



Model-Based Approach for Optimizing a Data Center Centralized Cooling System

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data center,
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COP, heat

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Abstract

A steady-state model for a centralized cooling system is developed to enable proper provisioning of cooling resources in capital intensive projects such as data centers. The model resolves the energy equations for a cooling tower and a centralized water-cooled chiller simultaneously. It requires inputs that are readily available to the design engineer. The user inputs are the ambient conditions, the cooling tower air flow rate, the condenser water flow rate, the evaporator water flow rate, the superheat and subcooling associated with the refrigeration cycle and the full load design conditions. The model utilizes an empirical relationship for the compressor power as a function of load and temperature and gives the user an option to select a constant speed chiller or a variable speed chiller. The outputs include the chiller coefficient of performance, compressor input power and compressor isentropic efficiency. The model results are validated with a manufacturer performance data [8] and compared to an experimental data collected at Hewlett-Packard Laboratories site for a 2110 kW (600 Ton) variable speed chiller and a 2286 kW (650 Ton) constant speed chiller. The model results are found to match the experimental data to within an acceptable deviation. For the constant speed chiller, the chiller efficiency increases with increasing heat load and peaks at full load. For the variable speed chiller, the chiller efficiency peaks between 50% and 70% loading depending on the ambient conditions.

Keywords

data center, heat transfer, cooling, chiller, cooling tower, COP

Nomenclature

Approach	the temperature difference between the cooling tower supply temperature and the ambient wet bulb temperature
COP	coefficient of performance
C_p	specific heat (kJ/(kg-K))
CPR	compressor power ratio
ER	efficiency ratio
h	enthalpy (kJ/kg)
\dot{m}	mass flow rate (kg/sec)
Q	heat transfer rate (Watts)
T	temperature ($^{\circ}$ C)
W	work (Watts)
x	evaporator heat load ratio
y	temperature difference ratio
ω	specific humidity
η	efficiency

Subscripts

chw	chilled water flow
chwr	chilled water return
chws	chilled water supply
comp	compressor
ctair	cooling tower air flow
ctw	cooling tower water flow
ctwr	cooling tower water return
ctws	cooling tower water supply
evap	evaporator
m	motor
mkw	makeup water
pol	polytropic
ref	refrigerant
v	vapor
wb	wet bulb

Introduction

The growth in information technology services has spurred the growth of high power density data centers. Data Centers are typically designed for heat load from compute, networking and storage equipment of 1 to 10 MW. The heat removal mechanism in most data centers includes centralized chillers with a chilled water loop to the air handler units in the machine room, and either air cooled condenser or water cooled condenser [1]. The power required by the cooling resources in a data center is typically equal to the power required by the hardware in the machine room [1].

There are over 100,000 chillers in operation in North America with varying sizes and efficiency. Centrifugal chillers are the preferred chillers when the cooling capacity larger than 1400 KW (400 tons) is required. In a centralized cooling system, the chiller's compressor power is a large component and with the trend of emphasizing energy savings, customers are paying careful attention to the total cost of ownership (TCO) when it comes to installing new or replacement chillers. This process consists of analyzing various available chillers options to determine the TCO and the annual recurring cost. The power consumed by the chiller's compressor varies with load while for constant speed water pumps - the power consumption remains unchanged. Therefore careful analysis of the system at part load is warranted for best selection. In addition, the cooling system energy requirement in data centers varies with time of the day, time of the year and the business requirements imposed by the compute services delivered. Thus, dynamic provisioning of the cooling system necessitates scalability in the cooling system and a deep technical understanding when making initial investment choices. Hence, a simple tool that can enable proper static provisioning and characterization of flexibility for dynamic operation is needed.

