Universidad de Puerto Rico
Recinto Universitario de Mayagüez
Department of Mechanical Engineering

INME 4032
Mechanical Engineering Lab II

Cooling Tower Experiment

By
Irvin E. De La Paz
Luis A. Plaza
Hiram González
José R. Vázquez
José M. Otaño

Submitted and Orally Presented to:
Dr. Orlando E. Ruiz Quiñones

Date of the Experiment: June 23, 2009
Date of Submission: June 24, 2009

Abstract

A cooling tower is a series of systems that are use to cool large structure or process. This cooling system utilizes a series of baffles inside a tower in order to permit a heat transfer between the atmospheric air and the water been cooled. A draft inducted system is one common configuration for rapid cooling effects using an air fan as the driving mechanism. The cooling tower experimental activity consisted in obtaining different parameters for the analysis of the system's performance. Such parameters as the speed of the fans and voltage were used to determine the power require for them. Apart from the fans, the temperatures of the water and air entering and exciting were used to calculate the thermodynamic balance in order to obtain the effectiveness of the system.

This experiment was carried out at the "Planta Fisica" building. This place is equipped with two chillers with several pumps and four cooling towers. These units are responsible for processing and distributing the cold water used for refrigeration purposes throughout the campus. The main goal of this experiment is to take measurements of a large scale, and functioning system on the field. The measurements taken include: psychometric data at the air entry and exit, temperature measurements of the system's water at different reference points and making a velocity profile for the cooling tower fan.

Experiences like this where the student comes in contact with the equipment that is used to perform a certain task, in this case the cooling towers, enriches the learning experience via a hands on approach. Additionally it exposes the student to a more industry-like scenario where a large scale system has to be analyzed in order to assess its performance. Having this in mind this also shows that the fundamental laws taught in the classroom will apply to real systems. Some examples of these laws are the conservation of mass and the conservation of energy, within others.

Table of Contents

Abstract i

Symbol List and Abbreviations iii

  Figure List iv

Introduction 1

Experimental Setup 2

Experimental Data 4

Analysis and Results 7

Discussion 13

Conclusion 15

References 16

Appendix 17

Symbol List and Abbreviations

Table 1 Symbol Table

Symbols:	Meaning
h_{1-4}	Enthalpy Air
L_{1-3}	Line Phase Voltage
$\dot{m}_a$	Mass Flow Rate Air (kg/s)
$\dot{m}_{evaporation}$	Mass Flow Rate of Evaporated Water (kg/s)
$\dot{m}_{makeup}$	Mass Flow Rate of Makeup Water (kg/s)
$\dot{m}_{water}$	Mass Flow Rate of Water (kg/s)
$\dot{q}_{actualheatload}$	Actual Heat Load (W)
Q_{air}	Volumetric Flow Rate of Air (CFM)
T_{db}	Dry Bulb Temperature (°C)
T_{wb}	Wet Bulb Temperature (°C)
U_0	Uncertainty of Measurements
U_D	Total System Error
U_C	Instrument Component Error
$V_{lineaverage}$	Voltage Line Average
$v_{@T_{airdb}}$	Specific Volume (m^3/kg)
$\dot{W}$	Fan Power (W)
ϕ_{1-3}	Line Phase Current
η	Cooling Tower Effectiveness
ω_{1-2}	Humidity Ratio

