ABSTRACT

STOUT, MALCOLM RUSSELL JR. Cooling Tower Fan Control for Energy Efficiency. (Under the direction of Dr. James W. Leach)

This thesis evaluates the economics of alternative cooling tower capacity control methods. Annual fan electrical energy requirements are calculated for towers with single-speed, two-speed, and variable-speed fans. Fan energy requirements are determined for counter-flow and cross-flow towers designed for low initial cost and for energy efficiency. Effectiveness-NTU equations are solved to predict cooling tower performance at various fan speeds. Natural convection, which determines the cooling capacity when the fan is off, is accounted for using a mean enthalpy difference. Ambient conditions are simulated using typical meteorological year data for five locations. The results show that potential savings are not strongly dependent on the approach temperature, but do increase with increasing range in colder climates. The potential for saving is greatest for cooling towers designed for low initial cost, and is generally higher in locations where the wet-bulb temperature remains relatively constant throughout the year. Two-speed fans that can operate at half speed are generally more suitable for low-cost cooling towers. Two-speed fans that can operate at two-third speed are better for towers designed for energy efficiency and are generally better when the range exceeds 10 °F.

COOLING TOWER FAN CONTROL FOR ENERGY EFFICIENCY

By

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He earned a Bachelor of Science Degree in Geology from Campbell University in May 1982. He worked as a chemist for a mining company for 2 years and as an engineering technician for a geotechnical engineering company for 4 years before returning to North Carolina State University in 1989. He earned a Bachelor of Science Degree in Mechanical Engineering in 1991. He worked as a plant engineer at Harris Nuclear Plant for 6 years and became a licensed Professional Engineer in North Carolina in 1996. In 1998, he returned to North Carolina State University to begin studies toward fulfillment of a Master of Science Degree in Mechanical Engineering. Graduate work focused on the thermal sciences including coursework in heat transfer and fluid mechanics. From May 1998 until August 2000, he worked at the Industrial Assessment Center as a research assistant. In October 2000, he began work at the NC Department of Correction as a Facility Mechanical Engineer and continued working there following completion of the requirements for MSME in May 2003.

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LIST OF SYMBOLS

ε_a	dimensionless variable defined by $(ha_o - ha_i) / (hsw_i - ha_i)$
ε_w	dimensionless variable defined by $(Tw_i - Tw_o) / (Tw_i - Ta_{wb})$
m*	dimensionless variable defined by $(m_a C_s) / (m_w Cp_w)$
NTU	dimensionless variable defined by $(h_c A) / (m_a C_s)$
C_s	effective specific heat of moist air defined by $(hsw_i - hsw_o) / (Tw_i - Tw_o)$
m_a	mass flow rate of air
m_w	mass flow rate of water
Cp_w	specific heat of water
h_c	heat transfer coefficient
A	heat transfer surface area
ha_o	enthalpy of air out
ha_i	enthalpy of air in
hsw_i	enthalpy of saturated air at the entering water temperature
hsw_o	enthalpy of saturated air at the leaving water temperature
Tw_i	temperature of water in
Tw_o	temperature of water out
Ta_{wb}	wet-bulb temperature of ambient air
m_{wi}	mass flow rate of water in
m_{wo}	mass flow rate of water out
W_o	humidity ratio of air in
W_i	humidity ratio of air out

cfm_0 volumetric airflow due to natural convection

cfm volumetric airflow capacity of cooling tower

Δh mean enthalpy difference with fan off

Δh_n mean enthalpy difference at nominal conditions with fan running

Δh_o enthalpy difference at top of tower defined by $hsw_i - ha_o$

Δh_i enthalpy difference at bottom of tower defined by $hsw_o – ha_i$

Cooling Tower Fan Control for Energy Efficiency

1.0 Objective

The objective of this work is to identify the conditions that make alternative capacity control methods for cooling towers cost effective. The annual fan electrical energy requirements for several types of cooling towers operating under various loads will be evaluated. The effects of range and approach temperatures will be established from parametric analysis. The potential savings depending on the operating schedule and the climatic conditions will be investigated using case studies. Ambient conditions for five locations are simulated using typical meteorological year data[1].

2.0 Introduction

2.1 Project Background

This project originated in work performed by the Industrial Assessment Center (IAC) at North Carolina State University. This thesis was originally written in the form of an article entitled "Cooling Tower Fan Control for Energy Efficiency" that was presented at the World Energy Engineering Congress 1999 and published in *Energy Engineering*[2]. Further analysis of climatic characteristics related to potential savings, discussion of an ideal second speed, and the information in the Appendix have been added to the published article.

The Center is sponsored by a grant from the United States Department of Energy and is one of more than two dozen centers at universities across the country. The objective of the

IAC is to identify, evaluate, and recommend opportunities to conserve energy, reduce waste, and increase productivity in small and medium-sized manufacturing facilities. In performing assessments of manufacturing facilities, the IAC team must recognize potential opportunities, collect necessary data and information, and estimate yearly cost savings and implementation costs. Opportunities with reasonable payback periods are reported to the plant as recommendations.

The purpose of this project is development of tools necessary to estimate yearly cooling tower fan energy usage for several capacity control options based on application, type of tower, and climate. These tools can be used by the IAC in evaluating a specific opportunity to upgrade to a more energy efficient capacity control option. Also, this project uses these tools to perform analysis on typical applications. Being able to discern viable opportunities early in the assessment process is important in allocating resources effectively.

2.2 Cooling Towers in Industry

Many manufacturing facilities use cooling towers to reject waste heat from process and air conditioning loads. Cooling tower performance depends on weather conditions, particularly ambient wet-bulb temperature. A cooling tower properly sized to meet the demand at design conditions has excess capacity for most of the year. Capacity control is usually accomplished by changing the airflow[3]. Thus, the excess capacity gives an opportunity to save fan energy. Fan cycling is a common method for controlling the capacity of small and medium-sized towers. Some units are equipped with two-speed fans.

Modulating dampers, variable speed fan motors, and variable pitch propeller fans are additional options.

A cooling tower in a manufacturing plant may be required to cool a constant flow of water to a prescribed constant outlet temperature throughout the year. In some cases the water inlet temperature may also be constant, but it is typically higher in the summer than in the winter. In HVAC applications, it has generally been considered desirable to cool a flow of water to the lowest possible temperature for given ambient conditions, as long as the water temperature remains above some minimum value. With more efficient chillers, the optimal control strategy for cooling towers in HVAC applications has changed to recognize the energy consumption of the tower[4]. A cooling tower must be selected that is large enough to satisfy the load in the summer when the ambient wet-bulb temperature is at its highest value. Capacity control is required to prevent the tower from cooling the water below the prescribed outlet temperature when the load or wet-bulb temperature is low.

Capacity control of cooling towers is an important consideration for effective energy management in manufacturing plants. It is well known that towers with variable speed fan motors consume significantly less electricity over the course of a year than towers with single-speed fans. However, because of the high initial cost of the variable speed drive, the overall economics may be unfavorable. The payback period depends on the design of the cooling tower, load, operating hours, and the climatic conditions.

3.0 Cooling Tower Performance Model

3.1 Cooling Tower Capacity

The size of a cooling tower is typically expressed in nominal tons. This rating specifies the amount of water that can be cooled from 95 °F to 85 °F at a wet-bulb temperature of 78 °F. A cooling tower that can cool 3 gallons per minute (gpm) of water at these conditions is said to have a capacity of 1 nominal ton, or 15,000 BTU/hr, corresponding approximately to the heat that would be rejected from a 1-ton refrigeration system[5]. The change in the temperature of the water in a cooling tower is called the range, and the difference between the water outlet temperature and the ambient wet-bulb temperature is called the approach. At nominal conditions, the range is 10 °F and the approach is 7 °F. The range and approach change as the ambient wet-bulb temperature and the airflow change. The values of range and approach given in this paper for different applications refer to design conditions.

The airflow at nominal conditions and consequently the required fan power vary according to the design of the cooling tower. Typical values of airflow from manufacturer literature[6, 7] range from about 200 standard cubic feet per minute per ton (scfm/ton) to about 300 scfm/ton. Towers designed for energy efficiency typically have small fans and large heat transfer surfaces. Towers designed for low initial cost usually have small heat transfer surfaces and large fans. Typical values of fan horsepower for induced draft towers range from about 0.04 hp/ton to about 0.08 hp/ton. Forced draft towers with exhaust ducts for indoor use[8] may require a fan horsepower of 0.18 hp/ton.

A cooling tower with a capacity of 100 nominal tons would have a water flow of 300 gpm when tested at standard conditions. Typically[6] this same tower would be capable of operating efficiently with water flows as low as 150 gpm, or as high as 600 gpm. The limits of the acceptable water flow depend on the design of the water distribution system[9]. If the water flow is too low, the distribution system cannot provide a uniform coverage of water on the heat transfer surfaces. If the flow is too high, the hot water basins overflow, or the spray nozzles do not operate correctly.

The actual capacity of a cooling tower is dependent on the operating conditions. The 100-ton tower designed for a flow of 300 gpm at standard conditions might be used in a manufacturing plant in which a flow of 215 gpm must be cooled from 92 °F to 82 °F at a wet-bulb temperature of 78 °F. In this case the approach is only 4 °F, and the rate of heat transfer between air and water is reduced. The actual capacity of the 100-ton tower is about 72 tons at these conditions. Equations for predicting cooling tower performance at off-design conditions are developed below.

3.2 Forced Draft - Effectiveness-NTU

In a cooling tower, heat and mass are transferred from water to air. A cooling tower with an effectiveness of 100% would exhaust saturated air from the tower at the temperature of the entering water. In actual towers, the air leaves at a temperature less than the entering water temperature and a relative humidity less than 100% because the airflow is too high and the heat transfer area is too small. Effectiveness, ε_a, is defined[10] in terms of the enthalpy of moist air as follows:

$$\varepsilon_a = (ha_o - ha_i) / (hsw_i - ha_i) \quad (1)$$

In Equation (1), ha_i and ha_o are the enthalpies of the air into and out of the tower, and hsw_i is the enthalpy of air saturated with water at the inlet water temperature.

Effectiveness is a dimensionless variable which can be determined from knowledge of only two additional dimensionless variables, m* and NTU.

$$m^* = (m_a C_s) / (m_w Cp_w) \quad (2)$$

$$NTU = (h_c A) / (m_a C_s) \quad (3)$$

The dimensionless variable, m*, can be determined from the known airflow and known water flow. In Equation (2), Cp_w is the specific heat of water, and C_s is the effective specific heat of moist air defined by:

$$C_s = (hsw_i - hsw_o) / (Tw_i - Tw_o) \quad (4)$$

From Equation (3), NTU depends on the heat transfer coefficient, h_c, the heat transfer surface area, A, the airflow rate, m_a, and the specific heat, C_s. Energy efficient cooling towers with large heat transfer surfaces and small fans might have nominal NTU values of 4.5 or more. Low initial cost towers with small heat transfer surfaces and large fans might have nominal NTU values of 1.5 or less.

Since h_c depends on the airflow rate and the water flow rate, the NTU value of a tower operating at off-design conditions will not be the same as the NTU value at design conditions. An empirical equation[11] useful for predicting NTU at off-design conditions is:

$$NTU = a\,(m_a)^m\,(m_w)^n \qquad (5)$$

The constants, a, m, and n, in Equation (5) are to be determined from published performance data. In some cases, this data is unavailable. A less accurate formula[12] for predicting the performance of cooling towers for which limited published data is available is:

$$NTU = c\,(m_w / m_a)^n \qquad (6)$$

Typical values[12] of n are in the range $0.4 < n < 0.6$. If a typical value of n is assumed, the value of c can be determined from m_a and m_w at nominal design conditions. Once c and n are known for a particular cooling tower, the cooling tower performance can be predicted at any operating condition given the water inlet temperature, Tw_i, the ambient air wet-bulb temperature, Ta_{wb}, and the flow rates, m_a and m_w.

Values of c and n were determined from performance data provided by several manufacturers. Calculations for a wide range of operating conditions showed that the performance characteristics of large induced draft single cell towers with axial fans could be predicted accurately with c = 2.66 and n = 0.634. These towers were all from the same manufacturer, and had nominal capacities in the range 200 tons to 1000 tons. The performance of a 16-ton tower produced by a different manufacturer could be predicted accurately with c = 1.33 and n = 0.4. The low value of c for this tower indicates that the heat transfer area is small for the airflow. The performance characteristics of a 68-ton induced draft tower with a large heat transfer surface could be predicted with c = 3.0 and n = 0.4.

The relationship between ε_a, m*, and NTU for counter-flow cooling towers is[10]:

$$\varepsilon_a = \{1- \exp\,[NTU\,(m^*-1)]\} / \{1-m^*\exp\,[NTU\,(m^*-1)]\} \qquad (7)$$

The temperature of the water leaving the tower can be determined from an energy balance. In English units, Cp_w = 1Btu/lb/°F, and the resulting equation is:

$$Tw_o = 32 + [m_{wi}\,(Tw_i - 32) - m_a(ha_o - ha_i)]/m_{wo} \qquad (8)$$

The exit water flow, m_{wo}, in Equation (8) can be determined from the humidity ratio, W, of the air entering and leaving the cooling tower.

$$m_{wo} = m_{wi} - m_a\,(W_o - W_i) \qquad (9)$$

Braun et al. (1989) recommend a procedure for calculating the humidity of the air leaving the tower, W_o. This procedure was followed in the calculations. However, as an approximation it can be assumed that the air leaving the cooling tower is saturated. The water outlet temperature calculated from Equation (8) is not sensitive to small errors in the value of W_o.

