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CREATING A NEW MODELTO PREDICT COOLING TOWER PERFORMANCE AND DETERMINING ENERGY SAVING OPPORTUNITIES THROUGH ECONOMIZER OPERATION

A Thesis Presented

by

PRANAV YEDATORE VENKATESH

Submitted to the Graduate School of the
University of Massachusetts Amherst in partial fulfillment
of the requirements for the degree of

MASTER OF SCIENCE

May 2015

Mechanical and Industrial Engineering

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ABSTRACT

CREATING A NEW MODEL TO PREDICT COOLING TOWER PERFORMANCE AND DETERMINING ENERGY SAVING OPPORTUNITIES THROUGH ECONOMIZER OPERATION

MAY 2015

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Cooling towers form an important part of chilled water systems and perform the function of rejecting the heat to the atmosphere. Chilled water systems are observed to constitute a major portion of energy consumption in air conditioning systems of commercial buildings and of process cooling in manufacturing plants. It is frequently observed that these systems are not operated optimally, and cooling towers being an integral part of this system present a significant area to study and determine possible energy saving measures. More specifically, operation of cooling towers in economizer mode in winter (in areas where winter temperatures drop to 40°F and below) and variable frequency drives (VFDs) on cooling tower fans [1] are measures that can provide considerable savings. The chilled water system analysis tool (CWSAT) software is developed as a primary screening tool for energy evaluation for chilled water systems. This tool quantifies the energy usage of the various chilled water systems and typical measures that can be applied to these systems to conserve energy. The tool requires

minimum number of inputs to analyze the component-wise energy consumption and incurred overall cost. The current cooling tower model used in CWSAT was developed by Benton [2]. A careful investigation of the current model indicates that the prediction capability of the model at lower wet bulb temperatures (close to 40°F and below) and at low fan power is not very accurate. This could be a result of the lack of data at these situations when building the model. A new model for tower performance prediction is imperative since economizer operation occurs at low temperatures and most cooling towers come equipped with VFDs. In this thesis, a new model to predict cooling tower performance is created to give a more accurate picture of the various energy conservation measures that are available for cooling towers. The weaknesses of the current model are demonstrated and prediction capabilities of the new model analyzed and validated. Further the economic feasibility of having additional cooling tower capacity to allow for economizer cooling, in light of reduced tower capacity at lower temperatures [3] is investigated.

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NOMENCLATURE

ANN	*Artificial Neural Network*
ARI	*Air-Conditioning & Refrigeration Institute*
CF	*Counter Flow*
CHWST	*CHilled Water Setpoint Temperature*
CT	*Cooling Tower*
CTI	*Cooling Tower Institute*
CWSAT	*Chilled Water System Assessment Tool*
CWT	*Condenser Water Temperature*
EWT	*Entering Water Temperature*
GPM	*Gallons Per Minute*
HVAC	*Heating Ventilation and Air Conditioning*
LWT	*Leaving Water Temperature*
PR	*Polynomial Regression*
VFD	*Variable Frequency Drive*
VSD	*Variable Speed Drive*
XF	*Cross Flow*

CHAPTER 1

INTRODUCTION

1.1 Cooling Towers

A cooling tower is a device that is used to cool a water stream while simultaneously rejecting heat to the atmosphere. In systems involving heat transfer, a condenser is a device that is used to condense the fluid flowing through it from a gaseous state to a liquid state by cooling the fluid. This cooling to the fluid flowing through a condenser, is generally provided by a cooling tower. Cooling towers may also be used to cool fluids used in a manufacturing process. As a result we see that cooling towers are commonly used in

1. HVAC (Heating Ventilation and Air-Conditioning) to reject the heat from chillers
2. Manufacturing to provide process cooling
3. Electric power generation plants to provide cooling for the condenser

Cooling tower operation is based on evaporative cooling as well as exchange of sensible heat. During evaporative cooling in a cooling tower, a small quantity of the water that is being cooled is evaporated in a moving stream of air to cool the rest of the water. Also when warm water comes in contact with cooler air, there is sensible heat transfer whereby the water is cooled. The major quantity heat transfer to the air is through evaporative cooling while only about 25% of the heat transfer is through sensible heat. Figure 1.1 taken from Mulyandasari [4] shows the schematic of a cooling tower.

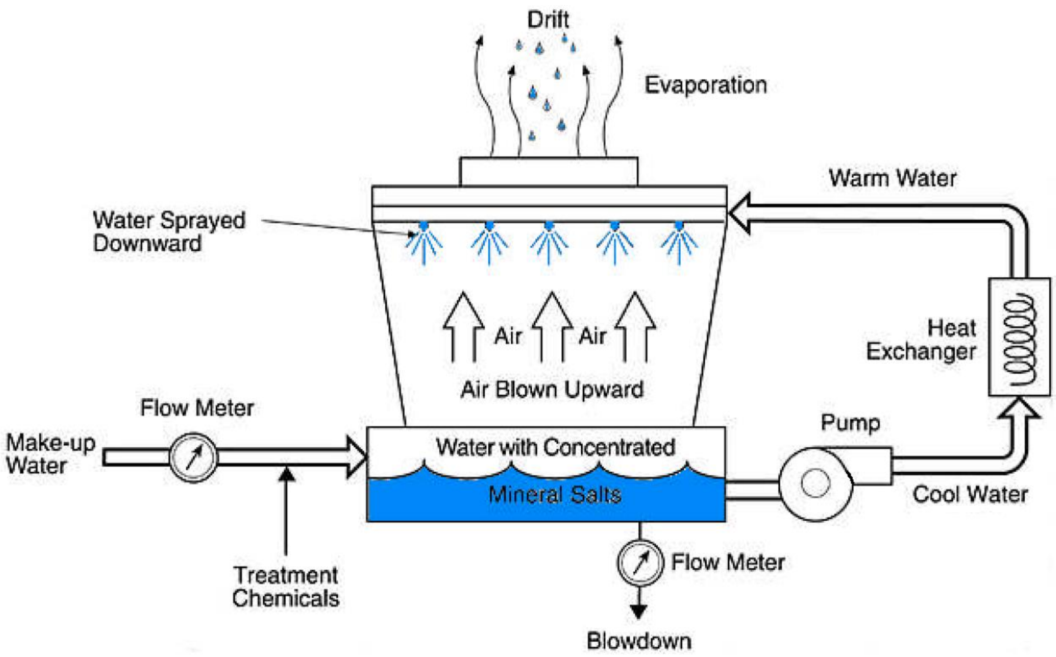


Figure 1.1 Schematic of a Typical Mechanical Draft Cooling tower

Some important terms relating to cooling towers as described by Stanford [5] are:

- Approach- It is the difference between the temperature of water leaving the cooling tower and the wet bulb temperature. It is used as an indicator of how close to wet bulb temperature the water exiting the tower is.
- Range- It is the difference between the temperature of water entering the tower and temperature of water leaving the tower.
- Capacity- The total amount of heat a cooling tower can reject at a given flow rate, approach and wet bulb temperature. It is generally measured in tons.
- Cell- It is the smallest tower subdivision that can operate independently. Each individual cell of a tower can have different water flow rate and air flow rate.
- Fill- The heat transfer media or surface designed to maximize the air and water surface contact area.

- Make up water- The additional water that needs to be added to offset water lost to evaporation, drift, blowdown and other losses.
- Dry bulb temperature – It is the temperature of air measured by a thermometer freely exposed to the air but shielded from moisture and radiation. In general when temperature is referred to, it is dry bulb temperature.
- Wet bulb temperature – It is the temperature of air measured by a thermometer whose bulb is moistened and exposed to air flow. It can also be said to be the adiabatic saturation temperature. The wet bulb temperature is always lesser than the dry bulb temperature other than the condition of 100% relative humidity when the two temperatures are equal.
- Free Cooling or Waterside Economizer Operation – It is the operation of the cooling tower in conditions where just the cooling tower is able to provide the required temperature cold water for HVAC or process needs without needing mechanical cooling from the chiller. This saves energy because while the chiller may utilize about 0.7 kW/ton, the tower is now able to provide the same cooling at about 0.2 kW/ton.

According to Hill [6] the factors influencing the performance of a cooling tower are:

1. The cooling range
2. The approach
3. The ambient wet bulb temperature
4. The flow rate of water through the tower
5. The flow rate of air over the water
6. The ambient temperature

7. The type of fill in the tower
8. Total surface area of contact between water and air

1.2 Cooling Tower Classification

Cooling towers can be classified in many different ways as follows

- ❖ Classification by build
  - ✓ Package type
  - ✓ Field Erected type
- ❖ Classification based on heat transfer method
  - ✓ Wet cooling tower
  - ✓ Dry Cooling tower
  - ✓ Fluid Cooler
- ❖ Classification based on type of Fill
  - ✓ Spray Fill
  - ✓ Splash Fill
  - ✓ Film Fill
- ❖ Classification based on air draft
  - ✓ Atmospheric tower
  - ✓ Natural Draft Tower
  - ✓ Mechanical Draft Tower
    - Forced Draft

- Induced Draft

- Classification based on air flow pattern
  - ✓ Crossflow
  - ✓ Counterflow

Figure 1.2 shows the graphical depiction of the tower classification.

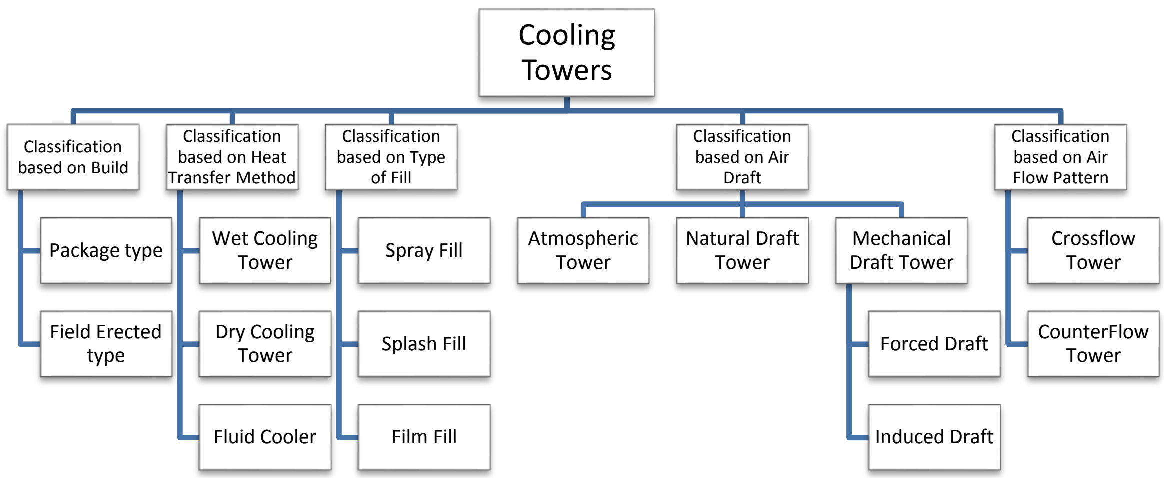


Figure 1.2 Classification of Cooling Towers

Classification by build

Package type cooling towers are preassembled and can be easily transported and erected at the location of use. These are generally suitable for applications where the heat load to be rejected is not very large (most HVAC and process load applications).

Field erected type of towers are usually much larger to handle the larger heat rejection loads and are custom built as per customer requirements. Most of the construction /assembly of the tower takes place at the site where the tower will be located.

This thesis will be to build a model that predicts the performance of package type towers. Although the performance of field erected type towers may be similar to package type towers, the predicted performance may not be very accurate as a result of the unique and requirement specific construction of field erected type towers.

Classification based on heat transfer method

Wet cooling towers are the most common type of cooling towers and the ones referred to when talking about cooling towers. As explained earlier, they operate on the principle of evaporative cooling. The water to be cooled and the ambient air come in direct contact with each other. This thesis will look mainly at the performance prediction of wet cooling towers since they are the most widely used.

In a dry cooling tower there is a surface (e.g. tube of a heat exchanger) that separates the water from the ambient air. There is no evaporative cooling in this case. Such a tower may be used when the fluid to be cooled needs to be protected from the environment.

In a fluid cooler water is sprayed over tubes through which the fluid to be cooled is flowing while a fan may also be utilized to provide a draft. This incorporates the mechanics of evaporative cooling in a wet cooling tower while also allowing the working fluid to be free of contaminants or environmental contact.

Classification based on type of Fill

In a spray fill tower the water is broken down into small droplets so that the area of contact between the water surface and air is increased. So in a way spray fill is not really a fill because there is no packing in the tower. Small droplets of water are created by spraying through nozzles, which are contained within the tower casing, through which there is airflow. The drawbacks of this kind of a fill are low efficiency, large tower size and large airflow requirement.

In a splash fill tower, there are slats of wood, PVC or ceramic material over which the water cascades down the tower. As the water splashes over the slats, it forms small droplets which allow for better tower performance. A splash fill is shown in Figure 1.3.

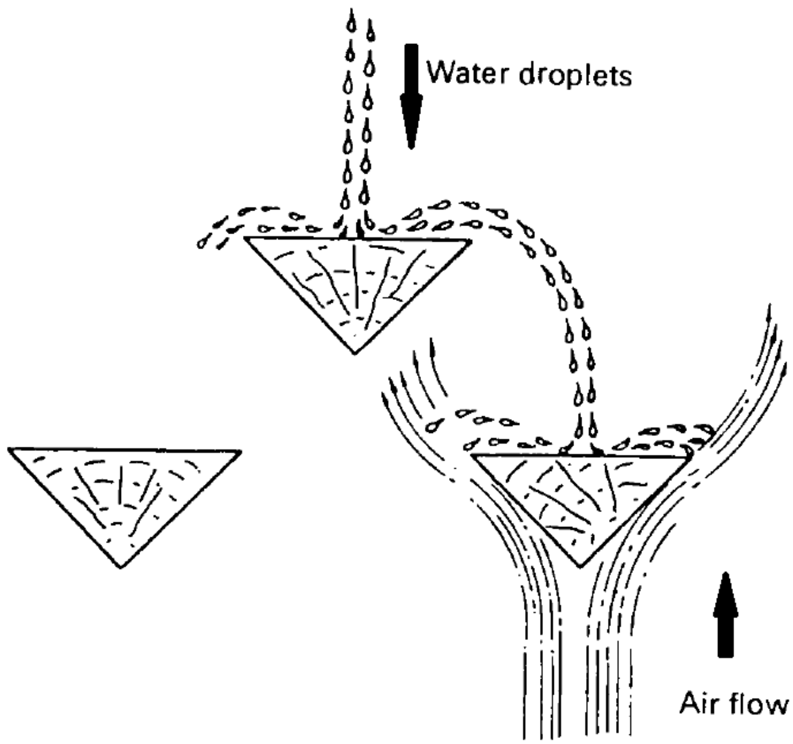


Figure 1.3 Triangular Slat Splash Fill

In a film fill, large surface area is provided for the water to flow over, which causes it to form a thin film. Because of this large contact area between the water surface and air, efficient evaporative cooling is seen. In this kind of a tower, the pressure drop as the air flows through the tower is lower as compared to the previous types and thus lesser fan power is needed to move the air through the tower. An example of a film fill for a cooling tower is shown in Figure 1.4. Film fill is less expensive and more efficient than splash fill and has resulted in its widespread use in cooling towers.

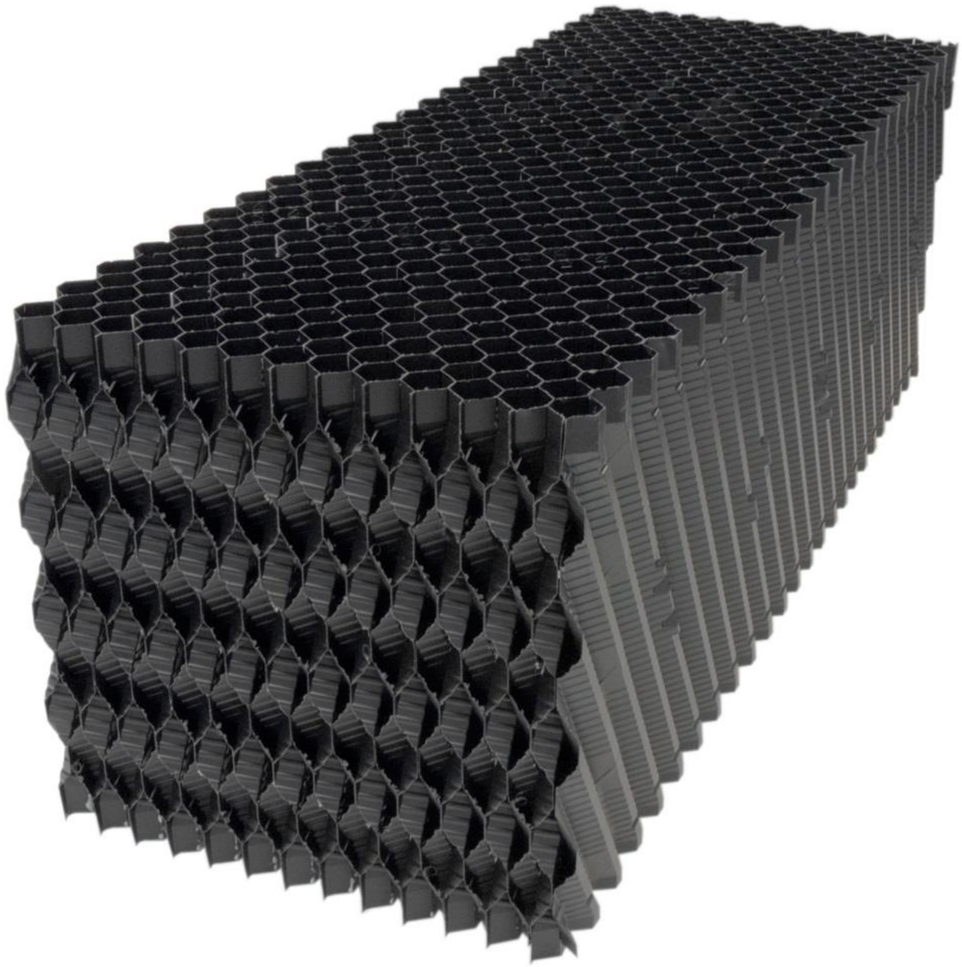


Figure 1.4 Typical Film fill for a Cooling Tower

Classification based on air draft

In an atmospheric tower the air enters the tower through louvers driven by its own velocity. This kind of a tower is inexpensive. Since the performance is greatly affected by wind conditions it is largely inefficient and is seldom used when accurate and consistent cold water temperatures are required.

A natural draft tower (also known as hyperbolic cooling tower) is similar to an atmospheric tower in that there is no mechanical device to create air flow through the tower. However it is dependable and consistent unlike an atmospheric tower. The air flow through the tower is a result of the density differential between hot and less dense air inside the tower as compared to the relatively cooler and denser ambient air outside the tower. The hot air rises up through the tower while cool ambient air is drawn in through inlets at the bottom of the tower. Natural draft towers are extensively used in electric power generation plants and areas where there is higher relative humidity. These towers are much more expensive as compared to other tower types and conspicuous by their hyperbolic shape which is so designed because

- The natural upward draft is enhanced by such a shape
- This shape provides better structural strength and stability

Sometimes natural draft towers are equipped with fans to augment the air flow and are referred to as fan assisted natural draft towers or hybrid draft towers.

Mechanical draft towers have one or more fans that are used to move the air through the tower to provide predictable and consistent performance making them the tower of choice in most HVAC and process applications. A mechanical draft tower can be subdivided in two types, namely, forced draft towers and induced draft towers.

The tower is termed a forced draft tower if the fans are arranged so as to blow air into the tower. Thus there is a positive pressure in the tower fill as compared to the outside. In this case the fans are generally located at the point where the air enters the tower.

The tower is termed an induced draft tower if the fans are arranged so as to push air out of the tower. Thus there is a negative pressure in the tower fill as compared to outside. The fan is located at the point where the air leaves the tower.

Figure 1.5 shows the configuration of the cooling tower for forced draft and induced draft fans as taken from Stanford [5].

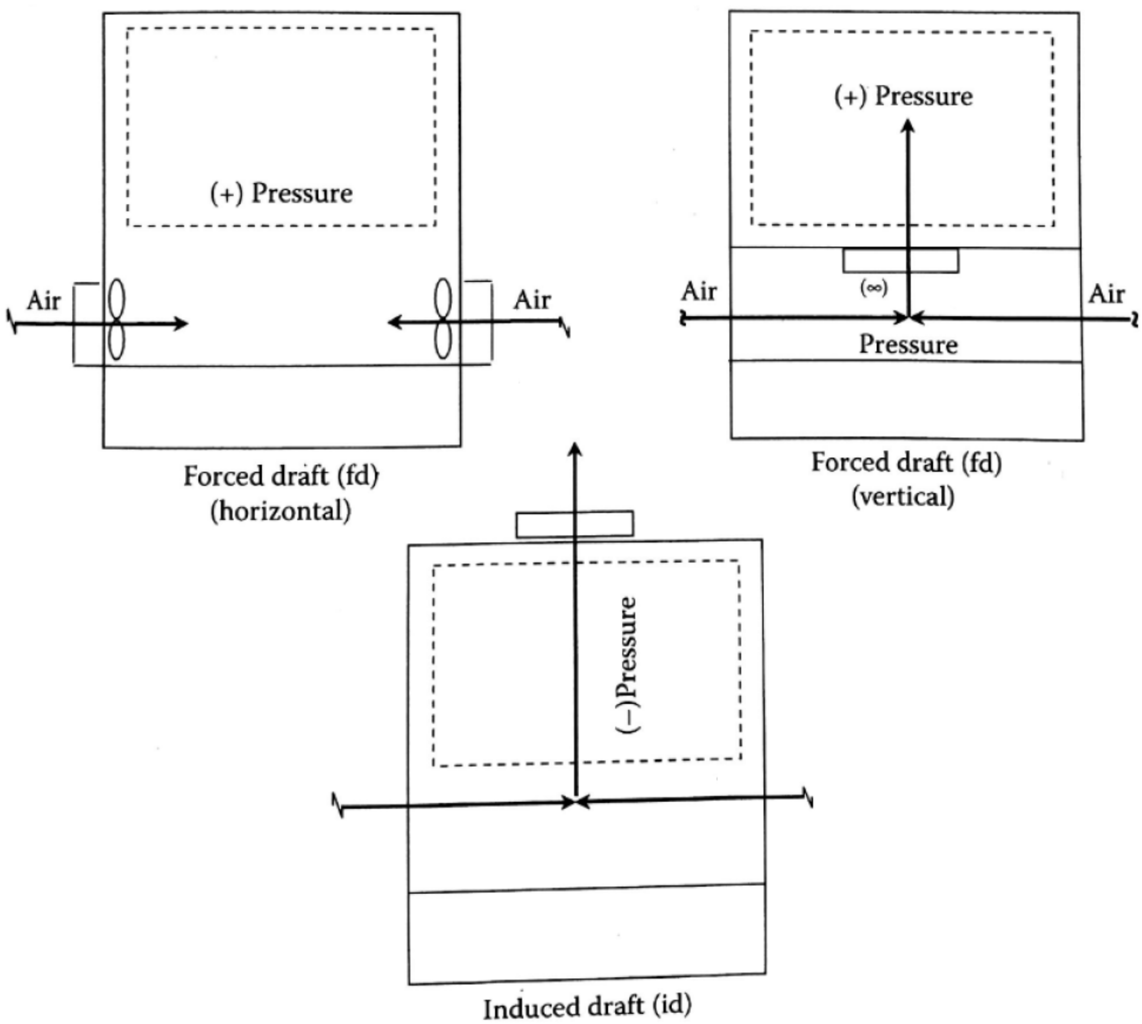


Figure 1.5 Schematic of Forced Draft and Induced Draft Cooling Towers

Classification based on air flow pattern

In a crossflow tower the direction of air flow is perpendicular to that of the water flow i.e. the water flows vertically downward through the fill, while the air flows horizontally thorough the fill.

In a counterflow tower the direction of air flow is directly opposite to that of the water flow i.e. as the water flows vertically downwards through the fill, the air flows vertically upwards through it.

Figure 1.6 shows the configuration of crossflow and counterflow cooling tower configurations as given in Stanford [5].