Figure and Table List

Figure List

FIGURE 1 - COOLING TOWER (LEFT PICTURE) AND SCHEMATIC OF COOLING TOWER (RIGHT)........1

FIGURE 2 - FAN WITH MEASURED LOCATIONS MARKED IN RED........2

FIGURE 3 - COOLING TOWER SIDE FACE IDENTIFICATION D AND C........3

FIGURE 4 - VELOCITY PROFILE GRAPH........7

FIGURE 5 PSYCHROMETIC CHART........9

Table List

TABLE 1 SYMBOL TABLE........III

TABLE 2 - AIR INLET MEASUREMENTS........4

TABLE 3 - AIR OUTLET TEMPERATURES........4

TABLE 4 - ANEMOMETER READING........5

TABLE 5 - INLET AND OUTLET TEMPERATURES OF WATER........6

TABLE 6 - MOTOR AND FAN DATA........6

TABLE 7 COOLING TOWER SPECIFICATIONS........7

TABLE 8 - VOLUME FLOW RATE LEAVING THE FAN........8

TABLE 9 PSYCHROMETRIC PROPERTIES........9

TABLE 10 - INLET AIR TEMPERATURE UNCERTAINTY RESULTS........11

TABLE 11 - OUTLET AIR TEMPERATURE UNCERTAINTY RESULTS........11

TABLE 12 - WATER MEASUREMENT UNCERTAINTY RESULTS........12

TABLE 13 - VOLUME FLOW RATE UNCERTAINTY RESULTS........12

Introduction

A cooling tower is a device designed to reject heat absorbed through a cooling system to the atmosphere. There are many configurations for this purpose, but the most common for small buildings is a small fabricated induced tower, such as

Figure 1. This tower uses a series of internal baffles to help transfer heat to the outside air by using a fan to induct the flow. Most of the water falls into a pool area below, while some of it evaporates to the atmosphere. The system compensates the loss of water with new water called “make-up water”. Figure 1 below shows the schematic configuration of this system.

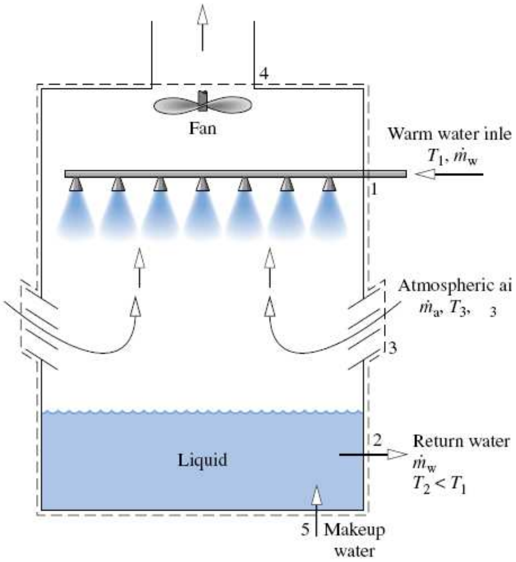


Figure 1 - Cooling tower (left picture) and schematic of cooling tower (right).

The major use for these systems is in the cooling of big buildings and industrialized processes that require a lot of heat removal. The theory behind this configuration lies in the design of thermodynamic concepts and fluid mechanics optimization. Chemicals are also aggregated to the pool in order to help maintain the performance of the pipes working in the system.

Experimental Setup

The experiment was conducted in the chilled water plant facility located inside the Mayaguez campus. Specialized equipment was needed to take the require measurements. The measurements were taken with the cooling tower operating in normal conditions and with the draft fan operating. An anemometer was used to measured win speed of the fan. The wind speed was measured radially from the center to the edge with 11 measurements in total. Figure 2 shows were the measurements were taken from. By using the guard rails of the fan as guide a more consistent measurement was achieved. This process was repeated twice and the average used for computations.

Figure 2 - Fan with measured locations marked in red

Thermocouple sensors were used to measure the inlet and outlet temperature of the water. The cooling tower was divided into two halves, side C and side D. Figure 3 illustrates how the cooling tower was divided. Six Measurements were taken from each side for both the inlet and outlet water temperature. A psychrometer was used to measure the wet-bulb and dry-bulb temperature of the inlet and outlet air. Again six measurements were taken at different parts of both the inlet and outlet air. A multi-meter was used to measure both the line voltage and line current feeding power to the fan.

