to disease; another important management decision for producers.

Evaporative cooling is of specific interest to engineers concerned with the efficiency and energy demands of post-harvest storage. While providing a humid environment required for storage, humidifiers also offer the potential to provide some cooling. However, this is only the case with mechanical humidifiers and not with traditional humidifiers that use a heating element to evaporate water. The theory behind cooling effect from evaporation is simple: as water evaporates it absorbs latent heat from the surroundings (notably air) and as a result the ambient temperature is reduced. This phenomenon is precisely why a human being feels cooler when sweating.

In the context of food storage, much research has been done to explore the effects that evaporative cooling may have for extended preservation by counteracting the previously discussed activities of stored produce. In a study which examined the benefits of evaporative cooling for tomato storage by Mordi and Olurunda (2003), an average drop of 8.2℃ in comparison to ambient temperature was observed. Also, an increase in relative humidity of 36% was experienced. These changes in air quality had a significant impact on the storage life of the tomatoes as within the evaporative cooling unit, the tomatoes were stored for 11 days in comparison to 4 days of ambient conditions storage. Another study by Dadhich et al. (2008) also demonstrated significantly improved storage life for a number of different commodities due to a simple evaporative cooling technology. In their comparative study between an evaporative cooling environment (0.7 m^3 brick structure) and ambient conditions, % weight loss and visual quality were the factors of concern. After 7 days, produce within the evaporative cooler retained much more of its moisture as compared to the ambient conditions. For example, carrots lost 5% and 50% of their mass between the respective conditions and coriander leaves lost 15% and 76% of their mass. In a study by Thiagu et al. (1990) it was demonstrated that moisture loss in tomatoes is 6.5 times as great in ambient conditions (28℃ – 33℃, 45%-65% RH) as in evaporative cooling conditions (20℃ – 25℃, 92%-95%). They also demonstrated that lycopene development is significantly higher (double in this case) in evaporative cooling storage conditions than ambient conditions.

Plastique Frapa Inc., a small humidifier manufacturer has been exploring the benefits of an ultrasonic mechanical humidifier. The unit itself produces droplets in the range of 1-5 microns compared to specialized fine nozzle diffusers that produce droplets in the range of 70-80 microns. The advantage of the smaller water droplets are less condensation and greater evaporation rate. The goal of this design project is to explore and test an ultrasonic unit in order to identify its potential for use in a storage environment. As such, an evaporative cooling system will be design and built the in order to determine the cooling effect, if any, of the unit. Again, this cooling effect has the potential to significantly reduce the energy requirements of a storage system. After such tests, the unit will also be tested for its ability to disperse humidity and droplets into a storage volume. A complete storage environment with a controlled atmosphere will, however not be created and such design details are beyond the scope of this project. It should be noted that the unit itself will only be tested for cooling effect, and the ability to provide humidity without making condensation. The hypothesis is that the humidifier will provide significant cooling and little condensation will occur when used.

METHOD AND MATERIALS

THE ULTRASONIC HUMIDIFYING UNIT

The ultrasonic humidifier, as already mentioned, is manufactured by a small humidifier company called Plastique Frapa. The unit comprises of a water reservoir, a fan, and a small mechanical vibrating transducer. The reservoir is attached to the transducer by a small hose and provides water, through a slight pressure head difference, at such a rate that there is always a thin layer of water sitting on top of the transducer. When the unit is operating, the transducer vibrates ultrasonically with a frequency of roughly 1.65MHz. These mechanical oscillations break the thin layer of water into extremely fine water droplets. The water in the reservoir completes a circuit with the transducer and the fan and is controlled through a simple pinch float mechanism. When the water level in the reservoir is too low, the circuit is broken and the unit is no longer in operation. The unit also has very low power consumption of 48 watts and more specifically operates at a voltage of only 48 volts and a current of only one amp.

DESIGN OF A MIXING CHAMBER

There are two reasons why a mixing chamber is a key component to a storage system design. The first is to continuously provide moisture and air into the control volume and the second is to properly disperse it. Properly dispersing the water droplets outside the control volume is hypothesised to reduce the amount of condensation occurring within the control volume. Dispersing the air and water droplets outside the control volume also has the advantage of keeping the heat loss from the mixing fan outside the control volume. Nevertheless, the hypotheses and assumptions about the benefits of using a mixing chamber were verified by designing and building an evaporative cooling system without a mixing chamber. The amount of cooling and condensation that occurred with and without a mixing chamber will be compared and evaluated. The results of these tests will be presented and discussed in the following section of the paper.

Design Concerns and Considerations for a Mixing Chamber

One major concern when building a mixing chamber is heat gain from the surrounding ambient air. The possibility of insulating the mixing chamber and the pipes was explored and examined; however, it was not implemented as it was deemed not necessary. At first the amount of condensation and thus cooling that occurred within the mixing chamber was not expected to be all that significant. This assumption however, was not the case as will be shown in one of the following graphs. With an air velocity of 4.5 m/s, the relative humidity at the outlet was 80% (see figure 3). Nevertheless, the amount of time that a given amount of air was inside the mixing chamber was not long at all and at a velocity of 4.5 m/s it was less than 2 seconds. Therefore, the amount of heat energy that could be transfer from slightly warmer ambient air in such a short amount of time was deemed to not be significant. Again, it should be noted that an evaporative cooling system was built without a mixing chamber to examine if any significant losses of the system occurred inside the mixing chamber and whether the assumptions made at this stage of the design were precise. A more significant source of heat gain or cold air temperature losses was direct losses of cold air if all the pipes and the fan were not properly sealed. Therefore, all the fan and pipes were connected and sealed using a substantial amount of Styrofoam and duct tape. Another design concern was avoiding or limiting condensation from occurring within the mixing chamber. There are two potential causes for condensation inside the mixing chamber: the accumulation of droplets along the pipe walls, and the air, within the pipes, reaching the dew point temperature. The greater concern of the two was the accumulation of droplets along the pipe walls. However, the only way to limit this is by was to minimize the amount of air flowing perpendicular to the pipe walls, which was done in the most initial stages of the mixing chamber design.

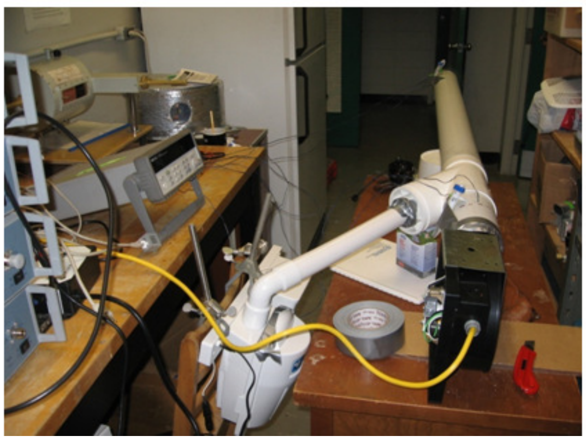
The placement of the inlet into the control volume for the air moisture mixture determines the distribution of the warm incoming air with the cold high moisture outgoing air. Since hotter air rises and colder air sinks, the inlet was placed high in the control volume. By contrast, the outlet was placed at the bottom of the control volume.

Furthermore, based on the psychometric chart calculation, a velocity of roughly 4.5 m/s in the 4 meter pipe is required in order to attain a relative humidity of 85%. The calculations are presented in the appendix. However, the velocity of the air exiting the fan was 10 m/s. Therefore a control for the fan had to be established in order to decrease the air output of the mixing fan. Because the fan was a non-shaded pole motor, it could not be connected to a variable speed control due to the fire hazard risk and the possibility of shorting out. Therefore, the fan had to be controlled physically by partly covering its outlet with duct tape.

After the right duct tape configuration was found in order to restrict the fan's air velocity to 4.5 m/s, a test was run in order to examine and compare the air quality at the inlet of the mixing chamber and the outlet of the mixing chamber. This initial test was run with a parallel configuration, in that the fan was connected to the mixing chamber in parallel with the humidifier. The following photo depicts the parallel and series configuration used for the mixing chamber.

Figure 1 - Images of parallel and series configurations

Parallel



Series

The following testing setup was used in order to examine the effect of the mixing chamber on the quality of air. Again, the setup of the experiments, is shown in the photo above.

Testing Setup

Materials

-4 Thermocouples

-data logger

-computer

-wet bulb readers

-2 x 4" pipe

-humidifier

-continuously running time data logger

As previously mentioned, the test was run with an air velocity of 4.5 m/s. However, the pressure created by the mixing fan was too great for the humidifier to operate and to more specifically, release water droplets into the mixing chamber. The test was then run at lower velocities however the same problem occurred. It was concluded that the pressure of the mixing chamber, mostly created by the fan, was far too great for the humidifying unit to operate. Therefore, the only plausible solution was to reconfigure the mixing chamber into a series configuration with the humidifier blowing directly into the fan. This orientation considerably lowers the pressure at the outlet of the humidifier and in fact created a negative pressure. In a series configuration, when the velocity of the fan was increased to 8 m/s, the negative pressure was so great that it actually sucked water straight out of the humidifier before it had a chance to transform into small droplets. Nevertheless, at 4.5 m/s, the unit operated efficiently. With a series configuration the greatest concern is that if the right amount of condensation accumulates on the motor of the mixing fan, it would short or burn the insulation of the motor. Nevertheless, this risk was minimal and deemed to be acceptable. During the first stages of testing, thorough monitoring was conducted in order verify whether this was in fact a minimal risk. The following graph represents the first test that was conducted using a series configuration with an air velocity of 4.5 m/s.

Figure 2 - Initial Test Results: RH and T_{dry} vs. time

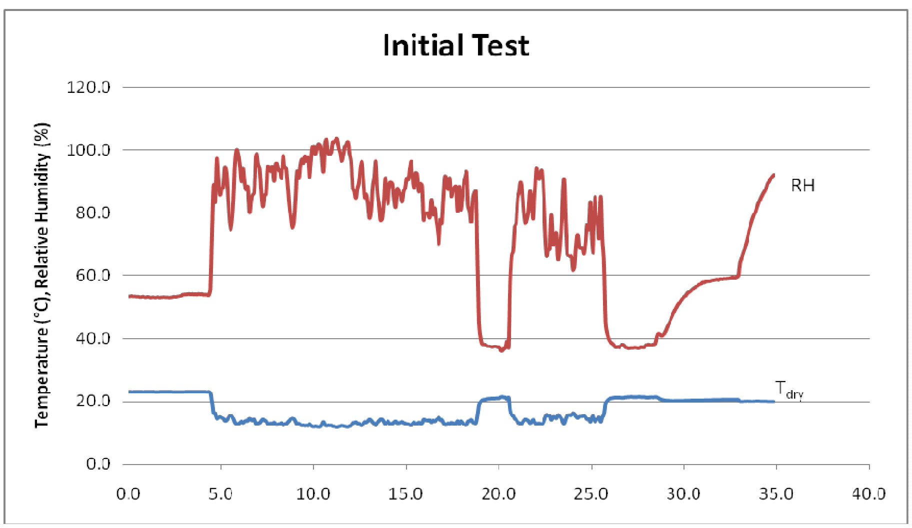
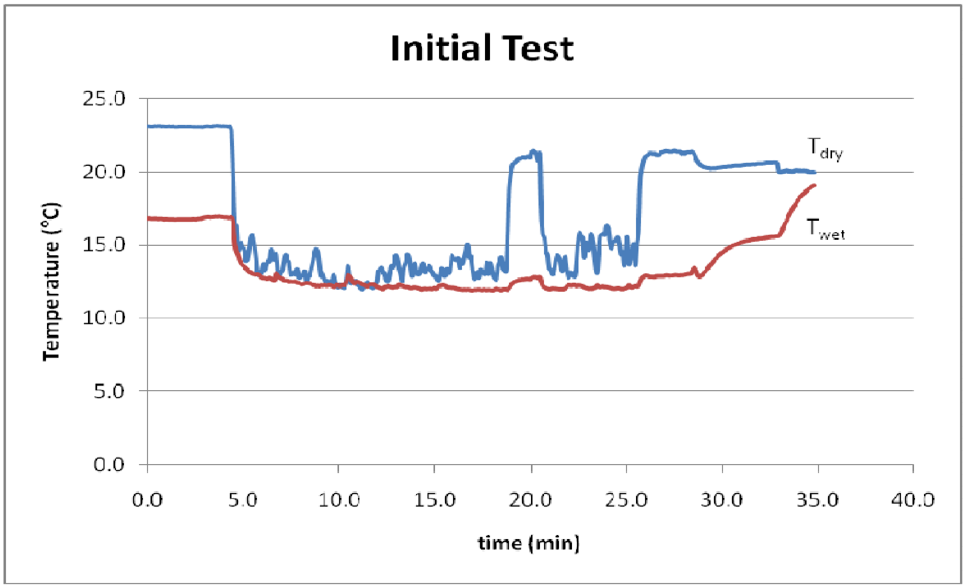