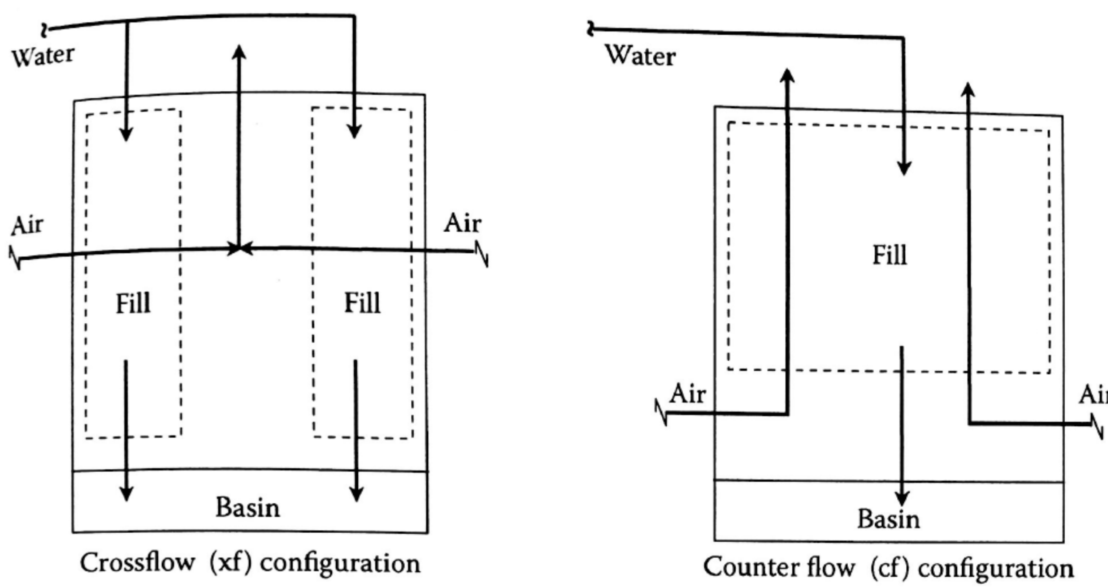


Figure 1.6 Schematic of Crossflow and Counterflow Cooling Towers

1.3 Literature Review

In 1925, Merkel [7] was one of the first to propose a theory to quantify the complex heat transfer phenomena in a counterflow cooling tower. Merkel made several simplifying assumptions so that the relationships governing a counterflow cooling tower could be solved much more easily. Benton [2] and Kloppers and Kroger [8] list the assumptions of the Merkel theory as follows

- The saturated air film is at the temperature of bulk water
- The saturated air film offers no resistance to heat transfer
- The vapor content of the air is proportional to the partial pressure of water vapor
- The force driving heat transfer is the differential enthalpy between saturated and bulk air
- The specific heat of the air water vapor mixture and heat of vaporization are constant.
- The loss of water by evaporation is neglected. (This simplification has a greater influence at elevated ambient temperatures)
- The air exiting the tower is saturated with water vapor and is characterized only by its enthalpy. (This assumption regarding saturation has a negligible influence above ambient temp of 68°F but is of importance at lower temperatures)
- The Lewis factor relating heat and mass transfer is equal to 1. (This assumption has a small influence but affects results at low temperatures.)

This model has been widely applied because of its simplicity. Baker and Shryock [9] give a detailed explanation of the procedure of arriving at the final equations of the Merkel theory and also list some of the shortcomings of the Merkel theory and suggest some corrections. Bourillot [10] developed a program called TEFERI to predict the performance of an evaporative cooling tower in 1983. Benton [11] developed the FACTS model in 1983 and compared it to test data. Benton [2] states that the FACTS model is widely used by the utilities to model cooling tower performance. Majumdar [12] reviewed the then existing methods of cooling tower performance evaluation and developed a new mathematical model that is embodied in a computer code called VERA2D. Majumdar [12] also gives a more detailed list of available mathematical models for analyzing wet cooling towers and this is shown in Figure 1.7.

		Applicability to the Types of Tower				Mathematical Models		
		Natural		Mechanical				
Authors	Year	Counter-flow	Cross-flow	Counter-flow	Cross-flow	Heat Transfer	Airflow	Coupled (?)
Chilton (1952)	1952	•				0D[(b)]	0D	Yes
Rish (1961)	1961	•				0D	0D	Yes
Singham and Spalding (1965)	1965	•				0D	0D	Yes
Foster Wheeler Corp. (1943)	1943			•		0D	0D	No
Pritchard and Co. (1957)	1957			•		0D	0D	No
Cooling Tower Inst. (1967)	1967			•		0D	0D	No
Mesarovic (1973)	1973	•				1D	0D	No
Zivi and Brand (1957)	1956				•	2D	Not Solved	—
Baker and Shyrock (1961)	1961				•	2D	Not Solved	—
Cross et al. (1976)	1976			•	•	1D for Counter 2D for Cross	Not Solved	—
Kelly (1976)	1970				•	2D	0D	No
Winniarski and Tichenor (1979)	1978	•	•			1D for Counter 2D for Cross	0D	Yes
Penney, Rosten, and Spalding (1979)	1979	•	•			2D	2D	Yes
I. A. Furzer (1978)	1978	•				2D	2D	Yes
Majumdar, Singhal, and Spalding (1983)	1981	•	•	•	•	2D	2D	Yes

(a) Adapted from Majumdar, Singhal, and Spalding (1983)
(b) 0D = Zero Dimensional (Global) Model.

Figure 1.7 Summary of models available for analyzing wet cooling towers

In 1989 Jaber and Webb [13] developed equations to apply the ϵ–NTU method of heat exchanger design to design cooling towers. The Merkel method and ϵ–NTU method with modifications are the methods generally used to predict tower performance. Bergsten [14] states that the ϵ–NTU method (with some modifications) is used in well-known and wide spread building simulation programs such as TRNSYS, EnergyPlus and the ASHRAE Primary HVAC Toolkit package. Poppe and Roegener [15] came up with the Poppe model also known as the exact model in 1991 which does not make the simplifying assumptions of Merkel's theory and is therefore more accurate. Kloppers and Kroeger [16] critically evaluate the Merkel theory by comparing it with the Poppe method. Kloppers and Kroger [8] give a detailed derivation of the Merkel, Poppe and Entu methods, their comparison and how to solve the governing equations in each of the methods. They conclude that the Poppe method is more accurate than the Merkel and ϵ–NTU methods and that the Merkel and ϵ–NTU methods give identical results since they are based on the same simplifying assumptions. With the advancement of computing power, computational Fluid Dynamics (CFD) models have been created to simulate performance of cooling towers [17].

Ebrahim et al [18] looked at the thermal performance of cooling towers under variable wet bulb temperature and report that as the wet bulb temperature increases, the approach, range and evaporation loss all increase considerably. Other information about low temperature tower performance is hard to come by.

The DOE-2, a widely used building energy analysis program predicts the cooling tower performance through a statistical model. Benton et al [2] say that the DOE2 uses a 12 parameter variable curve fit. They further develop a statistical model through multiple linear least squares regression of vendor data and compare it to the DOE2 model, Merkel model, ϵ–NTU model and Poppe model. They surmise that the statistically developed model is comparable to the analytically developed ones and is better than the DOE2 model while also being faster than the other models.

To conclude, all the past literature have served as driving factors for this research and show that there has been an effort to predict performance of cooling towers. The literature review also reveals that none of the models have been created for a regular user (e.g. a chilled water system operator) to be able to determine possible energy saving opportunities. All the aforestated models require a lot of computation and significant knowledge of the tower type, structure, materials used and thermodynamic properties. Further they do not give much information about cooling tower operation at low temperatures or about economizer operation of the cooling tower (also known as free cooling). Creating a new model to study cooling tower performance through a wider range of temperatures while also trying to keep the required parameters to a minimum without sacrificing accuracy will go a long way towards realizing opportunities for energy savings

1.4 Research Objectives

The objective is to create a new model to predict cooling tower performance over a larger operating range thereby allowing for better determination of energy saving measures like free cooling and VFDs on cooling tower fans. This will need to be carried out without making the process cumbersome so that an average user would be able to use the model, to achieve fairly accurate results without the model demanding too many inputs. Towards this objective, the following steps were carried out:

1) Collect a large range of cooling tower operating data from cooling tower manufacturers that is suitable to analyze the extent of validity of the current model as well as create a new model to simulate cooling tower performance
2) Compare the existing model in CWSAT to manufacturer tower data to identify and quantify the shortcomings of the current model
3) Create a new model and validate the results
4) Compare the new model with the existing model to demonstrate improvements

5) Demonstrate that the cooling tower performance reduces when operated in free cooling mode at lower wet bulb temperatures
6) Determine if installing oversized cooling towers is economically feasible to take advantage of economizer operation

1.5 Organization

Chapter 1 gives an introduction. Chapter 2 starts off with a discussion on the limitations of existing models as outlined in the literature review. Chapter 3 dives into the how the data for creating a new model was collected and what the sources were. Chapter 4 gives a detailed account of methods available to create new models and the polynomial regression method used to create a new model in this thesis. The model performance is verified and the results discussed. Chapter 5 addresses the reduction of cooling tower capacity when operated in a free cooling mode at low temperatures and the benefits of having a larger tower capacity to be able to incorporate free cooling operation in winter months. In Chapter 6 a summary is given and recommendations for future work are made.

CHAPTER 2

EXISTING MODELS AND THEIR LIMITATIONS

The purpose of a cooling tower model is to be able to predict the cooling tower performance. However not all models are suitable for an average user to utilize and determine energy use and possible saving measures. This thesis and chapter focusses on the existing models capability to meet the needs of a user to easily estimate cooling tower energy use and look at possible energy savings.

2.1 Drawbacks of Thermodynamic Models

In the literature review it was observed that there were numerous models to predict cooling tower performance through thermodynamics and heat transfer principles. Before moving on to create a new model, a few of the drawbacks of such models will be discussed.

Amongst the thermodynamic models found in literature, the simplest one was seen to be the Merkel method. According to Kloppers [16], the Merkel equation is given by

$$Me_M = \frac{h_d a_{fi} A_{fr} L_{fi}}{m_w} = \frac{h_d a_{fi} L_{fi}}{G_w} = \int_{T_{wo}}^{T_{wi}} \frac{c_{pw} dT_w}{\left(i_{masw} - i_{ma}\right)} \tag{2.1}$$

Where,

Me_M = Transfer coefficient or Merkel number

h_d = Mass transfer coefficient, $\mathrm{kg/m^2s}$

a_{fi} = Surface area of fill per unit volume of the fill, $\mathrm{m^{-1}}$

A_{fr} = Frontal Area; m^2

L_{fi} = Length of fill; m

m_w = Mass flow rate of water; kg/s

c_{pw} = Specific heat of water at constant pressure; J/kgK

i_{masw} = Enthalpy of mean air saturated with water; J/kg

i_{ma} = Enthalpy of mean air; J/kg

T_{wi} = Temperature of water at inlet of the tower; K

T_{wo} = Temperature of water at outlet of the tower; K

Kloppers [16] also states that it is difficult to evaluate the surface area per unit volume of fill due to the complex nature of the two phase flow in fills. However it is not necessary to explicitly specify the surface area per unit volume or the mass transfer coefficient since the value of the Merkel number can be obtained by integrating the right hand side of the equation above. Further it is to be noted that the exact state of the air leaving the fill cannot be calculated and is assumed to be saturated with water vapor so that temperature of water leaving the fill may be calculated. Bourillot [10] has stated that the Merkel method is simple to use and can correctly predict the cold water temperature when an appropriate value of coefficient is used but is insufficient for estimating the characteristics of warm air leaving the fill and for calculation of changes in the water flow rate due to evaporation. Using the equation above requires quite a few parameters that are not easily available to an average person and if we are looking at information on how air flow rate will affect the temperature of water leaving the tower (to determine savings possible through VFD operation of the tower fan) it is impossible to proceed without having even more information.

The ϵ–NTU method is very similar to the Merkel method in the solutions it gives because of the same simplifying assumptions. The equation for the ϵ–NTU according to Jaber and Webb [13] is

$$\frac{d\left(i_{masw}-i_{ma}\right)}{\left(i_{masw}-i_{ma}\right)}=h_d\left(\frac{\frac{di_{masw}}{dT_w}}{m_w c_{pw}}-\frac{1}{m_a}\right)dA \tag{2.2}$$

The equation above corresponds to the heat exchanger ϵ–NTU equation which is given by

$$\frac{d\left(T_h-T_c\right)}{\left(T_h-T_c\right)}=-U\left(\frac{1}{m_h c_{ph}}+\frac{1}{m_c c_{pc}}\right)dA \tag{2.3}$$

Where,

T_h = Temperature of the hot fluid in a heat exchanger; K

T_c = Temperature of the cold fluid in a heat exchanger; K

m_h = mass flow rate of hot fluid; kg/s

m_c = mass flow rate of cold fluid; kg/s

c_{ph} = Specific heat of hot fluid at constant pressure; J/kgK

c_{ph} = Specific heat of cold fluid at constant pressure; J/kgK

Now comparing the two equations we can simplify the cooling tower to a heat exchanger and use it to predict the temperature of water leaving the tower. Once again there are quite a few parameters involved, requiring in depth engineering knowledge to be able to use the ϵ–NTU model to model tower performance.

Next we look at the Poppe model. The governing equations for the heat and mass transfer in the fill for unsaturated air are given by the following equations.

$$\frac{Me_P}{dT_w} = \frac{c_{pw}}{i_{masw} - i_{ma} + \left(Le_f - 1\right)\left[i_{masw} - i_{ma} - \left(w_{sw} - w\right)i_v\right] - \left(w_{sw} - w\right)c_{pw}T_w} \tag{2.4}$$

$$Me_P = \int \frac{m_a\left(\frac{dw}{dT_w}\right)}{m_w\left(w_{sw} - w\right)} dT_w \tag{2.5}$$

$$Le_f = 0.865^{0.667} \frac{\left(\frac{w_{sw} + 0.622}{w + 0.622} - 1\right)}{\ln\left(\frac{w_{sw} + 0.622}{w + 0.622}\right)} \tag{2.6}$$

$$\frac{m_w}{m_a} = \frac{m_{wi}}{m_a}\left(1 - \frac{m_a}{m_{wi}}\left(w_o - w\right)\right) \tag{2.7}$$

Where,

Me_P = Merkel number according to Poppe method

Le_f = Dimensionless Lewis factor

w = Humidity Ratio of air; kg water vapor/ kg dry air

w_{sw} = Humidity Ratio of air saturated with water; kg water vapor/ kg dry air

w_o = Humidity Ratio of air at outlet of tower; kg water vapor/ kg dry air

One method of solving these governing differential equations is by the fourth order Runge-Kutta method according to Kloppers [8] and is clearly outlined in [8]. Further [8] says that the air outlet conditions can be calculated from the equations above and that since the value of w_0 is not known a priori, the equations are solved iteratively. Kloppers [8] also says that the Poppe method predicts the water content of the exiting air accurately and the results are consistent with full scale cooling tower test results.

Kloppers [16] concludes that if the temperature of water leaving the tower is of interest, then both the Merkel and Poppe model predict identical temperatures. Further the Merkel method predicts heat rejection rates and air outlet temperatures very accurately when the actual outlet air is supersaturated with water vapor but when the ambient air is relatively hot and dry, the outlet air may be unsaturated and the number predicted by the Poppe and Merkel model may differ significantly. The Merkel and ϵ–NTU model give almost identical answers because of the same underlying assumptions. The Poppe model gives overall better results since no assumptions regarding the state of exiting air or Lewis number are made but is comparatively more complex to solve than the Merkel method whose assumptions make solving it a simpler hand calculation.

As discussed in the literature review earlier, there are many other models which utilize similar equations based on thermodynamics to predict tower performance. Without droning on further about how these models are unsuitable, it can be concluded that all thermodynamic models require considerable information to be able to use them to predict tower thermodynamic performance. The reason for this is that these models are not meant for an average user to determine tower fan energy use but rather for tower designers in building and evaluating towers who have all of the information readily at hand. Thus we have a strong case for surrogate models which can do away with the unnecessary thermodynamics and use information that is more readily available to predict tower performance without losing accuracy.

2.2 Limitations of Existing Model

In the literature review it was identified that there are two metamodels, one is used in the DOE2 engine and the other was developed by Benton et al. [2]. Information about the model used in DOE2 was unavailable and [2] say that their model is more accurate than the DOE2 model. A metamodel is an engineering method used when an outcome of interest cannot be easily directly measured, so a model of the outcome is used instead. The advantages of a metamodel are that it takes into account the variables that affect the process and which are readily available to the user to use as predictors. Benton [2] choose the parameters for their model as the wet bulb temperature, the range, water flow, fan power and approach. These parameters represent the extent of the information available to an average user and thus represent a very good set of parameters for a surrogate model. These parameters are suitable because

a) Wet Bulb Temperature – For most locations, typical meteorological year (TMY) data is available that can be used to determine the wet bulb temperature on an hourly basis. Suitable sensors if present in the system can also obtain this information.

b) Range – The cooling tower user will have the need to obtain a certain temperature difference between the tower inlet and outlet.

c) Water flow – This information is also readily available to the user. If not directly available, it can be measured relatively easily or even a value closely estimated.

d) Fan power – This is once again information that is readily known or that can be measured.

e) Approach – A user would like to determine the approach based on previous four parameters. Approach is the dependent variable while the remaining four are the independent variables. This may be depicted in the form of an equation as follows

$$Approach = f\left(T_{wb}, Range, Water\ flow, Fan\ Power\right)$$

This metamodel is not only simple to use because of the easily available information but also gives results comparable to thermodynamic models [2]. In this way a surrogate model allows to predict the temperature of water leaving the cooling tower in a much simpler manner, while also keeping it relevant with the available parameters. This also allows us to incorporate the fan energy in the model more easily which is the parameter of most importance to an average user. CWSAT current utilizes this model to predict fan power usage.

As discussed earlier the opportunities for energy savings on a cooling tower are through implementation of a VSD on the cooling tower fan [1] and operating the cooling tower in free cooling mode [3, 5, 19]. There are also savings possible by changing the temperature requirement of cold water required from the tower. For e.g. during free cooling, the temperature of water from the tower required may not be as low as 45°F but only 55°F since the 45°F requirement is mostly to maintain the required level of humidity. Since the water content in the air is much lower in winter, the temperature of cold water required may not be as low as 45°F, but rather 55°F. Increasing the cold water temperature from the tower will decrease the fan energy usage as well as increase the number of hours when free cooling is possible and therefore energy savings. Figure 5.2 in Chapter 5 outlines the capacity of the tower at different values of cold water temperature.

It is seen that the existing model by Benton [2] does not perform very well at low fan power and at low wet bulb temperatures. This means that estimates of energy use during free cooling operation and predicted fan energy savings through VSD are not very accurate. Figures 2.1 to 2.4 show the increased error in prediction of approach as the fan speeds are reduced. Figure 2.5 and Figure 2.6 show reduced tower performance prediction capability at low wet bulb temperature for a counter flow tower. As the wet bulb temperature reduces the magnitude of error is seen to increase. The lowering of prediction capability at lower wet bulb temperature can also be seen in the Figures 2.1 to 2.4 for fan speed variation. This makes a strong case for creation of a new model that can better predict cooling tower performance at these conditions.