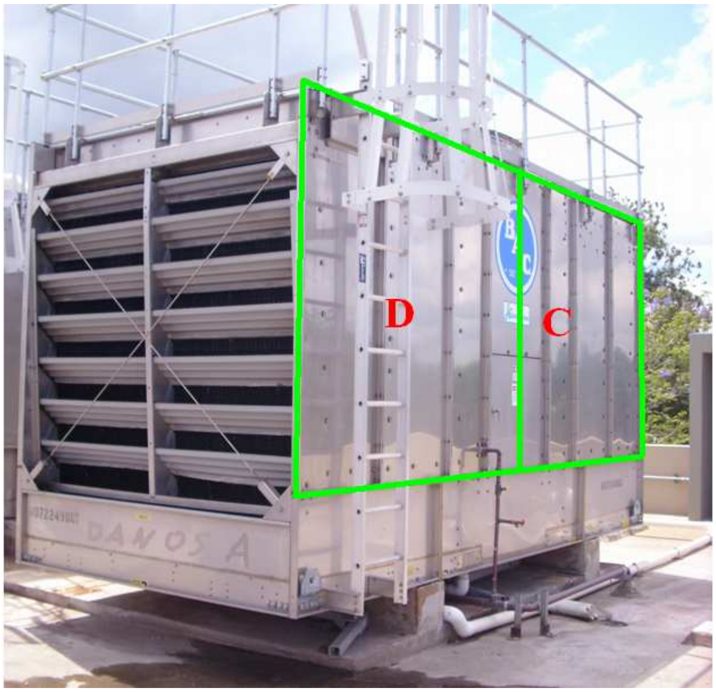


Figure 3 - Cooling tower side face identification D and C

Experimental Data

Measuring with a psychrometer Table 2 and Table 3 was obtained.

Table 2 - Air inlet measurements

Air Inlet Temperature (ºC)				
	T_{db}	T_{wb}	T_{db}	T_{wb}
TIME	Face D	Face D	Face C	Face C
1	28	20	28	20
2	28	20	28	20
3	28	20	28	20
4	28	20	28	20
5	28	20	28	20
6	28	20	28	20
		Average	28	20

Table 3 - Air outlet temperatures

Air Outlet Temperature (ºC)		
	T_{db}	T_{wb}
TIME		
1	30	22
2	30	22
3	30	22
4	30	22
5	30	22
6	30	22
Average	30	22

With an anemometer instrument Table 4 was obtained to create the velocity profile.

Table 4 - Anemometer Reading

Velocity Profile				
Position	**Distance from center (in)**	**Distance of side (in)**	**Velocity1 (CFM)**	**Velocity2 (CFM)**
1	0	46.5	465	467
2	5.5	41	440	535
3	10	36.5	626	600
4	14.5	32	628	697
5	19	27.5	823	613
6	23	23.5	2013	2346
7	28	18.5	2928	2996
8	32	14.5	3047	3024
9	36.5	10	3121	2867
10	40.5	6	3045	2617
11	44.5	2	2268	1321

With a multimeter Table 5 and Table 6 was obtained. Table 5 shows inlet and outlet temperatures measured with a thermocouple and Table 6 the experimental electrical properties of the fan's motor measured.

Table 5 - Inlet and outlet temperatures of water

WATER TEMPERATURES

TIME	INLET TEMP Face D (ºC)	OUTLET TEMP Face D (ºC)	INLET TEMP Face C (ºC)	OUTLET TEMP Face C (ºC)
1	29	24	29	23
2	29	24	29	24
3	29	23	29	24
4	29	24	29	24
5	29	24	29	23
6	29	24	29	24

Table 6 - Motor and Fan Data

Parameter	Values
L1-L2	474
L2-L3	479
L3-L1	465
L1	262V
L2	264V
L3	272V
ϕ1	25.6
ϕ2	24.4
ϕ3	26.7

Analysis and Results

To calculate range of the cooling tower, the following formula was used:

$$Range = T_{water\ IN} - T_{water\ OUT} = 29 - 23.75 = 5.25ºC$$

The approach of the cooling tower was calculated using:

$$Approach = T_{water\ OUT} - T_{wb\ air\ IN} = 23.75 - 20 = 3.75ºC$$

Effectiveness of the cooling tower was calculated using the following formula:

$$\eta = \frac{Range}{Range + Approach} x\ 100 = \frac{5.25}{3.75 + 3.75} x100 = 58.3\%$$

All this values are represented in Table 7.