3.3 Natural Draft – Mean Enthalpy Difference

The equations listed above may be solved to determine the water outlet temperature when the fan is running and the airflow is known. When the fan is off, air continues to flow through the tower because of natural convection. The cooling capacity of the tower due to natural convection must be estimated to determine the time that the fan remains off in capacity control options that involve fan cycling. In this work, the volumetric air flow due to natural convection, cfm_0, is calculated from the following equation:

$$cfm_0 = C_0\,(cfm)\,(\Delta h / \Delta h_n)^{0.20} \qquad (10)$$

where C_0 is a constant, cfm is the capacity of the fan, and Δh and Δh_n are mean enthalpy differences. At the top of the tower, $\Delta h_o = hsw_i - ha_o$. At the bottom, $\Delta h_i = hsw_o - ha_i$. The mean enthalpy difference is:

$$\Delta h = (\Delta h_o - \Delta h_i) / \log(\Delta h_o / \Delta h_i) \qquad (11)$$

In Equation (10), Δh is the enthalpy difference with the fan off and Δh_n is the enthalpy difference for the case when the fan is running with Tw_i = 95 °F, Tw_o = 85 °F, and Ta_{wb} = 78 °F. The constant, C_0, is the ratio of the airflow with natural convection to the airflow with the fan running. It must be determined for each cooling tower. Towers with large heat transfer areas and small fans would have relatively high values of C_0.

Equation (6) assumes forced convection, and does not apply when the fan is off. For natural convection, the heat transfer coefficient should increase in proportion to the airflow so that NTU remains constant.

3.4 Predicted Performance

Figure 3.1 illustrates how cooling tower performance is determined. To construct this figure, ε_a is calculated from Equation (7) for given values of m_a/m_w and NTU. The exhaust air enthalpy, ha_o, determined from Equation (1) is then substituted into Equation (8). The water outlet temperature, Tw_o, is plotted as a dimensionless ratio, ε_w. The curve for m_a/m_w = 0.6 is typical of well-designed cooling towers at nominal conditions. The curve for m_a/m_w = 0.30 represents a tower operating with high water flow, or with the fan at half speed. The lower curves represent a tower operating with the fan off.

The dashed lines in Figure 3.1 are determined from Equation (6). The dashed curve with c = 1 and n = 0.4 represents a tower designed for low initial cost. The dashed curve with c = 3 represents a tower designed for efficiency. The intersections of the dashed curves and the solid curves determine the water outlet temperature. For example, with c = 3 and $m_a/m_w = 0.6$, the point of intersection gives $\varepsilon_w = 0.585$. For $Tw_i = 95$ °F and $Ta_{wb} = 78$ °F, the water outlet temperature is $Tw_o = 85$ °F.

If this tower operates with the fan at half speed, the point of intersection at $m_a/m_w = 0.30$ gives $\varepsilon_w = 0.33$. In this case the calculated water outlet temperature is $Tw_o = 89.4$ °F. When the fan operates at half speed, the range is reduced from 10 °F to 5.6 °F. Thus, the capacity of the tower operating with the fan at half speed is 56% of the rated capacity. Calculations for the tower designed for low initial cost, c = 1, show that the capacity with the fan at half speed is 68% of the rated capacity.

Figure 3.2 compares predicted and published[9] data for a tower operating with a fixed load. The range is 10 °F. When the fan is on, the performance data is predicted accurately with c = 3.0, n = 0.4, and $m_a/m_w = 0.6$. When the fan is off, the airflow is calculated from Equation (10) with $C_0 = 0.134$. For a wet-bulb temperature of 78 °F and a water inlet temperature of 95 °F, the tower of Figure 3.2 has a capacity with natural convection that is about 23% of the rated capacity with the fan on. Other manufacturers[6] indicate that the capacity with natural convection is about 10% of the rated capacity. For these towers, the appropriate value is $C_0 = 0.056$.

The relationship[13] between effectiveness, m*, and NTU for cross-flow cooling towers is:

$$\varepsilon_a = (1 - \exp\{m^*[\exp(-NTU) - 1]\})/m^* \qquad (12)$$

Cross-flow towers cannot achieve a nominal approach temperature of 7 °F for $m_a/m_w < 0.8$. At nominal conditions, most cross-flow towers operate in the range $0.8 < m_a/m_w < 0.9$. A typical value of c in Equation (6) is in the range $2.5 < c < 5.5$. Although the airflow is relatively high in these towers, the required fan power is still in the range 0.04 hp/ton to 0.08 hp/ton. There is less fluid friction in cross-flow towers, so higher airflow can be maintained with the same fan power needed by a counter-flow tower. For c = 2.5, a cross-flow tower will maintain 65% of its nominal capacity when the fan runs at half speed.

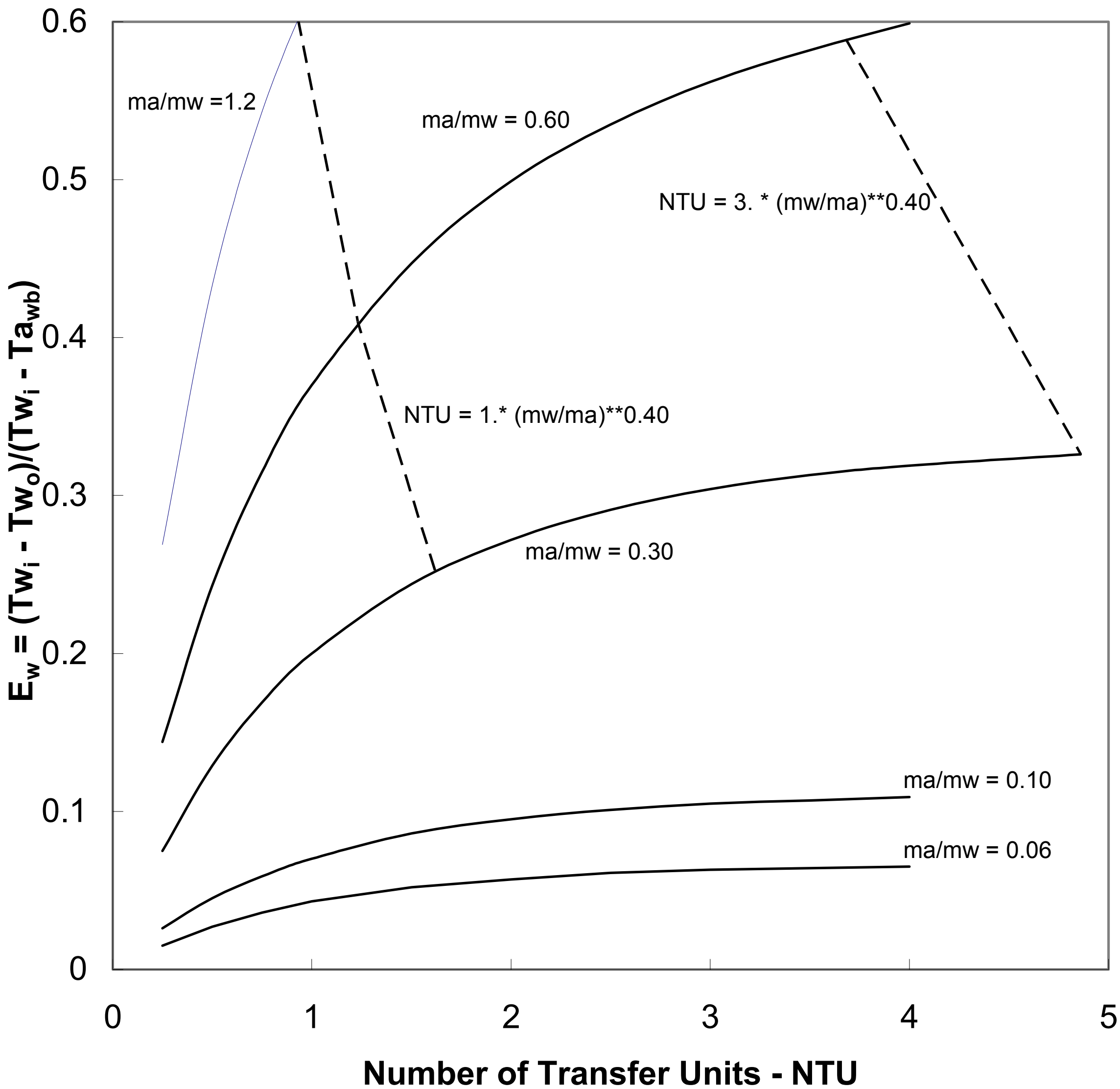

FIGURE 3.1 COOLING TOWER PERFORMANCE CALCULATIONS

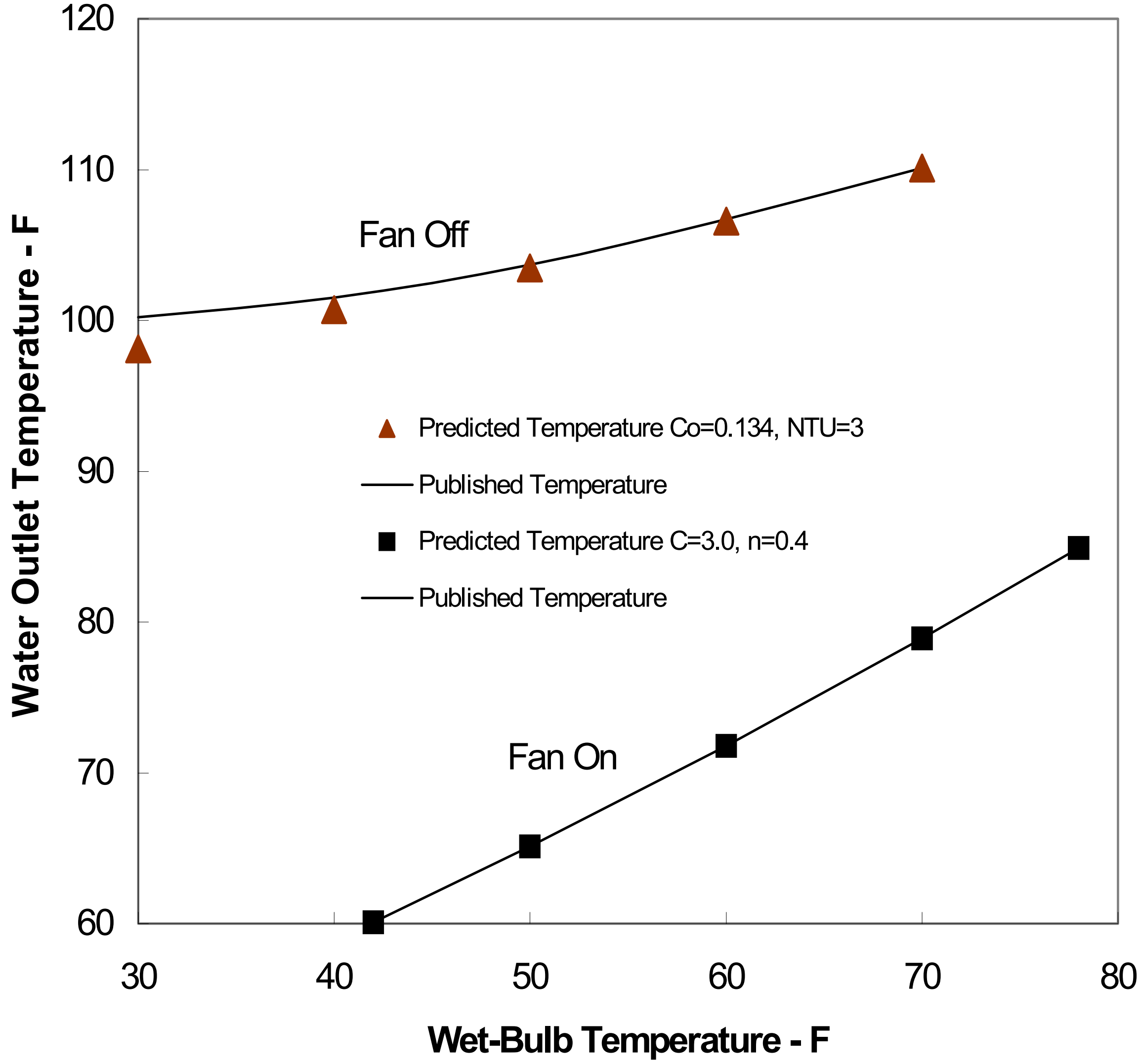

FIGURE 3.2 COOLING TOWER PERFORMANCE PREDICTIONS

4.0 Results and Discussion

4.1 Capacity Control Options and the Effect of Natural Convection

For towers with single-speed fans, the temperature of the cold-water basin is maintained within a prescribed dead band by cycling the fan on and off. The basin and the dead band must be large enough to prevent the fan motor from overheating. Large fan motors should not be started more than once or twice in an hour. Fans with two-speed motors can provide more precise capacity control, and also have potential for saving electricity.