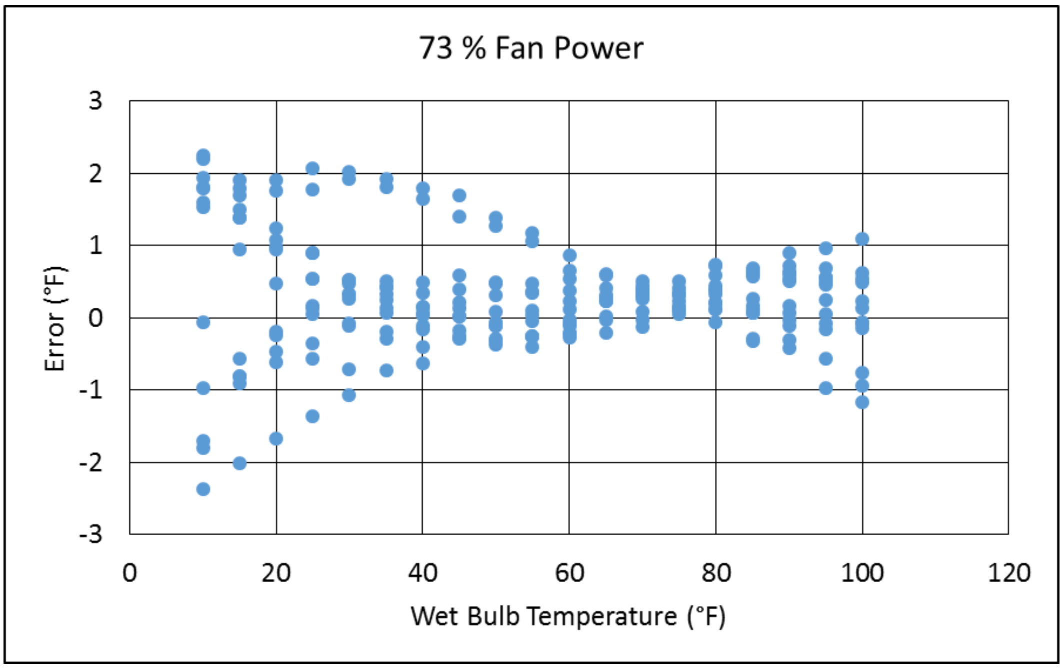


Figure 2.1: Error in Tower Performance Prediction with at 73% Fan Power

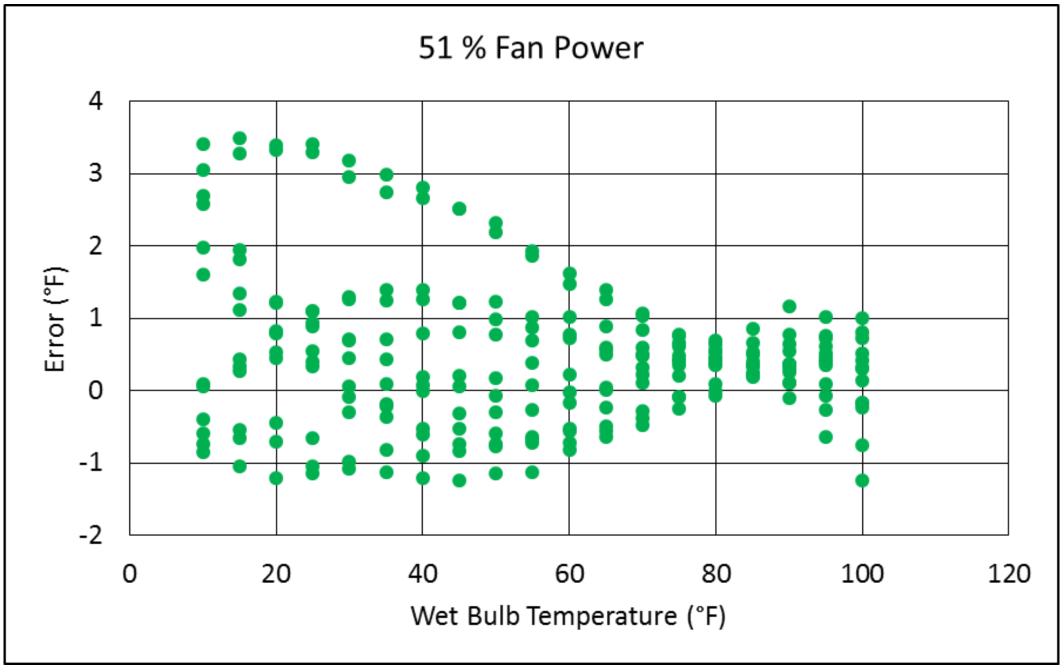


Figure 2.2: Error in Tower Performance Prediction with at 51% Fan Power

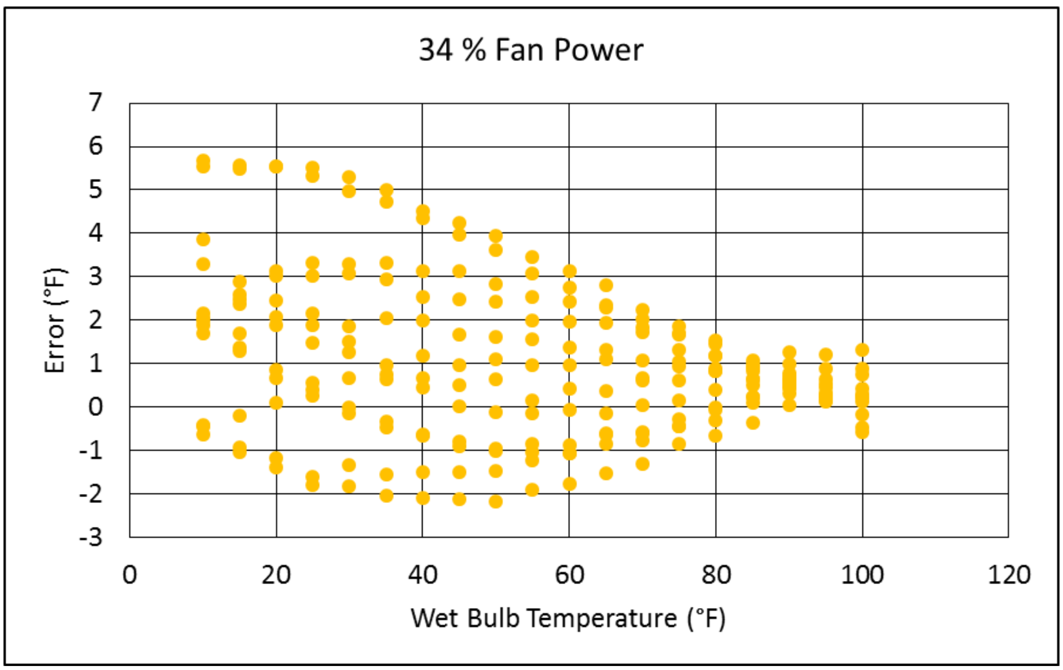


Figure 2.3: Error in Tower Performance Prediction with at 34% Fan Power

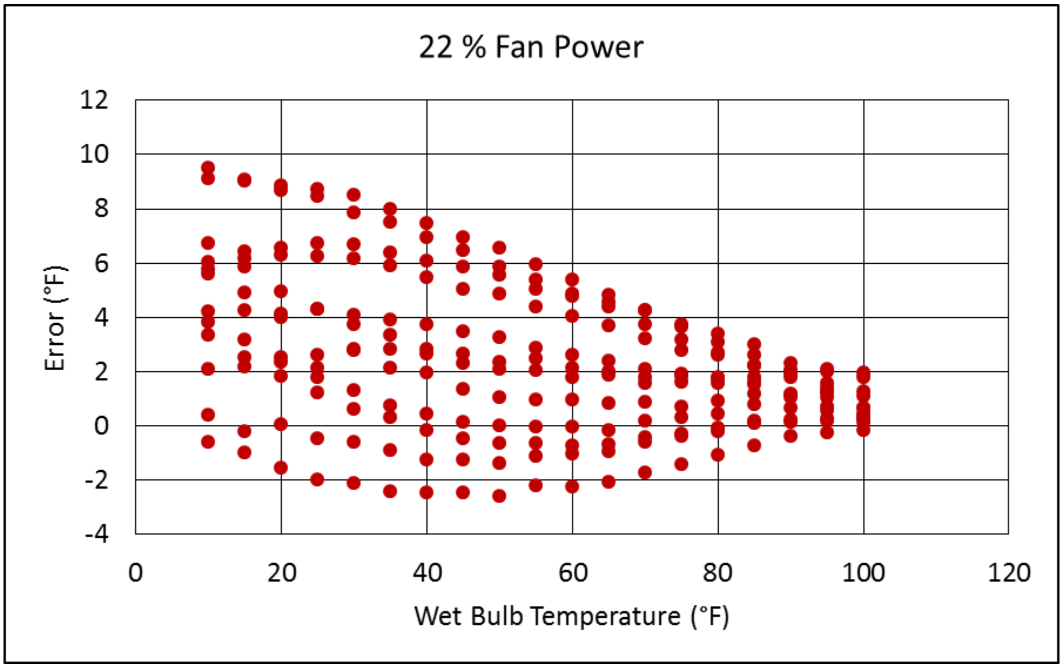


Figure 2.4: Error in Tower Performance Prediction with at 22% Fan Power

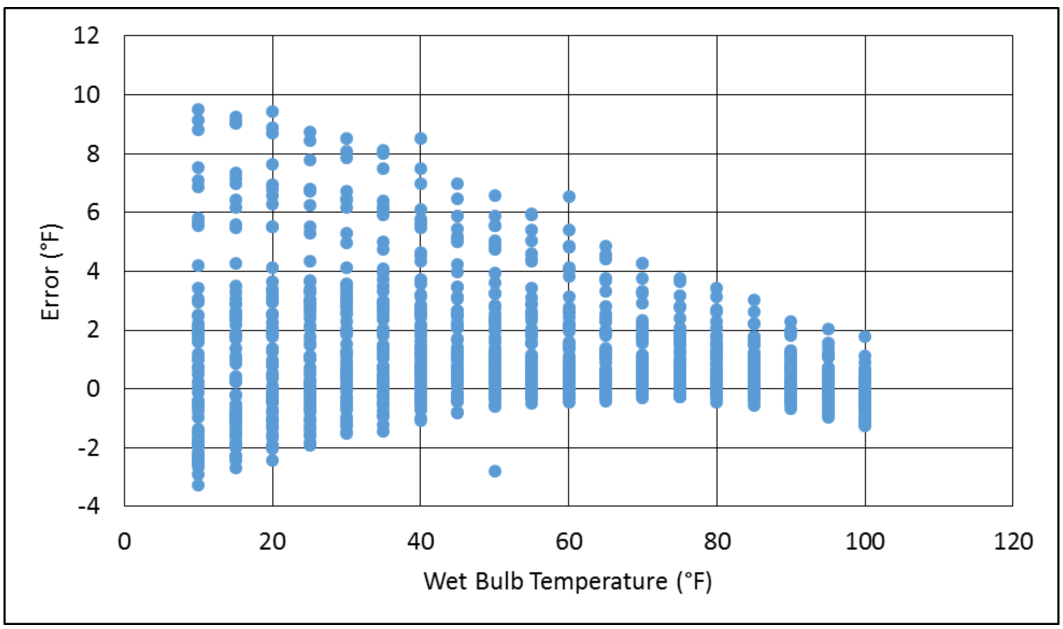


Figure 2.5: Variation in Tower Performance with wet bulb temperature for a Counter Flow Tower

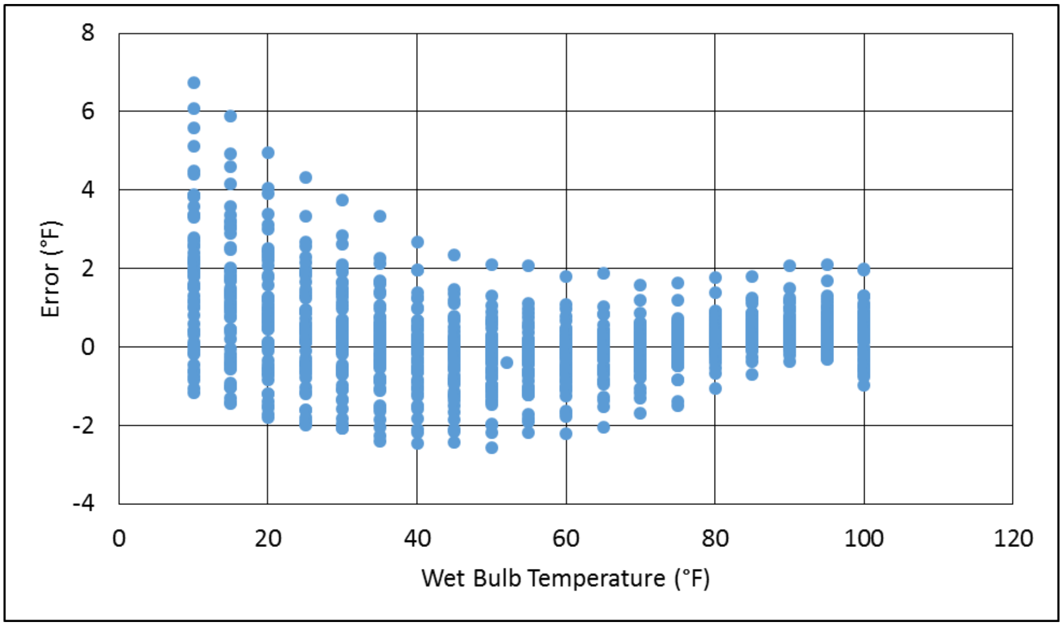


Figure 2.6: Variation in Tower Performance with wet bulb temperature for a Cross Flow Tower

CHAPTER 3

COOLING TOWER PERFORMANCE DATA COLLECTION

Cooling tower performance data is very difficult to access. Most cooling tower manufacturers have proprietary software that they use when providing customers help in selecting a tower. While the Cooling Tower Institute (CTI) certifies some cooling towers sold by many manufacturers on thermal performance, they do not take into account the tower performance in low temperature conditions. This chapter discusses the source and method for data collection required for creating a new model.

3.1 Data Collection Source

The cooling tower data collected needs to be expansive and easy to collect. It was found that Baltimore Aircoil Company (hereinafter referred to as Tower Manufacturer A) and Marley Cooling Towers (hereinafter referred to as Tower Manufacturer B) have a product selection software that generates graphs for the different conditions specified. A graph digitizer was used to gather data points accurately from the graphs (a sample graph is shown in Appendix A) for use in verifying the existing model and then creating a new model.

It was seen that the Tower Manufacturer A was able to provide a larger variation of parameters for wet bulb temperature and water flow as compared to Tower Manufacturer B. Table 3.1 shows the range of variation of different parameters allowed on both the selection software.

Table 3.1: Extent of Cooling Tower Selection Software Parameter Variation

	Tower Manufacturer A	Tower Manufacturer B
Wet Bulb Temperature	10°F - 100°F	20°F - 90°F
Range	2°F - 50°F	3°F - 55°F
Water	80% - 120%	90% - 110%
Fan Speed	0% - 100%	25% - 100%

3.2 Data Collection Approach

The tower performance data across as wide a range of operating conditions as was possible was collected from Tower Manufacturer A and Tower Manufacturer B. Data was collected across two different tower manufacturers to account for variability of tower performance with make.

To account for change in tower performance with type of tower, data was collected separately for counter flow towers and cross flow towers to be able to create a separate model for each. This also helped to verify the existing models for both counter flow towers and cross flow towers.

Tower types and tonnage vary greatly in the field but most packaged type towers range from 50 tons to 750 tons for single celled towers. Larger tonnages are accommodated by larger number of cells rather than a single cell, larger tower of the packaged type or by constructing a custom built tower. To check the impact of tower tonnage, data was collected from a 100 ton tower, a 300 ton tower as well as a 750 ton tower. The 100 ton tower and the 750 ton are compared to the 300 ton tower to see if the performance data for a tower tonnage of 300 tons would be suitable for the prediction model. Table 3.2 and Table 3.3 show the similarity of tower performance across tonnages for counter flow towers for Tower Manufacturer A and Tower Manufacturer B respectively. Table 3.4 and Table 3.5 show the similarity of tower performance across tonnages for cross flow cooling towers for Tower Manufacturer A and Tower Manufacturer B respectively. It is observed that a 300 ton tower is a good estimate for tower performance since performance doesn’t vary more than a few percent points for both the tower manufacturers across the range of tower tonnages. The water flow and fan power is maintained a 100% across tonnages. The performance remains the same since the cooling tower works on the principle of evaporative cooling where the water can be cooled only as low as the wet bulb. Thus no matter the tonnage, since the water flow in gpm/ton is constant and the fan speed is at a 100% the temperature of water leaving the tower tends to be the same. This also saves time on collecting identical data for a large number of tower tonnages for the same tower type.

Table 3.2: Tower Manufacturer A Performance across Various Tonnages for a CF tower

Wet bulb Temperature (°F)	Cold Water Temperature (°F)			Percent Difference in Cold Water Temperature	
	100 Ton Tower	300 Ton Tower	750 Ton Tower	100 Ton VS 300 Ton	750 Ton VS 300 Ton
100	103.2	103	102.8	-0.19%	0.19%
95	98.7	98.9	98.6	0.20%	0.30%
90	94.6	94.6	94.3	0.00%	0.32%
85	90.6	90.4	90.5	-0.22%	-0.11%
80	86.5	86.6	86.6	0.12%	0.00%
75	82.7	82.7	82.7	0.00%	0.00%
70	79	79	79.1	0.00%	-0.13%
65	75.2	75.3	75.7	0.13%	-0.53%
60	71.6	71.9	72.1	0.42%	-0.28%
55	67.9	68.3	68.8	0.59%	-0.73%
50	64.4	64.9	65.3	0.77%	-0.62%
45	61.1	61.5	62.2	0.65%	-1.14%
40	57.7	58.4	59	1.20%	-1.03%
35	54.3	55.1	55.9	1.45%	-1.45%
30	51	52	52.8	1.92%	-1.54%
25	47.6	48.5	49.6	1.86%	-2.27%
20	44	45.1	46.3	2.44%	-2.66%
15	40.6	41.6	42.7	2.40%	-2.64%
10	36.9	38.1	39	3.15%	-2.36%

Table 3.3: Tower Manufacturer B Performance across Various Tonnages for a CF tower

Wet bulb Temperature (°F)	Cold Water Temperature (°F)			Percent Difference in Cold Water Temperature	
	100 Ton Tower	300 Ton Tower	750 Ton Tower	100 Ton VS 300 Ton	750 Ton VS 300 Ton
100	102.3	102.6	102.5	0.29%	0.10%
95	98.4	98.2	98	-0.20%	0.20%
90	94.5	94.3	94.1	-0.21%	0.21%
85	90.5	90.5	90.4	0.00%	0.11%
80	86.6	86.5	86.4	-0.12%	0.12%
75	82.9	82.9	82.8	0.00%	0.12%
70	79.1	79.1	79.1	0.00%	0.00%
65	75.5	75.7	75.6	0.26%	0.13%
60	72	72.3	72.3	0.41%	0.00%
55	68.7	69.1	69	0.58%	0.14%
50	65.5	65.7	65.8	0.30%	-0.15%
45	62.2	62.5	62.7	0.48%	-0.32%
40	59	59.3	59.6	0.51%	-0.51%
35	55.7	56.4	56.7	1.24%	-0.53%
30	52.7	53	53.6	0.57%	-1.13%
25	49.4	50.3	50.6	1.79%	-0.60%
20	46.3	47.1	47.5	1.70%	-0.85%
15	43.2	44	44.4	1.82%	-0.91%
10	40	41	41.3	2.44%	-0.73%

Table 3.4: Tower Manufacturer A Performance across Various Tonnages for a XF tower

Wet bulb Temperature (°F)	Cold Water Temperature (°F)			Percent Difference in Cold Water Temperature	
	150 Ton Tower	300 Ton Tower	750 Ton Tower	150 Ton VS 300 Ton	750 Ton VS 300 Ton
100	102.9	103	102.9	0.10%	0.10%
95	98.8	98.7	98.7	-0.10%	0.00%
90	94.6	94.6	94.5	0.00%	0.11%
85	90.6	90.5	90.6	-0.11%	-0.11%
80	86.7	86.5	86.7	-0.23%	-0.23%
75	82.8	82.7	82.8	-0.12%	-0.12%
70	78.8	78.7	78.8	-0.13%	-0.13%
65	75.2	75.3	75.4	0.13%	-0.13%
60	71.6	71.7	71.7	0.14%	0.00%
55	68	68	67.9	0.00%	0.15%
50	64.6	64.6	64.5	0.00%	0.15%
45	61.2	61.1	61.1	-0.16%	0.00%
40	57.7	57.6	57.6	-0.17%	0.00%
35	54.5	54.5	54.4	0.00%	0.18%
30	51	51	51.1	0.00%	-0.20%
25	47.7	47.6	47.6	-0.21%	0.00%
20	44.2	44.2	44.3	0.00%	-0.23%
15	40.8	41.2	40.9	0.97%	0.73%
10	37.3	38.1	37.5	2.10%	1.57%

Table 3.5: Tower Manufacturer B Performance across Various Tonnages for a XF tower

Wet bulb Temperature (°F)	Cold Water Temperature (°F)			Percent Difference in Cold Water Temperature	
	150 Ton Tower	300 Ton Tower	750 Ton Tower	150 Ton VS 300 Ton	750 Ton VS 300 Ton
100	102.3	102.6	102.5	0.29%	0.10%
95	98.4	98.2	98	-0.20%	0.20%
90	94.5	94.3	94.1	-0.21%	0.21%
85	90.5	90.5	90.4	0.00%	0.11%
80	86.6	86.5	86.4	-0.12%	0.12%
75	82.9	82.9	82.8	0.00%	0.12%
70	79.1	79.1	79.1	0.00%	0.00%
65	75.5	75.7	75.6	0.26%	0.13%
60	72	72.3	72.3	0.41%	0.00%
55	68.7	69.1	69	0.58%	0.14%
50	65.5	65.7	65.8	0.30%	-0.15%
45	62.2	62.5	62.7	0.48%	-0.32%
40	59	59.3	59.6	0.51%	-0.51%
35	55.7	56.4	56.7	1.24%	-0.53%
30	52.7	53	53.6	0.57%	-1.13%
25	49.4	50.3	50.6	1.79%	-0.60%
20	46.3	47.1	47.5	1.70%	-0.85%
15	43.2	44	44.4	1.82%	-0.91%
10	40	41	41.3	2.44%	-0.73%

In all, 1,350 data points were collected for both counter flow cooling towers and cross flow cooling towers each. Of the 1,350 data points, 798 data points were obtained from Tower Manufacturer A and 552 data points from Tower Manufacturer B. Each data point corresponds to the approach of the tower based on the four parameters of wet bulb temperature, range, percent water flow and percent fan power. Once all of the data is collected, a new model is created as discussed in Chapter 4.

CHAPTER 4

NEW MODEL CREATION

This chapter deals with the creation of a new model for cooling tower performance. Available techniques of surrogate model creation are investigated and the method of polynomial regression is chosen as a suitable technique for this situation. All the steps involved in the creation of a new model are discussed. The results are then verified and improvements over the previous model are presented.

4.1 Model Creation Techniques

Forrester and Keane [20], Koziel et al. [21] and Queipo et.al. [22] give a detailed account of the methods suitable for constructing surrogate models. The methods generally used are

1. Polynomial Regression (PR)
2. Artificial Neural Networks (ANN)
3. Kriging
4. Radial Basis Functions (RBF)
5. Moving Least Squares (MLS)
6. Support Vector Regression (SVR)

Of these varied techniques, polynomial regression (PR) and artificial neural networks (ANN) are discussed. The reason for choosing polynomial regression is that the current model is based on polynomial regression and thus represents a good opportunity to create a new model in the same manner and compare it with the older one. The reason for choosing ANNs is that they represent a relatively new method of model creation and it would be useful to determine if this method gives good solutions and see its advantages and disadvantages as compared to polynomial regression.

Polynomial regression is a form of linear regression in which the relationship between the independent variables and the dependent variables is modelled as an n^{th} degree polynomial. According to Queipo et.al. [22] Polynomial Regression (PR) is a methodology that studies the quantitative association between a function of interest f, and N_{PRG} basis functions z_j, where there are N_S sample values of the function of interest f_i, for a set of basis functions $z_j^{(i)}$. For each observation i, a linear equation is formulated as given below where the errors ε_i are considered independents with expected value equal to zero and variance σ^2. The $\hat{\beta}$ are the estimated parameters (by method of least squares) are unbiased and have minimum variance.

$$f_i\left(\mathrm{z}\right)=\sum_{j=1}^{N_{PRG}}\beta_j z_j^{(i)}+E\left(\varepsilon_i\right)\ ,\quad E\left(\varepsilon_i\right)=0\ ,\ V\left(\varepsilon_i\right)=\sigma^2 \tag{4.1}$$

The same can be represented in a much simpler fashion as follows

$$\mathrm{f}=\mathrm{X}\beta+\varepsilon,\ \ E\left(\varepsilon\right)=0,\ \ V\left(\varepsilon\right)=\sigma^2 I \tag{4.2}$$

Where X is a $N_S\times N_{PRG}$ matrix of basis functions with the design variables for sampled points.

For this specific case of modelling approach as a function of four design variables x_1, x_2, x_3 and x_4 i.e. wet bulb temperature, range, percent water flow and percent fan power respectively, the complete equation for the model will yield 35 terms and is represented as follows,

$$
\begin{aligned}
Approach = {} & \beta_0 + \beta_1 x_1 + \beta_2 x_2 + \beta_3 x_3 + \beta_4 x_4 \\
& + \beta_5 x_1^2 + \beta_6 x_2^2 + \beta_7 x_3^2 + \beta_8 x_4^2 \\
& + \beta_9 x_1 x_2 + \beta_{10} x_1 x_3 + \beta_{11} x_1 x_4 + \beta_{12} x_2 x_3 + \beta_{13} x_2 x_4 + \beta_{14} x_3 x_4 \\
& + \beta_{15} x_1^3 + \beta_{16} x_2^3 + \beta_{17} x_3^3 + \beta_{18} x_4^3 \\
& + \beta_{19} x_1 x_2 x_3 + \beta_{20} x_1 x_3 x_4 + \beta_{21} x_1 x_2 x_4 + \beta_{22} x_2 x_3 x_4 \\
& + \beta_{23} x_1^2 x_2 + \beta_{24} x_1^2 x_3 + \beta_{25} x_1^2 x_4 \\
& + \beta_{26} x_2^2 x_1 + \beta_{27} x_2^2 x_3 + \beta_{28} x_2^2 x_4 \\
& + \beta_{29} x_3^2 x_1 + \beta_{30} x_3^2 x_2 + \beta_{31} x_3^2 x_4 \\
& + \beta_{32} x_4^2 x_1 + \beta_{33} x_4^2 x_2 + \beta_{34} x_4^2 x_3
\end{aligned} \tag{4.3}
$$

Artificial Neural Networks (also referred to as just neural networks) are computational models that are inspired by biological nervous systems and consist of neurons that perform operations. Koziel et.al. [21] state that the neuron performs an affine transformation followed by a nonlinear operation. If the inputs to a neuron are denoted as $x_1, x_2, \ldots, x_n$, then the neuron output is computed as

$$
y = \frac{1}{1 + e^{\left(\frac{-\eta}{T}\right)}} \tag{4.4}
$$

Where $\eta = w_1 x_1 + \ldots + w_n x_n + \gamma$, with $w_1, w_2, \ldots, w_n$ being regression coefficients, γ being the bias value of a neuron and T being the user defined slope or parameter. This is depicted in Figure 4.1 taken from Gershenson [23]. The equation 4.4 above is a sigmoid activation function. The other commonly used activation functions are the threshold and hyperbolic tangent. The sigmoid function is preferred in this case since it represents a smooth, continuous, nonlinear function. The most common neural network architecture is the multi-layer feed-forward network and is shown in Figure 4.2. Once a suitable network architecture is chosen, the next step is to train the ANN. The inputs and their corresponding outputs are given to the ANN which “learns” and adjusts the weights to be able to give the correct output. Koziel et.al. [21] further state that the network training can be stated as a nonlinear least squares regression problem for a number of training points and a popular technique for solving this regression problem is the error back-propagation algorithm.

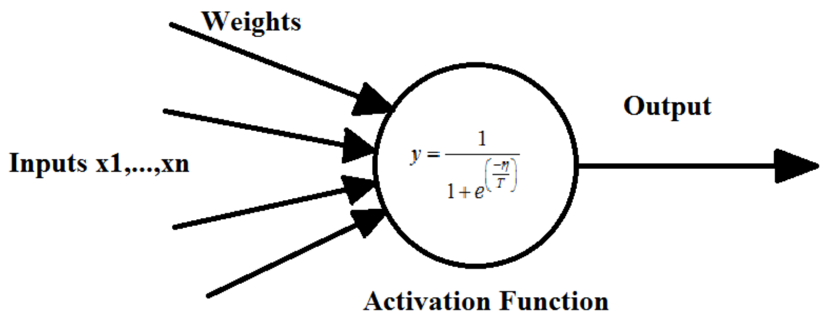


Figure 4.1: Basic Structure of an Artificial Neuron

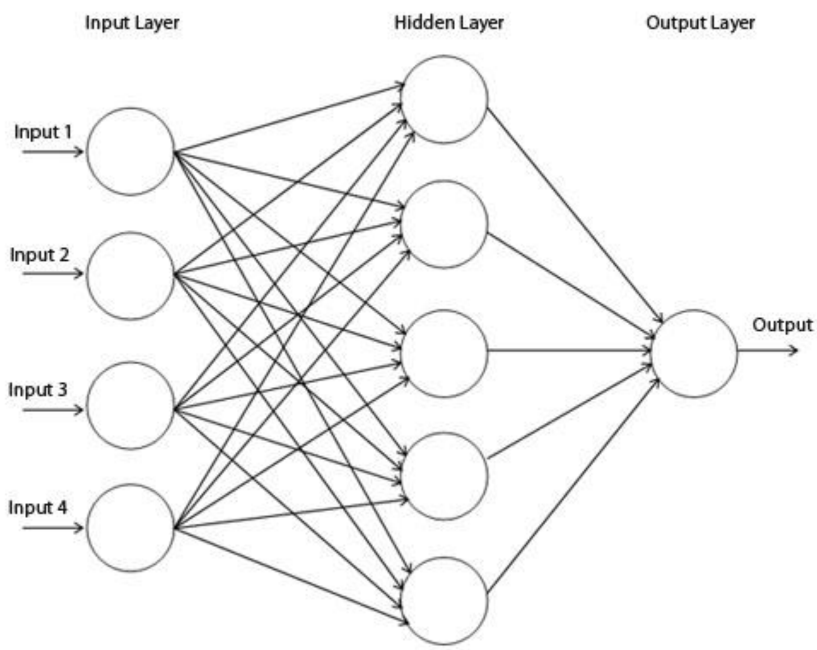


Figure 4.2: Two Layer Feed Forward Neural Network Architecture

4.2 New Model Creation through Polynomial Regression

The data collection process has been discussed. Also as discussed earlier, the new model is a third order polynomial regression with the approach as the dependent variable and wet bulb temperature, range, water flow and fan power as the independent variables. Before starting on the model creation, a note on the cross validation technique that will be employed to make sure we do not overfit the data [24].

Cross validation is a way of measuring the predictive performance of a statistical model. It is generally used in situations where the goal is prediction (as in this case). Model fit statistics are not completely indicative of the predictive performance of a model. It is easy to keep adding higher order terms in polynomial regression until we get an $R^2=1$ and yet this can adversely affect the prediction capability of the model. To overcome this, one way is to keep adding the higher order terms one by one and check if it improves model prediction performance. This is time consuming and requires a lot of effort. Another method is cross validation. Here the collected data is randomly partitioned into groups and one group (also called training set) is used to build the model and the other group (also called the validation set or testing set) is used to validate the model.