Several researchers have investigated chillers used for comfort air conditioning. Braun [4] developed a computer simulation tool for centrifugal chillers to investigate part load performance and used the results to optimally control the cooling system as a whole. Popovic et. al. [5] presented a simple analytical centrifugal compressor model to describe its performance. They built the model using empirical relations and compressor parameters to run for different

refrigerants. Browne and Bansal [6] presented a steady-state liquid-cooled centrifugal chiller model capable of analyzing both hermetic and open-drive compressors. The model was validated with experimental data collected on site. They showed the performance of the chiller increases with increased capacity. Wang et. al [7] developed a chiller model to simulate a centrifugal chiller performance. The model was based on real operational data and can be generalized to a system of multiple chillers. However, none of these studies incorporated the ambient conditions and the cooling tower effect on the chiller performance and the total energy consumption. In this study, an attempt to build a simplified model capable of examining the total cooling system (cooling tower and chiller) performance is presented. This is achieved by building a realistic centralized cooling model that is able to simulate the performance of the cooling system at various loading conditions. The system model, the experimental data and the model validation are presented. In the future, the model will be extended to enable detailed energy analyses. The emphasis in this paper is on centralized chillers with chilled water loop on the heat load side and water cooled condenser on the heat rejection side as shown in Fig. 1 .