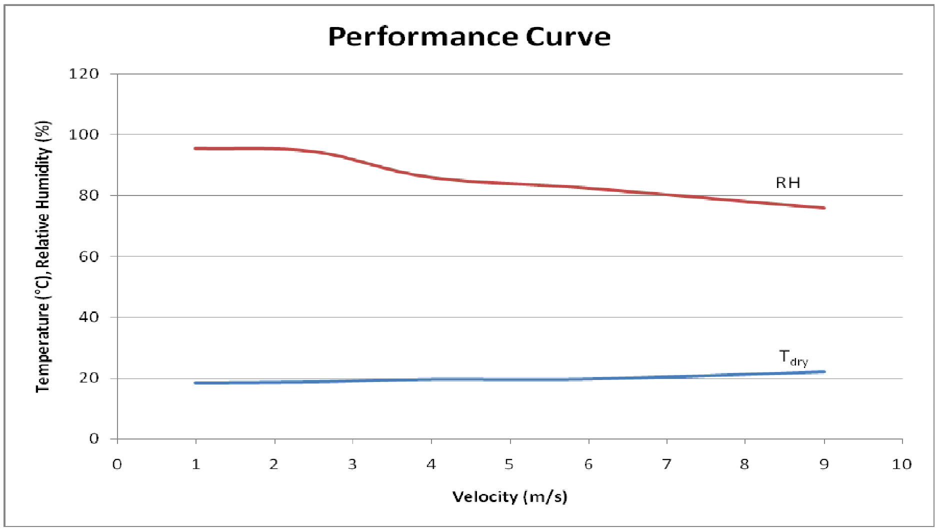
Figure 3 - Initial Test Results: T_{dry} and T_{wet} vs. time



The humidifier was only started after the 4 minute mark. This point is marked by the instantaneous drop in T_{wet} and T_{dry} in the above graph. The relative humidity was calculated by measuring both the wet bulb and dry bulb temperatures and then by calculating the relative humidity. However, measuring the wet bulb is not a simple task. In order to truly measure the wet bulb temperature, a wick on a thermocouple must be constantly saturated but not saturated to the point where droplets form on its surface. From the graph, before the humidifier was even started, the relative humidity was calculated to be 60%, which was far from the 22% value measure for ambient air. The dry bulb temperatures at the outlet of the mixing chamber were consistent with ambient dry bulb temperatures. Therefore, it was concluded that there were errors in precision of our wet bulb readings and another method for measuring relative humidity was needed. Data loggers were then used and placed at the outlet and the inlet of the mixing chamber. These data loggers measure relative humidity directly.

The quality of the air was measured at the outlet using these data loggers for various air velocities. The following performance curve was obtained. (See Appendix A for graphs detailing the tests which compose Figure 4.)

Figure 4 - Performance curve relating T_{dry} and RH to air velocity



At a velocity of 4.5 m/s, the relative humidity was roughly 80%.

EVAPORATIVE COOLING SYSTEM DESIGN

Design Parameters

As stated in the discussion of the design of the mixing chamber, the evaporative cooling system was designed to deliver a quasi-uniform mass of air in terms of temperature and relative humidity. The theoretical expectation for the design was that a control volume (in the present case, a cold storage room) would eventually be in equilibrium with the incoming air. Once the point of equilibrium was reached, it was postulated that the humidifying system would maintain the temperature and relative humidity at a relatively constant level.

Simply providing moisture to a control volume will provide a certain amount of cooling but only in a batch process: after saturation conditions are reached, no further cooling can be provided and the temperature will increase due to heat influx from the higher temperature ambient surroundings. For this reason, it was decided to provide an air flow

Evaporative Cooling Design Set-Up:

The set-up consisted of the following components:

- humidifying unit (MDFD-1)
- mixing chamber
- blower
- delivery pipe
- insulating foam to insulate door opening
- cold storage room at Bioresource Engineering Machine Shop

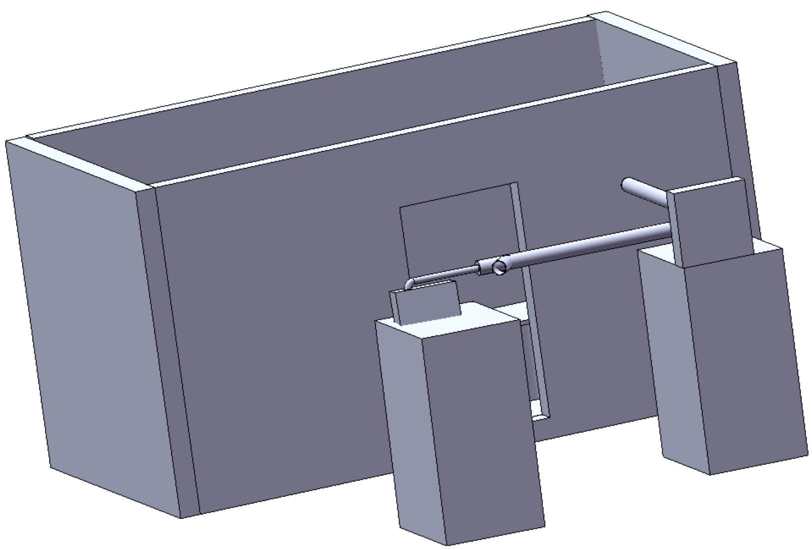
A representational model can be found in Appendix B which was created with Solid Works 3-D modelling software. This model does not reflect exact dimensions as some of the features are extremely difficult to model (i.e. the blower and the contours of the humidifier); because this is not a working model which is to be tested within the framework of Solid Works itself, it was decided to forego such detail.

As evaporative cooling works by removing heat from warm, dry ambient air to evaporate water inside a control volume, the cooling system was placed outside the cold storage room for access to the ambient air. For previous humidity control experiments, a delivery inlet had been placed in one of the walls of the cold room (See Figure 5). The inlet's diameter was a convenient dimension for the delivery pipe of this design, however it was difficult to access because of space restrictions in the antechamber (which was acting as a storage room) to the cold room. In order to overcome the challenge in accessing the delivery inlet, objects (such as a filing cabinet and fridge) were rearranged in the antechamber to facilitate proper spacing and direction for the evaporative cooling system. It is important to mention that these objects were not part of the initial design of the delivery apparatus. Unfortunately the physical design of the prototype evaporative cooler in this design was limited by space and budget. However, pursuant to this section is a suggested apparatus which will better suit most implementations in terms of space and budget.

Testing Procedure of Evaporative Cooling Unit

First Testing Setup

Figure 5 - First Setup: Humidifying unit outside control volume



Tests were previously performed to ascertain the performance curve of the delivery system in terms of minimum temperature obtained and maximum relative humidity of the inlet (to the control volume) air (See section entitled "Testing of Mixing Chamber"). The next step was to apply the overall set-up to the cold storage room to verify the hypotheses on humidification and cooling that were anticipated. These hypotheses related directly to the performance curve information (See Figure 4) as well as theoretical calculations based on psychrometric principles (See Appendix F).

In order to determine whether or not the set-up was effective or not, a threshold time value of approximately one day was chosen as the limit for time to observe cooling and humidifying effects and more importantly the stabilization of these effects.

The first test was run over a period of approximately 24 hours beginning on March 10, 2008 from approximately 15h00 to March 11, 2008 at 15h00. This test was run as a preliminary test in order to observe the general effects upon the quality of the air in the control room. A replicate test was run to verify these results beginning March 12, 2008 and ending March 23, 2008.

The parameters of this test were:

- full-time functioning of MDFD-1 unit
  - humidity control unit set to 100% RH and 5% variation
- air inlet velocity of approximately 2.5 m/s
- air inlet quality consisted of $T_{dry, min}$ = 19.5°C, and RH_{max} = 94%
- a small outlet was cut in the foam at the door to allow air to exit and reduce pressure build-up within the control volume

The expected end quality of the air was to be in equilibrium with the inlet air and be at a T_{dry} = 19.5 and RH = 94%. What was observed at the end of the trial period (24 hours later) was much different from the expected results. Based on the T_{dry} and T_{wet} readings, Relative Humidity was calculated using the method stipulated in Appendix X.

Dry-bulb temperature throughout this experiment did not drop as expected to 19.5°C as was being supplied. As can be seen in Figure X below, $T_{dry,CV}$ hovered between 22°C and 25°C throughout the entire experiment. There are a number of explanations for why the temperature did not drop as expected which follow the presentation of the graphs: fluctuation of ambient air temperatures, adiabatic nature of the cold-storage room, thermal load, and finally a flawed setup.

The results of the experiment yielded an increase in relative humidity to approximately 80% as shown in Figure 6. This part of the result can essentially be discarded because the method of measuring relative humidity relied partly on wet-bulb temperature measurements which were deemed inaccurate as discussed previously in the section detailing the performance curve derivation.

Figure 6 - Relative Humidity, $T_{dry,CV}$ vs. time with unit outside control volume