There are different methods of cross validation. In general they may divided into exhaustive cross validation and non-exhaustive cross validation. Exhaustive cross validation methods are those in which the original sample is split into a training set and validation set in all possible ways. In non-exhaustive cross validation the original sample is split into a predetermined number of training and validation sets.

The cross validation method used in this case is k-fold cross validation technique which is a type of non-exhaustive cross validation method. In k-fold cross validation, the data is randomly split into k subsets of fairly equal size. Then k-1 sample sets are used as training data and the remaining one sample is used as the validation set. This is repeated with all the k samples being used as the validation set once. Based on the results a suitable model from the k models may be selected or the results may be averaged over the k folds to get the final model parameters. More information on cross validation is found in [24-26]. There is no fixed value for k and for this project the value of k=6 is chosen. In this way all of the data plays a role in building the model and also validating it without leading to post hoc theorizing. Post hoc theorizing in this case would mean that we use the same data to build the model and test it against the same leading us to believe that the new model is suitable even when in fact it may not be and the cross validation technique as outlined earlier helps us prevent this.

Once the data consisting of 1,350 data points is randomly split into 6 subsets polynomial regression is carried out. The regression through least squares is carried out for each of the k folds. The statistical results for one such regression for a counter flow tower is shown in Figure 4.3. The 35 coefficients obtained for each regression over each fold for a counter flow tower and a cross flow tower are as shown in Table 4.1 and Table 4.2 respectively. It is seen that the coefficients are fairly similar on each regression test. This reinforces our belief that we are not overfitting the model. The results are then averaged out over the six folds and used to verify the model.

Linear regression model:
    y ~ [Linear formula with 35 terms in 4 predictors]

Estimated Coefficients:

	Estimate	SE
(Intercept)	0	0
x1	1.3288	0.187
x2	1.0147	0.99798
x3	-3.0506	0.85819
x4	7.2055	1.8002
x1^2	-0.0014936	0.00085066
x1:x2	-0.042032	0.0022939
x2^2	-0.076251	0.0063834
x1:x3	-0.023493	0.003279
x2:x3	0.067564	0.018868
x3^2	0.03859	0.012528
x1:x4	-0.00023224	0.00079031
x2:x4	-0.020455	0.0045169
x3:x4	-0.053773	0.017357
x4^2	-0.068989	0.014125
x1^3	1.4464e-05	2.63e-06
x1^2:x2	2.7682e-05	6.296e-06
x1:x2^2	0.00042493	1.2593e-05
x2^3	0.00058269	3.2156e-05
x1^2:x3	2.5157e-06	7.1785e-06
x1:x2:x3	5.9566e-05	2.1303e-05
x2^2:x3	-0.00012398	5.9045e-05
x1:x3^2	7.9591e-05	1.5641e-05
x2:x3^2	-0.00027695	9.2353e-05
x3^3	-6.0765e-05	4.0766e-05
x1^2:x4	-2.1732e-06	1.9806e-06
x1:x2:x4	2.0445e-05	5.0644e-06
x2^2:x4	0.00010345	1.2958e-05
x1:x3:x4	4.2426e-05	6.2132e-06
x2:x3:x4	-1.2573e-05	3.4543e-05
x3^2:x4	-0.00023082	2.3539e-05
x1:x4^2	-2.0949e-05	3.3762e-06
x2:x4^2	0.0001004	2.2092e-05
x3:x4^2	0.00073776	0.00014311
x4^3	-1.7654e-05	5.2543e-06

Number of observations: 1125, Error degrees of freedom: 1091
Root Mean Squared Error: 1.39
R-squared: 0.991, Adjusted R-Squared 0.991
F-statistic vs. constant model: 3.64e+03, p-value = 0

Figure 4.3 Statistical results of Regression over one of the folds

Table 4.1: Regression Coefficients over the six folds and average for Counterflow Tower

Co-efficient	Cross Validation folds						Average
	1	2	3	4	5	6	
ß1	0.000000	0.000000	0.000000	0.000000	0.000000	0.000000	**0.000000**
ß2	1.328789	1.424746	1.350502	1.195628	1.155017	1.166524	**1.270201**
ß3	1.014656	0.813050	0.667990	0.222533	1.233791	0.200159	**0.692030**
ß4	-3.050589	-3.378980	-3.359064	-2.900809	-2.924391	-3.304418	**-3.153042**
ß5	7.205482	7.834125	7.609858	7.003322	6.930440	7.959816	**7.423841**
ß6	-0.001494	-0.002396	-0.001200	-0.001512	-0.001611	-0.001711	**-0.001654**
ß7	-0.042032	-0.044831	-0.040212	-0.038066	-0.039175	-0.039863	**-0.040697**
ß8	-0.076251	-0.074129	-0.073371	-0.068565	-0.074885	-0.068633	**-0.072639**
ß9	-0.023493	-0.024086	-0.024310	-0.021647	-0.020687	-0.021413	**-0.022606**
ß10	0.067564	0.070630	0.071724	0.076425	0.059981	0.077717	**0.070674**
ß11	0.038590	0.043148	0.044409	0.037384	0.037246	0.042974	**0.040625**
ß12	-0.000232	-0.000290	-0.000408	0.000025	-0.000031	0.000871	**-0.000011**
ß13	-0.020455	-0.017507	-0.018630	-0.016803	-0.017316	-0.017586	**-0.018050**
ß14	-0.053773	-0.059359	-0.059873	-0.054087	-0.051652	-0.062677	**-0.056904**
ß15	-0.068989	-0.074463	-0.070555	-0.066095	-0.066736	-0.074359	**-0.070199**
ß16	0.000014	0.000016	0.000014	0.000014	0.000014	0.000016	**0.000015**
ß17	0.000028	0.000038	0.000030	0.000029	0.000033	0.000033	**0.000032**
ß18	0.000425	0.000427	0.000408	0.000396	0.000411	0.000417	**0.000414**
ß19	0.000583	0.000585	0.000591	0.000567	0.000582	0.000594	**0.000584**
ß20	0.000003	0.000006	-0.000001	0.000002	0.000003	0.000000	**0.000002**
ß21	0.000060	0.000077	0.000048	0.000036	0.000036	0.000038	**0.000049**
ß22	-0.000124	-0.000144	-0.000149	-0.000168	-0.000129	-0.000199	**-0.000152**
ß23	0.000080	0.000080	0.000086	0.000075	0.000071	0.000077	**0.000078**
ß24	-0.000277	-0.000287	-0.000287	-0.000300	-0.000230	-0.000298	**-0.000280**
ß25	-0.000061	-0.000074	-0.000088	-0.000063	-0.000061	-0.000079	**-0.000071**
ß26	-0.000002	-0.000002	-0.000001	-0.000002	-0.000001	-0.000002	**-0.000002**
ß27	0.000020	0.000020	0.000019	0.000013	0.000015	0.000020	**0.000018**
ß28	0.000103	0.000095	0.000100	0.000094	0.000099	0.000100	**0.000099**
ß29	0.000042	0.000040	0.000042	0.000038	0.000034	0.000030	**0.000038**
ß30	-0.000013	-0.000035	-0.000028	-0.000042	-0.000030	-0.000047	**-0.000032**
ß31	-0.000231	-0.000237	-0.000211	-0.000206	-0.000219	-0.000217	**-0.000220**
ß32	-0.000021	-0.000020	-0.000020	-0.000019	-0.000016	-0.000020	**-0.000019**
ß33	0.000100	0.000097	0.000100	0.000101	0.000093	0.000107	**0.000100**
ß34	0.000738	0.000792	0.000755	0.000706	0.000708	0.000791	**0.000748**
ß35	-0.000018	-0.000018	-0.000018	-0.000017	-0.000015	-0.000018	**-0.000017**

Table 4.2: Regression Coefficients over the six folds and average for Crossflow Tower

Co-efficient	Cross Validation folds						Average
	1	2	3	4	5	6	
ß1	0.000000	0.000000	0.000000	0.000000	0.000000	0.000000	**0.000000**
ß2	0.333130	0.328617	0.385409	0.392525	0.404719	0.308373	**0.358795**
ß3	2.210134	1.772512	1.919607	2.102291	2.385846	1.998358	**2.064791**
ß4	0.381950	0.311665	0.349521	0.291808	0.262469	-0.050123	**0.257882**
ß5	-1.172590	-0.919887	-1.111221	-1.098329	-1.008896	-0.171865	**-0.913798**
ß6	-0.001449	-0.000895	-0.001636	-0.001503	-0.001332	-0.000960	**-0.001296**
ß7	-0.032927	-0.032602	-0.032728	-0.032837	-0.033468	-0.032011	**-0.032762**
ß8	-0.049167	-0.047601	-0.048151	-0.048457	-0.049424	-0.048892	**-0.048616**
ß9	-0.007635	-0.008300	-0.008621	-0.008932	-0.009228	-0.007895	**-0.008435**
ß10	0.033218	0.040565	0.037429	0.034466	0.029120	0.036264	**0.035177**
ß11	-0.002737	-0.001667	-0.002016	-0.000731	-0.000721	0.003387	**-0.000748**
ß12	0.003427	0.003611	0.003631	0.003700	0.003617	0.003524	**0.003585**
ß13	-0.021173	-0.020198	-0.018996	-0.020498	-0.019790	-0.020359	**-0.020169**
ß14	0.007129	0.004590	0.005986	0.005447	0.005580	-0.001991	**0.004457**
ß15	0.010808	0.008723	0.010456	0.010695	0.009329	0.002531	**0.008757**
ß16	0.000011	0.000010	0.000011	0.000011	0.000011	0.000011	**0.000011**
ß17	0.000034	0.000035	0.000035	0.000042	0.000036	0.000031	**0.000036**
ß18	0.000382	0.000379	0.000381	0.000390	0.000386	0.000382	**0.000383**
ß19	0.000349	0.000357	0.000363	0.000358	0.000358	0.000365	**0.000358**
ß20	0.000006	0.000003	0.000007	0.000006	0.000005	0.000001	**0.000005**
ß21	-0.000027	-0.000031	-0.000030	-0.000041	-0.000026	-0.000033	**-0.000031**
ß22	-0.000202	-0.000225	-0.000225	-0.000220	-0.000209	-0.000213	**-0.000215**
ß23	0.000021	0.000027	0.000026	0.000029	0.000030	0.000027	**0.000026**
ß24	-0.000082	-0.000109	-0.000096	-0.000079	-0.000060	-0.000092	**-0.000086**
ß25	0.000009	0.000006	0.000006	-0.000002	0.000001	-0.000009	**0.000002**
ß26	-0.000003	-0.000002	-0.000002	-0.000002	-0.000003	-0.000002	**-0.000002**
ß27	0.000020	0.000021	0.000020	0.000022	0.000020	0.000020	**0.000021**
ß28	0.000101	0.000107	0.000109	0.000105	0.000106	0.000100	**0.000105**
ß29	0.000006	0.000005	0.000004	0.000004	0.000005	0.000003	**0.000004**
ß30	0.000004	-0.000009	-0.000005	-0.000002	0.000005	0.000001	**-0.000001**
ß31	-0.000007	-0.000007	-0.000002	0.000003	-0.000009	-0.000014	**-0.000006**
ß32	-0.000019	-0.000020	-0.000020	-0.000020	-0.000020	-0.000018	**-0.000019**
ß33	0.000087	0.000086	0.000075	0.000084	0.000075	0.000083	**0.000082**
ß34	-0.000052	-0.000031	-0.000049	-0.000052	-0.000037	0.000029	**-0.000032**
ß35	-0.000022	-0.000021	-0.000021	-0.000021	-0.000021	-0.000021	**-0.000021**

Figure 4.4 and 4.5 show the graph of actual values of approach versus the predicted values of approach for the old and new model respectively for a counter flow tower. Figure 4.6 and 4.7 show the graph of actual values of approach versus the predicted values of approach for the old and new model respectively for a counter flow tower. Table 4.3 and Table 4.4 show the comparison between the old model and the new model in terms of average error, maximum error and standard deviation of error for counter flow towers and a cross flow towers respectively. It is clearly seen that the

average error of the new model is lower, maximum error of the new model is lesser as is the standard deviation.

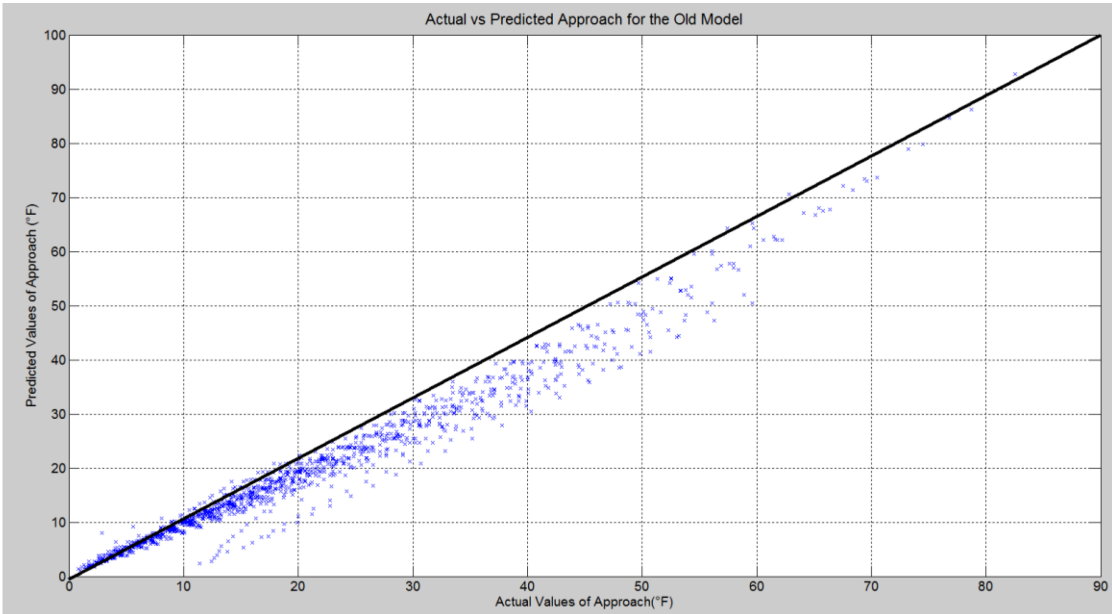


Figure 4.4: Actual Approach vs Predicted Approach of the Old Model for a CF Tower

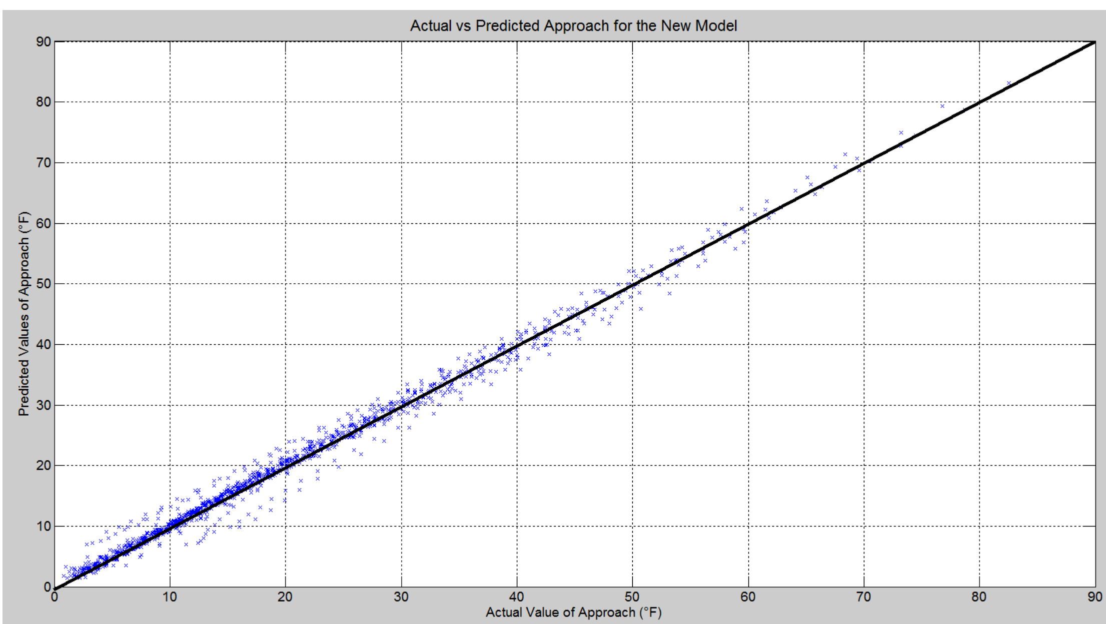


Figure 4.5: Actual Approach vs Predicted Approach of the New Model for a CF Tower

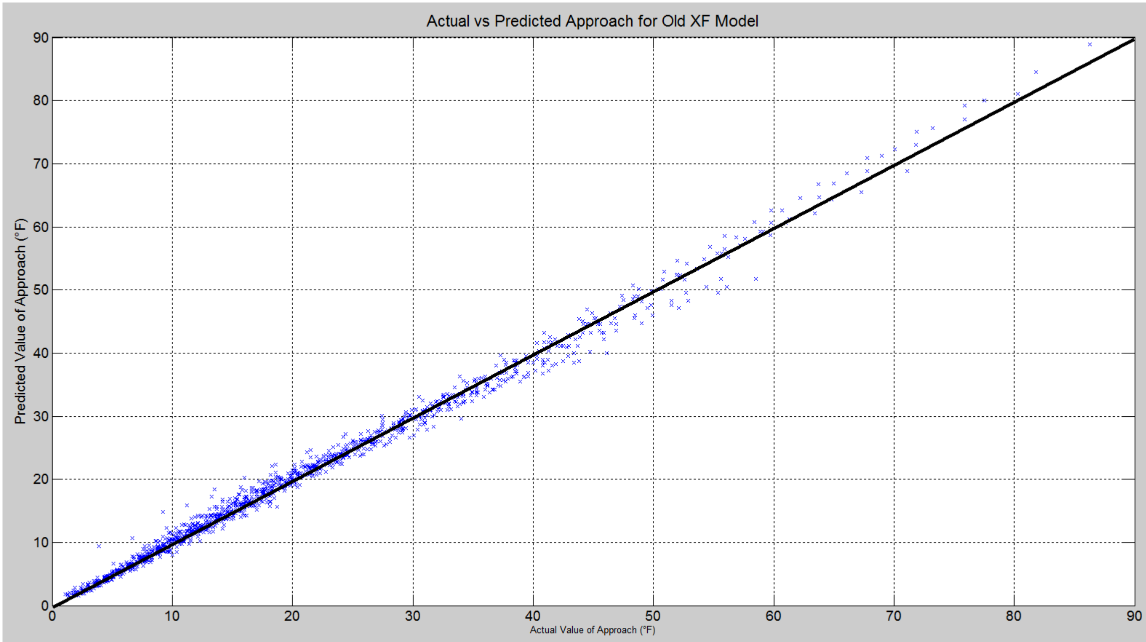


Figure 4.6: Actual Approach vs Predicted Approach of the Old Model for a XF Tower

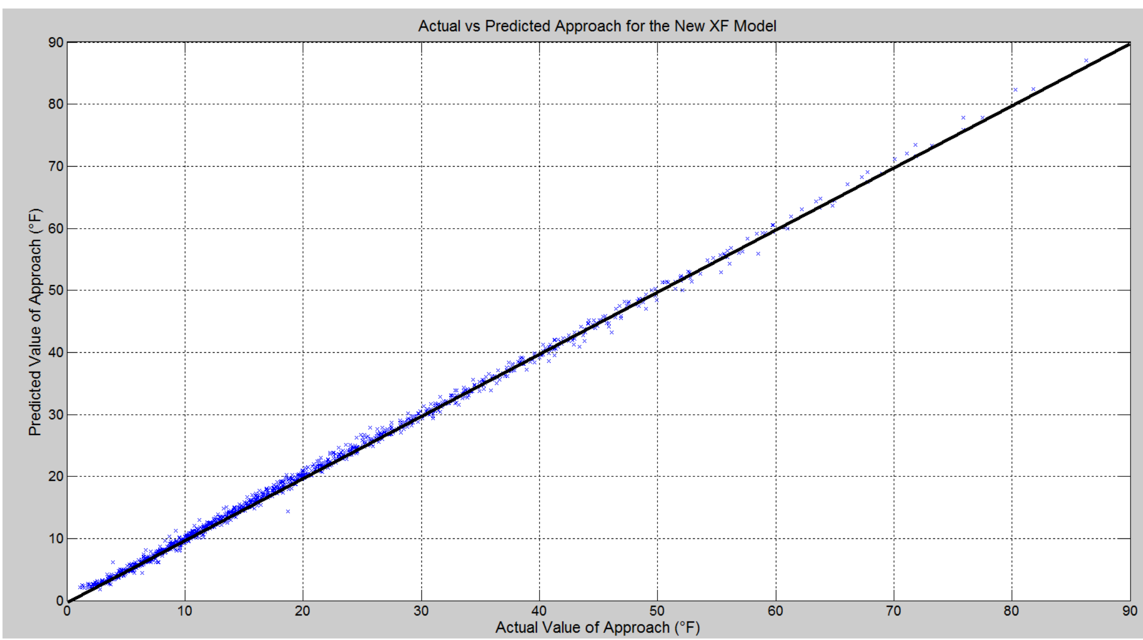


Figure 4.7: Actual Approach vs Predicted Approach of the New Model for a XF Tower

Table 4.3: Comparison of the Old and New Models for a Counter Flow Tower

	Old Model	New Model
Average Error	1.3072	0.0001
Maximum Error	10.1455	6.2159
Standard Deviation	2.8807	1.3656

Table 4.4: Comparison of the Old and New Models for a Cross Flow Tower

	Old Model	New Model
Average Error	0.1050	0.0036
Maximum Error	6.7416	4.3904
Standard Deviation	1.2279	0.6063

Figures 4.8 through 4.13 show the error with variation in wet bulb temperature for a given range for the new model as well as the old model. The improvement in the new model over the old model is evident from the graphs across the different wet bulb temperatures and ranges.