Table 7 Cooling tower specifications

Cooling Tower Specifications		
Range	5.25	ºC)
Approach	3.75	(ºC)
Efectiveness	58.33333	%

The Figure 4 was obtained from Table 4.

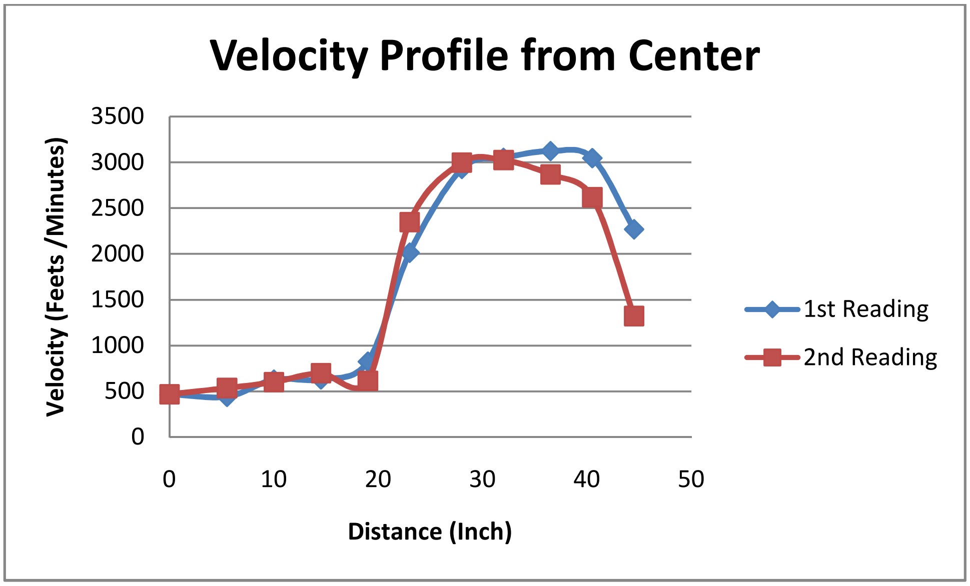


Figure 4 - Velocity profile graph

To determine the volume flow rate of the air leaving the cooling tower

- first volumetric flow rate was calculated using the following formula:

$$\dot{Q}_{AIR} = \int V * A\, dr = \sum_1^{11} \overline{Vel}_{i,j} * \pi(r_{i,j}^2 - r_{i-1,j-1}^2) - = \frac{(465+467)}{2} * \frac{\pi(0^2 - 0)}{144} + \cdots = 417{,}613.73 CFM$$

Were $\overline{Vel}_{i,j}$ is the average of the two readings taken at a given radius. Cooling tower manual (Baltimore Air Coil) was consulted to compared values of CFM.

Formula above allowed Table 8 to be calculated for each radius.

Table 8 - Volume flow rate leaving the fan

Position	Area	Velocities1	Velocities2	Volumetric flow Rate (CFM)
1	0	465	467	0
2	0.659953	440	535	321.7269039
3	2.177079	626	600	1334.549158
4	4.571825	628	697	3028.833956
5	7.844049	823	613	5632.027517
6	11.48652	2013	2346	25034.86405
7	17.02446	2928	2996	50426.44812
8	22.22199	3047	3024	67454.84765
9	28.91087	3121	2867	86559.13529
10	35.58393	3045	2617	100738.1169
11	42.95524	2268	1321	77083.18265
			Total	**417,613.73**

To determine the cooling water entering the tower, equation used is provided in experiment manual.