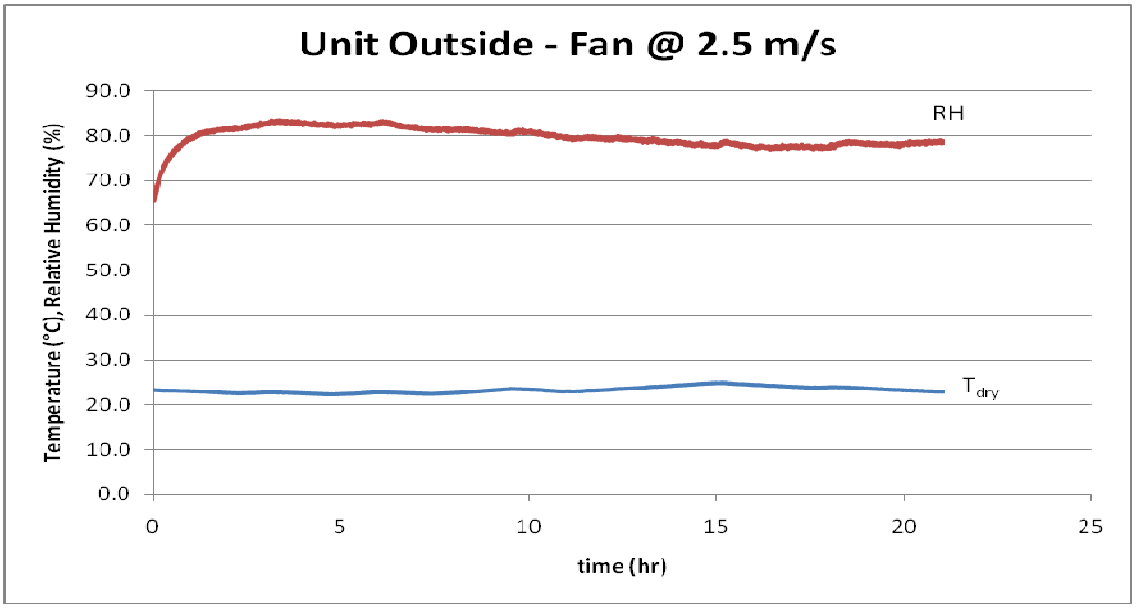


Figure 7 - $T_{dry,ambient}$ vs. time for unit outside control volume

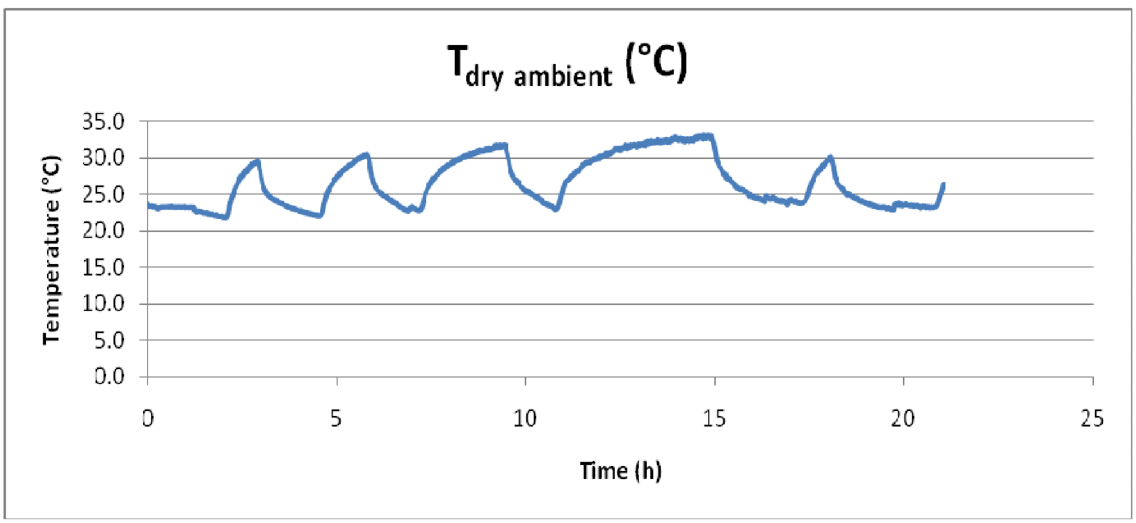
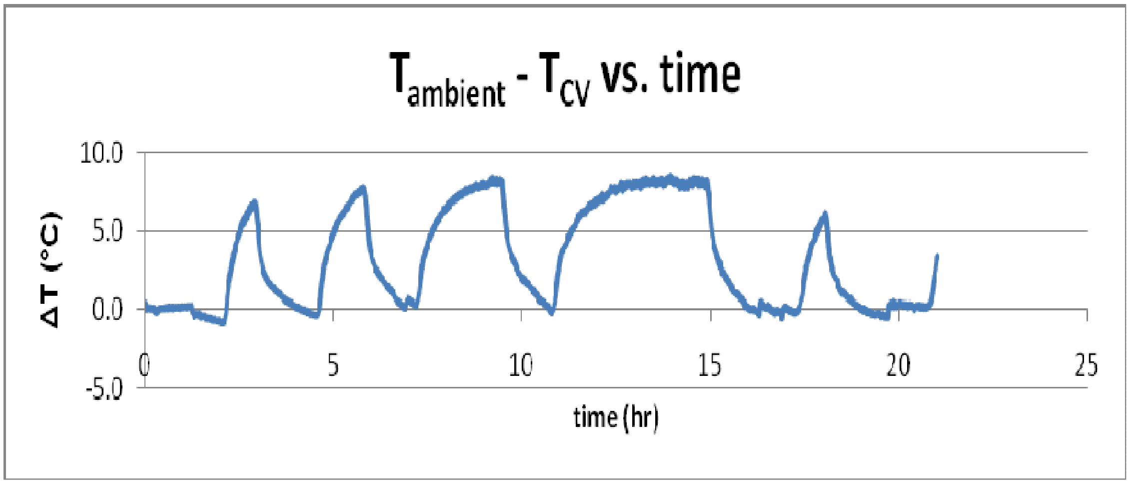


Figure 8 – Tdry,ambient – Tdry,CV vs. time for unit outside control volume



Temperature fluctuations:

As shown in Figure X above, $T_{dry,ambient}$ fluctuates greatly as it ranges between approximatley 22°C and 33°C. The system relies on humidifying and cooling ambient air which is then input into the control volume; it logically follows that as the ambient air warms and cools, the inlet air and consequently the control volume should warm and cool accordingly. In this experiment the previous statement is not accurate because the temperature of the control volume varies only slightly, though it does match the pattern of the ambient air fluctuations. This is apparent when attempting to remove the "noise" (i.e. ambient temperature fluctuations) to prove the effectiveness of the cooling system. In an effective system, the difference between ambient temperatures and control volume temperatures is expected to be relatively stable; however, in the case of this test the difference between the two temperatures has a very unstable, tooth-like shape.

Adiabatic cold-storage room:

The adiabatic nature of the cold-storage room is significant enough to resist changes to temperature as it presents a significant barrier between ambient conditions and control volume conditions. In non-adiabatic conditions, the control volume is somewhat transparent to the effects of ambient conditions and will consequently be influenced by said conditions. In this case, the cold-storage room is designed to effectively limit heat entrance and thus maintain stable temperatures.

Thermal Load:

After the first day long test was performed, very little cooling of the control volume air was observed. Part of the resistance to temperature change can be related to the large thermal mass that was within the cold storage room; the thermal mass was in the mostly in the form of PVC pipe sections but also some other miscellaneous articles (eg. wood chair, plastic tubing, foam). To obtain cooling from this system, it is important to recognize the system in its entirety. More plainly stated the evaporative cooling system must be concerned with cooling the mass inside the room as well as the air.

The cold room is designed to be (more or less) an adiabatic unit, but for the purposes of cooling its inside it cannot be considered an adiabatic system. There is a transfer of heat between the components inside the cold storage room: air, water, and objects. To simplify the system, the interactions can be broken up into two sections: i) air-water, and ii) air-objects. While this may not be an exact reflection of what is occurring, it does allow some analysis. In separating the interactions in this fashion, it is possible to derive an approximation for how much heat energy the thermal mass is supplying to the air. Because the driving force of heat transfer is a heat gradient – such as in Newton's Law of Cooling ($Q = hA[T_s-T_\infty]$) or a thermodynamic analysis ($Q = mc_p[T_2-T_1]$)- as the air in the control volume is cooled by transferring its heat energy to the water supplied by the humidified air stream, a temperature gradient between the air and the thermal mass is created. Consequently, the thermal mass loses heat to the air and the temperature of the air rises in response. In effect, the supplied moisture must cool both the air and the thermal mass in the control volume.

It is postulated that the thermal load within the cold storage room was too significant for the evaporative cooling system. (See Appendix X for an analysis.) The results of this analysis show that cooling the thermal mass requires 6 times as much energy as cooling the air within the same cold-storage room. This enhanced energy requirement may play a significant role in limiting overall temperature drop as well as rates of temperature drop.

Note: The thermal mass is estimated to be wholly due to the PVC pipe sections based on visual approximation.

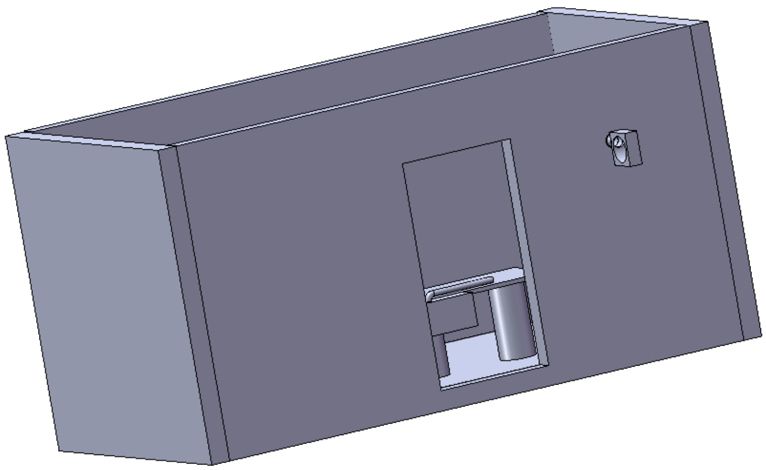
Flawed setup:

The setup used was conceived to obtain constant, equilibrium temperature and humidity conditions. The principle idea was that if cool, humid air was supplied to the control volume at higher inlet flow rates than was allowed to exit, then the temperature and humidity conditions would reach an equilibrium – the inlet conditions. However, this setup proved insufficient as shown by the results. One explanation for poor results based on setup can be related to the principles behind evaporative cooling. In this setup, ambient air is being cooled and supplied to the control volume which does not directly remove heat energy from the control volume itself.

The more effective setup (discussed in the next section) is to supply the moisture directly to the control volume air itself, which removes heat energy from the air in the control volume; thus, the air is cooled directly.

Second Testing Set-Up

Figure 9 - Second Setup: Humidifying unit inside control volume



As very little cooling was observed over the course of the first testing procedure, it was decided to place the humidifying unit within the cold-storage room for the same time duration. This second test was designed to provide a more direct contact application between the water from the humidifier stream and the air contained in the control volume. The intention was to derive a more rapid cooling effect from this initial contact.

This second test was run over the course of approximately a 24-hr period beginning at 15h30 on March 13, 2008 and ending at 15h30 on March 14, 2008. The setup parameters of this test were as follows:

- control unit of MDFD-1 set to target 100% RH with 5% variability in sensing
    - chosen to obtain maximum humidity and temperature drop.
- delivery system was removed (i.e. blower and delivery pipe) although fan was left to provide air flow for continuous cooling.
- control volume was sealed to limit entrance of warm air.

In this scenario, the relative humidity was expected to near or attain 100%. Based on experience with the previous test, a final temperature was not forecasted. Although, under normal conditions, where the intial $T_{dry,CV}$ and relative humidity were 24°C and 20% respectively, the anticipated $T_{dry,CV}$ (at near 100% RH) would approach 10°C (See Appendix). The purpose of this test was to find the final minimum $T_{dry,CV}$.

Results:

The $T_{dry,CV}$ obtained temperatures lower than in the first testing procedure where it was measured between 18.5°C and 22°C as shown in Figure X. Also in this figure, the relative humidity reached very high levels as shown by attaining 100% RH for extended periods of time. There was a problem with this setup however as the humidity control unit was set to 100% RH. Having the threshold value set this high enabled the humidifying unit to continuously supply water to the control volume. The problem occurred when the control volume air became saturated however the unit kept supplying water at which point condensation occurred. When the test was ended, upon entry to the cold-storage room significant condensation was found on the floor directly beneath the humidifying unit. The condensation was an indication of a flawed setup insofar as the humidity control unit was set too high.