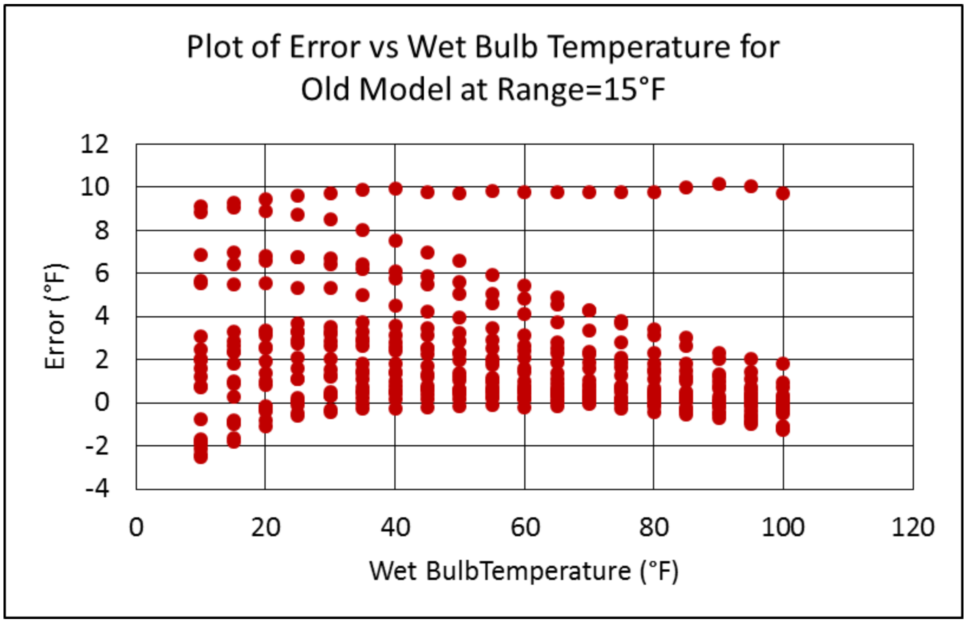


Figure 4.8: Comparison of Error between the old and new model at Range=15°F for a Counter Flow Tower

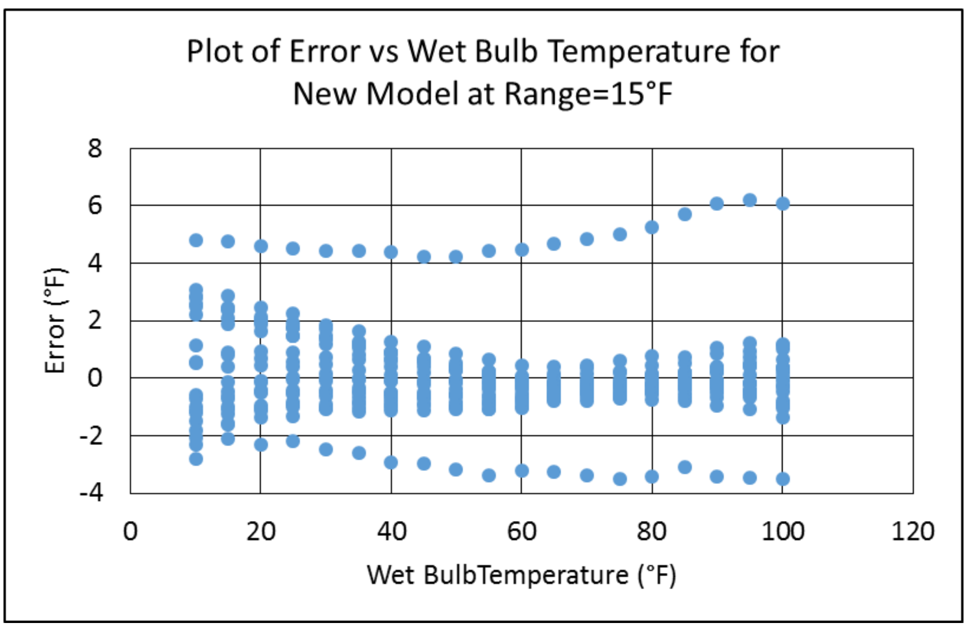


Figure 4.9: Comparison of Error between the old and new model at Range=15°F for a Counter Flow Tower

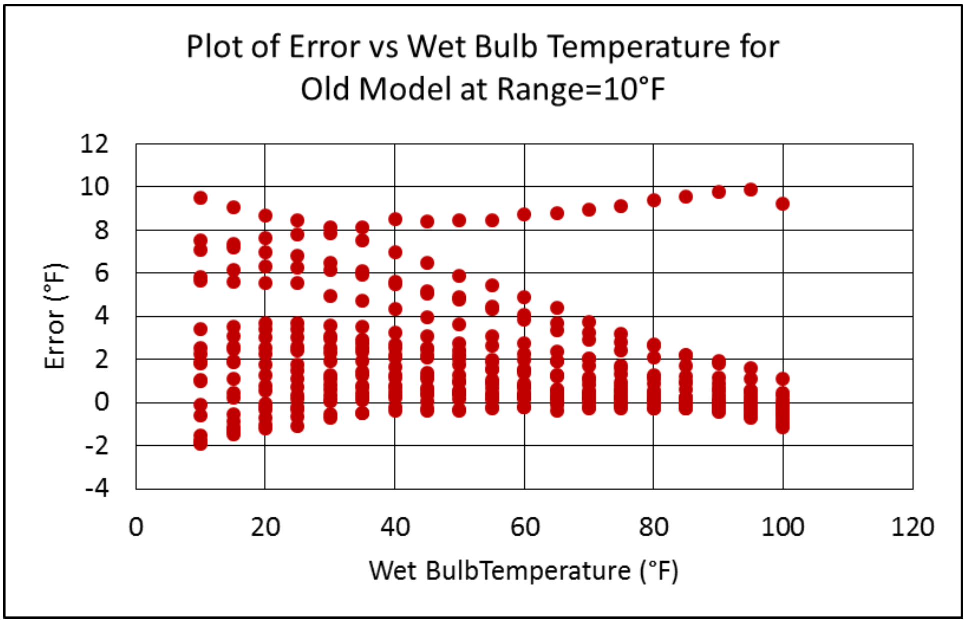


Figure 4.10: Comparison of Error between the old and new model at Range=10°F for a Counter Flow Tower

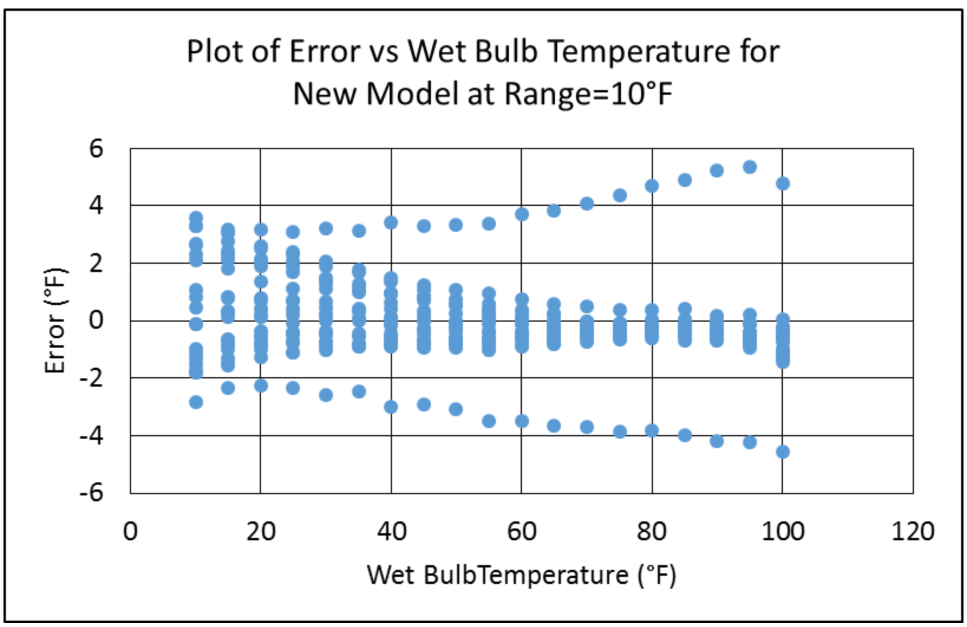


Figure 4.11: Comparison of Error between the old and new model at Range=10°F for a Counter Flow Tower

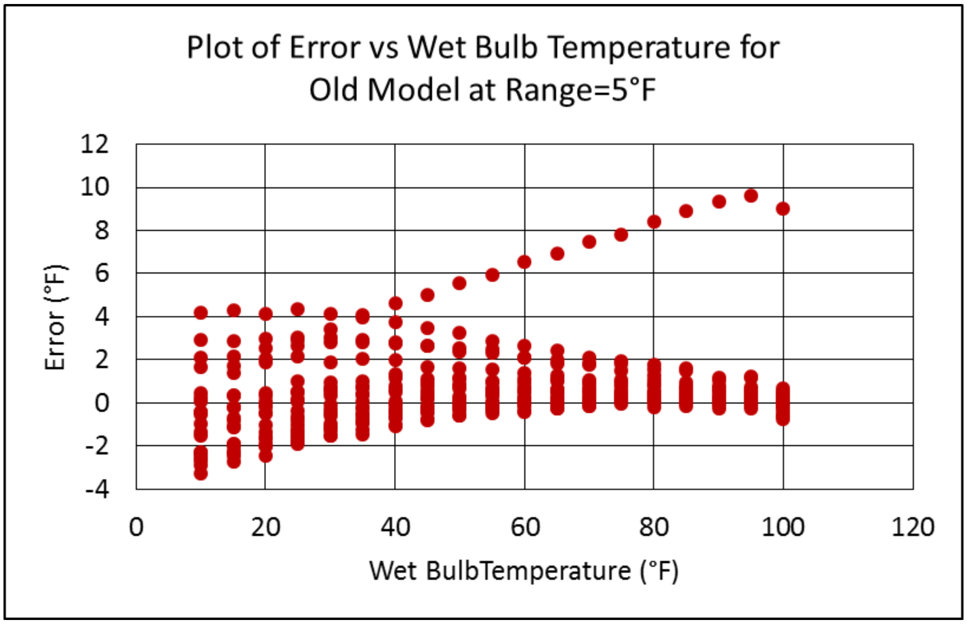


Figure 4.12: Comparison of Error between the old and new model at Range=5°F for a Counter Flow Tower

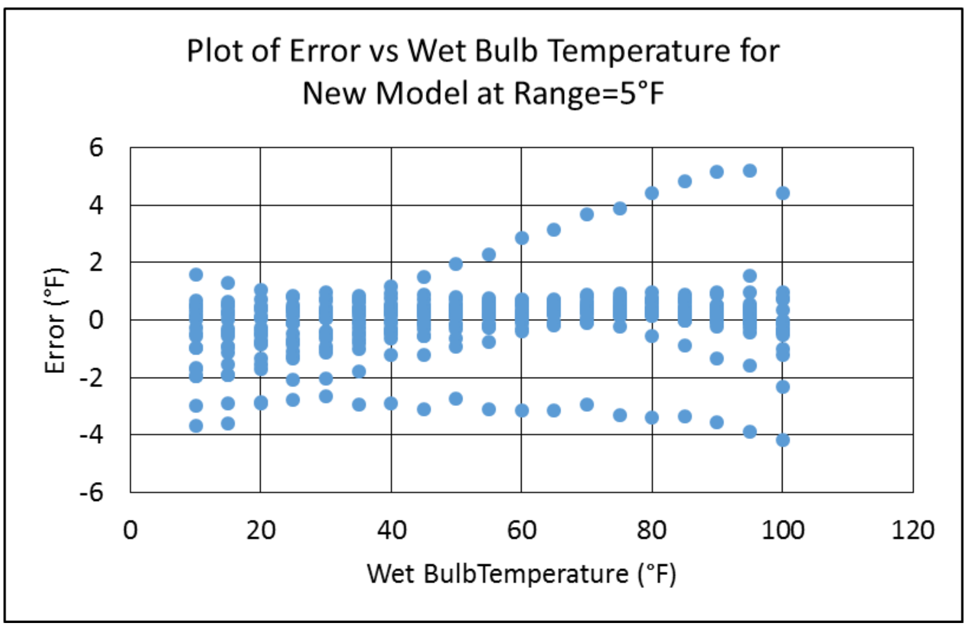


Figure 4.13: Comparison of Error between the old and new model at Range=5°F for a Counter Flow Tower

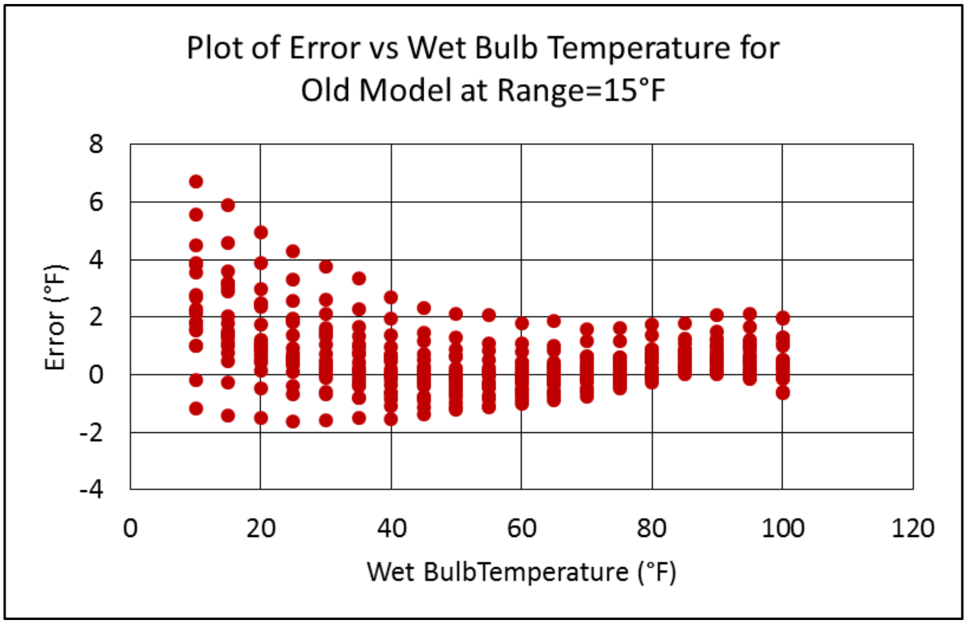


Figure 4.14: Comparison of Error between the old and new model at Range=15°F for a Cross Flow Tower

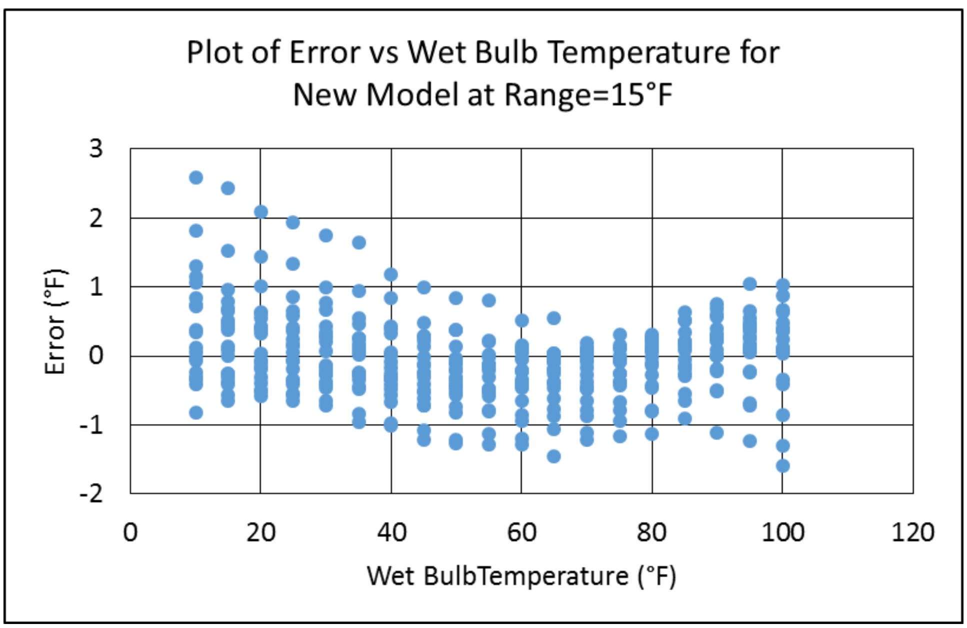


Figure 4.15: Comparison of Error between the old and new model at Range=15°F for a Cross Flow Tower

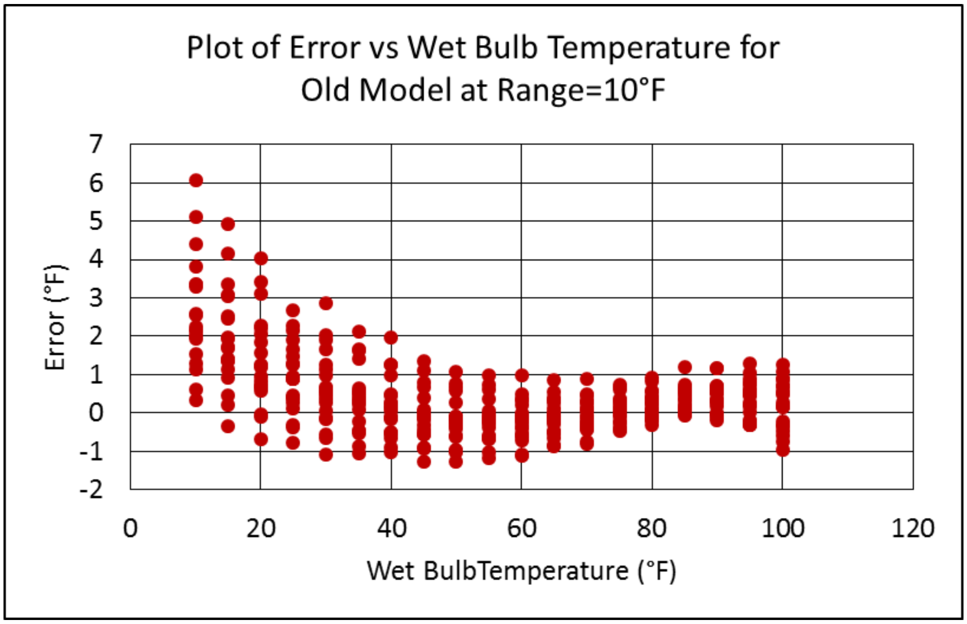


Figure 4.16: Comparison of Error between the old and new model at Range=10°F for a Cross Flow Tower

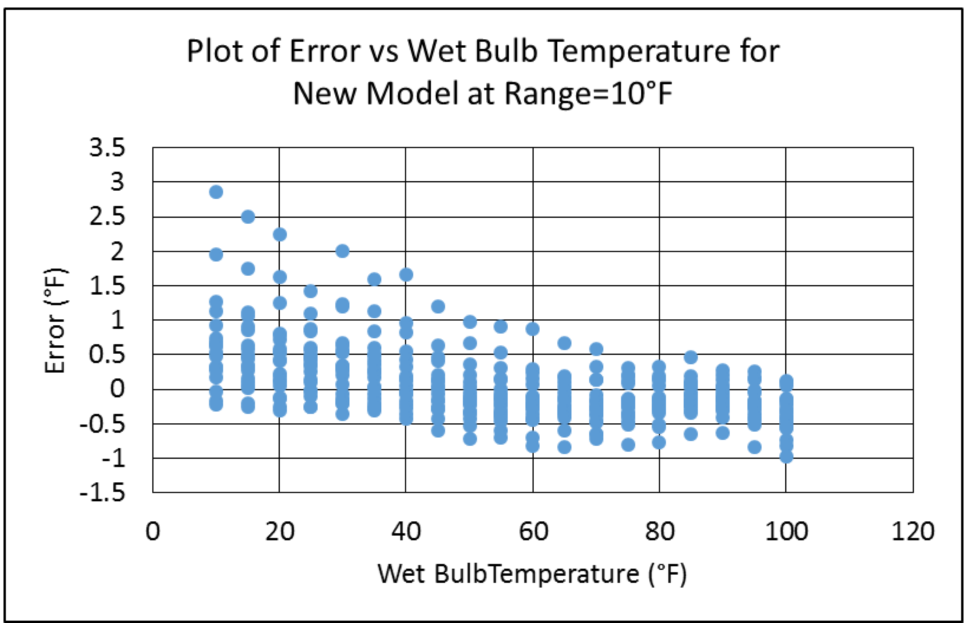


Figure 4.17: Comparison of Error between the old and new model at Range=10°F for a Cross Flow Tower

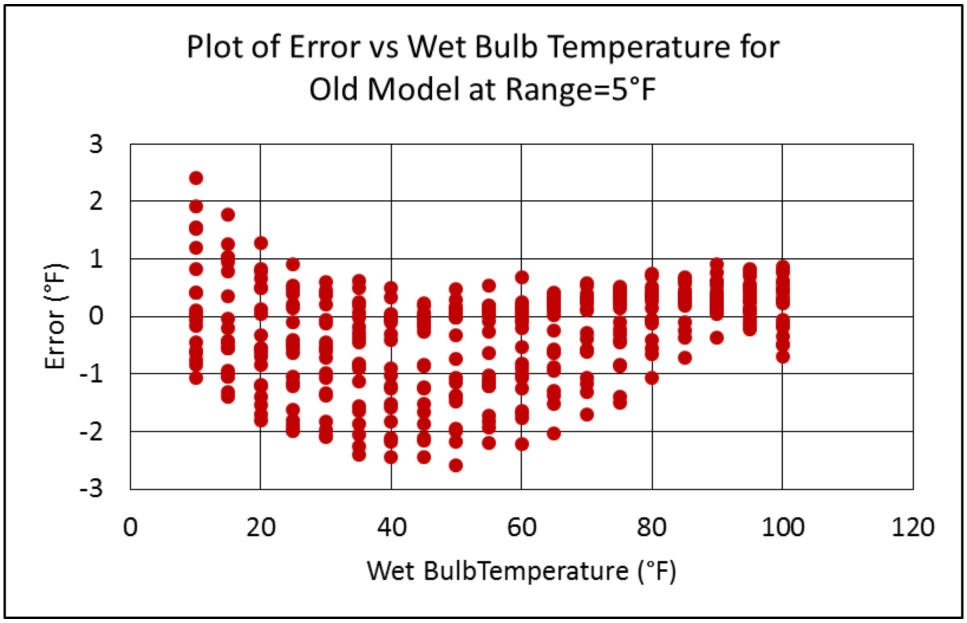


Figure 4.18: Comparison of Error between the old and new model at Range=5°F for a Cross Flow Tower

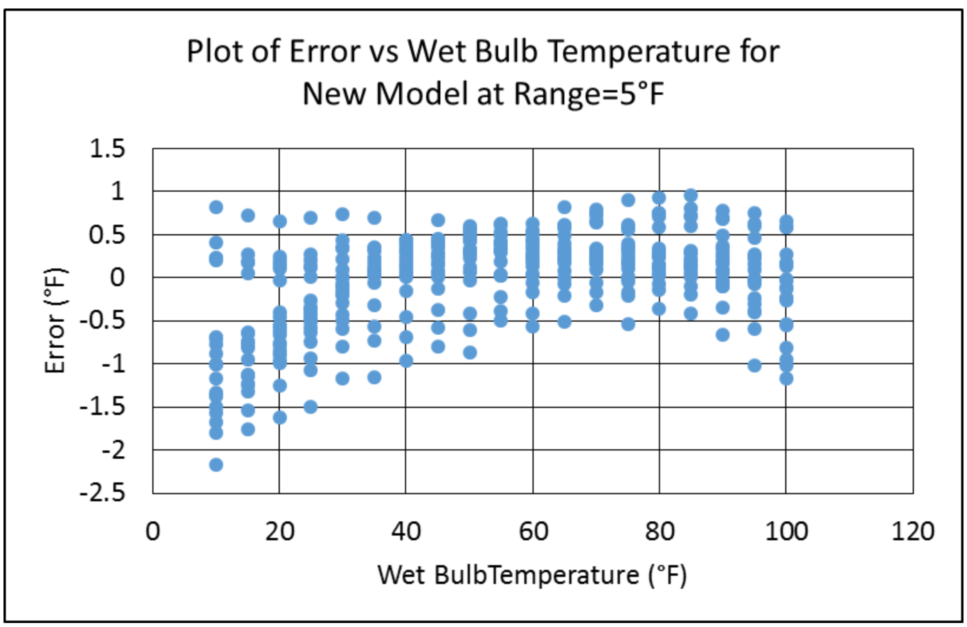


Figure 4.19: Comparison of Error between the old and new model at Range=5°F for a Cross Flow Tower

4.3 New Model Creation through Artificial Neural Networks

To check if ANNs are a feasible method to create a prediction model for cooling towers, counter flow cooling tower data will be used to create a new model and the results compared. The neural network toolbox in MATLAB was used for creating ANNs. A neural network fitting tool was used. In a way similar to cross validation, the data is split into 3 subsets of training data (70%), validation data (15%) and testing data (15%). The ANN was trained with the training data during which the network weights and functions are adjusted. Next the ANN is validated with the validation data to measure network generalization and to halt training when generalization stops improving. The testing data has no effect on training and is used to provide an independent measure of network performance during and after training. The ANN is created with one hidden layer of neurons. If there is no hidden layer of neurons, only linear separable functions can be represented using that ANN. Having one layer of hidden neurons can approximate most continuous functions. Adding additional hidden layers may increase accuracy in some cases, but will require greater computational effort and may also lead to over fitting. The ANN was trained using the Levenberg-Marquardt backpropagation algorithm. The results obtained for a neural network model for counter flow tower with one hidden layer are shown below. Figure 4.20 shows the error histogram, Figure 4.21 shows the performance of the neural network and Figure 4.22 gives the regression plots.