The equation was modified to account for power input of fan in cooling tower and became:

$$\dot{m}_a = \frac{\dot{Q} - \dot{W} + \dot{m}_3(h_3 - h_4)}{h_2 - h_1 - (\omega_2 - \omega_1)h_5}$$

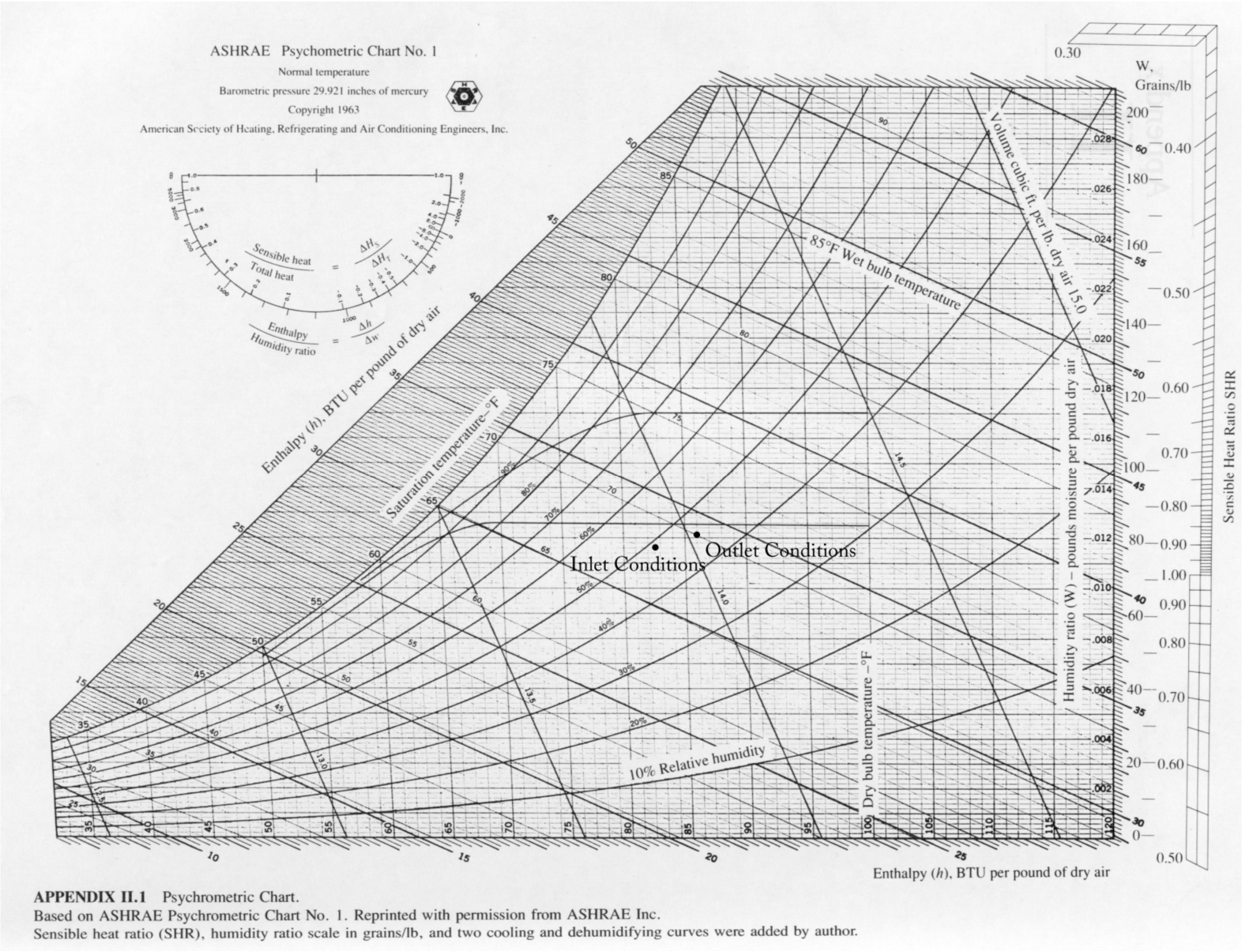
Where

$$\dot{m}_a = \frac{\dot{Q}}{v_{@T_{airdb}}} = \frac{417{,}613 cfm}{.876\,\frac{m^3}{kg}} = \frac{224.99\, kg}{s}$$