Figure 10 – Tdry,CV vs. time with unit inside control volume

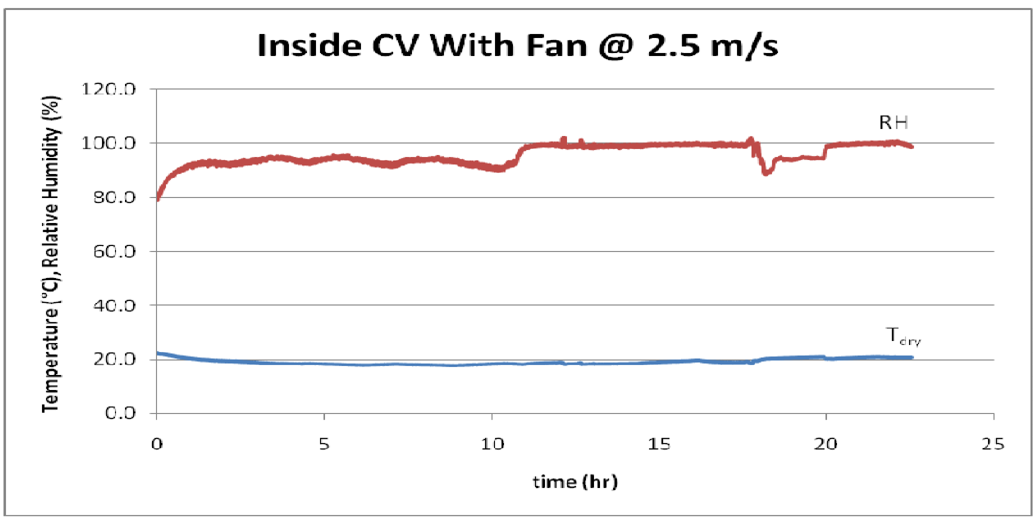


Figure 11 – Tdry,ambient vs. time for unit inside control volume*

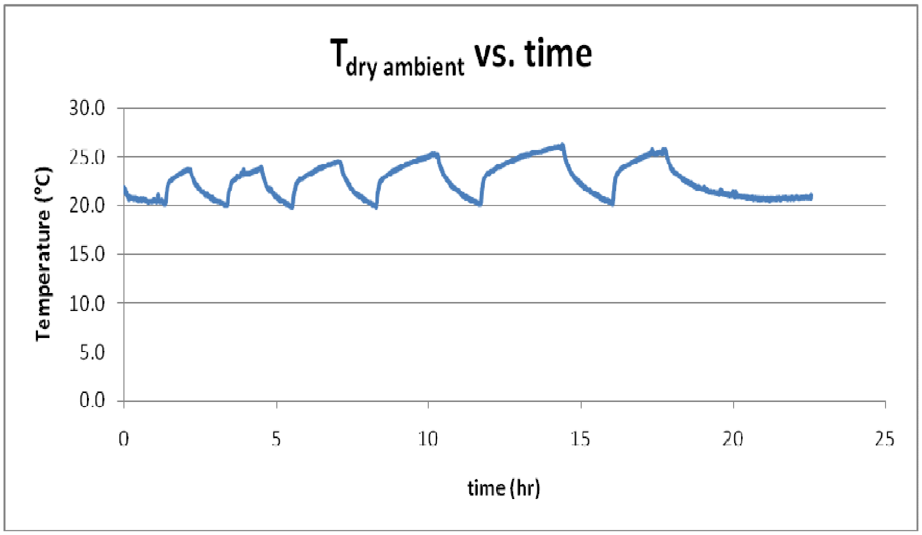
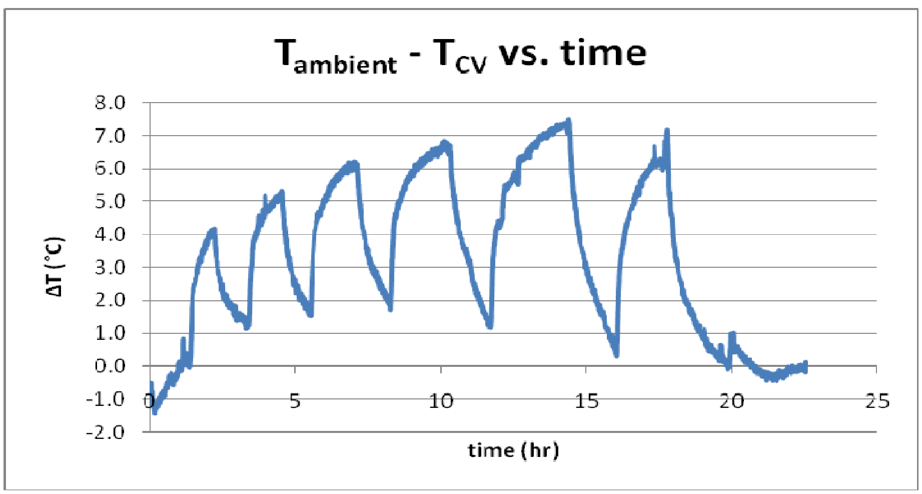


Figure 12 - Tdry,ambient – Tdry,CV vs. time with unit inside control volume*



*A discussion of the above two graphs is not necessary given the discussion of the initial test setup.

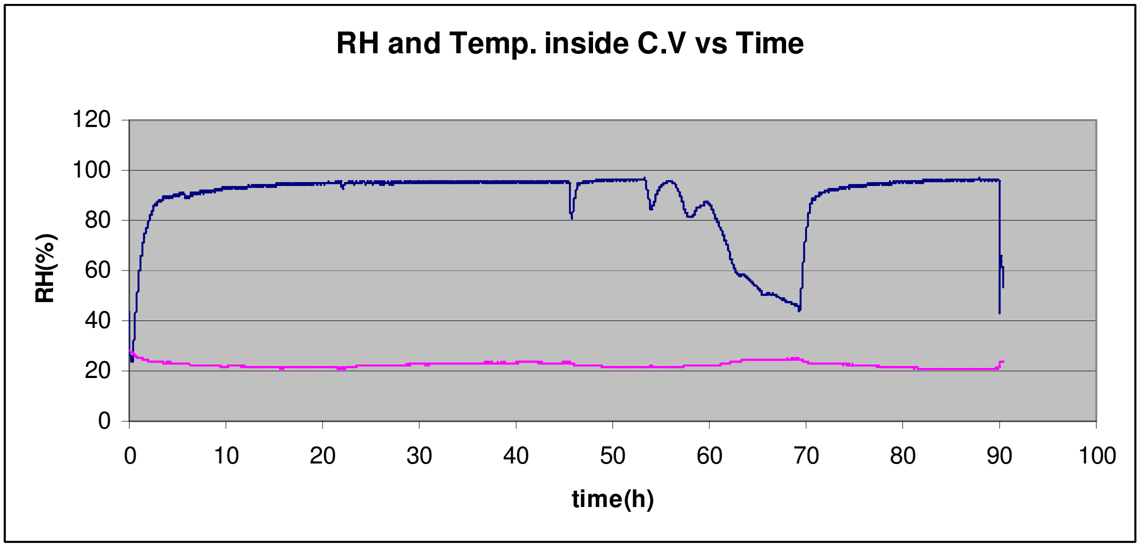
The following product quality test was a measure of how well the humidifying unit dispersed humidity. More specifically, this test allowed some quantification of the effect of that this unit had on product mass loss. Three different kinds of produce were tested: carrots, radish, and spinach. Mass measurements were taken at the beginning of the test and at every 24 hour interval after that. A control test was done for ambient conditions. Due to time constraints however, the test could not be tested for a period longer than 4 days. The results are presented in the following table and will be compared a similar experiment conducted by Dadhich et al. (2008).

Table 1 – Overall Percent Weight (%) loss of produce after 4 days

Product	Inside C.V	Outside C.V
Carrots	21.4%	57.3%
Spinach	40.8%	78.1%
Radish	30.2%	85.1%

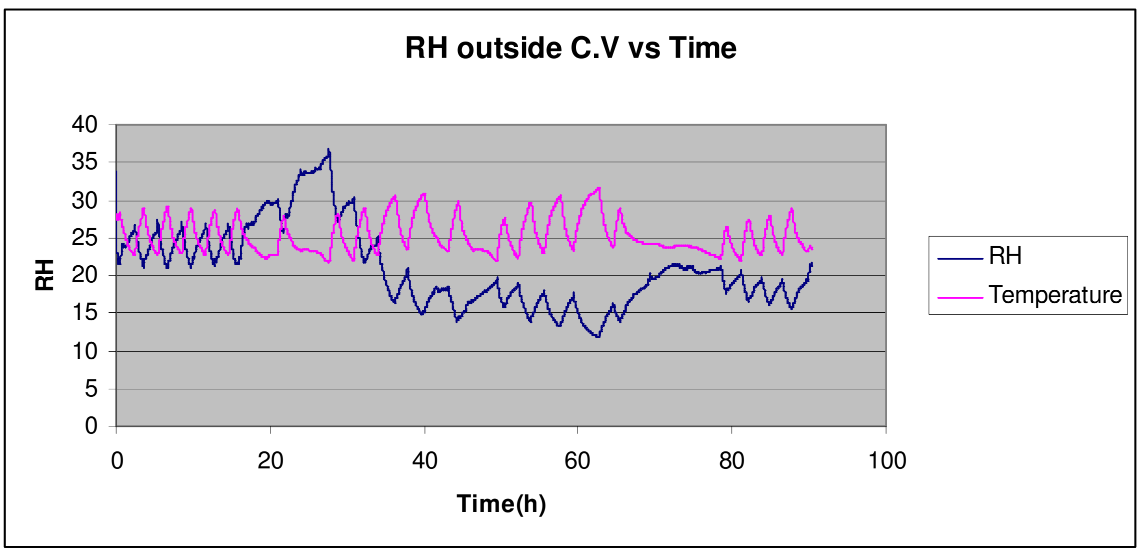
At one point during the experiment, the water delivery system malfunctioned and it did not provide a sufficient amount of water to the humidifier. As a result, the relative humidity inside the control volume dipped well below 90%. The graph on the following page shows the relative humidity versus time inside the control volume and exactly at what point during the experiment this dip in relative humidity occurred. In order to minimize the error from this drop in relative humidity, the values for percentage mass weight loss during this dip in relative humidity were interpolated from other mass weight loss values.

Figure 13 - RH and T_{dry} inside control volume vs. time



From Figure 13, the dip occurred during day 3 interval. So the mass loss values measure for day 3 will be replaced with interpolated values based on day 2 and day 4 measurements. The ambient conditions are also important to consider and note. The following graph illustrates both the temperature and the relative humidity of the ambient air.

Figure 14 - RH and T_{dry} vs. time outside control volume



The following results include interpolated weight loss values for day 3.Note that the values for outside the control volume are the exact same as the previous data set.

Table 2 - Percent Weight (%) loss of produce after 4 days -- Interpolated

Product	Inside C.V.	Outside C.V.
Carrots	11.2%	57.3%
Spinach	28.4%	78.1%
Radish	13.9%	85.1%

The following graphs show the results with interpolated values for day 3, measured data for day 3 and ambient conditions for all of the produce tested.