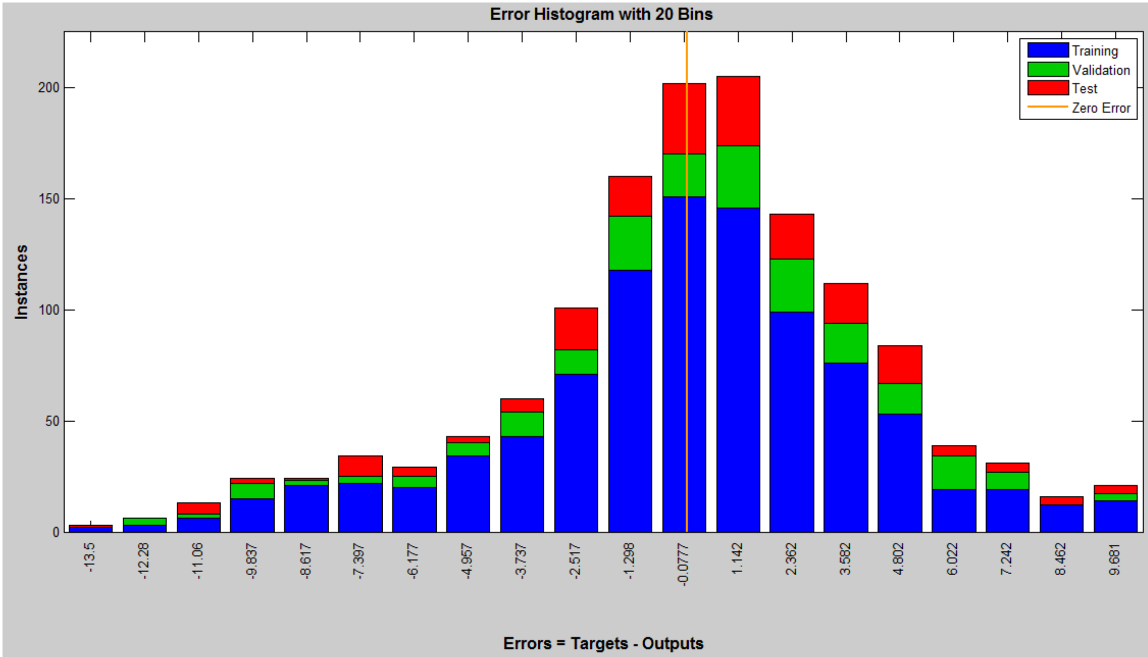


Figure 4.20: Error Histogram for Neural Network Fitting with One Hidden layer of Neurons

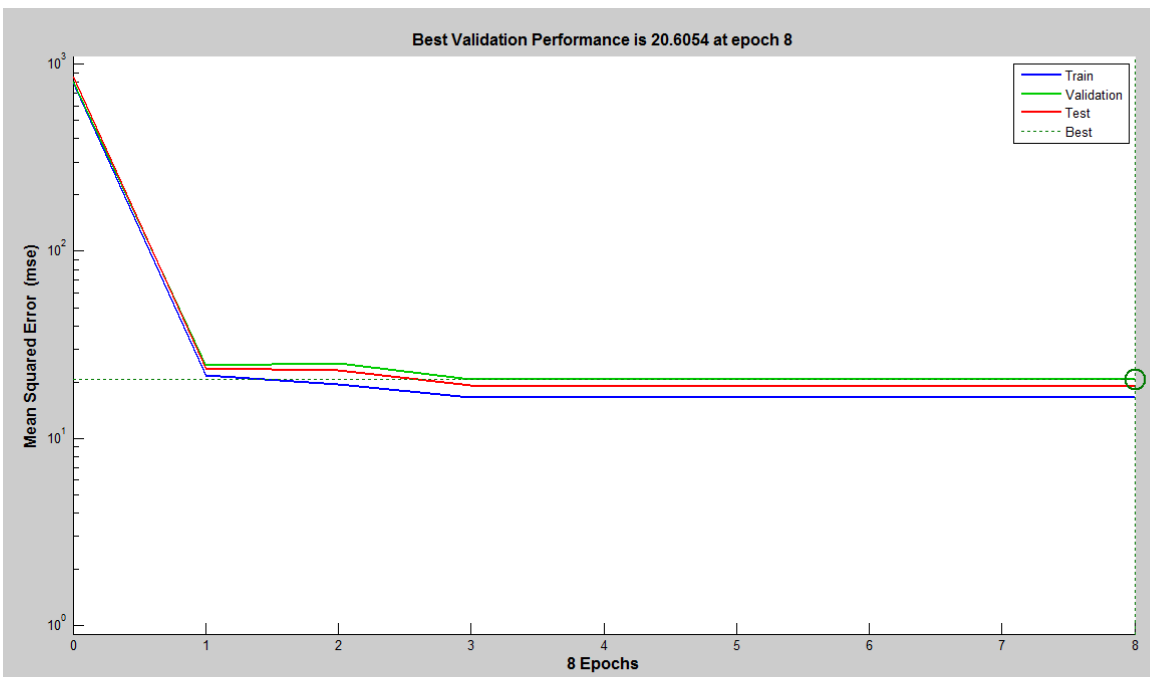


Figure 4.21: Performance for Neural Network Fitting with One Hidden layer of Neurons

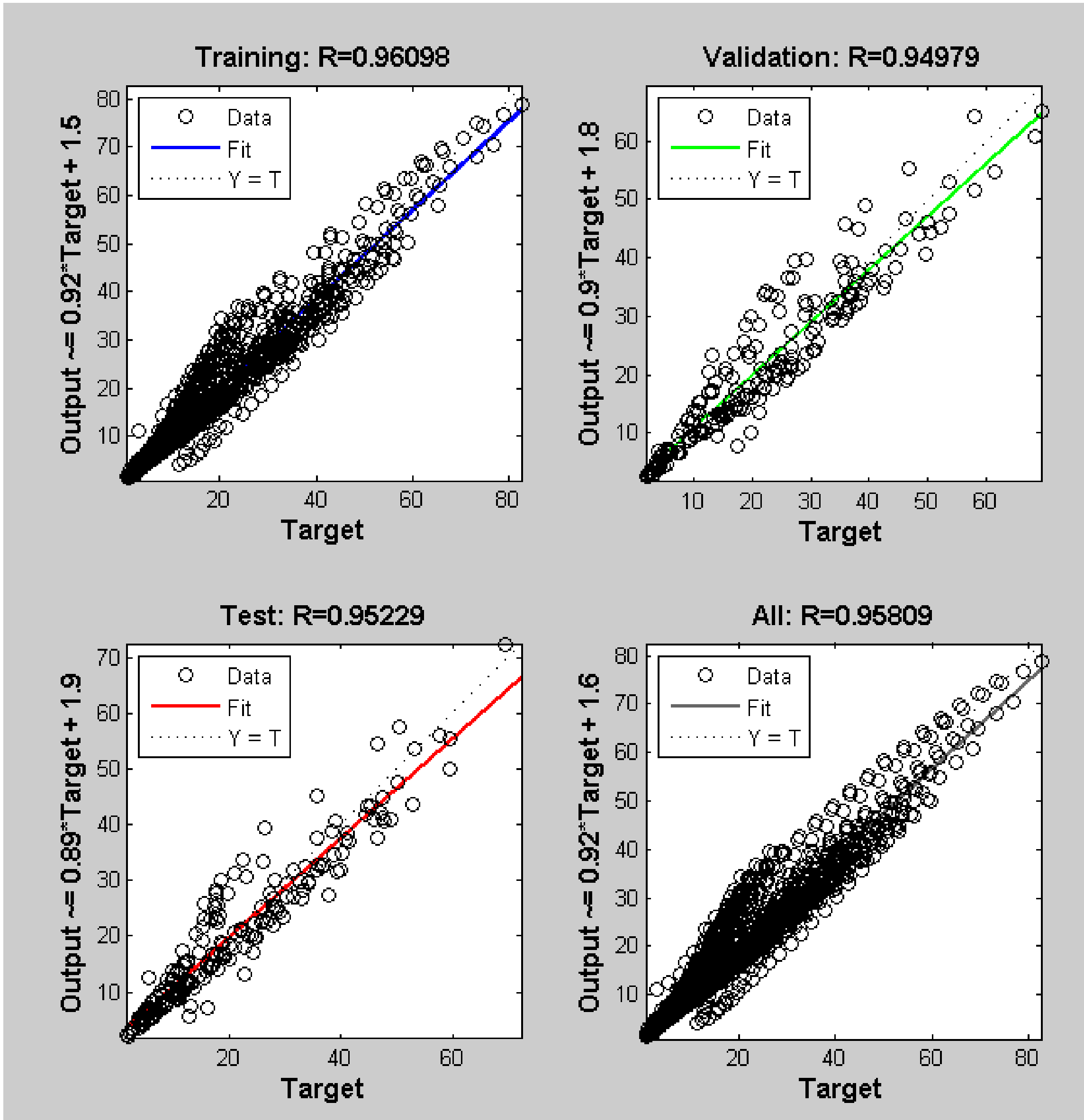


Figure 4.22: Regression Plots for Neural Network Fitting with One Hidden layer of Neurons

The ANN is created again, but now with two layers. The results obtained are as follows

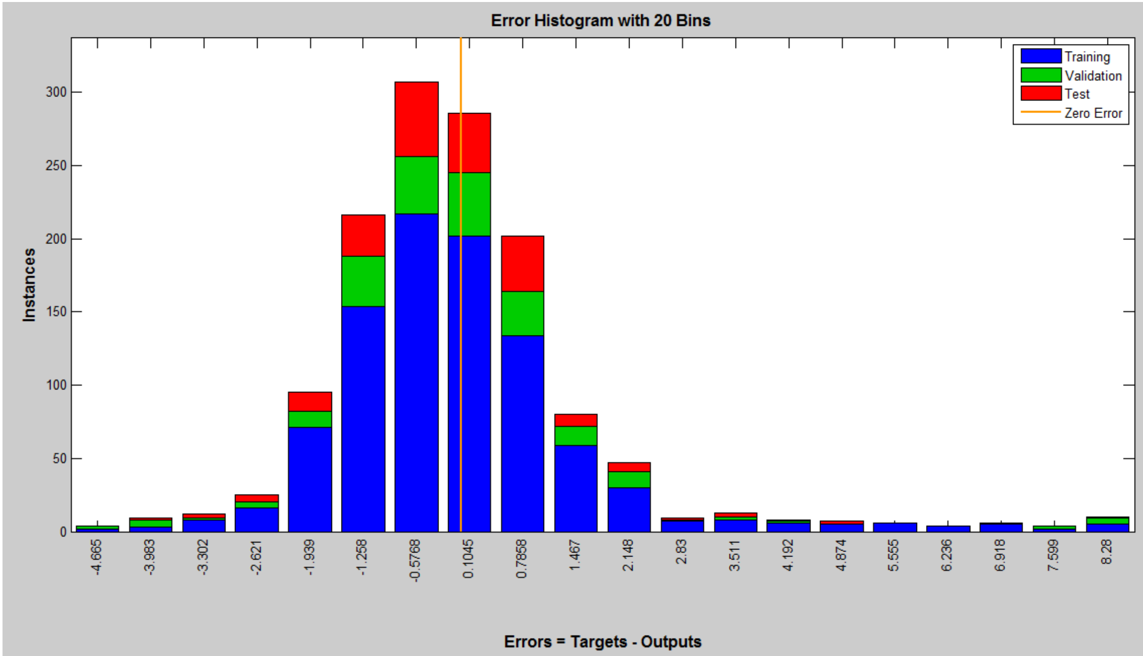


Figure 4.23: Error Histogram for Neural Network Fitting with Two Hidden layers of Neurons

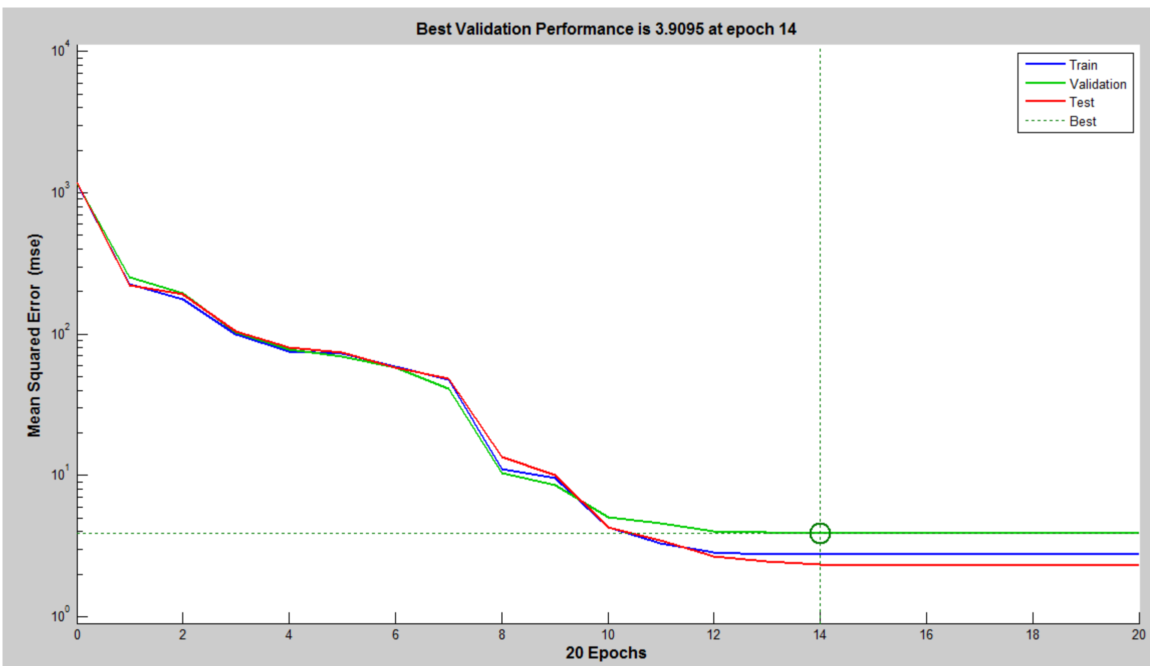


Figure 4.24: Performance for Neural Network Fitting with Two Hidden layers of Neurons

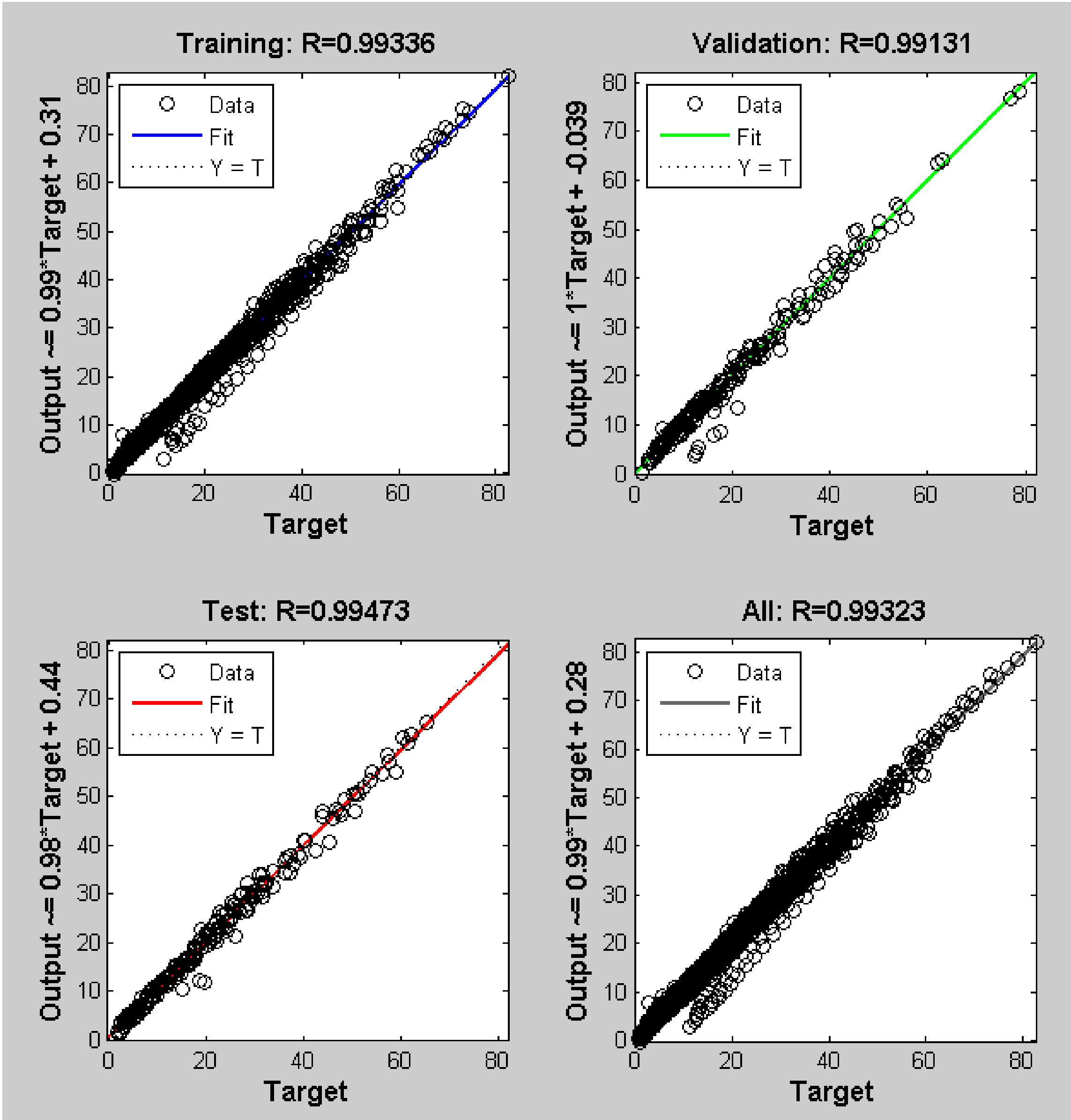


Figure 4.25: Regression Plots for Neural Network Fitting with Two Hidden layers of Neurons

It is seen that the prediction error was decreased by increasing the hidden layer of neurons to two and a better prediction is achieved. According to MATLAB, an epoch is a measure of the number of times all of the training vectors are used once to update the weights. For batch training all of the training samples pass through the learning algorithm simultaneously in one epoch before weights are updated. Thus by increasing the number of layers of hidden neurons, the computational effort increased. It was seen that increasing the number of neurons further did not greatly increase prediction capability.

Upon comparing these results to the results obtained through polynomial regression, we see that the results are quite similar. For e.g. Figure 4.24 gives the mean squared error as 3.9095 which makes the root mean squared value as 1.9772. Comparing this to the value of 1.3656 as obtained through polynomial regression we see that both are similar. Now amongst PR and ANN, one needs to be chosen to move ahead.

4.4 Comparison of Techniques

Jin et.al. [27] suggest that the following metrics be considered when comparing metamodeling techniques.

- Accuracy: the capability of predicting the system response over the design space of interest.
- Robustness: the capability of achieving good accuracy for different problems. This metric indicates whether a modelling technique is highly problem-dependent.
- Efficiency: the computational effort required for constructing the metamodel and for predicting the response for a set of new points by metamodels.
- Transparency: the capability of providing the information concerning contributions of different variables and interactions among variables.
- Conceptual simplicity: ease of implementation. Simple methods should require less user input and be easily adapted to each problem.

The accuracy of the two techniques was compared and seen that the PR method gave slightly better results but on the whole results were comparable. More detailed and different approaches to building neural networks may yield better results through ANNs.

As seen in the literature review earlier PR technique has been used before (Benton et.al and DOE2 engine) to create surrogate models for this particular situation to good effect and is thus a good choice for moving forward. It is seen that ANN is also a suitable method to create a prediction model for cooling tower performance based on results obtained.

PR is a more widely used technique compared to ANN and also requires lesser computational effort [21]. Further PR is much simpler and quicker to implement than ANN.

The CWSAT currently uses a model created through third order polynomial regression. Utilizing a similar new model will allow for a much simpler improvement of CWSAT as well. While the ANN technique allows another method of creating a model and has the advantage of being able to train the neural network with newer data as available, implementing a neural network model outside (e.g. in CWSAT) of software built specifically for neural networks would require additional effort. Thus the PR technique will be employed to create a new model at this stage for the sake of simplicity and possibility of easier implementation in CWSAT.

CHAPTER 5

WATER SIDE ECONOMIZER / FREE COOLING

This chapter explains the reduction in capacity of cooling towers when operated in free cooling mode. The new and old models are used to simulate tower performance over a year and differences are noted. Also the economic benefits of having larger cooling tower capacity to accomplish free cooling are analyzed.

5.1 Reduction in Cooling Tower Capacity at Low Temperatures

It seems counter intuitive that the cooling tower capacity drops when the tower is operated in free cooling mode at low temperatures. The reason this occurs goes back to the point addressed in the introduction that the cooling tower works primarily on the principle of evaporative cooling. As the temperature drops, the amount of water that the air can hold also drops significantly. Since the tower can no longer reject as much heat to atmosphere through the evaporation of water, the tower capacity drops. It is also to be noted that there is some heat transfer to atmosphere through sensible heat transfer as a result of the temperature difference. During free cooling conditions the sensible heat transfer is not large because the water entering the cooling tower is quite close to the atmospheric dry bulb temperature. This is the reason for reduced cooling tower capacity when operating the tower in free cooling mode.

This can also be illustrated with some numbers. Consider the standard conditions that a tower is rated at; 95°F entering water temperature (EWT), 85°F leaving water temperature (LWT) and a wet bulb temperature of 78°F. For the location of Boston, TMY2 (Typical Meteorological Year) data was obtained and the average relative humidity (RH) for the summer months from May to October was found to be 68.5%. At a wet bulb temperature of 78°F and a RH of 68.5% we find the humidity ratio to be 0.01875 lb/lb (the units are pounds moisture per pound dry air). Assuming that evaporative cooling is taking place and the air leaves the cooling tower in a saturated condition i.e. 100%RH (one of the assumptions of Merkel theory) we have the humidity ratio of air leaving the tower as 0.02078 lb/lb. The amount of moisture increase is $2.03*10^{-3}$.

It is seen that the winter months from November to April have an average RH of 63%. In most cases the chilled water set point temperature (CHWST) for the chiller is 45°F. If we have to substitute the chiller with the tower, the tower should be able to produce water at 45°F. For the ideal case we can assume that the water reaches the wet bulb temperature and we also need to account for the temperature drop across a heat exchanger that may be used.

$$FCT_{wb} = CHWST - HEX$$

If we assume a temperature drop of 4°F across the heat exchanger, then the wet bulb temperature that is required for us to be able to perform free cooling is 41°F. At a wet bulb temperature of 41°F and a RH of 63% we find the humidity ratio to be 0.00418 lb/lb. Assuming that evaporative cooling is taking place and the air leaves the cooling tower in a saturated condition we have the humidity ratio of air leaving the tower as 0.0054 lb/lb. The amount of moisture increase is $1.22*10^{-3}$. This is almost half of that in the earlier case. Table 5.1 shows the conditions of air entering and leaving the tower for both summer and winter to present the number in a clear and concise manner. Figure 5.1 shows the conditions of air entering and leaving the tower in summer and winter on a psychrometric chart to better illustrate the decreased amount of water vapor the air can hold in winter. The diagonal line with arrow represents the process the air is going through. The other sides of the triangle are just to bring out visually the size difference between summer and winter.

Table 5.1: Conditions of Air entering and leaving the Cooling Tower

	Summer		Winter	
	Entering	Leaving	Entering	Leaving
T_{WB} (°F)	78	78	41	41
RH	68.5	100	63	100
T_{DB} (°F)	86.6	78	46.4	41
Humidity Ratio (lb/lb)	0.01875	0.02078	0.00418	0.0054

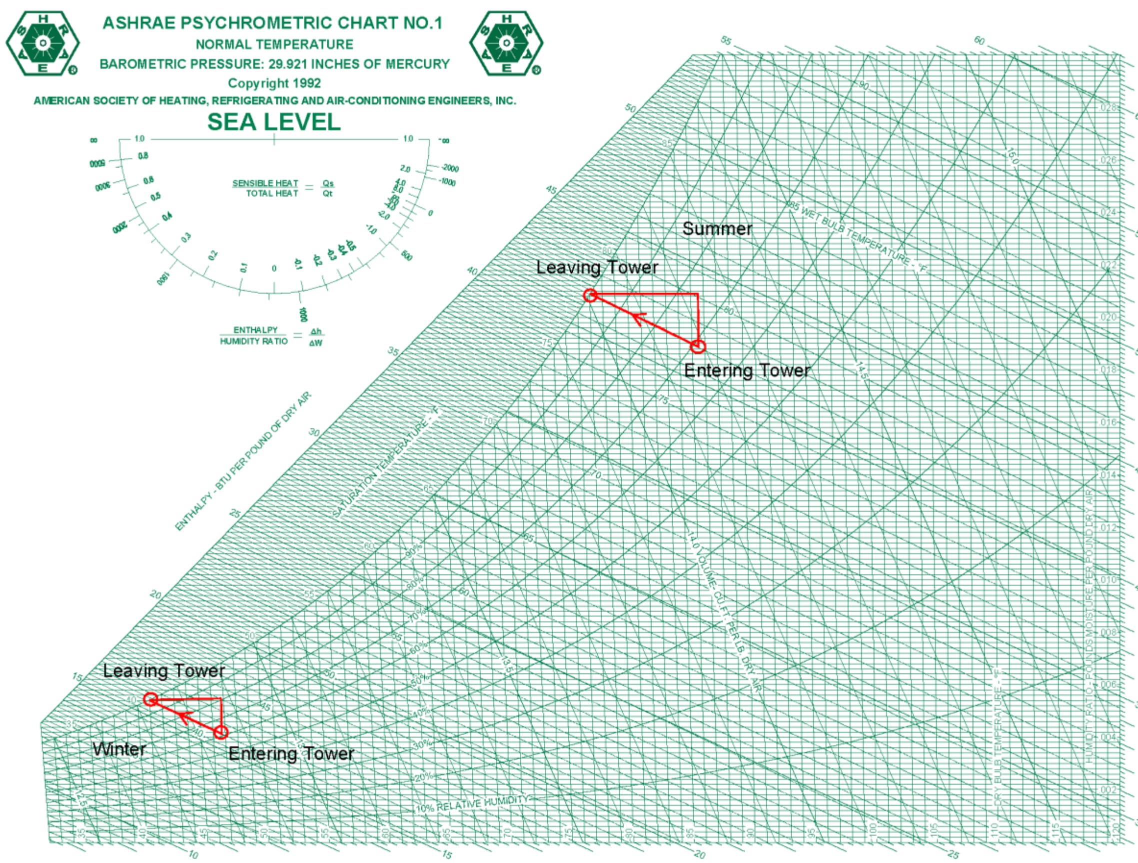


Figure 5.1: Conditions of Air Entering and Leaving the tower on a Psychrometric Chart

Since the cooling tower primarily relies on evaporative cooling, the capacity of the tower drops to about half as corroborated in [3], [5] and [19]. This drop in capacity is when the wet bulb temperature is close to the required temperature of water exiting the cooling tower. If the wet bulb temperature drops much lower than the temperature of cold water exiting the tower, then the capacity once again increases. Figure 5.2 shows the change in tower capacity with wet bulb temperature for different temperatures of cold water exiting the tower (CWT) for a 300 ton tower.