Figure 4.1 shows how fan power changes with ambient wet-bulb temperature for a counter-flow tower that operates with a constant load. This is a hypothetical case that neglects natural convection when the fan is off. The upper curve is for on-off fan cycling. When Ta_{wb} = 52 °F, the fan operates one half of the time, and the average power requirement is one half of the rated fan power. The lower curve is for a two-speed fan motor. When Ta_{wb} = 58 °F, the two-speed motor operates at half speed and uses one-eighth of the rated fan power. When Ta_{wb} is above 58 °F, the motor cycles between full speed and half speed. When Ta_{wb} is below 58 °F, the motor cycles between half speed and off. The electrical savings made possible by the two-speed motor can be determined from the difference between the two curves in Figure 4.1.

Natural convection is accounted for in Figure 4.2. With on-off cycling at Ta_{wb} = 52 °F, the average fan power has now dropped to about 33%. Natural convection helps when the fan is off. Thus, it helps the tower with on-off cycling over the entire temperature range. It has no effect on the two-speed fan until Ta_{wb} drops below 58 °F. At lower temperatures, the average power of the two-speed motor is low, and the potential for savings is relatively small. Thus, the overall effect of natural convection is to reduce the advantage of the two-speed fan. This is apparent from the smaller area between the curves of Figure 4.2.

Figure 4.3 compares the fan motor power requirements for a counter-flow cooling tower operating with a constant load. For this case, the motor that operates at 2/3 rated speed is better at ambient wet-bulb temperatures above 62 °F while the motor that operates at half speed is better at ambient wet-bulb temperatures below 62 °F.

Figure 4.4 compares several control alternatives for a case when the range is relatively large. The hot water enters the tower at 122 °F throughout the year, and the fans cycle to maintain the water outlet temperature at 82 °F. The 2/3-speed motor is better than the half-speed motor for wet-bulb temperatures above about 24 °F in this application, and is almost as good as the motor with a variable speed drive. When the range is large, the mean temperature difference between the water and the air is not strongly dependent on the ambient conditions. The heat transfer rate at moderate ambient conditions is not much greater than the rate at hot conditions. In Figure 4.4 the single-speed motor still runs half of the time when the wet-bulb temperature is 40 °F. As the range increases, there is more opportunity for energy savings.