Figure 15 - Carrot weight loss vs. time

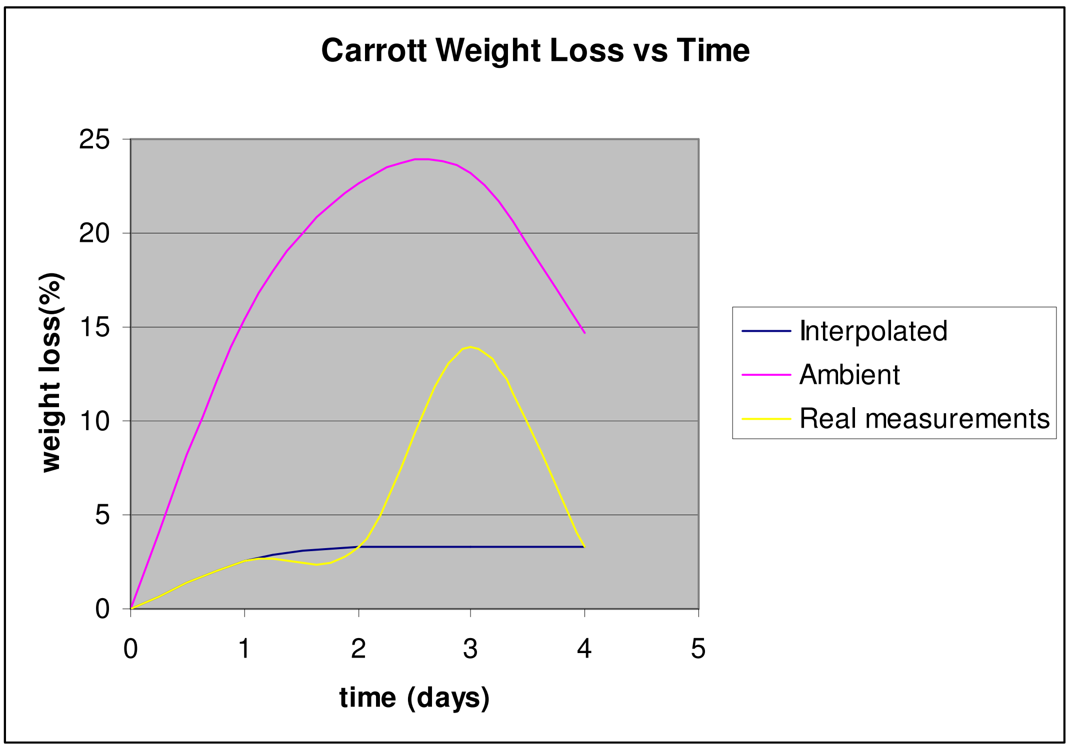


Figure 16 - Radish weight loss vs. time

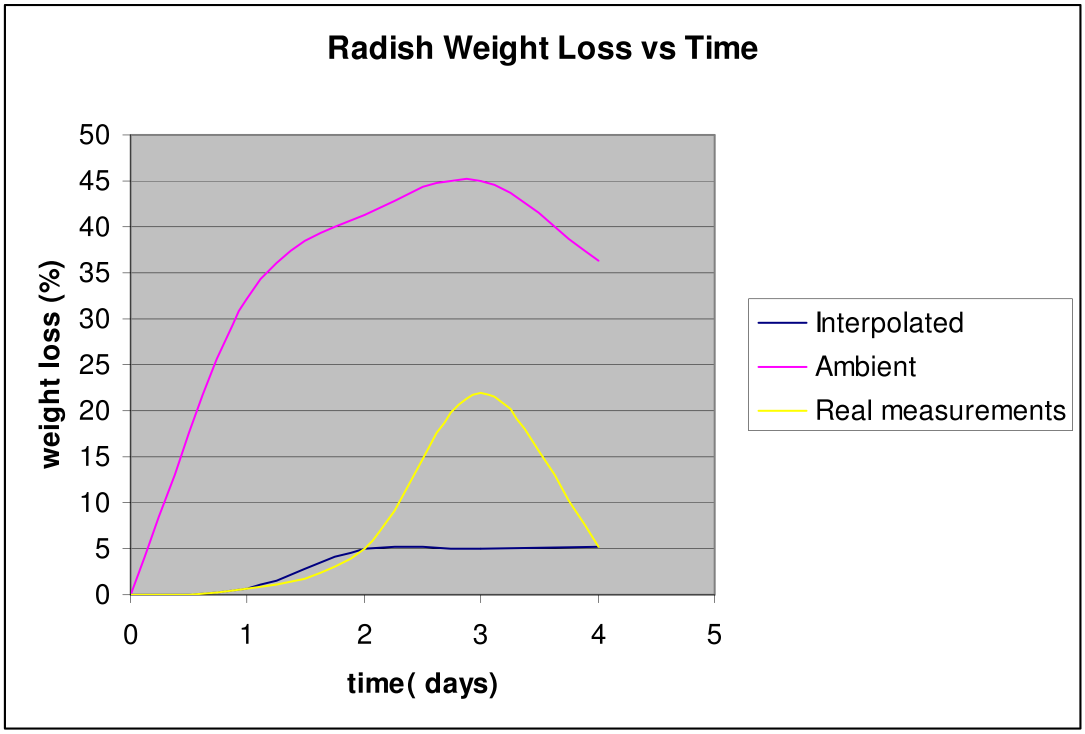
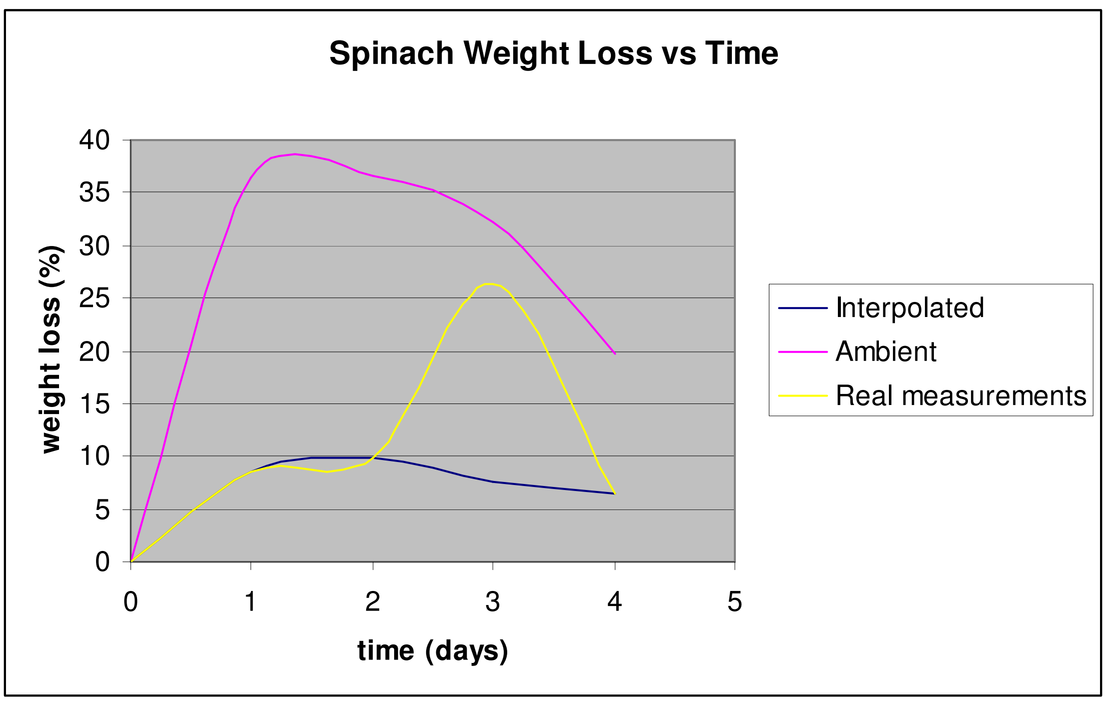


Figure 17 - Spinach weight loss vs. time



The interpreted results might in fact not necessarily reduce the error of the change in relative humidity inside the control volume. The real measurements and ambient measurements of all the produce have similar concave shapes that peak around the same point in time. However, the only precise way to settle any uncertainty in the results and their interpretation is to simply conduct another test. However, due to time constraints this is not possible.

The results from a similar evaporative cooling experiment by Dadhich et al. (2008) are presented in the following table.

Table 3 - Percent Weight (%) loss of produce after 7 days

Product	Inside C.V	Outside C.V
Carrots	5%	50%
Spinach	8%	49%
Radish	12%	55%

These values are weight loss values are considerably lower than the weight loss obtained in the present experiment. In fact, in Dadhich et al. (2008) the weight losses are for 7 days of storage as opposed to only 4. However, it must be noted that the experiments were not the same. The first major source of error to consider would be difference in ambient conditions and in ambient weight loss. The ambient temperatures in Dadhich et al. (2008) experiment varied from 12℃ to 23℃ while the ambient temperature in our experiment varied from 23℃ to 31℃. Furthermore, the ambient relative humidity varied from 45% to 60% while for our experiment it varied from 12% to 37%. Therefore, the ambient conditions in our experiment were considerably dryer and warmer than for Dadhich et al. (2008). In a dryer and warmer environment, moisture loss is greater. In addition, the conditions inside the control volume are different for both experiments. The relative humidity is however practically the same but the dry bulb temperatures are quite different. In our experiment the temperature ranged from 24℃ to 20℃ while for their experiment it ranged from 12℃ to 23℃. Again, for higher temperatures, moisture loss is greater. However, it remains uncertain as to whether these differences in temperatures and relative humidity could result in the considerable difference in percent moisture loss values measured. One possible source of error in the values that should be noted is the fact that when the produce was initially massed, there was a considerable amount of moisture on the surfaces of the vegetables. Perhaps, a more conservative approach would have been to let the surface moisture dissipate before initial mass measurements were taken.

LIMITATIONS AND SETBACKS

This project was fraught with setbacks of all kinds rooted in a number of different causes. Some of the setbacks resulted from lack of knowledge and experience (ex. wet-bulb thermometer inaccuracy, oversaturation, etc.) whereas others were a result of equipment failure (ex. humidifier fan burn-out, water delivery system malfunction, etc.) or time and budget constraints. This statement is included in order to show that more time would allow for more complete examination of the evaporative cooling effects that were attempted to be demonstrated.

CONCLUSIONS

In conclusion, the ultrasonic unit exerted a very small cooling effect when integrated into an evaporative cooling system. It is hypothesised that the reason why the evaporative system did not cool significantly was due to the high thermal load in the system. Another test could be run with a reduced thermal load to verify whether this is really the case; however, such a test would most likely not reflect conditions (notably a fairly high thermal load) found in commercial storage. Furthermore, quantifying the cooling effect of the unit for commercial storage is one the objectives of this design project. In addition, the use of a mixing chamber seemed to have very little effect on the amount of cooling occurring inside the control volume. Finally, the percentage weight loss results, when compared to the experiments conducted by Dadhich et al. (2008) were high. However, to properly compare weight loss values to another experiment, all of the testing environments should be the same. In the test conducted in this design project, both the ambient and control volume temperatures were significantly higher than that of Dadhich et al. (2008). In addition, the relative humidity in the ambient conditions was also significantly higher in the Dadhich et al. (2008). Finally, to truly examine the value of the ultrasonic unit as a humidity source, and to quantify its effect of produce weight loss, the relative humidity and dry bulb temperature in the control volume should reflect and simulate the relative humidity and dry bulb temperature in a commercial storage environment and not that of a warm dry machine shop.