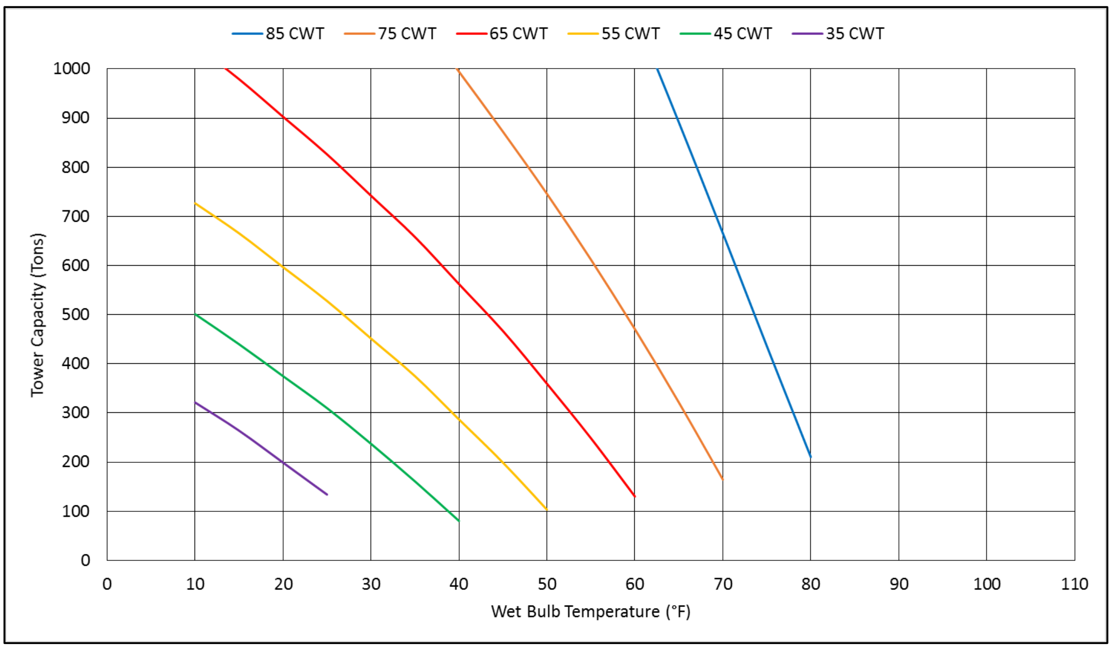


Figure 5.2: Variation in Tower Capacity with Wet Bulb Temperature

Referring to the Figure 5.2 it is seen that for a CWT of 45°F, the tower capacity is about 150 tons, half of the rated tower capacity, at a wet bulb temperature 35°F. Further at the same CWT of 45°F, the tower is capable of taking the rated load only at temperatures of 25°F and lower. This reduction in capacity needs to be taken into account when determining if the tower can operate in free cooling mode based on the current load on the tower.

5.2 Comparison of Prediction Capability of the different Models

To compare the old and new models, code is written in MATLAB and given in Appendix B. This code simulates the tower performance round the year. The same code is utilized to simulate tower performance for both the old and new models. A comparison is made with the results obtained from CWSAT to ascertain if the model is appropriately implemented (Note: The old model and the CWSAT utilize the same regression model for a crossflow tower. The reason for choosing to compare the old model outside of CWSAT is to determine if CWSAT is working appropriately). The tower is assumed to be a cross flow tower since that is the regression model that is used in the CWSAT program.

The chiller is assumed to be loaded according to the Air-Conditioning & Refrigeration Institute (ARI) schedule which imposes a corresponding load on the tower. Table 5.2 shows the number of hours the chiller is at a particular load annually according to the ARI schedule. The input parameters are given in Table 5.3. CWSAT is also used to simulate tower performance for the same conditions. A 300 ton tower is needed for a 250 ton chiller because the tower needs to reject an additional amount of heat apart from the load on the chiller which is the heat of compression of the chiller. For the current case, a heat of compression of 20% of the current load is utilized which yields a tower size requirement of 300 tons. From looking at specifications of 300 ton cooling towers, it is determined that they have a fan of 25 hp.

Table 5.2: ARI Loading Schedule

Load	Hours	Percent Annual Hours
0%	0	0%
10%	0	0%
20%	95	1%
30%	437	5%
40%	1138	13%
50%	2016	23%
60%	2273	26%
70%	1670	19%
80%	790	9%
90%	258	3%
100%	83	1%
Total	**8760**	**100%**

Table 5.3: Parameters considered to simulate tower performance

Parameter	Value
Location	Boston, MA
Chilled water supply temperature	45°F
Temperature of water exiting tower	75°F, Constant
Chiller capacity	250 tons, helical rotary
Full load efficiency	0.7 kW/ton
Tower capacity	300 tons
Fan motor	25 hp
Heat Exchanger Approach temperature	2°F
Water flow rate to tower	3 gpm, const

Table 5.4 shows a comparison of the predicted fan energy for different situations and for different models when the chiller is subjected to an ARI load. It is seen that

CWSAT consistently predicts higher tower energy usage, even with the addition of a VSD to the fan. The old and new model predict similar values which is to be expected based on the results seen in the previous chapter.

Table 5.4: Comparison of Tower Energy Prediction across models and CWSAT for a Tower with ARI Loading schedule

		CWSAT Predicted tower fan energy use (kWh)	Old Model Predicted Tower fan Energy Use (kWh)	New Model predicted tower fan energy use (kWh)
No Free Cooling	Single Speed Fan	38,211	36,394	32,460
	Variable Speed Fan	24,677	19,104	17,313
With Free Cooling	Single Speed Fan	77,778	63,973	57,864
	Variable Speed Fan	63,653	33,385	28,365

Table 5.5 and Table 5.6 give the predicted energy savings when going from a cooling tower without a VSD on a fan and without free cooling to a tower with both. Table 5.5 gives the predictions for a tower subjected to ARI load and the Table 5.6 gives the predictions for a tower subjected to a process load. The process load is one where the chiller is subjected to a constant design load (rated chiller capacity) for all of the 8,760 hours in a year. The predicted chiller energy savings, the predicted tower energy usage and the total energy savings coming from implementing both VSD and free cooling operation are detailed.

From Table 5.5 it is seen that for the tower subjected to ARI load, the predicted chiller energy savings are similar for the different models. The tower energy predicted by CWSAT seems to be high. Upon further investigation of the results of CWSAT, it is seen that it determines the fan power to be a 100% for large parts when free cooling is taking place. Also from Table 5.6 it is seen that CWSAT predicts a lot more chiller energy savings as compared to the old and new models.

Table 5.5: Comparison of energy savings predicted thorough the models and CWSAT for a Tower with ARI Loading Schedule

	Tower Energy Usage without VFD and free cooling (kWh)	Chiller Energy Usage without free cooling (kWh)	Tower Energy Usage with VFD and free cooling (kWh)	Chiller Energy Usage with VFD and free cooling (kWh)	Tower energy savings (kWh)	Chiller energy savings (kWh)	Total energy savings (kWh)
CWSAT	38,211	762,796	63,653	569,392	-25,442	193,404	167,962
Old model	36,394	762,796	33,385	564,427	3,009	198,369	201,378
New model	32,460	762,796	28,365	573,640	4,095	189,156	193,251

Table 5.6: Comparison of energy savings predicted thorough the models and CWSAT for a Tower with Process Loading

	Tower Energy Usage without VFD and free cooling (kWh)	Chiller Energy Usage without free cooling (kWh)	Tower Energy Usage with VFD and free cooling (kWh)	Chiller Energy Usage with VFD and free cooling (kWh)	Tower energy savings (kWh)	Chiller energy savings (kWh)	Total energy savings (kWh)
CWSAT	65,575	1,244,314	92,265	924,571	-26,690	319,743	293,053
Old model	62,467	1,244,314	51,812	1,050,139	10,655	194,175	204,830
New model	65,575	1,244,314	51,335	1,077,695	14,240	166,619	180,859

Table 5.7 shows the predicted fan energy, predicted chiller energy and number of free cooling hours. It is seen that CWSAT predicts the same free cooling hours despite the change in loading conditions. This is because CWSAT doesn't take into account the fact that the tower capacity is reduced during free cooling.

Table 5.7: Comparison of free cooling hours predicted thorough the models and CWSAT

	Predicted Fan Energy (kWh)			Number of Annual Free Cooling Hours			Predicted Chiller Savings (kWh)		
	CWSAT	Old Model	New Model	CWSAT	Old Model	New Model	CWSAT	Old Model	New Model
ARI Load	63,653	33,385	28,365	2,251	2,395	2,297	193,404	198,369	189,156
Process Load	92,265	51,812	51,335	2,251	1,367	1,173	319,743	194,175	166,619

It is seen that considerable energy savings are possible through the use of a VSD and operating the tower in economizer mode. Assuming a marginal cost of electricity at $0.08/kWh, the results from the previous tables can be converted to cost savings as shown in Table 5.8.

Table 5.8: Predicted Cost Savings associated with VSD and Free Cooling

	Predicted Total Cost Savings		
	CWSAT	Old Model	New Model
ARI Load	$13,437	$16,110	$15,460
Process Load	$23,444	$16,386	$14,469

Figures 5.3 to 5.5 show the variation in wet bulb temperature, percent fan power, free cooling condition and the load on the chiller as predicted by CWSAT, old model and new model respectively for a few hours in winter. Referring to Figure 5.3 for CWSAT prediction, we see that the check for free cooling is completely based on temperature. If the temperature is below a certain determined number, (that is constant based on the required cooling water temperature and the drop across the heat exchanger) only then is the tower assumed to be able to perform free cooling. Further, it is seen that as the load on the chiller changes and the wet bulb temperature changes, there is almost no change in the predicted fan power which is close to maximum (a value of greater than 100% for fan power is seen because the VFD on the fan adds some losses). It can be seen between the hours 27 and 39 that the load is continuously increasing while the wet bulb temperature is fairly constant. CWSAT predicts maximum fan power usage at hour 27. If this is true, increasing the load would mean that the tower would not be able to provide the required cooling even at 100% fan power but CWSAT assumes that this is the case anyway which

would be incorrect. Conversely if the fan power prediction of 100% at hour 39 is correct, then the previous hours at lower load would be expected to require lesser fan power. Thus there clearly is some error in prediction by CWSAT. Figure 5.4 and Figure 5.5 are very similar and show the applicability of free cooling with change in load and wet bulb temperature. The fan power is also observed to change according to variations in the in load and wet bulb temperature. The new model predicts fewer free cooling hours and the fan power predicted is slightly lesser than the old model.

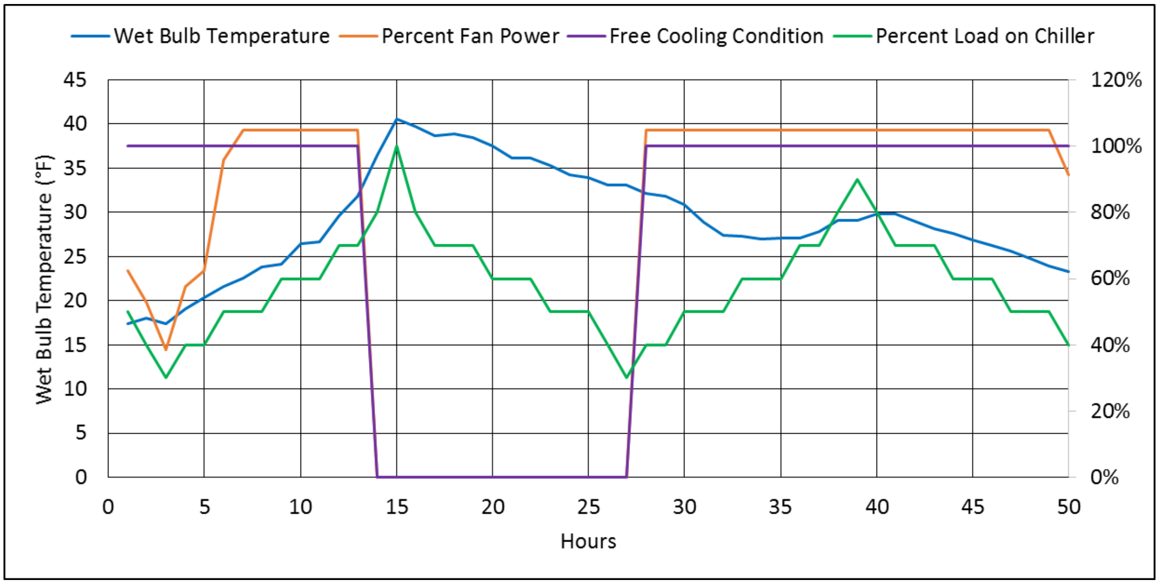


Figure 5.3: Variation in Tower Fan Power with Load and Wet Bulb Temperature as predicted by CWSAT

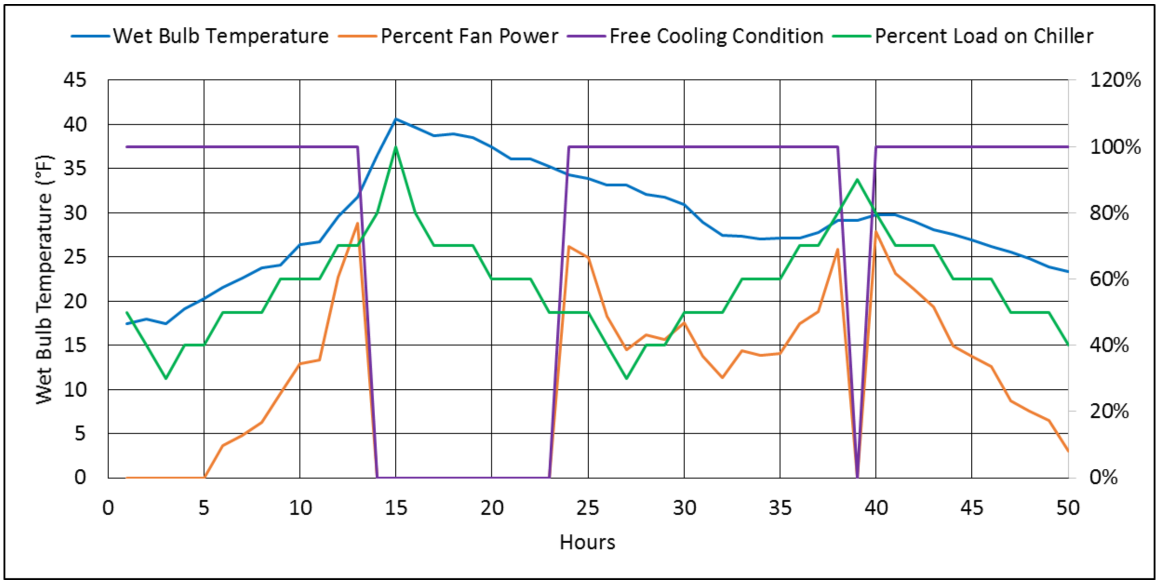


Figure 5.4: Variation in Tower Fan Power with Load and Wet Bulb Temperature as predicted by the Old Model

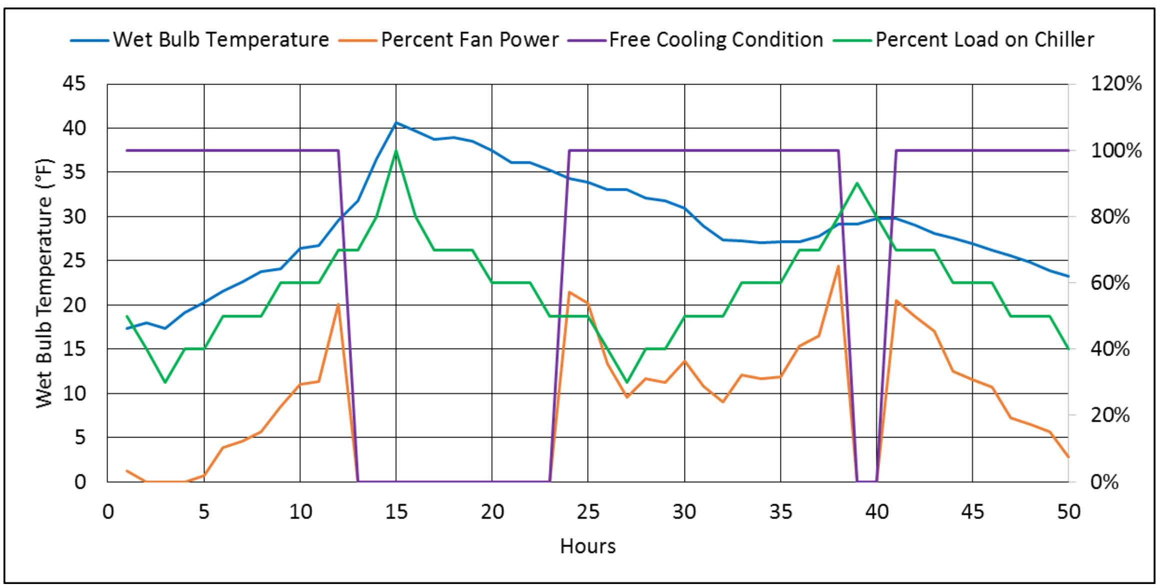


Figure 5.5: Variation in Tower Fan Power with Load and Wet Bulb Temperature as predicted by the New Model

5.3 Benefits of Larger Cooling Tower Capacity to meet Winter Load

We see from the Table 5.2 earlier that the annual the load on the tower is 50% or lesser for 42% of the year. If we want to be able to carry out free cooling in winter when the load on the tower is greater than the tower can take, we will need additional tower capacity. From Table 5.7 we see that there are approximately 2,300 annual hours when free cooling can be carried out for an ARI load and only about 1,200 hours if it is a process load. Now an analysis for both situations is done to see if adding additional tower capacity through another tower will be economically viable.

A new tower of the same size is purchased that is only used in conditions of free cooling when the main tower cannot handle the load. The analysis is conducted (with two towers and both having VFDs) and the results are shown in Table 5.9 and Table 5.10. There is an increase in tower fan energy usage from earlier but also more chiller energy savings.

Table 5.9: Increase in Chiller Energy Savings with additional Tower for ARI loading schedule

Condition	Energy Usage (kWh)			Associated Cost
Tower energy use when free cooling with one tower	28,365			$2,269
Free cooling energy savings on chiller with one tower	189,156			$15,132
Tower energy use when free cooling with two towers	Tower 1	Tower 2	Total	$2,379
	28,171	1,571	29,742	
Free cooling energy savings with two towers	222,403			$17,792
Increase in cooling tower fan energy usage	1,377			$110
Increase in chiller energy savings	33,247			$2,660
Total increase in savings with additional tower	31,870			$2,550

Table 5.10: Increase in Chiller Energy Savings with additional Tower for a constant process load

Condition	Energy Usage (kWh)			Associated Cost
Tower energy use when free cooling with one tower	51,335			$4,107
Free cooling energy savings on chiller with one tower	166,619			$13,330
Tower energy use when free cooling with two towers	Tower 1	Tower 2	Total	$5,103
	52,590	11,197	63,787	
Free cooling energy savings with two towers	358,806			$28,704
Increase in cooling tower fan energy usage	12,452			$996
Increase in chiller energy savings	192,187			$15,375
Total increase in savings with additional tower	179,735			$14,379

Table 5.11: Economic Benefit of an additional cooling tower

ARI Loading	Additional Tower Cost Savings	$2,550
	Implementation Cost	$45,000
	Simple Payback	17.6 years
Process Loading	Cost Savings	$14,379
	Implementation Cost	$45,000
	Simple Payback	3.1 years

Table 5.11 shows the economic benefit of the additional tower for both situations. It is seen that while the additional tower does not make much economic sense for an ARI loaded tower, the tower with a process load is greatly benefitted by the additional tower leading to considerable cost savings and a short payback period of only 3.1 years. The life of a cooling tower is generally between 15 to 20 years making this a viable option. Thus if the load on the tower is greater than 50% for large parts of free cooling season,

there is great opportunity for savings through the installation of a larger cooling tower to accommodate economizer operation.

It is observed that maximum savings are possible from operating in free cooling mode rather than just through a VSD on the cooling tower fan. However the ease of installing a VSD on a cooling tower fan with minimal costs and its applicability even in areas where the temperatures are not suitable for free cooling make it prime target to achieve energy savings. Figure 5.2 shows the variation in tower performance for different fan speeds. It is observed that tower performance varies very little with variation in fan speed between 100% fan speed and 60% fan speed. In comparison, reducing fan speed lower than 50% is seen to more drastically influence tower performance. Thus there is huge opportunity for reducing fan speed and this leads to savings in energy because of the fan affinity laws where the power consumption varies as the cube of the fan speed. This means that if the fan speed is reduced from 100% (corresponding to a power consumption of 100%) to 70% fan speed (corresponding to a power consumption of 34.3%) there is a savings in power consumption of 65.7%. To put the cost of a VSD in perspective, a VSD for the 25 hp fan considered in the earlier cases is $5,375 (including material and labor according to RSMeans Electrical Cost Data 2014) which makes installing a VSD relatively inexpensive and not a capital intensive measure. However it should be noted that a VSD has an efficiency and the efficiency of VSDs of a few sizes at different speeds is shown in Figure 5.3. For the purposes of calculation in the previous analysis a constant VSD efficiency of 95% was considered which is largely correct as seen in the figure.

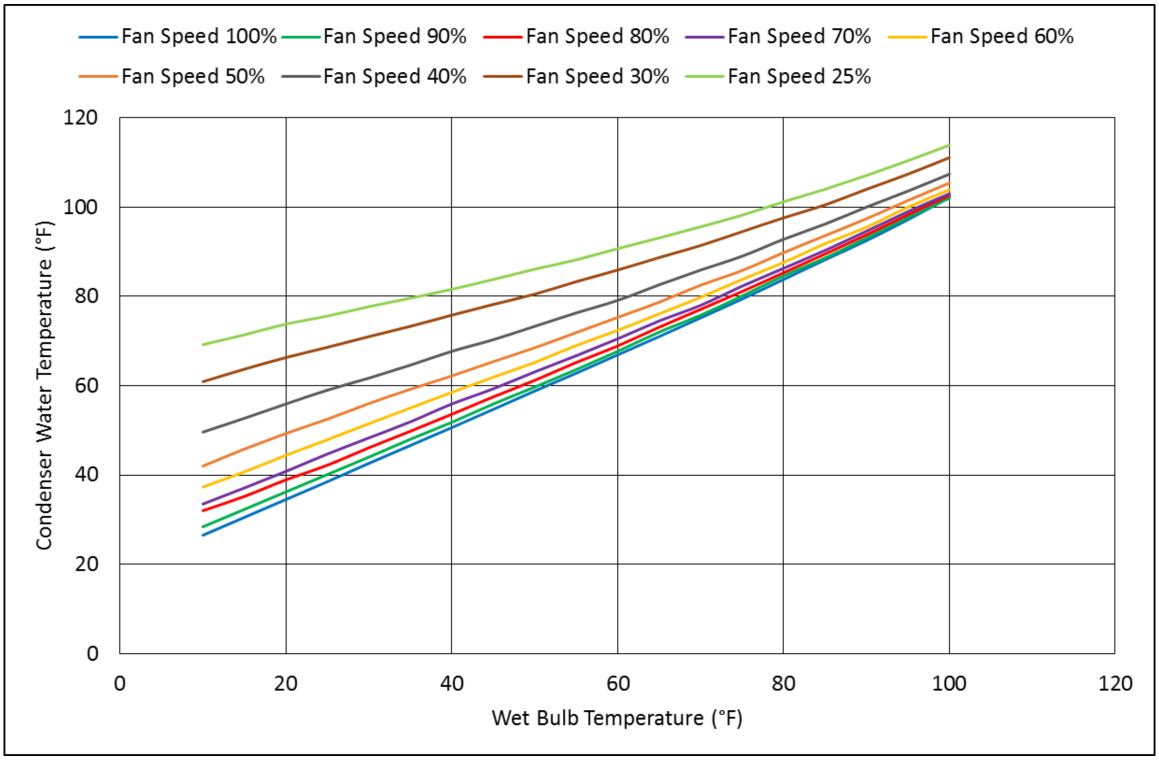


Figure 5.2: Tower Performance Variation with Fan Speeed

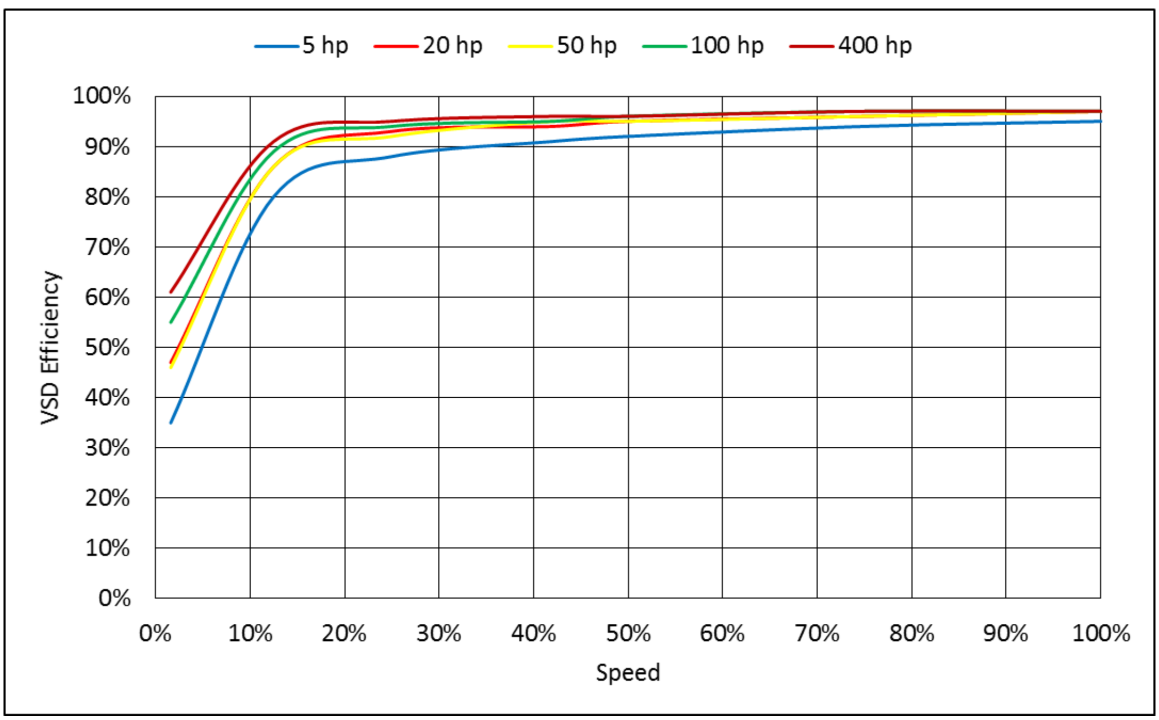


Figure 5.3: VSD Efficiencies at varying Motor Speeds

CHAPTER 6

CONCLUSIONS

This chapter includes recommendations for future work to the tool

5.1 Summary

In Chapter 2, the drawbacks of thermodynamic models and the need for simple surrogate models was outlined. A simple to use model that requires easily available information to predict tower fan energy use would be of great use to an average user. Chapter 3 talked about the method and sources of data collection to verify the existing model given by [2] and create a new model. Despite the difficulty of tower performance data availability over a wide range of parameters, suitable information was collected to be able to verify the existing model and create a new model.