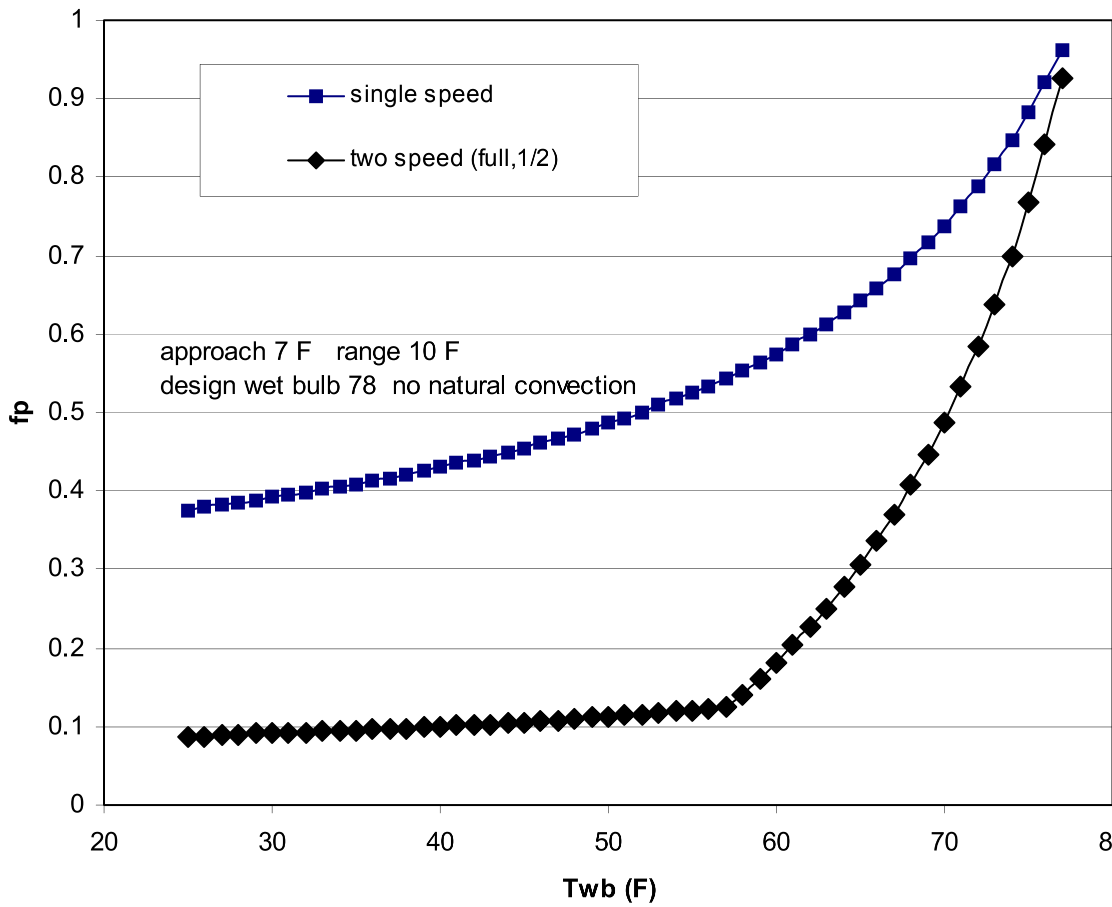

FIGURE 4.1 - CAPACITY CONTROL WITH NO FREE CONVECTION

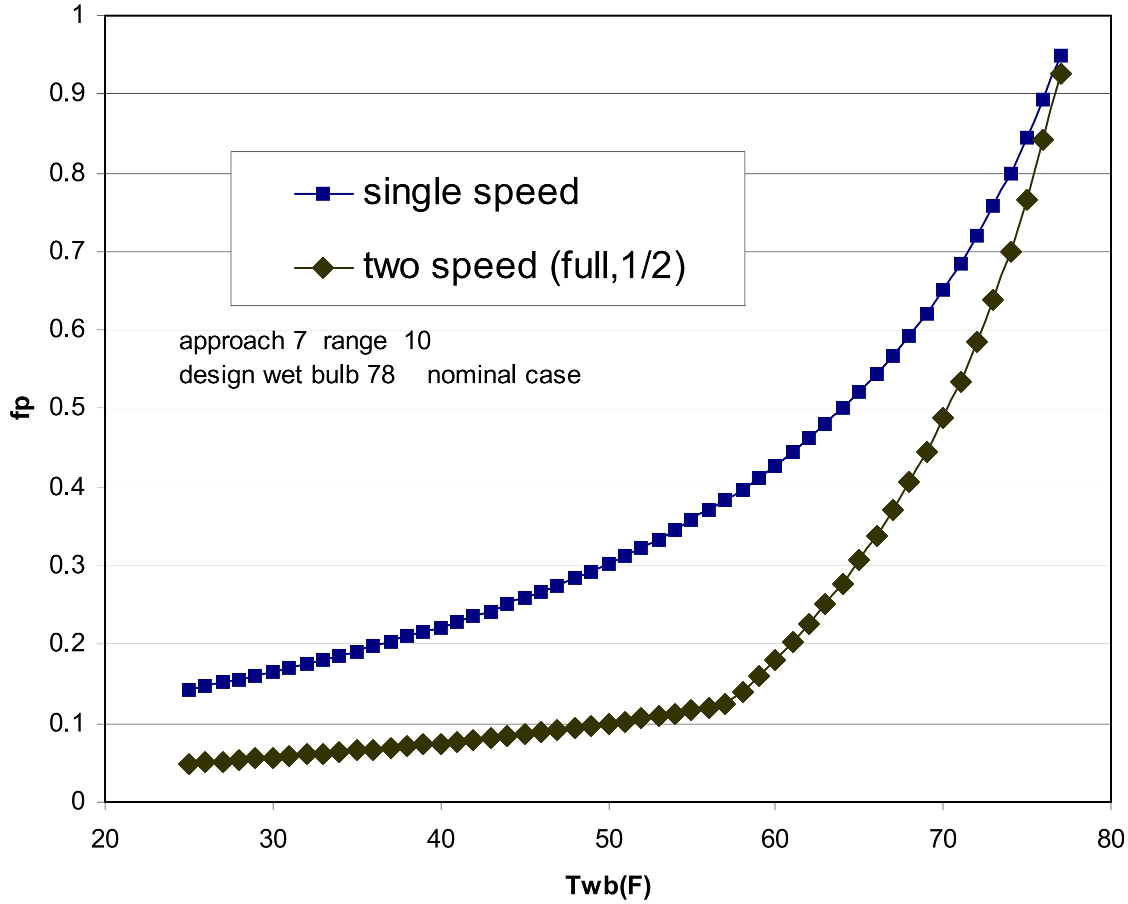

FIGURE 4.2 - THE EFFECT OF FREE CONVECTION

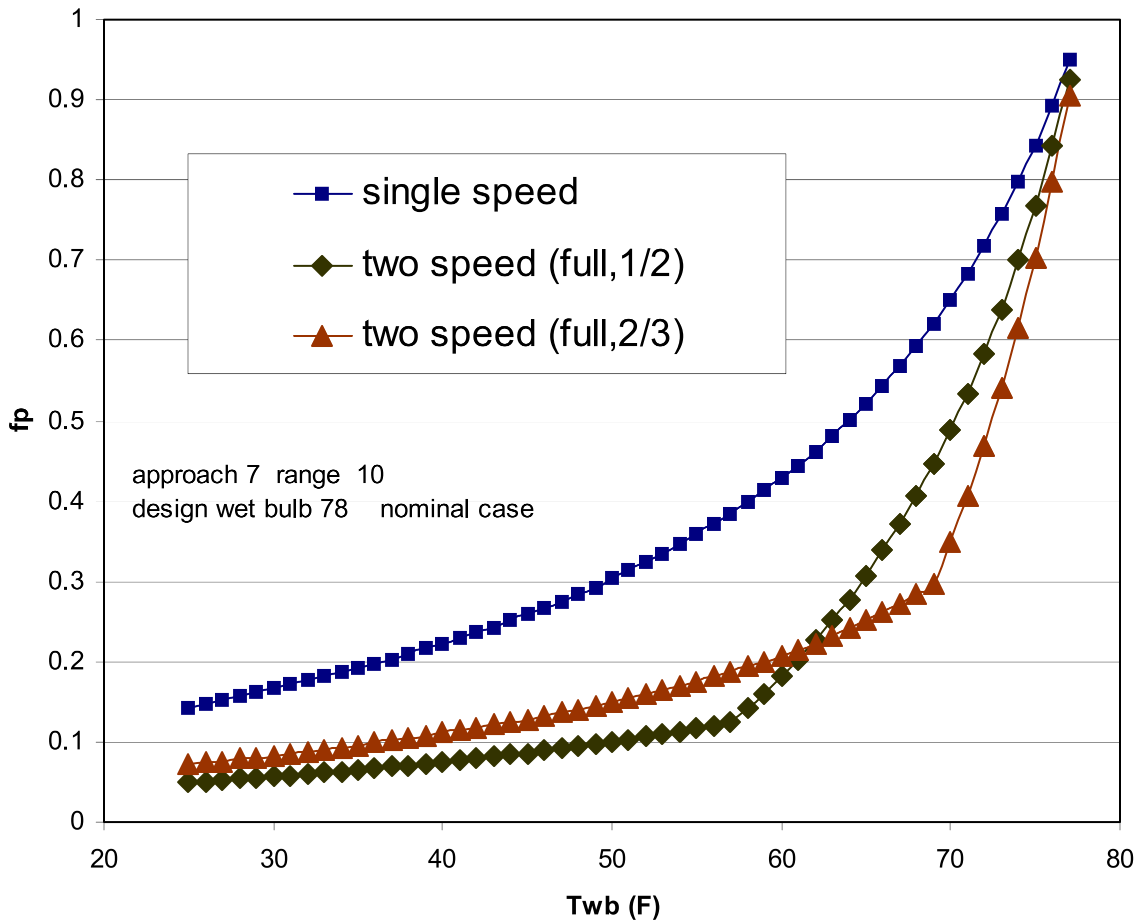

FIGURE 4.3 - FAN-SPEED OPTIONS, RANGE = 10 °F

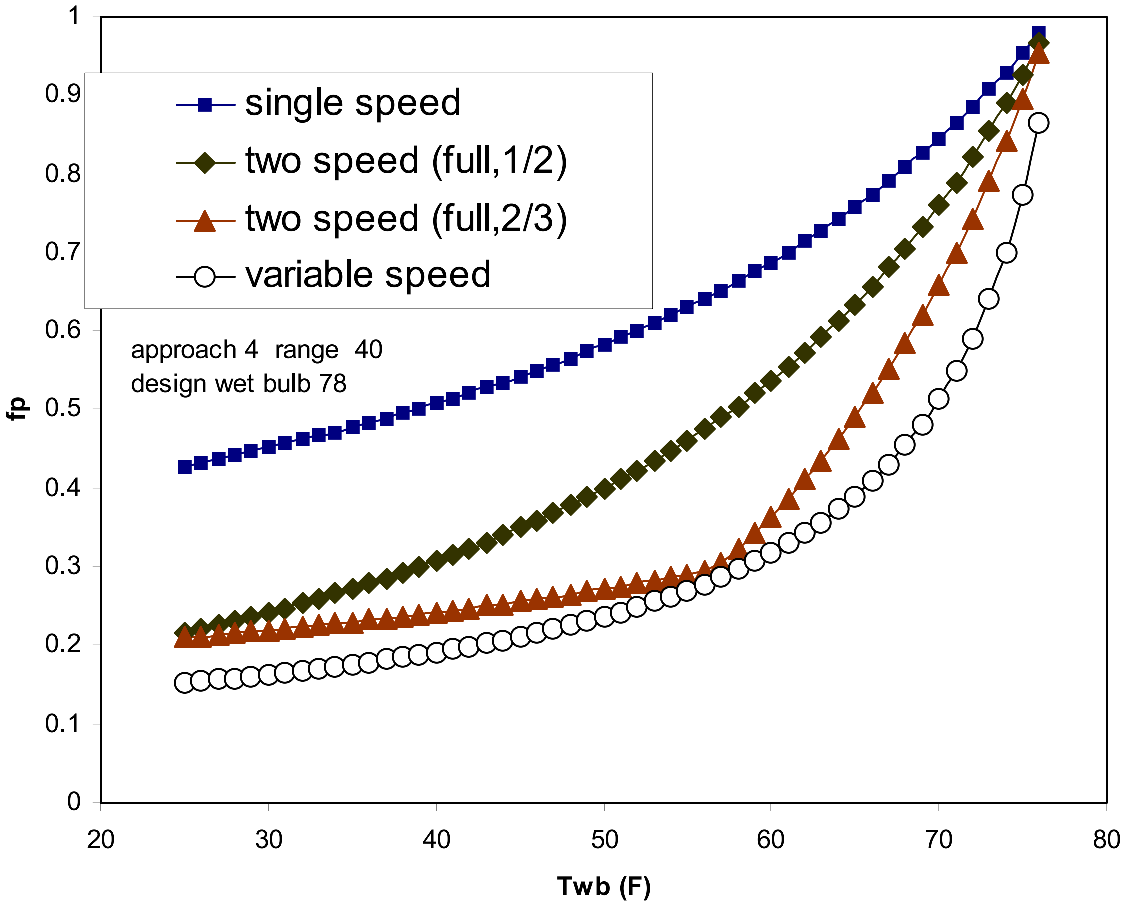

FIGURE 4.4 - FAN-SPEED OPTIONS, RANGE = 40 °F

4.2 Constant Load Applications

The effect of range on annual fan energy requirements for a counter-flow tower is shown in Figure 4.5. The results are for a system that returns cold water at 85 °F throughout the year. The entering hot water temperature is also constant, and depends on the range. The cooling tower is a counter-flow designed for efficiency, $c = 3$, $n = 0.4$, $C_0 = 0.134$, operating in moderate climatic conditions. The fan power at nominal design conditions is 0.0455 hp per nominal ton, which is typical for this type of tower. The figure shows that the fan electric energy consumption increases with range. This increase is because the fan remains on for longer periods of time during the winter, as shown in Figure 4.4. The potential savings, which depend on the differences between the various curves, are not strongly dependent on range for this climate. At each range evaluated for this tower, climate, and approach, the two-speed motor that can run at 2/3 speed consumes less energy during a typical year than the two-speed motor that can run at 1/2 speed. As expected the motor with the variable speed drive consumes the least energy of any fan control option. The advantage of the two-speed (full, 2/3) motor over the two-speed (full, 1/2) motor increases as the range increases.

An electric motor efficiency of 0.90 is factored into the yearly usage and savings data presented in this paper. Variable speed drive losses, belt drive losses, and the typically lower efficiency of a two-speed motor at the slower speed are neglected. These assumptions should not impact the results substantially. For a particular application, known efficiencies can easily be taken into account and incorporated in the calculations.

Table 4.1 lists the energy requirements for a single-speed fan motor and the annual savings for different capacity control options at several values of range and approach. The fan with a variable speed drive saves between 52% and 62% of the electricity of a fan that cycles on and off except in the two cases evaluated with approach above 7 °F. At approach of 20 °F and range of 10 °F, the variable speed drive savings drop to 35% of the single-speed fan usage. Comparing the savings available between the two-speed fan motor options listed in Table 1, the two-speed (full, 2/3) is better than the two-speed (full, 1/2) in every operating condition evaluated except one with savings as much as 76% higher. In the one case where the two-speed (full, 1/2) has the advantage, the savings are 5% higher. The potential for fan energy savings decreases at approach temperatures above about 12 °F, and increases slightly as the range increases. The opportunity for savings is a little higher for counter-flow towers than for cross-flow towers.

Table 4.2 shows the effects of load variations and tower design variations not considered in Table 4.1. All of the cases in Table 4.2 have an approach of 7 °F. Cases 1, 2, 5, and 6 have a range of 10 °F. The range is modified in Cases 3 and 4 as described. The savings listed are the difference between the electrical energy requirements of a tower operating with on-off fan cycling and a tower operating with a two-speed or variable speed fan. Case 1 is listed for reference purposes. It is a constant load case for an energy efficient tower. Case 2 is for a tower that has half as much airflow due to natural convection. The high potential savings for Case 2 demonstrate the importance of natural convection to energy efficiency.

Case 3 applies to a plant in which a manufacturing process has been eliminated so as to reduce the load on the cooling tower by one-third. As a result, the cooling tower is oversized, and the fan cycles on and off even during the hottest day of the summer. The results show that there is less benefit from two-speed or variable speed fans for this case. Interestingly, this operating condition favors a two-speed motor that operates at half of its rated speed.

In all of the cases considered to this point, the load has been held constant throughout the year. However, even with a constant heat load from process equipment, the load on the cooling tower may decrease in the winter due to line losses. To model this type of load, the entering hot water temperature was programmed to change in accordance with the ambient dry bulb temperature, Ta. For Case 4, a linear relationship between Tw_i and Ta is assumed. The load on the tower is reduced by one-half on the coldest day of the winter.

Case 5 is for a plant that operates one shift, five days per week. The cooling tower is assumed to operate between 7 AM and 5 PM. The results show that the potential energy savings are reduced roughly in proportion to the operating hours. Daytime wet-bulb temperatures are usually higher than at night so the tower energy usage and potential savings are slightly above a proportional reduction according to hours only (about 30% versus 24%).

Case 6 applies to a tower designed for low initial cost. The fan power for this case is 0.0625 hp per nominal ton, and c = 1.33. Opportunities for energy savings increase significantly for this type of tower compared to the more energy efficient tower.

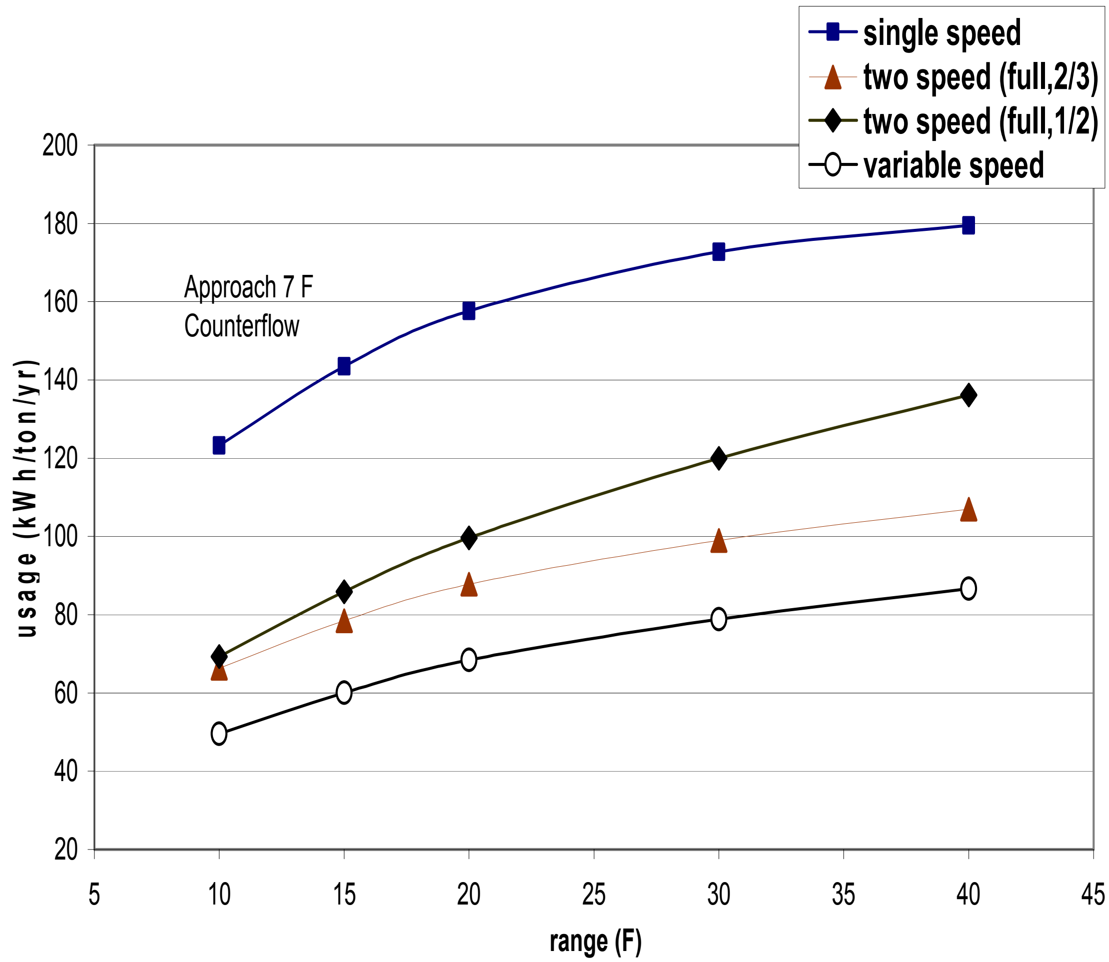

FIGURE 4.5 – ANNUAL FAN ENERGY CONSUMPTION, APPROACH = 7 °F
RALEIGH, NC

TABLE 4.1 – FAN ENERGY SAVINGS FOR CONSTANT LOAD
RALEIGH, NC

Type	**Approach** (F)	**Range** (F)	**single speed usage** (kWh/ton/yr)	**two-speed (2/3) savings** (kWh/ton/yr)	**two-speed (1/2) savings** (kWh/ton/yr)	**variable speed savings** (kWh/ton/yr)
counter-flow	7	10	123.2	56.9	53.8	73.7
counter-flow	7	15	143.5	65.0	57.5	83.4
counter-flow	7	20	157.6	69.8	57.9	89.1
counter-flow	7	30	172.7	73.7	52.8	93.8
counter-flow	7	40	179.5	72.5	43.4	92.9
counter-flow	4	10	92.9	48.4	51.0	63.1
counter-flow	7	10	123.2	56.9	53.8	73.7
counter-flow	12	10	155.9	58.7	47.8	76.4
counter-flow	20	10	190.1	49.5	28.1	66.9
cross-flow	7	10	95.0	46.4	45.6	59.1
cross-flow	7	15	110.0	53.6	50.0	67.6
cross-flow	7	20	121.5	58.7	52.4	73.7
cross-flow	7	30	136.5	64.3	53.0	80.5
cross-flow	7	40	143.2	66.2	50.3	82.6

TABLE 4.2 – FAN ENERGY SAVINGS FOR COUNTERFLOW TOWERS
RALEIGH, NC

Case	**Approach** (F)	**Range** (F)	**single speed usage** (kWh/ton/yr)	**two-speed (2/3) savings** (kWh/ton/yr)	**two-speed (1/2) savings** (kWh/ton/yr)	**variable speed savings** (kWh/ton/yr)	**Description**
1	7	10	123.2	56.9	53.8	73.7	reference
2	7	10	150.7	76.6	79.0	101.2	low natural conv.
3	7	10	70.8	36.6	42.0	48.8	oversized
4	7	10	93.3	44.2	43.0	57.1	weather dep. load
5	7	10	39.0	17.1	15.8	22.3	operating hours
6	7	10	202.9	118.0	135.2	159.9	low first cost

4.3 Similar Applications in Various Climates

The impact of climate on cooling tower fan energy usage and potential energy savings is estimated using typical meteorological year data for five locations listed in Table 4.3. The fan energy savings in this table represent a tower characterized by Equation (6) with $c = 3$ and $n = 0.40$. The heat transfer by natural convection is calculated from Equation (10), assuming $C_0 = 0.134$. The fan power is 0.0455 hp/ton. In each case, the load is held constant throughout the year. It is assumed that the tower is correctly sized for the load so that the fan does not cycle at design point conditions. The design point wet-bulb temperature for each city is listed in the table.

The energy savings in Table 4.3 are for a counter-flow tower that operates with a steady load for 8760 hours per year. As an approximation, the savings for towers that operate fewer hours can be reduced in proportion to the operating hours. For the towers and operating conditions represented in Table 4.3, fan motors that operate at 2/3 speed save more energy than motors that operate at 1/2 speed. The advantage of the two-speed (full, 2/3) fan motors over the two-speed (full, 1/2) increases as the range increases, as the approach increases, and is more pronounced in some locations.

For variable speed motors and for motors that run at 2/3 speed, the potential savings are insensitive to the approach temperature in every location. However, it is expected that the savings would decrease if the approach were to exceed 12 °F. In Los Angeles and

Houston, the potential savings are insensitive to range. In Raleigh, and to a larger extent, in Columbus and Denver, the savings increase with range.