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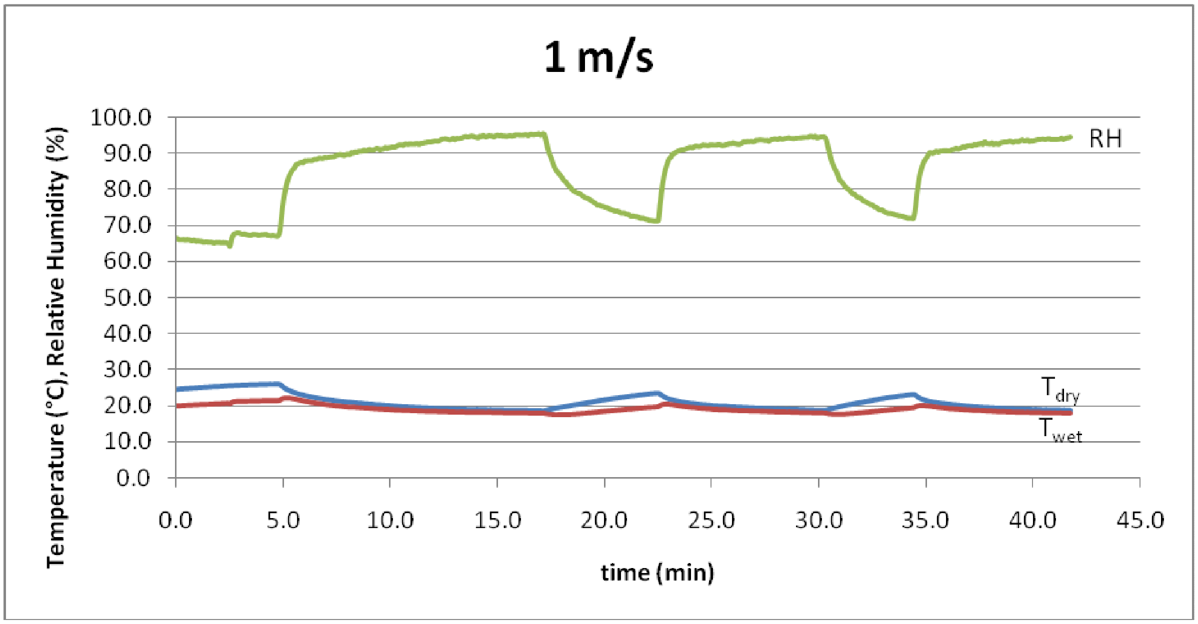
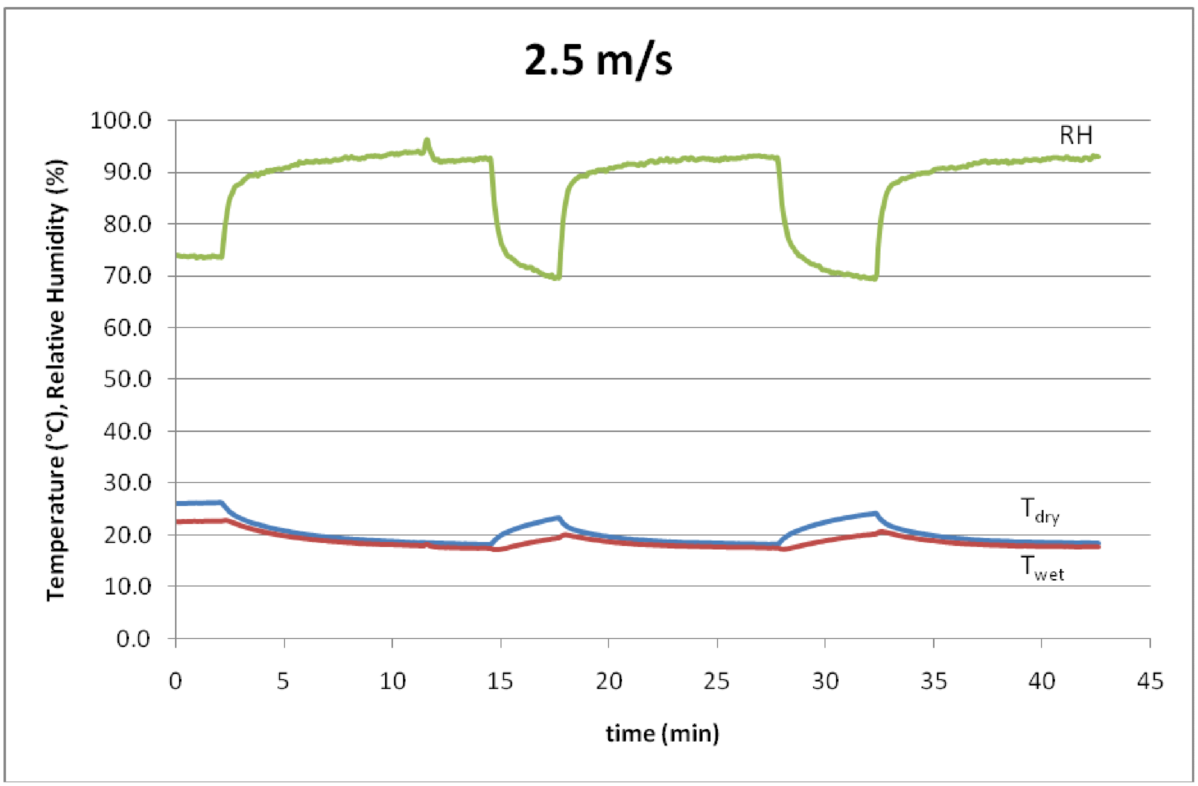
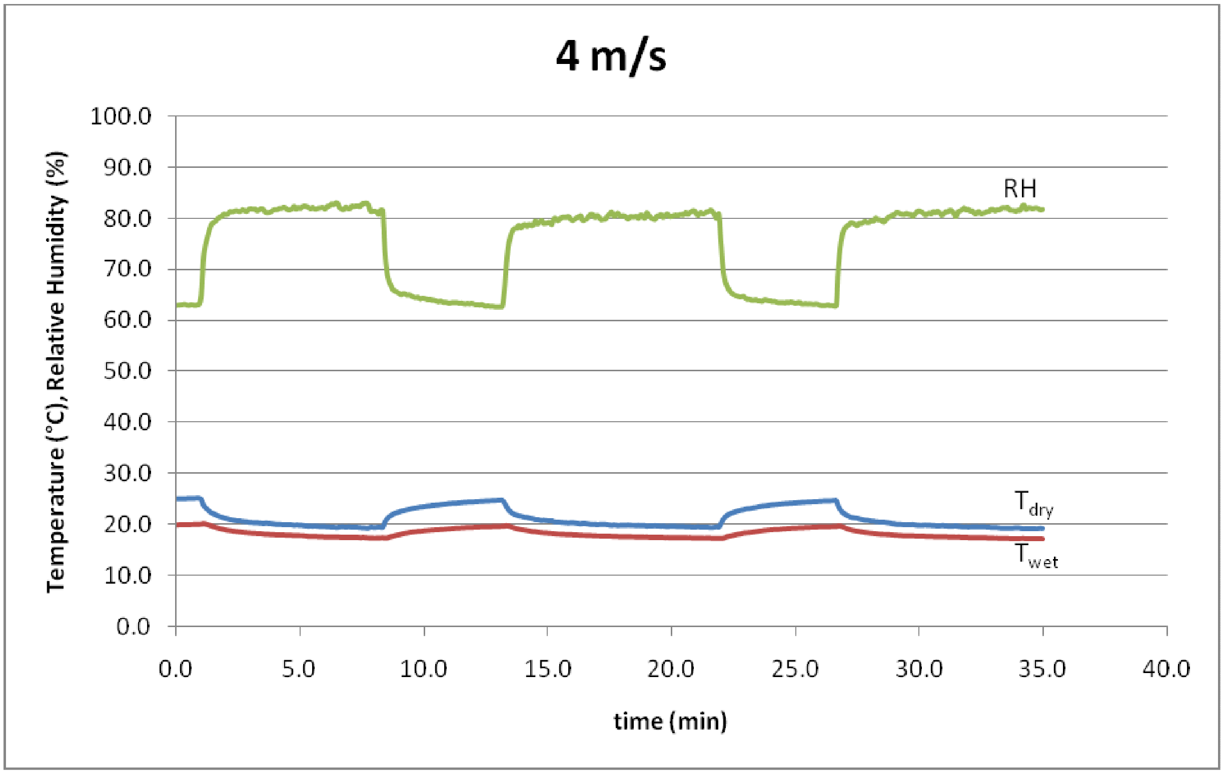
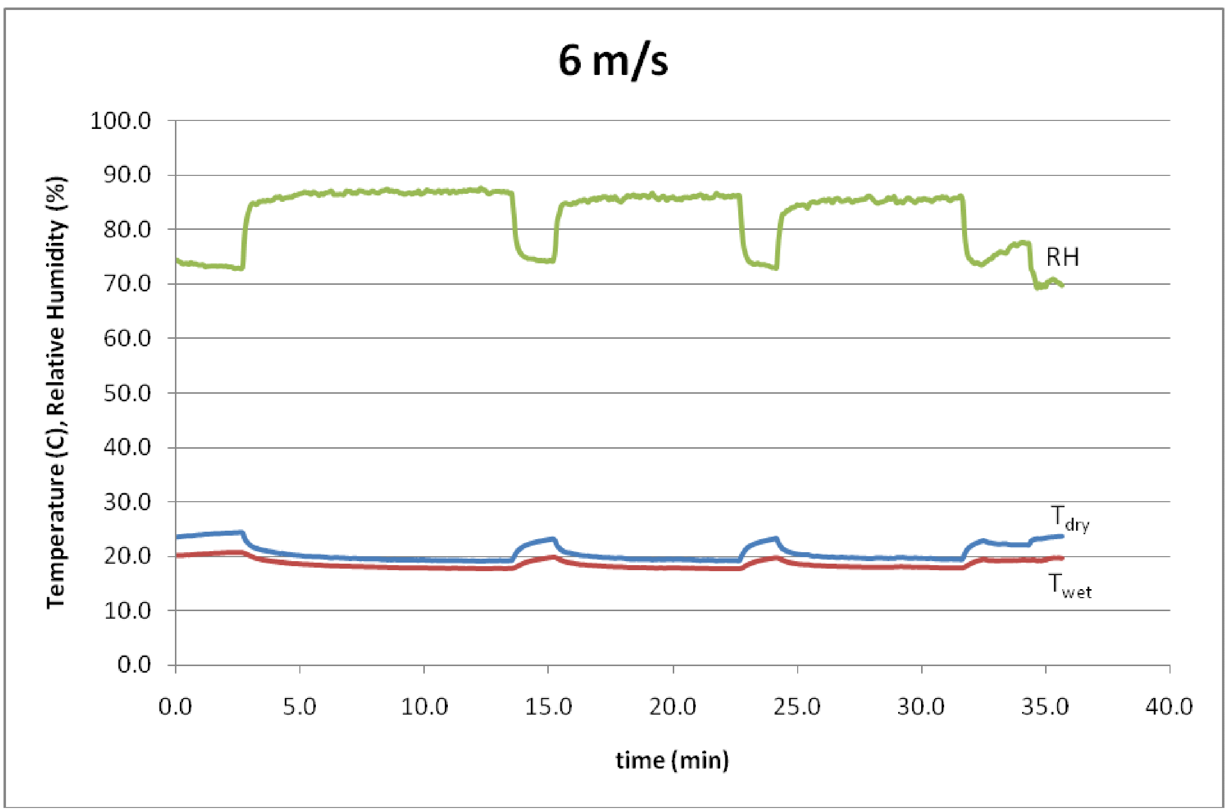
APPENDIX A – Performance Analysis of Mixing Chamber at Varying Maximum Air Velocities

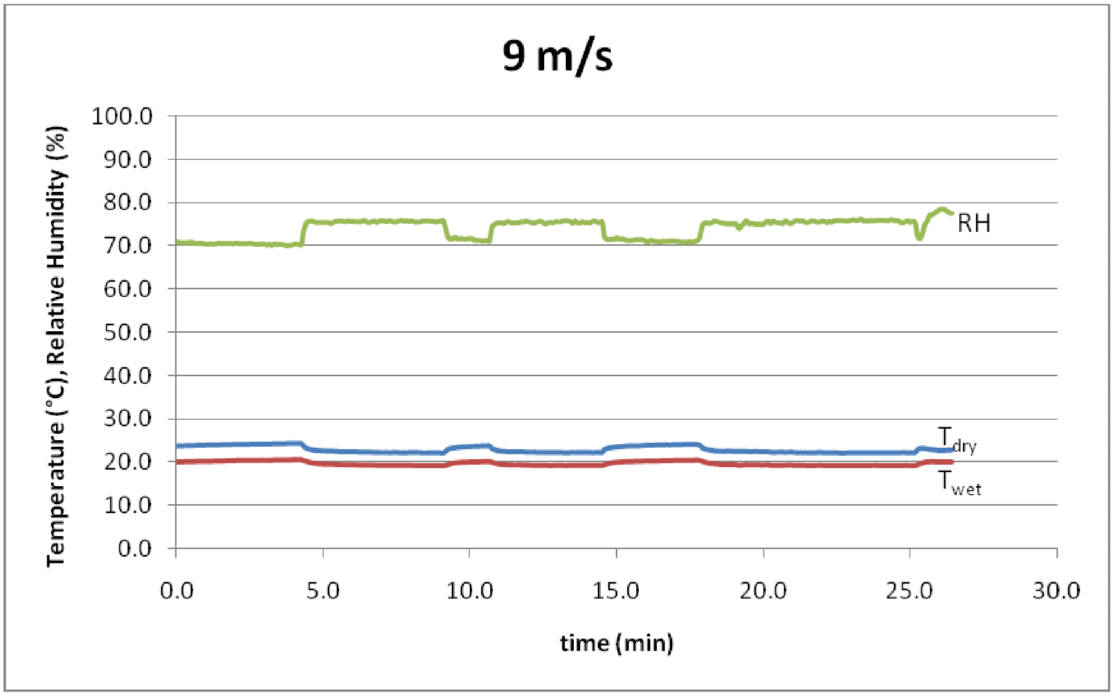
Data obtained March 10, 2008.