The power work consumed by the fan is calculated using the average for voltage and current presented in table, the equation is as followed:

$$\dot{W} = \sqrt{3} * V_{lineaverage} * I_{lineaverage} = \sqrt{3} * 472.6 * 25.6 = 20931 Watts$$

Moist states are provided in Figure 5 Psychrometic Chart and psychometric values, which were obtained thru (Sugar Engineers' Library), are showed in Table 9



APPENDIX II.1 Psychrometric Chart.
Based on ASHRAE Psychrometric Chart No. 1. Reprinted with permission from ASHRAE Inc.
Sensible heat ratio (SHR), humidity ratio scale in grains/lb, and two cooling and dehumidifying curves were added by author.

Figure 5 Psychrometic Chart

Table 9 Psychrometric Properties

Air Inlet Temperature				**Air Outlet Temperature**			
T_{db}	28(ºC)	T_{wb}	20(ºC)	T_{db}	30(ºC)	T_{wb}	22(ºC)
Relative Humidity	48	%		Relative Humidity	49.98	%	
Entalphy	57.117	kJ/kg		Entalphy	64.16	kJ/kg	
Specific Volume	0.867	m^3/kg		Specific Volume	0.876	m^3/kg	
Humidity Ratio	0.011	kg/kg		Humidity Ratio	0.013	kg/kg	

Using psychometric properties available in the appendix also, equation reduced to

$$\dot{m}_a = \frac{\dot{Q}-\dot{W}+\dot{m}_3(h_3-h_4)}{h_2-h_1-(\omega_2-\omega_1)h_5} = \frac{0-20\dot{9}31+\dot{m}_3(121570-99630)}{64160-57117-(0.013-0.011)99630} => \dot{m}_3 = \frac{62.2kg}{s} = \frac{493659\,lbm}{hr}$$

Actual heat load of cooling tower was calculated using the following formula:

$$\dot{q}_{actualheatload} = \dot{m}_a(h_3 - h_4) = 224.99(121\dot{5}70 - 99630) = 493.62KW$$

Water loss by evaporation was calculated using the following formula:

$$\dot{m}_{evaporation} = \dot{m}_a * (w_1 - w_2) = 224.99(0.013 - 0.011) = 3571\,\frac{\mathrm{lbm}}{\mathrm{hr}}$$

An estimation of makeup water was obtained using (Beychok, 1967) and (Process Equipment Global Spec) which stated:

$$\dot{m}_{makeup} = 0.005 * m_{w\dot{a}ter} = 0.005 * 493659 = 2468.3\,\frac{\mathrm{lbm}}{\mathrm{hr}}$$

Uncertainty:

To calculate the uncertainty for each of the measure variables, six measurements were taken for each except for wind speed. From this data the standard deviation was calculated.

Table 10 summarizes the results obtained for the inlet air temperature taken with a psychrometer with a resolution of 5 °C. From the results it can be seen that no precision errors were present in the measurements. Because each thermometer is hand calibrated the Bias error is assumed to be 0. Therefore the uncertainty of all the measurements taken with the psychrometer is equal to only its zero order uncertainty which is given by;

$$U_0 = \mp \frac{1}{2} Resolution = \mp \frac{1}{2} * 0.50\ ^\circ\mathrm{C} = \mp 0.25\ ^\circ\mathrm{C}.$$

Table 10 - Inlet air temperature Uncertainty results

AIR INLET TEMPERATURES (ºC)				
	T_{db}	T_{wb}	T_{db}	T_{wb}
Mean	28	20	28	20
Standard Deviation	0	0	0	0
Resolution	5	5	5	5
Uncertainty ±	0.25	0.25	0.25	0.25

For the air outlet temperature, the approach is the same. The measurements taken are analyzed for measurement errors. As seen from Table 11 no measurements errors were observed. Because the instrument used in the measurements is the same, uncertainty is ± 0.25 °C.