The cities in Table 4.3 are arranged in order of decreasing design wet-bulb temperature. By examination of the results, it is apparent that the single-speed usage and potential savings for various fan speed options is unrelated to the design wet-bulb temperature. In order to study the effect of climate on the results, the mean wet-bulb temperature for the typical meteorological year data was calculated and compared to the design wet-bulb temperature at each location. The location with the smallest difference between mean wet-bulb and design wet-bulb and consequently the highest single speed energy consumption is Los Angeles. Figure 4.6 is a plot of single-speed usage from Table 4.3 versus location with the locations arranged in order of increasing difference between mean wet-bulb temperature and design wet-bulb temperature. This plot shows a good correlation between cooling tower single-speed fan motor consumption and the difference between design wet-bulb temperature and mean wet-bulb temperature for a particular location.

Figure 4.7 is a plot of potential fan energy savings at nominal conditions versus location. The locations are arranged in order of increasing difference between design wet-bulb temperature and mean wet-bulb (increasing single-speed fan motor consumption). This plot shows that for this tower and operating conditions the locations with the highest consumption generally have the highest potential savings and the type of two-speed fan with the most energy savings depends on location.

Figure 4.8 plots the potential savings for alternative capacity control methods for various locations when the range is 40 °F and the approach is 7 °F. The fan motors that operate at 2/3 speed are more energy efficient than the motors that operate at 1/2 speed for each location. The plot shows that in general the advantage for the 2/3 speed motor over the 1/2 speed motor increases as the range increases.

In Figure 4.8, the locations are again arranged in order of increasing difference between design wet-bulb temperature and mean wet-bulb (increasing single-speed fan motor consumption). Under these operating conditions, the correlation between the most savings being available for the towers with the highest consumption begins to break down. In particular, the potential savings for Houston are much less than the other locations as a percentage of single-speed fan motor usage. To investigate, the standard deviation of the hourly wet-bulb temperatures was calculated for each location's typical meteorological year data and compared to the difference between the design wet-bulb temperature and the mean wet-bulb temperature. The location where the standard deviation is closest to the difference in design and mean wet-bulb is Houston. The standard deviation being close to the difference in design and mean wet-bulb suggests that the wet-bulb temperature is relatively often near the design wet-bulb and, likewise, relatively often far below the design wet-bulb. These conditions limit the amount of savings available for the fan speed control options. In Figure 4.9, the distributions of hourly wet-bulb temperatures during a typical year are shown for the subject climates with the highest (Columbus) and the lowest (Houston) percentage potential savings based on single-speed fan consumption.

Figure 4.10 plots the variable speed savings as a percentage of single-speed motor consumption for various operating conditions with the locations arranged in order of increasing difference between design wet-bulb minus mean wet-bulb and standard deviation. This plot shows a good correlation between percent savings and this parameter. Also, where the wet-bulb is relatively often near the design wet-bulb, the two-speed (full, 2/3) should have the advantage over the two-speed (full, 1/2). Results in Table 4.3 show that the locations where the two-speed (full, 2/3) is most advantageous are Houston and Los Angeles.

Figure 4.11 shows that the 1/2 speed fan is better than the 2/3 speed fan for cooling towers designed for low initial costs with moderate loads (range = 10 F). However, as the range increases, the 2/3 speed fan becomes more advantageous relative to the 1/2 speed fan. Figure 4.12 shows that the best choice for towers designed for low initial cost is dependent on location for high loads (range = 40 F).

The savings in Table 4.3 are in the range of 50 to 100 kWh/ton/yr. For a fan power of 0.0455 hp/ton, a small tower with a 1 hp fan would have a nominal capacity of about 22 tons. Thus, the electrical energy savings for the small tower would be in the range of 1100 to 2200 kWh/yr. If electricity cost $.05/kWh, the dollar savings are in the range of $55/yr to $110/yr. The dollar savings for a 1650-ton tower with a 75 hp motor are in the range of $4100/yr to $8200/yr.

TABLE 4.3 – FAN ENERGY SAVINGS FOR VARIOUS LOCATIONS

City	Design Wet Bulb (F)	Approach (F)	Range (F)	single speed usage (kWh/ton/yr)	two-speed (2/3) savings (kWh/ton/yr)	two-speed (1/2) savings (kWh/ton/yr)	variable speed savings (kWh/ton/yr)
Houston	79	7	10	160.5	59.9	50.8	80.3
	79	7	40	204.4	60.9	32.4	84.6
	79	12	10	189.1	53.2	38.0	73.0
Raleigh	78	7	10	123.2	56.9	53.8	73.7
	78	7	40	179.5	72.5	43.4	92.9
	78	12	10	155.9	58.7	47.8	76.4
Columbus	75	7	10	111.1	50.8	52.8	67.4
	75	7	40	172.3	75.6	50.1	93.2
	75	12	10	145.1	58.2	51.5	75.0
Los Angeles	69	7	10	168.4	79.4	68.6	99.6
	69	7	40	219.1	84.8	42.7	109.1
	69	12	10	196.6	73.1	46.1	91.1
Denver	63	7	10	134.0	60.2	60.1	78.9
	63	7	40	194.8	84.1	59.6	104.9
	63	12	10	165.7	64.3	57.0	84.1