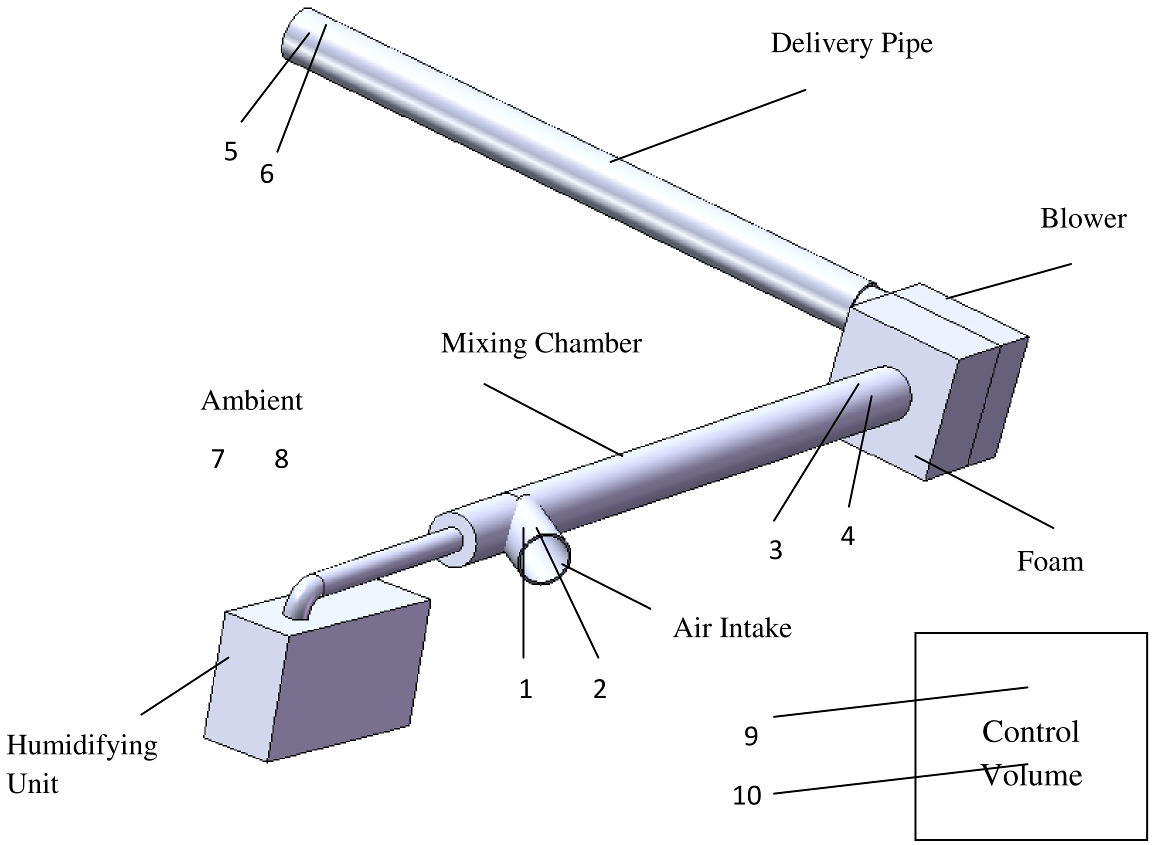



4 m/s
Temperature (°C), Relative Humidity (%)
time (min)
RH
T dry
T wet
0.0
10.0
20.0
30.0
40.0
50.0
60.0
70.0
80.0
90.0
100.0
0.0
5.0
10.0
15.0
20.0
25.0
30.0
35.0
40.0


6 m/s
Temperature (C), Relative Humidity (%)
time (min)
RH
T dry
T wet
0.0
10.0
20.0
30.0
40.0
50.0
60.0
70.0
80.0
90.0
100.0
0.0
5.0
10.0
15.0
20.0
25.0
30.0
35.0
40.0


9 m/s
Temperature (°C), Relative Humidity (%)
100.0
90.0
80.0
70.0
60.0
50.0
40.0
30.0
20.0
10.0
0.0
0.0
5.0
10.0
15.0
20.0
25.0
30.0
time (min)
RH
T_{dry}
T_{wet}

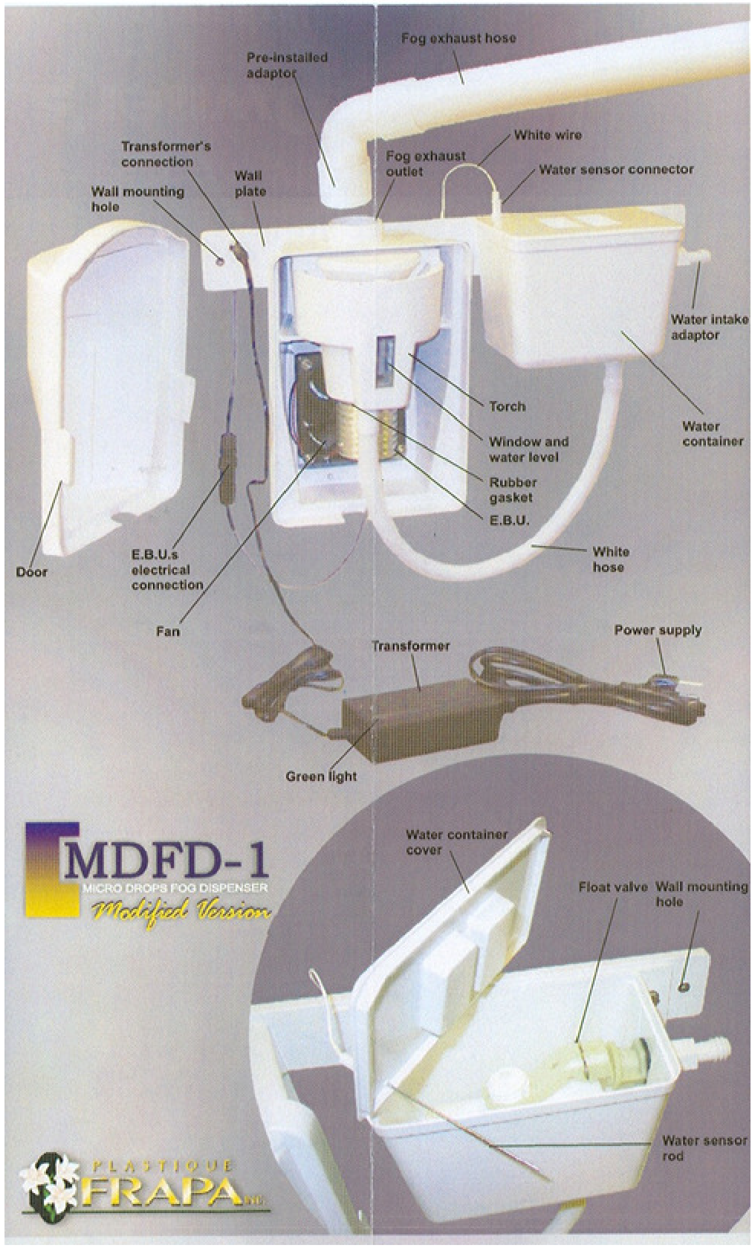
APPENDIX B – Solid Works Representational Model



The numbers 1 through 6 signify the positions of thermocouples which were used to measure T_{dry} (odd) and T_{wet} (even) throughout the experiments.

- 1 and 2 were used to take readings of pre-mixing quality.
- 3 and 4 were used to take readings of post-mixing chamber quality, but pre-blower.
- 5 and 6 were used to take readings of post-mixing and post-blower quality.
- 7 and 8 were used to take reading of ambient air quality.
- 9 and 10 were used to take readings of control volume air quality.

APPENDIX C – Diagram of MDFD-1 Ultrasonic Humidifier by Plastique Frapa Inc.



APPENDIX D – Thermal Load Analysis

In this example analysis, the T_{dry} is assumed to decrease from 22 °C to 15 °C.

• 22 PVC pipe secions

› L = 23” = 0.584 m

› d_{out} = 10-3/4“ = 0.273 m -> r_{out} = 0.1365 m

› d_{in} = 9-14/16” = 0.251 m -> r_{in} = 0.1255 m

• 40 PVC pipe caps

› d_{cap} = 9-14/16” = 0.251 m -> r_{cap} = 0.1255 m

› t = 1/4" = 0.00635 m

• $\rho_{PVC} \approx 1380\ kg\cdot m^{-3}$

• $c_{p,PVC} \approx 0.9\ kJ\cdot kg^{-1}\cdot K^{-1}$

The equation for heat loss from the pipe to the air will be:

$Q = m_{pipe}c_{p,pipe}(T_2\text{-}T_1)$

Where,

$m_{total} = V_{total}\rho = (aV_{pipe} + bV_{caps})\ \rho_{PVC} = [a\pi(r_{out}^2\text{-}r_{in}^2)\ L + b\pi r_{cap}^2 t]\rho_{PVC} = (0.116 + 0.0123)m^3\cdot 1380\ kg\cdot m^{-3}$
$= 179.5$ kg

NOTE: “a” represents 22 pipe sections and “b” represents 40 pipe caps.

Therefore,

$$Q = 179.5\,kg \cdot 0.9\,\frac{k\text{J}}{kg\cdot \text{K}} \cdot 7\text{ K} = 1130.85\,k\text{J}$$

This analysis in itself is not particularly descriptive and it must be put in relation to the energy required to cool the air in order to provide perspective.

Again,

$$Q = m_{air}\,c_{pair}\left(T_2 - T_1\right)$$

Where,

• $m_{air} = \rho V = \frac{1}{v}\,V = \frac{1}{0.83\,\frac{m^3}{kg}} \cdot 20.34\,m^3 = 24.51\text{ kg}$

- $c_{pair} = 1.006 \dfrac{k\mathrm{J}}{kg \cdot K}$

- Dimensions of control volume: L = 172” = 4.37 m; W = 88” = 2.24 m; H = 83” = 2.08 m

So, over the same temperature drop from 22℃ to 15℃,

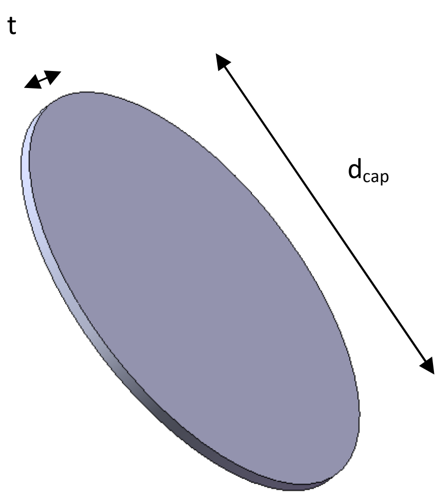
$$Q = 24.51\, kg \cdot 1.006 \frac{k\mathrm{J}}{kg \cdot \mathrm{K}} \cdot 7\ \mathrm{K} = 172.6\, k\mathrm{J}$$

In comparing this value to the energy required for decreasing the temperature of the PVC piping:

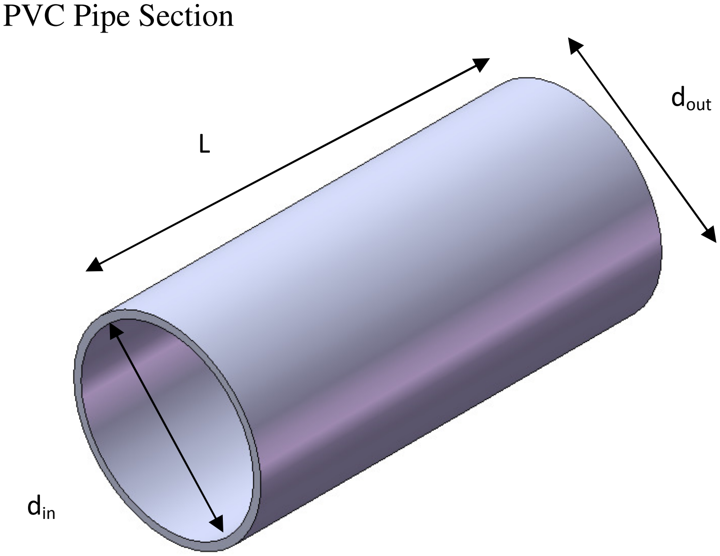
$$\frac{Q_{PVC}}{Q_{qir}} = \frac{1130.85\, k\mathrm{J}}{172.6\, k\mathrm{J}} = 6.55$$

So, by comparison it can be seen that the PVC piping will play a significantly larger role in resisting temperature changes within the control volume than the air, and it can be considered as a thermal mass which may be a large reason that the temperature does not drop quickly or significantly.