Table 11 - Outlet air temperature uncertainty results

AIR OUTLET TEMPERATURES (ºC)		
	T_{db}	T_{wb}
Mean	30	22
Standard Deviation	0	0
Resolution	5	5
Uncertainty ±	0.25	0.25

The water inlet and outlet were measured with K thermocouples with an instrument resolution of 1°C. Again the standard deviation was used to calculate the instrument measurement error. The results are summarized in Table 12. The Bias uncertainty of a class 1 K thermocouple (Thermocouple Technical Information, 2009) is ± 1.5 °C. The zero order uncertainty is;

$$U_0 = \mp\frac{1}{2}Resolution = \mp\frac{1}{2} * 1\,^\circ\mathrm{C} = \mp 0.5\,^\circ\mathrm{C}$$

For the outlet temperature in face D the instrument component error is given by;

$$U_c = \mp\sqrt{e_1^2 + e_2^2} = \sqrt{0.4^2 + 1.5^2} = \mp 1.55\,^\circ\mathrm{C}$$

And the total stage uncertainty is given by;

$$U_d = \mp\sqrt{U_0^2 + U_c^2} = \sqrt{0.5^2 + 1.55^2} = \mp 1.63\ °\mathrm{C}$$

Table 12 - Water measurement uncertainty results

WATER TEMPERATURES				
	Inlet Temp Face D	Outlet Temp Face D	Inlet Temp Face C	Outlet Temp Face C
Mean	29.0	23.8	29.0	23.7
Standard Deviation	0.0	0.4	0.0	0.5
Resolution	1.0	1.0	1.0	1.0
Uncertainty ±	1.58	1.63	1.58	1.66

The air speed measurements uncertainty was calculated differently because only two data sets of were taken for each of the 11 data points. The total air flow (CFM) was calculated for each of the data set and then the average and standard deviation calculated. Thus the uncertainty of the airspeed was not calculated directly but only for the total CFM. Bias errors and resolution errors are insignificant when compared to measurement errors and thus are excluded in the total CFM analysis. The results are summarized in Table 13.

Table 13 - Volume flow rate uncertainty results

Volume Flow Rate Leaving the fan (ft^3/min)	
CFM Set 1	447,667
CFM Set 2	387,561
Mean	417,614
Standard Deviation	42,501
Uncertainty ±	42,501

Discussion

The efficiency for the cooling tower was examined during this laboratory session it was determined to be of 58%. This efficiency is calculated by determining the range and the approach of such system. The range is the difference between the inlet and outlet temperature of the water and the approach is the difference between the cooled water exiting the tower and the wet bulb temperature. The lower the wet bulb temperature the better, a low wet bulb temperature will mean that the air is very dry; thus a higher amount of water can be evaporated. This ideal scenario of a very dry air is very difficult to find in Puerto Rico where the humidity is always very high, hindering the efficiency of these type of systems. A way to counter this effect would be to add an additional system that would take the moisture of the incoming air; however this would increase operational costs. It is recommended to make a trade off analysis in order to see if it is viable to continue operating at a lower efficiency or to install a dehumidifying unit to “pretreat” the air that enters the cooling tower.