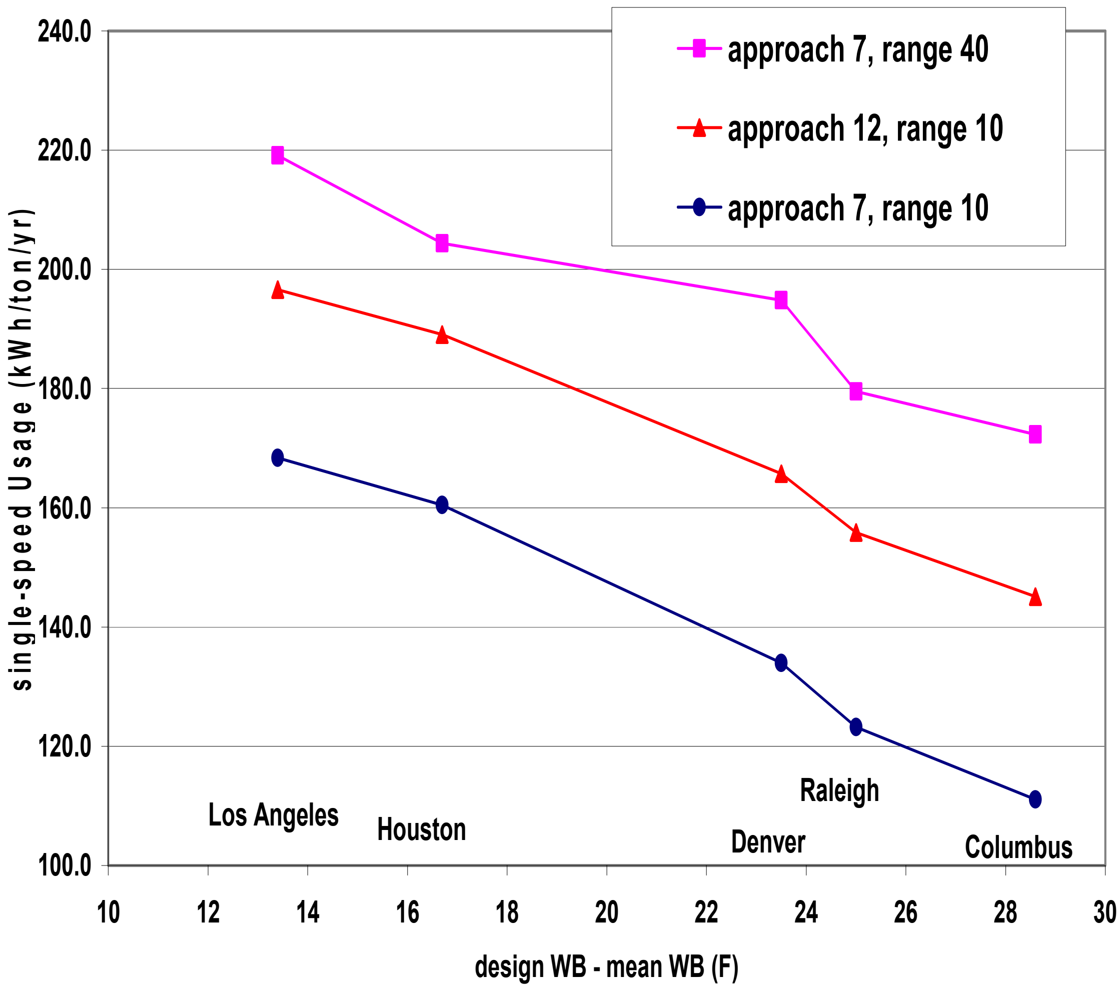

FIGURE 4.6 - SINGLE-SPEED FAN ENERGY USAGE FOR VARIOUS LOCATIONS

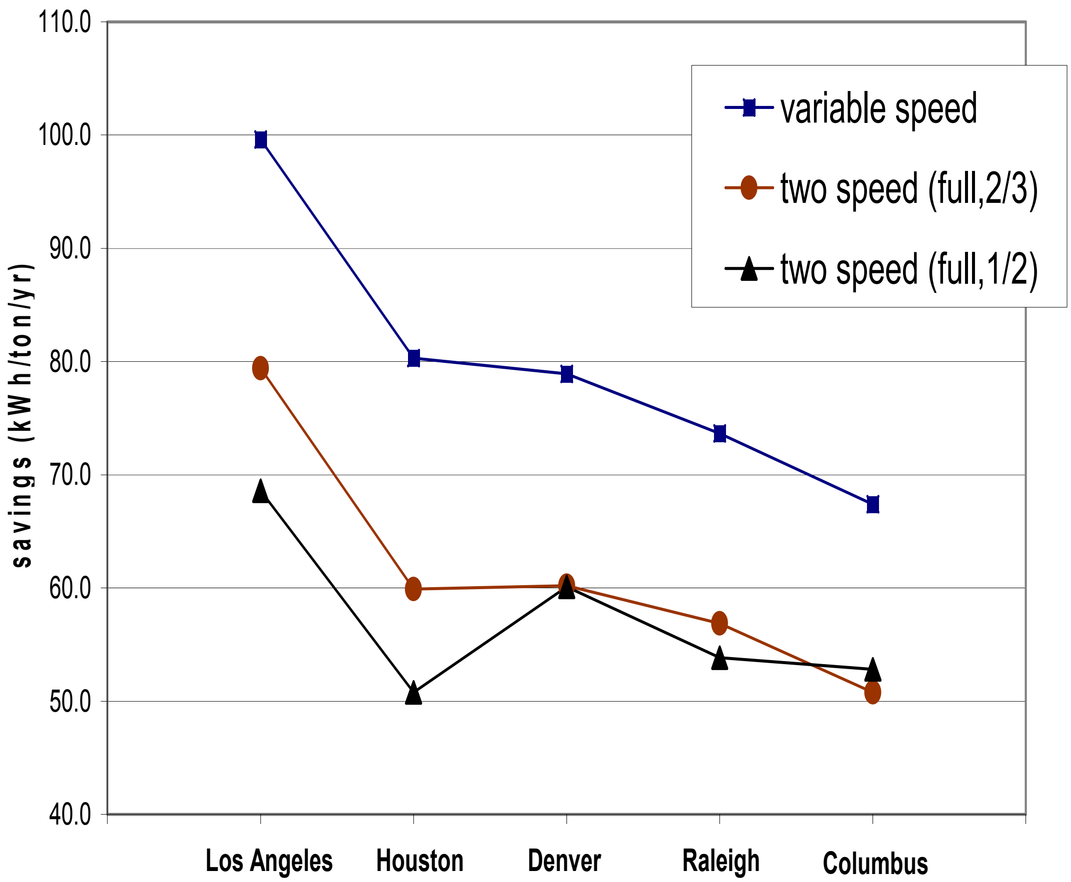

FIGURE 4.7 - FAN ENERGY SAVINGS, RANGE =10 °F, APPROACH =7 °F, c=3, n=0.4, C_o=0.134

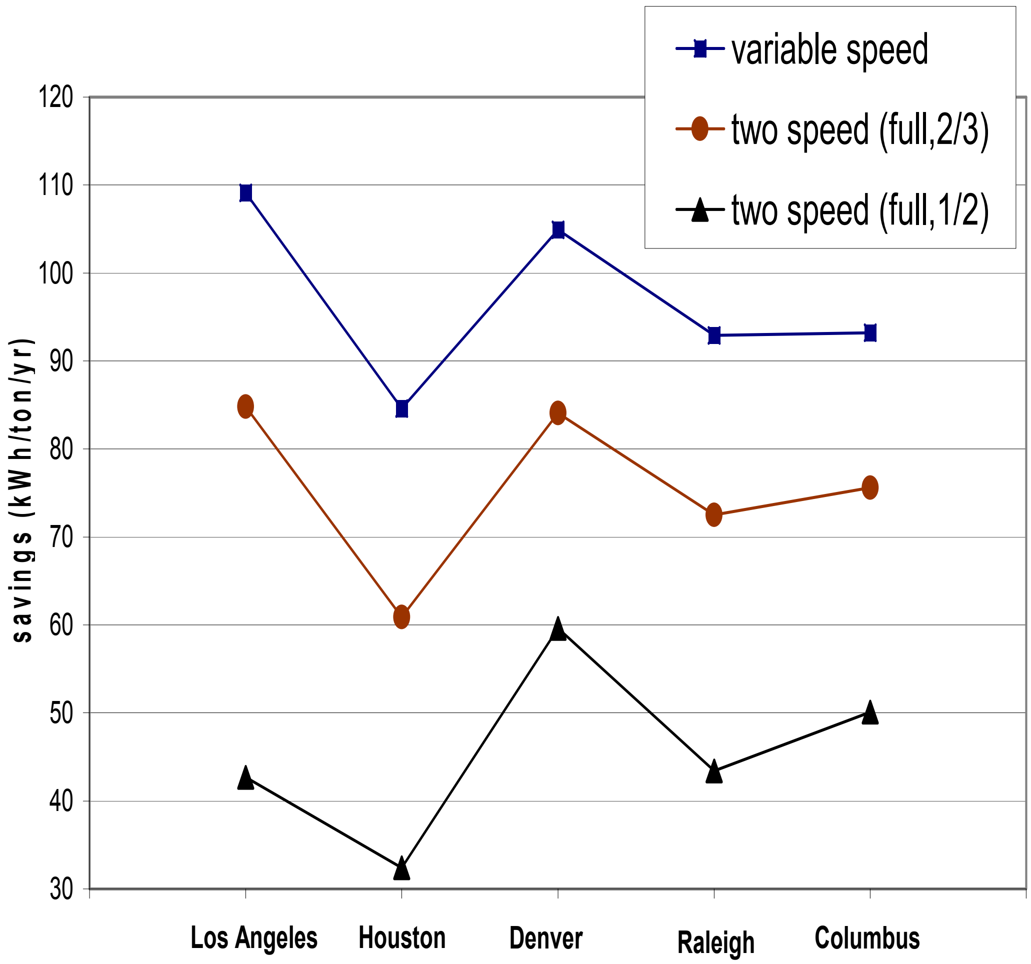

FIGURE 4.8 - FAN ENERGY SAVINGS, RANGE =40 °F, APPROACH =7 °F, c=3, n=0.4, C_o=0.134

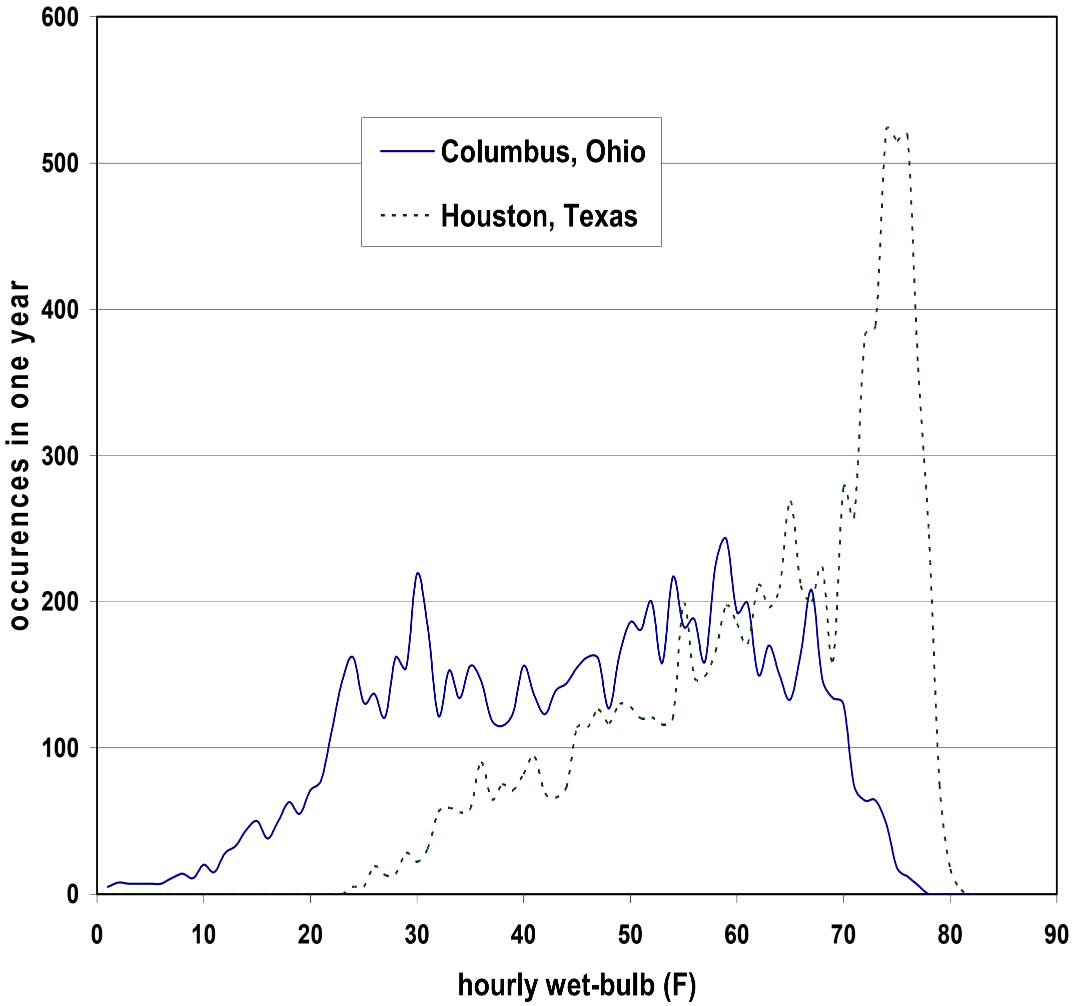

FIGURE 4.9 - WET-BULB DISTRIBUTION FOR TYPICAL YEAR IN HOUSTON AND COLUMBUS

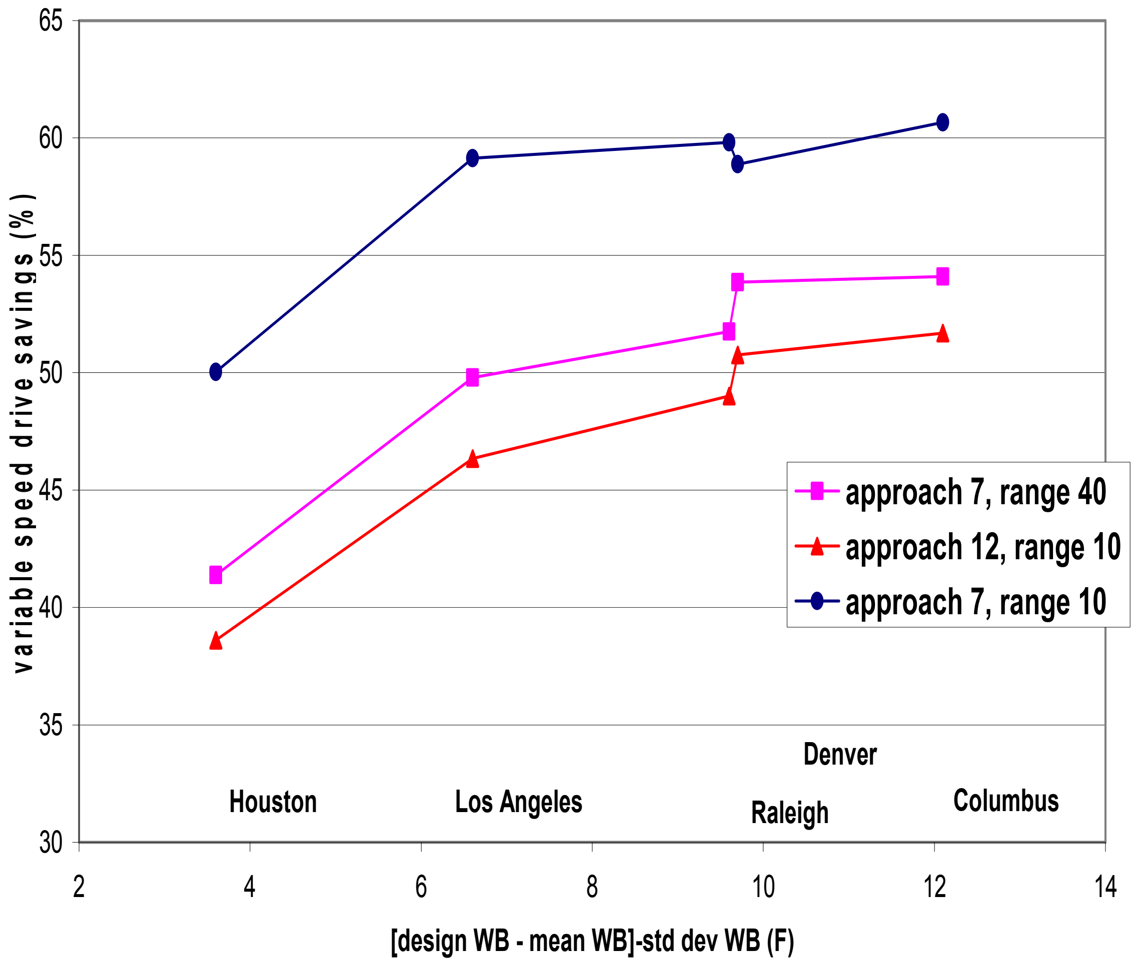

FIGURE 4.10 - VARIABLE SPEED DRIVE SAVINGS AS PERCENTAGE OF SINGLE-SPEED CONSUMPTION FOR VARIOUS LOCATIONS AND OPERATING CONDITIONS