PVC Cap



PVC Pipe Section



APPENDIX E – Calculation of Relative Humidity from T_{dry} and T_{wet}

The following method for calculating Relative Humidity was taken from a web article at the following address: http://joseph-bartlo.net/articles/070297.htm. It was chosen over other methods because of the accuracy it demonstrated in calculating Relative Humidity.

A number of equations rely on other information regarding the system such as T_{DP} or Humidity Ration (H) in order to determine RH (gorshamschaffler.com, 2008). Unfortunately, in the context of this experiment this information is not given and these equations cannot be used. Even using the equations found in ASHRAE's Handbook: Principles (ASHRAE, 2005) in the section concerning psychrometrics, the RH which is calculated is inaccurate.

For example, for the conditions of T_{dry} = 30°C and T_{wet} = 15°C, using the equations in ASHRAE (2005) yields a RH = 21%. In contrast, using the method in question which was outlined by Mr. Joseph Bartlo yields RH = 17.3%. Upon inspection of a psychrometric chart (See Appendix F), the latter value seems to be very accurate as the intersection point on the chart is approximately RH = 17.5% (See psychrometric chart in Appendix X).

Description of method:

During the wet-bulb process, air, water and water vapour coexist; some vapour may be condensed or water may be evaporated into the air, saturating it. These are isobaric processes where, as water vapour content in the air changes, the air volume also changes. In these processes, it is the latent heat of vapourization that is responsible for the temperature changes. The process is governed by the following equation:

$$(M_d C_{pd} + M_v C_{pv})\, dT = -L_v\, dM_v \qquad (1)$$

Where,

M_d = dry air mass
M_v = water vapour mass
C_{pd} = dry air specific heat, constant pressure
C_{pv} = water vapour specific heat, constant pressure
L_v = vapourization of latent heat
T = temperature

Dividing (1) by M_d gives the following:

$$(C_{pd} + RC_{pv})\, dT = -L_v\, dR \qquad (2)$$

Where,

$R = M_v/M_d$, which is recognized as the water vapour mixing ratio.

The author isolates dR in (2) and integrates with respect to T_{dry} and T_{wet} which is very difficult given that R and L_v are functions of T_{dry}. As an approximation, the author uses a representative value for R and T_{dry}:

$R = R_{wet} + ((C_{pd} + R''C_{pv})/L_v)(T_{wet}-T_{dry})$ (3)

$R'' = R_{wet}/2$ (4)

$T'' = (T_{dry}+T_{wet})/2$ (5)

$L_v = 2500800 - 2370\, T''$ (6)

Saturation vapour pressure is given by the Wexler equation as follows:

$E_s = 6.112\, e^{(17.67\, T/(243.5+T))}$ (7)

And the mixture ratio is related to the vapour pressure by:

$R = zE/(P-E)$ (8)

$E = RP/(z+R)$ (9)

Where,

Z = water vapour molecular mass/ dry air molecular mass = 0.62197
E = water vapour pressure
P = pressure (measured in mbar)

Finally, RH can be found from:

$RH = E/E_s * 100\%$ (10)

Sample calculation:

Given $T_{dry} = 30°C$ and $T_{wet} = 15°C$.

i) Calculate L_v using T_{wet} and (6).
   $L_v = 2500800 - 2370(15) = 2465250$ J

ii) Calculate $E_{s,wet}$ using T_{wet} and (6).
   $E_{s,wet} = 6.112\, e^{(17.67(15)/(243.5+15))} = 17.04049$ Pa

iii) Calculate R_{wet} using T_{wet} and (8).
   $R_{wet} = (0.62917)(18.169)/(1000-18.169) = 0.010782$ kg H_2O/kg dry air

iv) Find R" using (4).
   $R'' = 0.011510/2 = 0.005391$ kg H_2O/kg dry air

v) Find R using (3) – NOTE: $C_{pd} \approx 1006.3$ J/kg/K and $C_{pv} \approx 1850$ J/kg/K
   R = 0.010782 + ((1006.3+(0.005391)(1850)/2465250)(15-30) = 0.004599 kg H_2O/kg dry air

vi) Find E using (9).
   E = (0.004599)(1000)/(0.62197 + 0.004599) = 7.339688 Pa

vii) Find E_s using T_{dry} and (6).
   $E_s = 6.112\ e^{(17.67(30)/(243.5+30))} = 42.45575$ Pa

viii) Finally, find RH using (10).
   RH = (7.339688)/(42.45575) * 100% = 17.28785%

APPENDIX F – Daily Weight Loss of Produce

Daily Percent Weight Loss of Carrots

	Inside C.V	**Outside C.V**
Day 1	2.6%	15.4%
Day 2	3.3%	22.7%
Day 3	13.9%	23.2%
Day 4	3.3%	14.7%

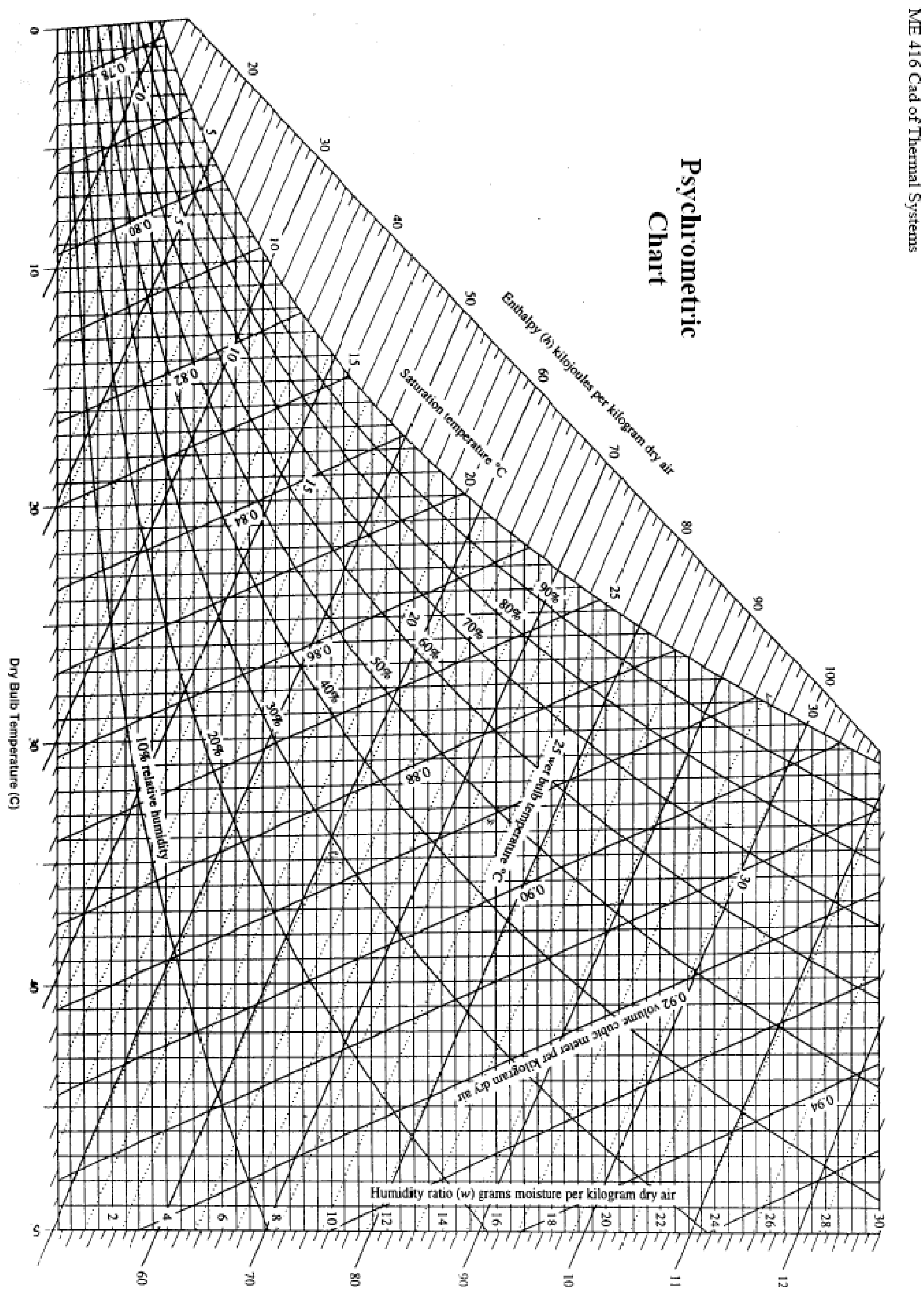
Daily Percent Weight loss of Spinach

	Inside C.V	**Outside C.V**
Day 1	8.5%	36.4%
Day 2	9.9%	36.6%
Day 3	26.4%	32.2%
Day 4	6.5%	19.7%

Daily Percent Weight loss of Radish

	Inside C.V	**Outside C.V**
Day 1	0.55%	32.1%
Day 2	5.1%	41.2%
Day 3	22%	45.0%
Day 4	5.2%	36.2%

APPENDIX G – Psychrometric Chart



APPENDIX H - Ideal Mixing Chamber Velocity

The ambient air in testing environment is 25% RH, and dry bulb T of 25˚C. To humidify this air to 85%, the dry bulb temperature from psychometric chart is 16.5˚C.

-The humidity ratio of initial air (@ 25% RH, and dry bulb T of 25˚C) = 0.0055kg water / kg dry air.

-The humidity of final air (@ 85% RH, and dry bulb T of 16.5℃) = 0.0095 kg water/ kg dry air.

Therefore, delivery system must add:

$$0.0095\, kg\, \frac{water}{kg\, dry\, air} - 0.0055\, kg\, \frac{water}{kg\, dry\, air} = 0.0040\, kg\, \frac{water}{kg\, dry\, air}$$

The average operating amount of water the humidifier adds to the system is 0.5 L/h or

0.5kg / h.

This translates into 1.4 E-4 kg water/h.

$$\therefore\ 1.4\, E - 4\, \frac{kg\, water}{h} \cdot \frac{\cdot}{X} = 0.0040\, kg\, \frac{water}{kg\, dry\, air}$$

$$X = 0.035\, \frac{kg\, air}{s} \quad = 0.035\, \frac{m^3}{s}$$

This assumes an average air density of 1 $\frac{kg}{m^3}$.

Area of 4 inch pipe used in mixing chamber:

Area = $\pi\, \frac{d^4}{4} = 0.0081\, m^2$

$$\frac{X}{area} = 0.035\, \frac{\frac{m^3}{s}}{0.0081}\, m^2 = 4.3\, \frac{m}{s}$$