Another important factor to mention is the design of the current system is one that has the fan on top of the tower. The fan produces a low entering velocity and a high air exiting velocity. This is very desirable to diminish the amount of high humidity air that could be recirculated to the air inlet. The velocity profile for the fan was plotted and it was seen that its behavior was very similar to a fluid inside a pipe. Fluids inside a pipe will suffer from the effect of the no-slip condition at the pipe wall and develop their maximum velocity approximately at their center. In this case the lowest velocities where found on the center of the fan blade, (this corresponds to the pipe wall) and at the tip of the fan (also corresponds to the pipe wall). The highest velocities where tabulated approximately at the center of the blade with values near to 3024 FPM. The power needed to maintain this kind of operation was determined by measuring the voltages from line to line of each of the phases of the three-phase motor and the line current. The value was calculated to 20.9KW which translates to 28hp. The actual value of the horsepower, according to the specifications in the motor’s plate, is of 30hp which by taking into account small variations that occur in voltage and current is within the expected value.

The heat load of the cooling tower was estimated by calculating the amount of air that is passing through the control volume and by determining the change in enthalpy of the water entering and leaving this same control volume. This change in enthalpy reflects the amount of energy that is being dissipated to the surroundings, by multiplying this difference in enthalpy by the amount of air passing through the cooling tower, finally it sums to 493KW.

In order to determine the amount of water being evaporated the heat dissipation and the losses due to drift where taken into consideration. This was done by calculating the 0.005% of the water that is being supplied to the cooling tower. The percentage above was taken from (Beychok, 1967). This water that is being evaporated has to be compensated in the system. This returned water is the makeup water.

Conclusion

This experiment was carried out with very simple measuring equipment. Overall the measuring equipment used can be summarized as a measuring tape, an anemometer, thermocouples, a psychrometer and a current and voltage metering device. With the first two set of equipment mentioned the velocity profile of the cooling tower was constructed. The unit studied in this experiment can be classified as an induced draft cross flow cooling tower (Process Equipment Global Spec). This type of arrangement has the fan located on top of the cooling tower. Another characteristic of this arrangement is that it guarantees low recirculation of the air that is exiting the cooling tower; this is due to the high velocity at the exit and the low velocity at entry. The benefits of having the fan located on top are directly reflected on the power consumption needed to drive the fan. A forced draft configuration on which the fan is located on the bottom requires a more powerful motor and is more prone to recirculation of exiting air.

The efficiency of the cooling tower was determined to be of 58%. By accessing the companies' website (Baltimore Air Coil) the team found out that they certify that their products will comply with ASHRAE standard 90.1 (Ashrae). According to this standard there is an index that determines if the cooling tower is operating upon the specified conditions. This index is determined from dividing the amount of water that is being supplied into the cooling tower divided by the fan's horsepower. This index value should be less or equal than 38.2. For the measurement taken during this experimental session the performance was determined to be 32.8, which certainly complies with the industry's standard. Regarding the velocity profile, it was observed that it follows the same behavior of a fluid that is passing through a pipe. This behavior shows a minimum velocity, due to no-slip condition, and a maximum velocity at 32 inches from the center (see Figure 4) then declines quickly to zero. This same conduct is seen along the radial direction where near the half of the radius of the velocity is maximum and at radius equal to zero or radius equal to maximum the velocity profile drops significantly.

References

Ashrae. (n.d.). Retrieved from http://www.ashrae.org/technology/page/548

Baltimore Air Coil. (n.d.). Retrieved from http://www.baltimoreaircoil.com/english/info_center/pubs/hb_iii/sections/HBiii_D32_CT_Series_1500.pdf

Beychok, M. R. (1967). *Aqueous Wastes from Petroleum and Petrochemical Plants (1st. Edition).* John Wiley and Sons.

Process Equipment Global Spec. (n.d.). Retrieved from http://process-equipment.globalspec.com/Specifications/Manufacturing_Process_Equipment/Heat_Transfer_Equipment/Cooling_Towers

Sugar Engineers' Library. (n.d.). Retrieved June 23, 2009, from Psychrometric Calculations: http://www.sugartech.co.za/psychro/index.php

Thermocouple Technical Information. (2009, June 23). Retrieved from Peak Sensors: http://www.peaksensors.co.uk/thermocoupledatasheets.html

Appendix