Chapter 4 mapped out the method to create a new model and the results were shown to be better than the previous model. Polynomial regression was found to be the simplest yet accurate method to create a new model. The improvements though marginal are found to be at those conditions where energy savings measures are possible leading to a better prediction of savings.

In Chapter 5, CWSAT, the old model and the new model were used to predict tower fan energy use over a year and the results were compared. Further the economic benefit of adding an additional cooling tower was analyzed and found that greater the load on the tower during times when the wet bulb temperature is lower than the required cold water temperature, an additional tower clearly adds economic benefit.

5.2 Recommendations for Future Work

To check if CWSAT was implementing the old model correctly, separate code was written to compare the results. It was seen that when the tower was not used in economizer mode, the results obtained were close between the old model and CWSAT (although CWSAT tended to predicted a higher value of fan power). When the tower is operated in free cooling mode, it is seen that CWSAT does not take into account the load on the tower but just the temperature. Making sure that CWSAT correctly accounts for the load the tower can take will go a long way towards making more accurate predictions of chiller energy savings. Further when hour by hour predicted fan power values were analyzed, it was seen that during free cooling operation CWSAT always tended to predict that the fan power required was almost always a 100% (much higher than needed). Making sure that CWSAT correctly implements the model is of utmost importance in being able to predict tower performance and thereby possible savings accurately.

As seen in Figure 5.2, the cooling tower performance is reduced considerably when the fan speed is reduced below 60% (corresponds to 21.6% fan power). During model building, data only until 60% fan speed was utilized since including lower fan speeds tended to make predictions at higher speeds inaccurate. Thus when the model predicts a low fan power (meaning low fan speeds), it might not always be possible in reality to reduce the fan speed that low or turn off the fan completely. While this doesn't make much difference if the tower fan has a VSD, the fan power may add up over time for a single speed fan. Going ahead it would be important to be able to determine what is the lowest possible fan speed required without degrading tower performance, through actual data collected from the field. Since ANNs were seen to give comparable accuracy to PR, greater depth in looking at ANNs to create a prediction model might be beneficial.

The current version of CWSAT doesn't allow choosing the kind of tower (i.e counter flow or cross flow) and adding an option for that may be useful to be able to predict tower performance more accurately. Finally adding an option in CWSAT to predict additional savings possible through the use of an additional tower, especially for process loads would make it an invaluable and complete tool to predict energy savings associated with cooling towers.

Although CWSAT doesn't need to be improved solely because of a new model (the old and new model tend to give similar results), it needs to be spruced up to better implement the existing model to more accurately predicted tower fan energy use. The recommendations for future work may be summarized as follows

1. Check that CWSAT implements the tower performance prediction correctly and makes sure that free cooling is possible by checking the maximum load that the tower can take.
2. Determine how low the fan speed can actually be turned down since tower performance degrades greatly at fan speeds lower than 50%.
3. Add an option in CWSAT to check if adding an additional tower is economically viable.
4. Creating a new model from data collected from actual towers may give a more realistic picture of tower performance in the field rather than just relying on data provided by tower manufacturers. It can also be looked into creating the new models with ANNs or other model building methods.

APPENDIX A

SAMPLE TOWER PERFORMANCE GRAPH

Baltimore Aircoil Company, Inc.

Cooling Tower Selection Program

Version:	8.5.1 NA
Product data correct as of:	October 20, 2014

Project Name:	
Selection Name:	
Project State/Province:	
Project Country:	United States
Date:	January 02, 2015

Model & Fan Motor

Product Line:	New Series 1500
Model:	S15E-1212-09JN
Number of Units:	1
Fan Motor:	(2) 7.50 = 15.00 HP/Unit
Total Standard Fan Power:	100% of Full Speed, 15.00 BHP/Unit

Model Accessories

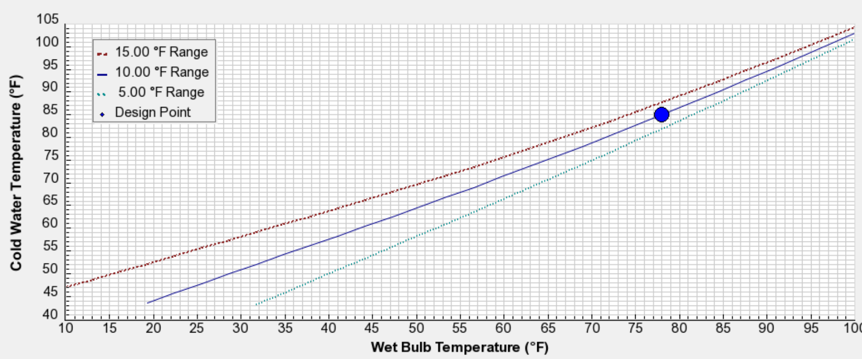
Intake Option:	None
Internal Option:	None
Discharge Option:	None
Fan Type:	Standard Fan

Design Conditions @ Standard Total Fan Motor Power per Unit (15.00 HP)

Thermal performance at design conditions and standard total fan motor power is certified by the Cooling Technology Institute (CTI).

Flow Rate:	984.00 USGPM
Hot Water Temp.:	95.00 °F
Cold Water Temp.:	85.00 °F
Wet Bulb Temp.:	78.00 °F

Predicted Performance
Fan Motor Alternative = 100% of Full Speed, 15.00 BHP
Flow Rate = 984.00 USGPM (100.00% of Design)



These performance curves are based on constant fan power.

Warnings	Applies to Design Conditions	Applies to Off Design Conditions
1. One or more selection parameters are outside of CTI Certification limits.	No	Yes
2. Potential for freezing. Cold water temperature less than 43 °F (6.11 °C).	No	Yes

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APPENDIX B

MATLAB CODE USED FOR TOWER PERFORMANCE PREDICTION

Initial declarations

```
chillercapacity=250;%chillercapacity=input('Enter Chiller Capacity in Tons');
numberofchillers=1;
towercapacity=300;%towercapacity=input('Enter tower capacity in tons');
numberoftowers=1;%Adjust in case of multiple towers
ratedwater=3*towercapacity; %rated water flow for the tower, generally 3gpm/ton
condenserwaterSPtemp=75;%condenserwaterSPtemp=input('Enter condenser water Set point
temperature')
condenserwatergpmpertonknown=input('Enter the condenser water flow rate in gpm/ton. Enter 3 if
unknown'); %actual water flow in gpm per ton through tower
fanhp=25; %Horsepower of tower fan
HEXtemp=2; %Drop in Temperature across Heat Exchanger
CHWST=45; %Chilled water set point temperature
fantype=input('Single speed [s], Two Speed [m], Variable Speed[v]', 's');
chillerenergysaved=0; %counter for chiller energy saved
FCON=0; % Free cooling condition, 1 means free cool on and 0 means free cool off
%Have the lists of wetbulb temperature, hourly chiller load and chiller
%power for the year loaded in MATLAB workspace. Also have the model coefficients loaded
```

Preallocations to speed up the program

```
waterlist=zeros(8760,1);
rangelist=zeros(8760,1);
approachlist=zeros(8760,1);
fanenergy=zeros(8760,1);
```

Finding the gpm of water every hour of the year

```
for i=1:8760
    waterlist(i,1)=(condenserwatergpmpertonknown*towercapacity); % if vfd on tower pump make
suitable adjustments of changing flow within limits of 80% and 120% of rated as load changes
end
```

Finding the range for ever hour of the year

```
for i=1:8760
    if freecoolfinal(i)==1 && FCON==1

rangelist(i,1)=(((hourlyloadlist(i,1))*chillercapacity*numberofchillers))/((waterlist(i,1)*8.333*
0.005)*numberoftowers);
    else rangelist(i,1)=
((hourlyloadlist(i,1)*1.2*chillercapacity*numberofchillers))/((waterlist(i,1)*8.333*0.005)*number
oftowers); %totalheatload=(gpm*range*8.333*0.005) where 0.005 is to covert from btu/min to tons,
also 1.2 if no free cooling to account for heat of compression
    end

end
```

Generating the approach every hour

```
for i=1:8760
    if freecoolfinal(i)==1 && FCON==1
        approachlist(i,1)=CHWST-twblist(i); %Approach changes based on wet bulb temperature and
whether free cooling is turned on or off
    else approachlist(i,1)=condenserwaterSPtemp-twblist(i,1);
    end
end
```

Actually solving for fan power every hour

```
for i=1:8760

    twb=twblist(i,1);
    range=rangelist(i,1);
    water=(waterlist(i,1)/ratedwater)*100;
    desiredapproach=approachlist(i,1);

    if hourlyloadlist(i)>0
        fantemp=1;
        itercounter=0;
       %The benton model equation
        approach(i,1)=(ben(1,1))+ ...
        (twb*ben(2,1))+...
        (range*ben(3,1))+...
        (water*ben(4,1))+...
        (fantemp*ben(5,1))+...
        (twb*twb*ben(6,1))+...
        (twb*range*ben(7,1))+...
        (range*range*ben(8,1))+...
        (twb*water*ben(9,1))+...
        (range*water*ben(10,1))+...
        (water*water*ben(11,1))+...
        (twb*fantemp*ben(12,1))+...
        (range*fantemp*ben(13,1))+...
        (water*fantemp*ben(14,1))+...
        (fantemp*fantemp*ben(15,1))+...
        (twb*twb*twb*ben(16,1))+...
        (twb*twb*range*ben(17,1))+...
        (twb*range*range*ben(18,1))+...
        (range*range*range*ben(19,1))+...
        (twb*twb*water*ben(20,1))+...
        (twb*range*water*ben(21,1))+...
        (range*range*water*ben(22,1))+...
        (twb*water*water*ben(23,1))+...
        (range*water*water*ben(24,1))+...
        (water*water*water*ben(25,1))+...
        (twb*twb*fantemp*ben(26,1))+...
        (twb*range*fantemp*ben(27,1))+...
        (range*range*fantemp*ben(28,1))+...
        (twb*water*fantemp*ben(29,1))+...
```

```
    (range*water*fantemp*ben(30,1))+...
    (water*water*fantemp*ben(31,1))+...
    (twb*fantemp*fantemp*ben(32,1))+...
    (range*fantemp*fantemp*ben(33,1))+...
    (water*fantemp*fantemp*ben(34,1))+...
    (fantemp*fantemp*fantemp*ben(35,1));
while abs((approach(i,1)-desiredapproach)/desiredapproach)>0.01
    itercounter=itercounter+1;
    fantemp=fantemp+0.1;
    approach(i,1)=(ben(1,1))+ ...
    (twb*ben(2,1))+...
    (range*ben(3,1))+...
    (water*ben(4,1))+...
    (fantemp*ben(5,1))+...
    (twb*twb*ben(6,1))+...
    (twb*range*ben(7,1))+...
    (range*range*ben(8,1))+...
    (twb*water*ben(9,1))+...
    (range*water*ben(10,1))+...
    (water*water*ben(11,1))+...
    (twb*fantemp*ben(12,1))+...
    (range*fantemp*ben(13,1))+...
    (water*fantemp*ben(14,1))+...
    (fantemp*fantemp*ben(15,1))+...
    (twb*twb*twb*ben(16,1))+...
    (twb*twb*range*ben(17,1))+...
    (twb*range*range*ben(18,1))+...
    (range*range*range*ben(19,1))+...
    (twb*twb*water*ben(20,1))+...
    (twb*range*water*ben(21,1))+...
    (range*range*water*ben(22,1))+...
    (twb*water*water*ben(23,1))+...
    (range*water*water*ben(24,1))+...
    (water*water*water*ben(25,1))+...
    (twb*twb*fantemp*ben(26,1))+...
    (twb*range*fantemp*ben(27,1))+...
    (range*range*fantemp*ben(28,1))+...
    (twb*water*fantemp*ben(29,1))+...
    (range*water*fantemp*ben(30,1))+...
    (water*water*fantemp*ben(31,1))+...
    (twb*fantemp*fantemp*ben(32,1))+...
    (range*fantemp*fantemp*ben(33,1))+...
    (water*fantemp*fantemp*ben(34,1))+...
    (fantemp*fantemp*fantemp*ben(35,1));
if itercounter>=1001
        break
end

%Check to see if fan power is closer to 100% or 25%
                        if fantemp>=100
                            fantempH=100;
                            fantempL=25;
```

```
approachH(i,1)=(ben(1,1))+ ...
    (twb*ben(2,1))+...
    (range*ben(3,1))+...
    (water*ben(4,1))+...
    (fantempH*ben(5,1))+...
    (twb*twb*ben(6,1))+...
    (twb*range*ben(7,1))+...
    (range*range*ben(8,1))+...
    (twb*water*ben(9,1))+...
    (range*water*ben(10,1))+...
    (water*water*ben(11,1))+...
    (twb*fantempH*ben(12,1))+...
    (range*fantempH*ben(13,1))+...
    (water*fantempH*ben(14,1))+...
    (fantempH*fantempH*ben(15,1))+...
    (twb*twb*twb*ben(16,1))+...
    (twb*twb*range*ben(17,1))+...
    (twb*range*range*ben(18,1))+...
    (range*range*range*ben(19,1))+...
    (twb*twb*water*ben(20,1))+...
    (twb*range*water*ben(21,1))+...
    (range*range*water*ben(22,1))+...
    (twb*water*water*ben(23,1))+...
    (range*water*water*ben(24,1))+...
    (water*water*water*ben(25,1))+...
    (twb*twb*fantempH*ben(26,1))+...
    (twb*range*fantempH*ben(27,1))+...
    (range*range*fantempH*ben(28,1))+...
    (twb*water*fantempH*ben(29,1))+...
    (range*water*fantempH*ben(30,1))+...
    (water*water*fantempH*ben(31,1))+...
    (twb*fantempH*fantempH*ben(32,1))+...
    (range*fantempH*fantempH*ben(33,1))+...
    (water*fantempH*fantempH*ben(34,1))+...
    (fantempH*fantempH*fantempH*ben(35,1));

approachL(i,1)=(ben(1,1))+ ...
    (twb*ben(2,1))+...
    (range*ben(3,1))+...
    (water*ben(4,1))+...
    (fantempL*ben(5,1))+...
    (twb*twb*ben(6,1))+...
    (twb*range*ben(7,1))+...
    (range*range*ben(8,1))+...
    (twb*water*ben(9,1))+...
    (range*water*ben(10,1))+...
    (water*water*ben(11,1))+...
    (twb*fantempL*ben(12,1))+...
    (range*fantempL*ben(13,1))+...
    (water*fantempL*ben(14,1))+...
    (fantempL*fantempL*ben(15,1))+...
    (twb*twb*twb*ben(16,1))+...
    (twb*twb*range*ben(17,1))+...
```

```
                                        (twb*range*range*ben(18,1))+...
                                        (range*range*range*ben(19,1))+...
                                        (twb*twb*water*ben(20,1))+...
                                        (twb*range*water*ben(21,1))+...
                                        (range*range*water*ben(22,1))+...
                                        (twb*water*water*ben(23,1))+...
                                        (range*water*water*ben(24,1))+...
                                        (water*water*water*ben(25,1))+...
                                        (twb*twb*fantempL*ben(26,1))+...
                                        (twb*range*fantempL*ben(27,1))+...
                                        (range*range*fantempL*ben(28,1))+...
                                        (twb*water*fantempL*ben(29,1))+...
                                        (range*water*fantempL*ben(30,1))+...
                                        (water*water*fantempL*ben(31,1))+...
                                        (twb*fantempL*fantempL*ben(32,1))+...
                                        (range*fantempL*fantempL*ben(33,1))+...
                                        (water*fantempL*fantempL*ben(34,1))+...
                                        (fantempL*fantempL*fantempL*ben(35,1));
                                    if abs(desiredapproach-approachL(i,1)) < abs(desiredapproach-
approachH(i,1))
                                        fanenergy(i)=0;
                                    else fanenergy(i)=100;
                                    end
                                    else fanenergy(i)=fantemp;
                                end


    end
    end
end
```

Adjust the actual fan power to be what it would based on fan types (e.g. two speed fan, VFD, single speed)

```
for i=1:8760

    if fantype=='s'
        if fanenergy(i)<=100
            vfdtosingle=(fanenergy(i)/100)^(1/3);
            fanenergy(i)=vfdtosingle*100;
        else fanenergy(i)=fanenergy(i);
        end

    end


    if fantype=='v'
        fanenergy(i)=fanenergy(i)/0.95; %assumed const vfd efficiency of 0.95
    end

end
```

Summing up fan power over the year

```
for i=1:8760
    checkfanenergy(i,1)=((fanenergy(i))/100)*fanhp*0.746*numberoftowers/0.9;
end
checkit=sum(checkfanenergy);
totalfanenergy=(sum(fanenergy)/100)*fanhp*0.746*numberoftowers/0.9; %because fanpower is in
percent form; 0.746 is for conversion from hp to kW  0.9 is motor efficiency
```

Finding how much chiller energy was saved by free cooling

```
for i=1:8760
    if freecoolfinal(i)==1
        chillerenergysaved=chillerenergysaved+hourlypowerlist(i);
    end
end
```

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```
%%Code to generate the freecooling possibility for each hour
%Initial Declarations
CWT=75;
HEXtemp=2;
CHWST=45; %Chilled water setpoint Temperature
flowrate=900; %gpm for a 300 ton tower
FCtempcheck=CHWST-HEXtemp-3; %can add -3 at he end because the best approach for the lowest load
of 0.1 is 2.8f i.e approx 3 and anything greater just let the load take care of it
freecooltempcheck=zeros(8760,1);
chillercapacity=250; %Chillercapacity

%Generating design water flow and max fan power for each hour
for i=1:8760
    fanlist2(i,1)=100;
    waterlist2(i,1)=100;
end
```

Checking if free cooling possible if just based off hex and wet bulb temp

```
for i=1:8760
if twblist(i)<FCtempcheck
    freecooltempcheck(i,1)=1;
else freecooltempcheck(i,1)=0;
end
end
```

Checking if free cooling possible based on load

```
for i=1:8760
if freecooltempcheck(i,1)==1
   approachlist2(i,1)=CHWST-twblist(i)-HEXtemp;
else approachlist2(i,1)=CWT-twblist(i);
end
end


    for i=1:8760
        twb=twblist(i,1);
        fantemp=fanlist2(i,1);
        water=waterlist2(i,1);
        desiredapproach=approachlist2(i,1);
        range=0.2;


        itercounter=0;
       %The new model equation
        approach(i,1)=(ben(1,1))+ ...
        (twb*ben(2,1))+...
        (range*ben(3,1))+...
        (water*ben(4,1))+...
        (fantemp*ben(5,1))+...
```

```
        (twb*twb*ben(6,1))+...
        (twb*range*ben(7,1))+...
        (range*range*ben(8,1))+...
        (twb*water*ben(9,1))+...
        (range*water*ben(10,1))+...
        (water*water*ben(11,1))+...
        (twb*fantemp*ben(12,1))+...
        (range*fantemp*ben(13,1))+...
        (water*fantemp*ben(14,1))+...
        (fantemp*fantemp*ben(15,1))+...
        (twb*twb*twb*ben(16,1))+...
        (twb*twb*range*ben(17,1))+...
        (twb*range*range*ben(18,1))+...
        (range*range*range*ben(19,1))+...
        (twb*twb*water*ben(20,1))+...
        (twb*range*water*ben(21,1))+...
        (range*range*water*ben(22,1))+...
        (twb*water*water*ben(23,1))+...
        (range*water*water*ben(24,1))+...
        (water*water*water*ben(25,1))+...
        (twb*twb*fantemp*ben(26,1))+...
        (twb*range*fantemp*ben(27,1))+...
        (range*range*fantemp*ben(28,1))+...
        (twb*water*fantemp*ben(29,1))+...
        (range*water*fantemp*ben(30,1))+...
        (water*water*fantemp*ben(31,1))+...
        (twb*fantemp*fantemp*ben(32,1))+...
        (range*fantemp*fantemp*ben(33,1))+...
        (water*fantemp*fantemp*ben(34,1))+...
        (fantemp*fantemp*fantemp*ben(35,1));
    while abs((approach(i,1)-desiredapproach)/desiredapproach)>0.01
        itercounter=itercounter+1;
        range=range+0.01;
        approach(i,1)=(ben(1,1))+ ...
        (twb*ben(2,1))+...
        (range*ben(3,1))+...
        (water*ben(4,1))+...
        (fantemp*ben(5,1))+...
        (twb*twb*ben(6,1))+...
        (twb*range*ben(7,1))+...
        (range*range*ben(8,1))+...
        (twb*water*ben(9,1))+...
        (range*water*ben(10,1))+...
        (water*water*ben(11,1))+...
        (twb*fantemp*ben(12,1))+...
        (range*fantemp*ben(13,1))+...
        (water*fantemp*ben(14,1))+...
        (fantemp*fantemp*ben(15,1))+...
        (twb*twb*twb*ben(16,1))+...
        (twb*twb*range*ben(17,1))+...
        (twb*range*range*ben(18,1))+...
        (range*range*range*ben(19,1))+...
        (twb*twb*water*ben(20,1))+...
```

```
        (twb*range*water*ben(21,1))+...
        (range*range*water*ben(22,1))+...
        (twb*water*water*ben(23,1))+...
        (range*water*water*ben(24,1))+...
        (water*water*water*ben(25,1))+...
        (twb*twb*fantemp*ben(26,1))+...
        (twb*range*fantemp*ben(27,1))+...
        (range*range*fantemp*ben(28,1))+...
        (twb*water*fantemp*ben(29,1))+...
        (range*water*fantemp*ben(30,1))+...
        (water*water*fantemp*ben(31,1))+...
        (twb*fantemp*fantemp*ben(32,1))+...
        (range*fantemp*fantemp*ben(33,1))+...
        (water*fantemp*fantemp*ben(34,1))+...
        (fantemp*fantemp*fantemp*ben(35,1));
    rangelist2(i,1)=range;
     if itercounter>=10001
            break
     end
    end
    end

    for i=1:8760
        maxtowerload(i,1)=rangelist2(i,1)*flowrate*8.333*0.005; %generating the max load the
tower can take for a given range
        percentload(i,1)=maxtowerload(i,1)/chillercapacity; %Expressing the max load as a percent
to compare withthe load on chiller
    end

for i=1:8760
if percentload(i)<hourlyloadlist(i)
checkit(i,1)=1;
else checkit(i)=0;
end
end
```

Generating the final freecooling possibility based on if freecooling possible based on temperature as well as based on load

```
counter=1;
testing=0;
for i=1:8760
    if percentload(i)>hourlyloadlist(i)
        tester(counter,1)=percentload(i)-hourlyloadlist(i);
        counter=counter+1;
        freecoolloadcheck(i,1)=1;
    else freecoolloadcheck(i,1)=0;
    end
    if percentload(i)<hourlyloadlist(i) && freecooltempcheck(i)==1
     testing=testing+1;
 end
end
```

```
for i=1:8760
    if freecoolloadcheck(i,1)==1 && freecooltempcheck(i,1)==1
     freecoolfinal(i,1)=1;
    else freecoolfinal(i,1)=0;
 end
end
```

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