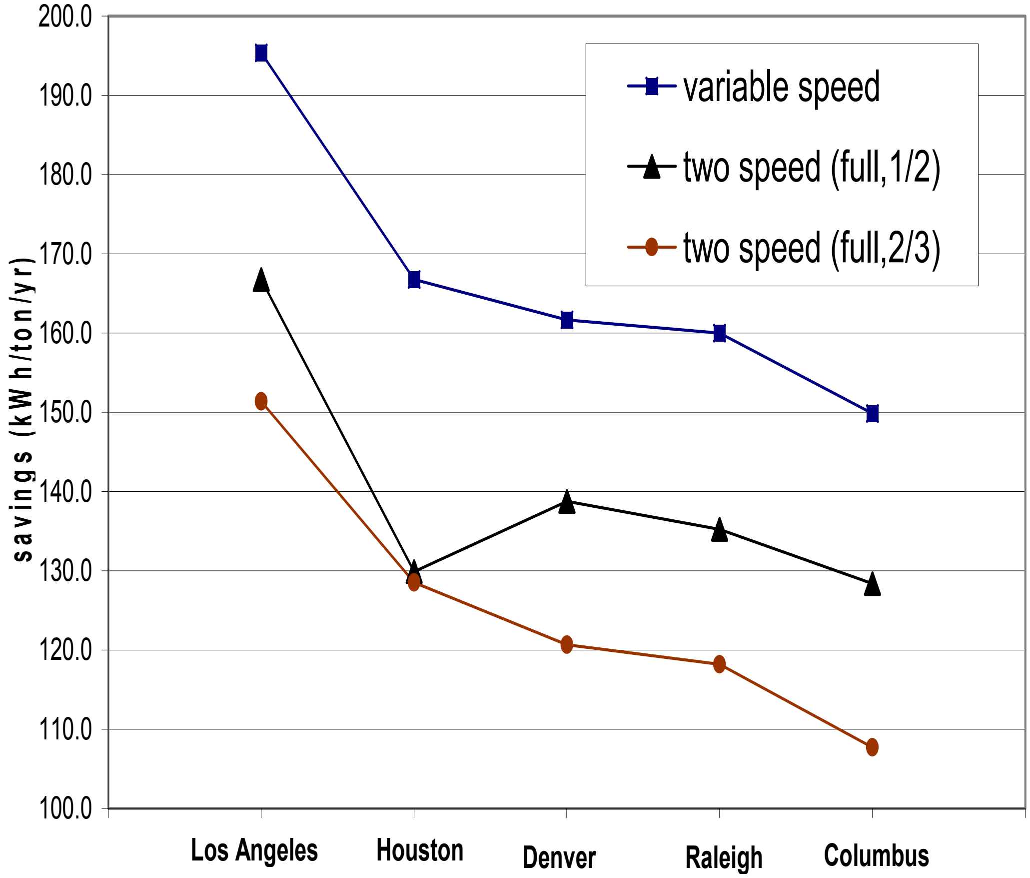

FIGURE 4.11 - FAN ENERGY SAVINGS, RANGE =10 °F, APPROACH =7 °F, c=1.33, n=0.4, C_o=0.056

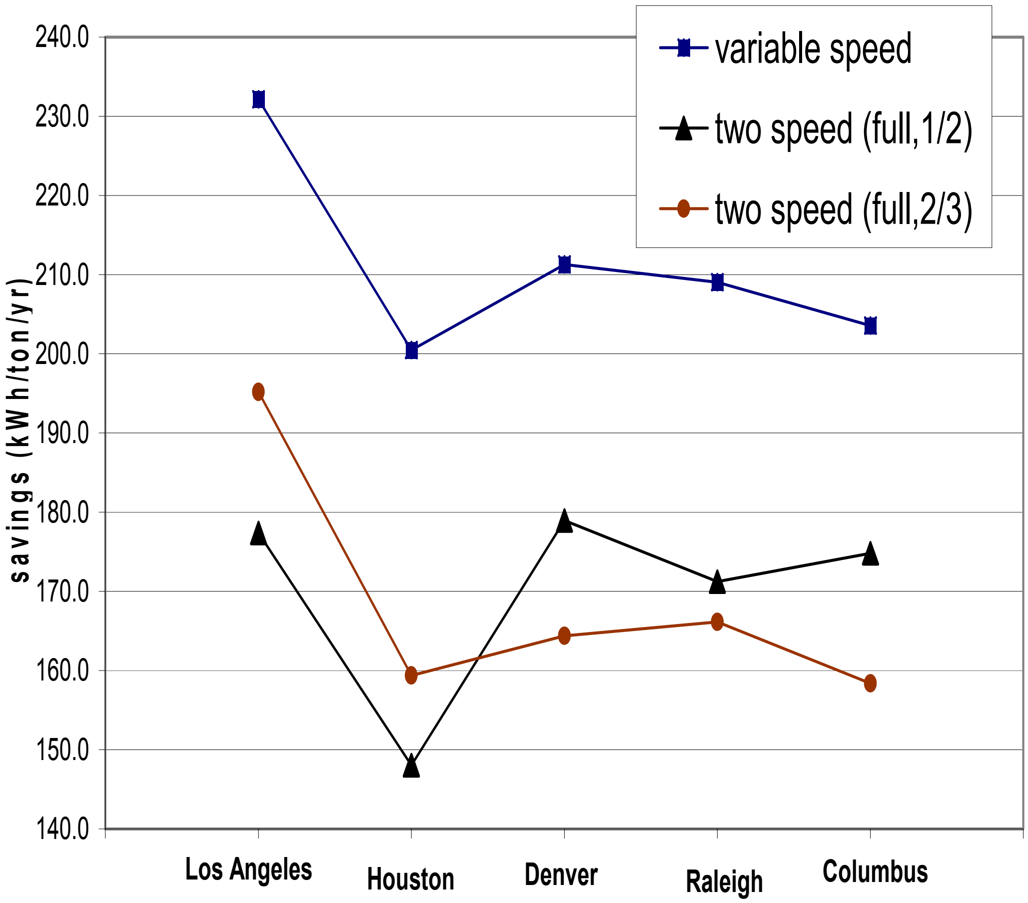

FIGURE 4.12 - FAN ENERGY SAVINGS, RANGE =40 °F, APPROACH =7 °F, c=1.33, n=0.4, C_o=0.056

4.4 Ideal Second Speed

The parametric analysis of the effects of both the climate and the operating conditions (load, approach) has shown a clear advantage of different speeds in certain applications. These findings suggest that an ideal second speed can be found that maximizes savings for the two-speed fan control option. Two-speed motors are readily available only in discrete steps with the slower speed usually 1/2 or 2/3 of full speed. This fact limits from a practical standpoint the slower speed that is used. Typically the slower speed in commercial cooling towers is either 1/2 or 2/3 full speed depending on the manufacturer. There is one design that allows selection of any speed easily. This design uses a second independent motor called a pony motor that is connected to drive the fan shaft by a separate belt and pulley. With this configuration, the cooling tower could be tuned to a specific application in a specific climate by selecting the ideal second speed. To investigate the dependence of savings on the selected second speed, energy usages were calculated over a range of second speeds for the nominal case and several cases for which savings seemed to be particularly sensitive to the slower speed. Figure 4.13 shows that the ideal second speed for a two-speed fan varied from 1/2 to 3/4 for the particular towers and applications represented. It is expected that the ideal second speed would almost always be in this range.

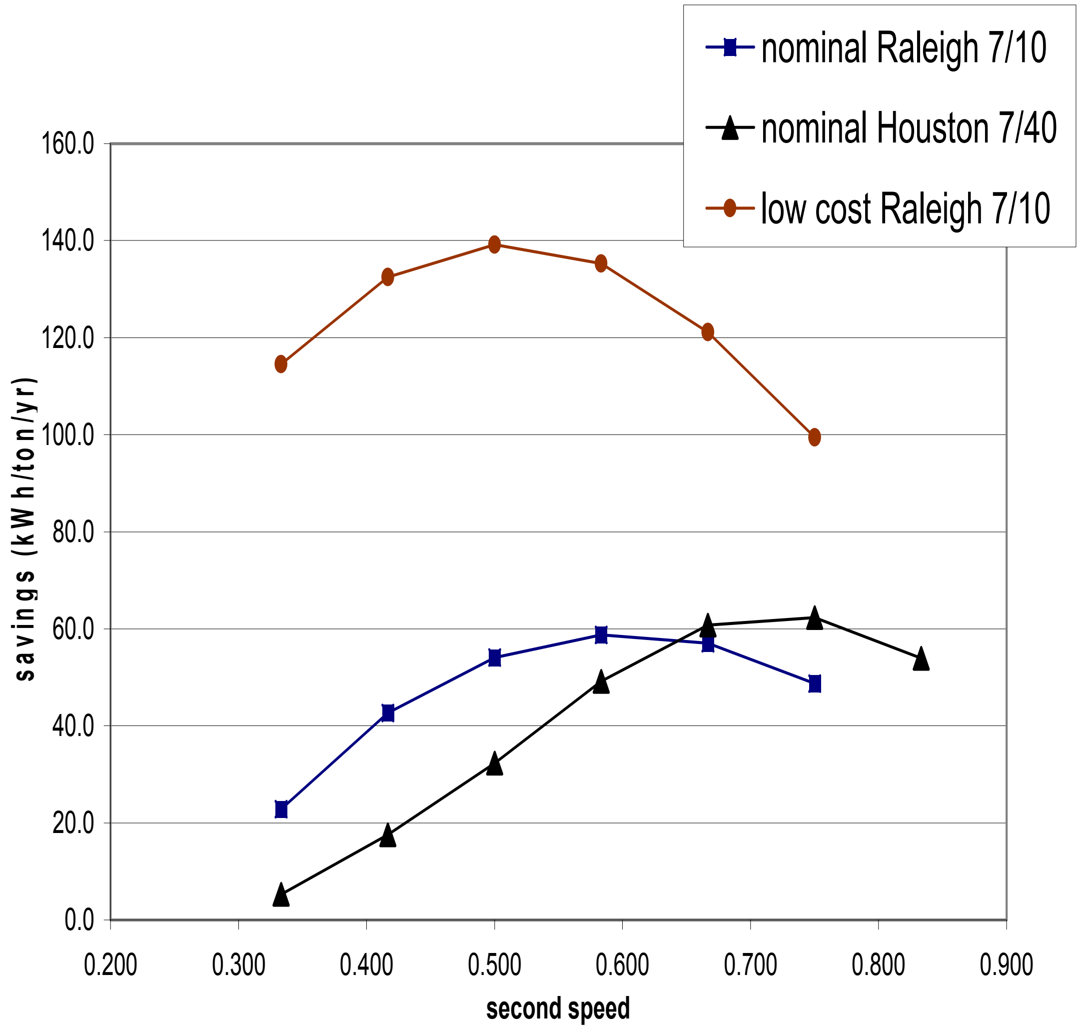

FIGURE 4.13 - IDEAL SECOND SPEED

4.5 Comparison of Equipment Cost and Estimated Savings

The cost for installing two-speed motors or variable speed drives on cooling towers depends somewhat on the cooling tower capacity. A single-speed 1hp motor for a small tower might cost $150/hp, while a 75 hp motor for a large tower might cost only $50/hp. Two-speed motors can be expected to cost twice as much. Variable speed drives will probably cost $800/hp for small motors and $200/hp for large motors.

Thus, the cost to replace a 1 hp single-speed fan motor with a two-speed motor would probably be in the range of $500. The payback period is 5 to 10 years. The cost to replace the single-speed 75 hp motor is probably in the range of $7500, and the payback period is in the range of 1 to 2 years. For smaller towers with belt driven fans, the most cost effective solution is to install a second motor to drive the fan at slower speed. For example, a tower with a 25 hp motor would require a 7.5 hp motor to drive the fan at 2/3 speed. The economics will be better in locations where the cost of electricity is higher than $.05/kWh, and for towers with small heat transfer areas for which the nominal fan power would be considerably greater than 0.0455 hp/ton. The economics will be worse for plants that operate one shift and in plants where the tower is oversized.

5.0 Conclusions

Calculations based on typical meteorological year data show that fan energy savings for alternative capacity control methods do not depend strongly on the approach temperature. In colder climates, the potential savings increase by 25% to 40% when the range increases from 10 °F to 40 °F. The potential savings are strongly dependent on the amount of natural convection that occurs when the fan is off. The greatest potential for savings occurs in towers designed for low cost. The fan power for such a tower would be in the range of 0.08 hp/nominal-ton, and the airflow due to natural convection would be small in comparison to the fan capacity. A tower with a fan power of 0.04 hp/ton benefits more from natural convection, and has less potential for energy savings. There is less potential for savings in cross-flow towers than in counter-flow towers. Also, the potential savings are lower when the cooling tower is oversized, or when the plant operates one shift instead of three shifts.

Two-speed fans that operate at 1/2 of the rated speed are suitable for low cost towers at moderate loads. Two- speed fans that operate at 2/3 of the rated speed are a better choice for energy efficient towers in most locations, especially at higher operating loads. At nominal conditions of approach = 7 °F and range = 10 °F, the single-speed energy usage and thus the potential savings are highest in locations where the wet-bulb temperature remains close to the design value through much of the year. The potential energy savings at nominal conditions in Los Angeles are about 50% higher than the savings in Columbus, Ohio. As a percentage of the single-speed fan energy usage, potential savings are a function of the distribution of the wet-bulb temperatures during the year.

6.0 References

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APPENDICES

7.1 Fortran Model for Counter-flow Cooling Tower - Single Operating Condition

```
c     Single Point Operation - Counterflow
      dimension tout(4)
  100 format(5F8.3)
      open(unit=11,file="data",status="old")
      write(*,*) 'enter range, approach, ratio,designWB'
      read(*,*) range,approach,ratio,twbd
      write(11,*) 'approach',approach,'    range',range
      write(11,*) 'design wet bulb',twbd
      write(11,*) '    wetbulb-hp1-hp23-hp12-hpv'
      two=twbd+approach
      twi=two+range
      do 10 i=1,53
      twb=24.+i
      tdb=twb
      tdp=twb
c      write(*,*)' enter tdb,tdp,twi,ratio'
c      read(*,*) tdb,tdp,twi,ratio
      call cycle(tdb,tdp,twi,ratio,tout)
c      write(*,*) 'two ',tout(1),tout(2),tout(3),tout(4)
      f=(two-tout(1))/(tout(4)-tout(1))
      hp1=f
      IF (two.lt.tout(3)) THEN
           f=(two-tout(3))/(tout(4)-tout(3))
         hp23=f+(1-f)*(.667**3.)
      ELSE
           f=(two-tout(1))/(tout(3)-tout(1))
         hp23=f*(.667**3.)
      END IF
      IF (two.lt.tout(2)) THEN
           f=(two-tout(2))/(tout(4)-tout(2))
         hp12=f+(1-f)*(.5**3.)
      ELSE
           f=(two-tout(1))/(tout(2)-tout(1))
         hp12=f*(.5**3.)
      END IF
      call vspeed(tdb,tdp,twi,ratio,two,f)
      hpv=f**3.
   10 write(11,100) twb,hp1,hp23,hp12,hpv
      stop
      end
      subroutine vspeed(tai,tdp,twi,ratio,twout,f)
      dimension cfm(2),diff(2)
      astd=990.
      wmdoti=astd/ratio
      c=3.
      call psych(tai,tdp,wai,hai)
      call psych(twi,twi,wwi,hwi)
      cfm(1)=13400.
      cfm(2)=1340.
      count=0.
 5    count=count+1.
      do 20 i=1,2
      v=.02519*(1.+1.61*wai)*(tai+460.)/(1.+wai)
      do 10 k=1,3
      amdot=cfm(i)/v
```

```
      xntu=c*(wmdoti/amdot)**.4
      call ctower(amdot,tai,tdp,wmdoti,twi,xntu,two,wmdoto,wao,hao)
      call psych(two,two,wwo,hwo)
      twbo=14.24+1.43*hao
10    v=.02519*(1.+1.61*wao)*(twbo+460.)/(1.+wao)
20    diff(i)=twout-two
      cfm(2)=(cfm(2)*diff(1)-cfm(1)*diff(2))/(diff(1)-diff(2))
      cfm(1)=.98*cfm(2)
      test=abs(diff(2))
      if(count.gt.10) write(*,*)'vspeed failed'
      if(count.gt.10) go to 30
      if(test.gt..01) go to 5
30    f=cfm(2)/13400.
      return
      end
      subroutine cycle(tai,tdp,twi,ratio,tout)
      dimension tout(4),cfm(4)
      astd=990.
      wmdoti=astd/ratio
      xntu=3.
      c=3.
      call psych(tai,tdp,wai,hai)
      call psych(twi,twi,wwi,hwi)
      dh=hwi-hai
      v=.02519*(1.+1.61*wai)*(tai+460.)/(1.+wai)
      do 10 k=1,3
      cfm0=2000.*dh**.20
      amdot=cfm0/v
      call ctower(amdot,tai,tdp,wmdoti,twi,xntu,two,wmdoto,wao,hao)
      call psych(two,two,wwo,hwo)
      twbo=14.24+1.43*hao
      v=.02519*(1.+1.61*wao)*(twbo+460.)/(1.+wao)
      dti=hwo-hai
      dto=hwi-hao
      dh=(dti-dto)/alog(dti/dto)
10    tout(1)=two
      cfm(2)=13200./2.
      cfm(3)=13200.*2./3.
      cfm(4)=13200.
      do 30 n=2,4
      v=.02519*(1.+1.61*wai)*(tai+460.)/(1.+wai)
      do 20 k=1,3
      amdot=cfm(n)/v
      xntu=c*(wmdoti/amdot)**.4
      call ctower(amdot,tai,tdp,wmdoti,twi,xntu,two,wmdoto,wao,hao)
      twbo=14.24+1.43*hao
20    v=.02519*(1.+1.61*wao)*(twbo+460.)/(1.+wao)
30    tout(n)=two
      return
      end
      subroutine psych(ta,tdp,w,h)
      psat=exp(14.8305-7362.08/(tdp+394.67))
      y=psat/14.696
      w=.6219*y/(1.-y)
      h=.249*(ta-2.5)+1100.*w
```

```
      return
      end
      subroutine ctower(amdot,tai,tdp,wi,twi,xntu,two,wo,wao,hao)
      dimension tsw(2),diff(2)
      wmdoti=wi
      call psych(tai,tdp,wai,hai)
      call psych(twi,twi,wswi,hswi)
      tsw(1)=(tai+tdp)/2.
      tsw(2)=tsw(1)+1.
      count=0.
 10   count=count+1
      do 20 i=1,2
      twb=tsw(i)
      call psych(twb,twb,wwb,hwb)
 20   diff(i)=hai-hwb
      tsw(2)=(tsw(1)*diff(2)-tsw(2)*diff(1))/(diff(2)-diff(1))
      tsw(1)=tsw(2)-1.
      if(count.gt.10) write(*,*) 'twb failed'
      if(count.gt.10) go to 30
      test=abs(diff(2))/hai
      if(test.gt..0005) go to 10
 30   two=twb
c      write(*,*)' twb,hwb =',twb,hwb
c      write(*,*)'approx water outlet temp. =',two
      do 70 j=1,5
      call psych(two,two,wswo,hswo)
      cs=(hswi-hswo)/(twi-two)
      smdot=amdot*cs/wmdoti
      texp=exp(xntu*(smdot-1.))
      eff=(1.-texp)/(1.-smdot*texp)
      hao=hai+eff*(hswi-hai)
      hswe=hai+(hao-hai)/(1.-exp(-xntu))
      tsw(1)=(twi+two)/2.
      tsw(2)=tsw(1)+1.
      count=0.
 40   count=count+1
      do 50 i=1,2
      ta=tsw(i)
      call psych(ta,ta,wswe,h)
 50   diff(i)=hswe-h
      tsw(2)=(tsw(1)*diff(2)-tsw(2)*diff(1))/(diff(2)-diff(1))
      tsw(1)=tsw(2)-1.
      if(count.gt.10) write(*,*) 'hswe failed'
      if(count.gt.10) go to 60
      test=abs(diff(2))/hswe
      if(test.gt..0005) go to 40
 60   wao=wswe+(wai-wswe)/exp(xntu)
      wmdoto=wmdoti-amdot*(wao-wai)
 70   two=32.+(wmdoti*(twi-32.)-amdot*(hao-hai))/wmdoto
      eff2=(twi-two)/(twi-twb)
      evap=(wmdoti-wmdoto)/wmdoti
      wo=wmdoto
      return
      end
```

7.2 Fortran Model for Cross-flow Cooling Tower – Yearly Energy Usage

```
***********************************************************************
******PROGRAM ESTIMATES YEARLY COOLING TOWER FAN ENERGY REQUIREMENTS USING
******DIFFERENT CAPACITY CONTROL OPTIONS - SINGLE SPEED MOTOR, TWO SPEED
******MOTORS (ONE-HALF AND TWO-THIRDS SPEED), AND VARIABLE SPEED
      dimension tout(4),hp(4)
  100 format(I4,3F8.2)
******ACCESS HOURLY TMY DATA FILE AND SET DESIGN PARAMETERS FOR THE
******COOLING TOWER, SUM VARIABLES USED TO SUM HP REQUIREMENTS
      open(unit=11,file="hou.dat",status="old")
      write(*,*) 'enter range, approach, ratio,designWB'
      read(*,*) range,approach,ratio,twbd
      sum1=0.
      sum23=0.
      sum12=0.
      sumv=0.
      two=twbd+approach
      twi=two+range
      do 10 i=1,8760
      read(11,100) nhr,tdb,tdp,twb
      call cycle(tdb,tdp,twi,ratio,tout,hp)
      IF (tout(1).lt.two) THEN
        hp(1)=0.
        hp(2)=0.
        hp(3)=0.
        hp(4)=0.
      END IF
      f=(two-tout(1))/(tout(4)-tout(1))
      hp1=f*hp(4)
      sum1=hp1+sum1
      IF (two.lt.tout(3)) THEN
         f=(two-tout(3))/(tout(4)-tout(3))
         hp23=f*hp(4)+(1-f)*hp(3)
      ELSE
         f=(two-tout(1))/(tout(3)-tout(1))
         hp23=f*hp(3)
      END IF
      sum23=hp23+sum23
      IF (two.lt.tout(2)) THEN
         f=(two-tout(2))/(tout(4)-tout(2))
         hp12=f*hp(4)+(1-f)*hp(2)
      ELSE
         f=(two-tout(1))/(tout(2)-tout(1))
         hp12=f*hp(2)
      END IF
      sum12=hp12+sum12
      call vspeed(tdb,tdp,twi,ratio,two,f,hpv)
      IF (tout(1).lt.two) THEN
          hpv=0.
      END IF
   10 sumv=hpv+sumv
      write(*,*) '  design wet bulb  approach  range hours'
      write(*,*) twbd,approach,range,nhr
      write(*,*) '  sum1  sum23   sum12 sumv  '
      write(*,*) sum1,sum23,sum12,sumv
      stop
```

```
      end
**************************************************************************
******SUBROUTINE TO DETERMINE HP REQUIREMENT FOR VARIABLE SPEED FAN MOTOR
******OPTION.  USES TRIAL AND ERROR TO FIND CFM THAT WILL MATCH DESIRED
******COLD WATER OUT TEMPERATURE.  ASSUMES FAN LAW TO DETERMINE HP
******REQUIREMENT.
      subroutine vspeed(tai,tdp,twi,ratio,twout,f,hpv)
      dimension cfm(2),diff(2)
      astd=990.
      wmdoti=astd/ratio
      c=3.
      call psych(tai,tdp,wai,hai)
      call psych(twi,twi,wwi,hwi)
      cfm(1)=13200.
      cfm(2)=1320.
      count=0.
 5    count=count+1.
      do 20 i=1,2
      v=.02519*(1.+1.61*wai)*(tai+460.)/(1.+wai)
      do 10 k=1,3
      amdot=cfm(i)/v
      xntu=c*(wmdoti/amdot)**.4
      call ctower(amdot,tai,tdp,wmdoti,twi,xntu,two,wmdoto,wao,hao)
      call psych(two,two,wwo,hwo)
      twbo=14.24+1.43*hao
 10   v=.02519*(1.+1.61*wao)*(twbo+460.)/(1.+wao)
 20   diff(i)=twout-two
      cfm(2)=(cfm(2)*diff(1)-cfm(1)*diff(2))/(diff(1)-diff(2))
      cfm(1)=.98*cfm(2)
      test=abs(diff(2))
      if(count.gt.10) write(*,*)'vspeed failed'
      if(count.gt.10) go to 30
      if(test.gt..01) go to 5
 30   f=cfm(2)/13200.
      hpv=13.33/v*f**3.
      return
      end
**************************************************************************
******SUBROUTINE TO DETERMINE HP REQUIREMENT FOR SINGLE SPEED AND TWO
******SPEED FANS.  DETERMINES COLD WATER TEMPERATURE FOR THE PERTINENT CFM
******BASED ON SPEEDS AND THE FAN LAWS.  SEEKS AVERAGE COLD WATER
******TEMPERATURE TO MATCH DESIRED COLD WATER TEMPERATURE.  ASSUMES FAN
******LAW TO DETERMINE HP REQUIREMENT
      subroutine cycle(tai,tdp,twi,ratio,tout,hp)
      dimension tout(4),cfm(4),hp(4)
      astd=990.
      wmdoti=astd/ratio
      xntu=3.2
      c=3.
      call psych(tai,tdp,wai,hai)
      call psych(twi,twi,wwi,hwi)
      dh=hwi-hai
      v=.02519*(1.+1.61*wai)*(tai+460.)/(1.+wai)
      do 10 k=1,3
      cfm0=2000.*dh**.20
```

```
      amdot=cfm0/v
      call ctower(amdot,tai,tdp,wmdoti,twi,xntu,two,wmdoto,wao,hao)
      call psych(two,two,wwo,hwo)
      twbo=14.24+1.43*hao
      v=.02519*(1.+1.61*wao)*(twbo+460.)/(1.+wao)
      dti=hwo-hai
      dto=hwi-hao
      dh=(dti-dto)/alog(dti/dto)
 10   tout(1)=two
      cfm(2)=13200.*6./12.
      cfm(3)=13200.*8./12.
      cfm(4)=13200.
      do 30 n=2,4
      v=.02519*(1.+1.61*wai)*(tai+460.)/(1.+wai)
      do 20 k=1,3
      amdot=cfm(n)/v
      xntu=c*(wmdoti/amdot)**.4
      call ctower(amdot,tai,tdp,wmdoti,twi,xntu,two,wmdoto,wao,hao)
      twbo=14.24+1.43*hao
 20   v=.02519*(1.+1.61*wao)*(twbo+460.)/(1.+wao)
      hp(n)=13.33/v*(cfm(n)/13200)**3
 30   tout(n)=two
      return
      end
************************************************************************
******SUBROUTINE TO DETERMINE HUMIDITY RATIO AND ENTHALPY OF AIR BASED ON
******EMPIRICAL RELATIONSHIPS GIVEN AMBIENT TEMPERATURE AND DEWPOINT
******TEMPERATURE
      subroutine psych(ta,tdp,w,h)
      psat=exp(14.8305-7362.08/(tdp+394.67))
      y=psat/14.696
      w=.6219*y/(1.-y)
      h=.249*(ta-2.5)+1100.*w
      return
      end
************************************************************************
******SUBROUTINE TO DETERMINE COOLING TOWER EXIT WATER TEMPERATURE, EXIT
******WATER MASS FLOWRATE, EXIT AIR SPECIFIC HUMIDITY, AND EXIT AIR
******ENTHALPLY GIVEN ENTERING WATER TEMPERATURE, AMBIENT AIR CONDITIONS,
******COOLING TOWER NTU'S, AND ENTERING WATER AND AIR MASS FLOWRATES
      subroutine ctower(amdot,tai,tdp,wi,twi,xntu,two,wo,wao,hao)
      dimension tsw(2),diff(2)
      wmdoti=wi
      call psych(tai,tdp,wai,hai)
      call psych(twi,twi,wswi,hswi)
      tsw(1)=(tai+tdp)/2.
      tsw(2)=tsw(1)+1.
      count=0.
 10   count=count+1
      do 20 i=1,2
      twb=tsw(i)
      call psych(twb,twb,wwb,hwb)
 20   diff(i)=hai-hwb
      tsw(2)=(tsw(1)*diff(2)-tsw(2)*diff(1))/(diff(2)-diff(1))
      tsw(1)=tsw(2)-1.
```

```
      if(count.gt.10) write(*,*) 'twb failed'
      if(count.gt.10) go to 30
      test=abs(diff(2))/hai
      if(test.gt..0005) go to 10
 30   two=twb
c      write(*,*)' twb,hwb =',twb,hwb
c      write(*,*)'approx water outlet temp. =',two
      do 70 j=1,5
      call psych(two,two,wswo,hswo)
      cs=(hswi-hswo)/(twi-two)
      smdot=amdot*cs/wmdoti
c      texp=exp(xntu*(smdot-1.))
c      eff=(1.-texp)/(1.-smdot*texp)
      texp=exp(-xntu)
      texp=exp(smdot*(texp-1.))
      eff=(1.-texp)/smdot
      hao=hai+eff*(hswi-hai)
      hswe=hai+(hao-hai)/(1.-exp(-xntu))
      tsw(1)=(twi+two)/2.
      tsw(2)=tsw(1)+1.
      count=0.
 40   count=count+1
      do 50 i=1,2
      ta=tsw(i)
      call psych(ta,ta,wswe,h)
 50   diff(i)=hswe-h
      tsw(2)=(tsw(1)*diff(2)-tsw(2)*diff(1))/(diff(2)-diff(1))
      tsw(1)=tsw(2)-1.
      if(count.gt.10) write(*,*) 'hswe failed'
      if(count.gt.10) go to 60
      test=abs(diff(2))/hswe
      if(test.gt..0005) go to 40
 60   wao=wswe+(wai-wswe)/exp(xntu)
      wmdoto=wmdoti-amdot*(wao-wai)
 70   two=32.+(wmdoti*(twi-32.)-amdot*(hao-hai))/wmdoto
      eff2=(twi-two)/(twi-twb)
      evap=(wmdoti-wmdoto)/wmdoti
      wo=wmdoto
      return
      end
```