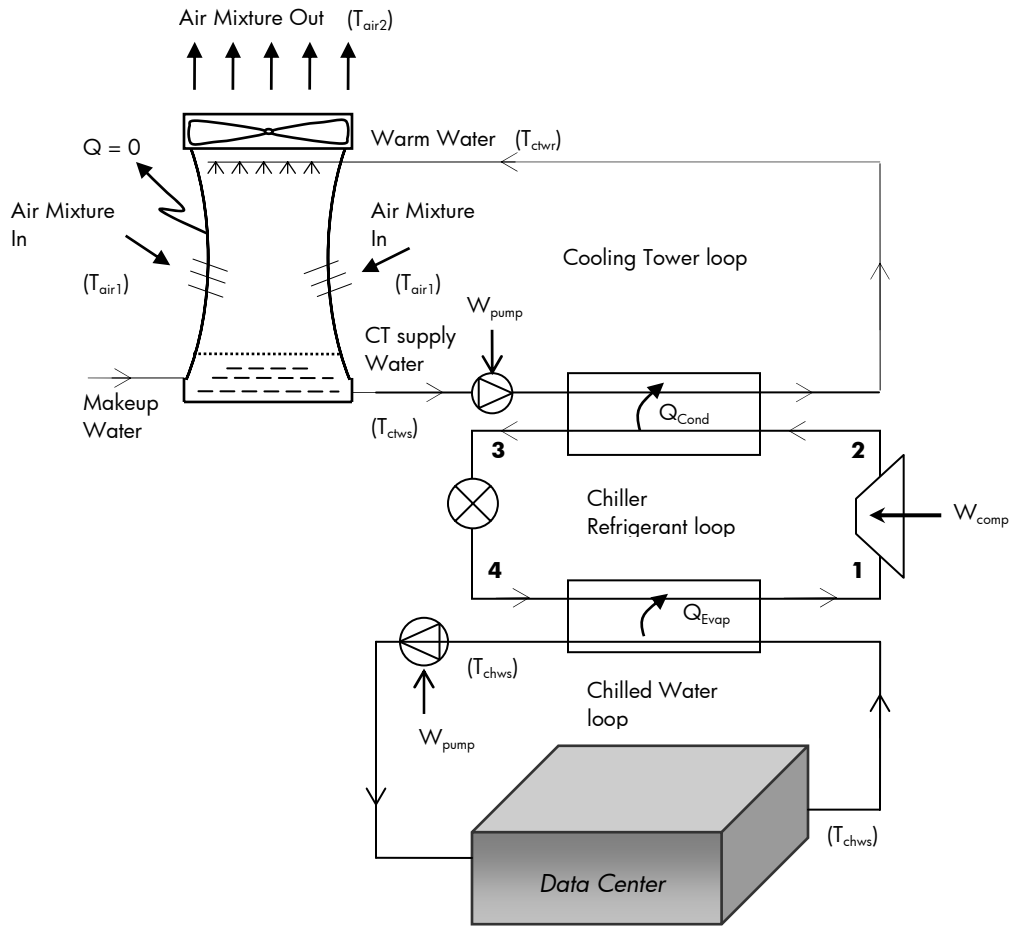


FIG. 1
Schematic for a typical water-cooled centralized chiller system

Current Design Approach

There are several issues with the current industry design approach and rules of thumbs developed decades ago that do not take into account some of the new technologies, features and advancement integrated into modern centralized chiller systems. As an example, the approximation applied for water flow rates – 3 gpm/ton for the condenser and 2 gpm/ton for the evaporator - can lead to over provisioning. It will be more efficient to specify the water flow rate based on the heat load, system pressure drop and on the temperature difference between the inlet and the exiting water temperature. In addition, the cooling tower manufacturers use of cooling tower capacity which is not technically correct - defining cooling tower “ton of cooling capacity” as 15,000 Btu/hr (4.396 kW) which is not in agreement with the universal definition of 12,000 Btu/hr per 1 ton (3.5168 KW) of cooling capacity. The cooling tower industry justifies their approach by an old estimate for chiller coefficient of performance (COP) of 4 and thus adding the additional heat load imposed by the compressor. Today’s water-cooled centrifugal chillers, for example, can have COP of 8 or more depending on the control features and on their environmental conditions.

Standard Rating Conditions for Water-Cooled Chillers

Based on the Air-Conditioning and refrigeration Institute ARI standard 550/590 [2], the rating condition for water cooled chiller is as follows:

- Evaporator leaving chilled-water temperature: 6.7 °C (44F).
- Entering water-cooled condenser water temperature: 29.4 °C (85F).
- Evaporator chilled-water flow rate: 2.0 gpm/ton.
- Water-cooled condenser water flow rate: 3.0 gpm/ton.
- Evaporator chilled-water temperature difference: 5.6 °C (10F).
- Water-cooled condenser water temperature difference: 5.1 °C to 5.6 °C (9.1F to 10F).

A simplified cooling system model would be beneficial in the design and selection of the proper sized water-cooled chillers and as a system performance analyzer at various load and ambient conditions. In this paper, a simplified model for two centrifugal chillers, one with variable frequency drive motor and the other with constant speed motor is presented. The model solves the mass and energy equations for the chiller components: the compressor, the condenser, the expansion device and the evaporator, and for the cooling tower heat transfer process. The model uses the chiller design load data to determine the system variables at (off-design) conditions different than the design conditions: alternate ambient conditions (temperature and relative humidity), various condenser water flow rate, various entering condenser water temperature and variable loading conditions. The model can be used as an optimization tool for best overall cooling system efficiency.

Model Presentation

The model utilizes the Engineer Equation Solver (EES) [2] software package with built-in refrigerants properties and uses the mass and the energy balance to determine the variables for

the interrelated water, refrigerant and air cycles. With reference to Fig. 1., the condenser and evaporator relations can be described by the following equations:

$$Q_{\text{evap}} = C_{\text{pchw}} \dot{m}_{\text{chw}} (T_{\text{chwr}} - T_{\text{chws}}) \quad (1)$$

$$Q_{\text{evap}} = \dot{m}_{\text{ref}} (h_1 - h_4) \quad (2)$$

$$\dot{m}_{\text{ref}} = \frac{Q_{\text{evap}}}{(h_1 - h_4)} \quad (3)$$

$$Q_{\text{cond}} = C_{\text{pctw}} \dot{m}_{\text{ctw}} (T_{\text{ctwr}} - T_{\text{ctws}}) \quad (4)$$

$$Q_{\text{cond}} = \dot{m}_{\text{ref}} (h_2 - h_3) \quad (5)$$

The ideal compressor work can be calculated from:

$$W_{\text{comp-s}} = \left(\frac{n}{n-1} \right) \dot{m}_{\text{ref}} P_2 v_2 \left[\left(\frac{P_2}{P_1} \right)^{\left(\frac{n-1}{n} \right)} - 1 \right] \quad (6)$$

The compressor polytropic efficiency is evaluated by:

$$\eta_{\text{pol}} = \frac{1}{\text{CPR}} \left[\frac{W_{\text{comp-s}}}{\eta_{\text{m}} W_{\text{comp-design}}} \right] \quad (7)$$

and the actual compressor work is defined by:

$$W_{\text{comp}} = \frac{W_{\text{comp-s}}}{\eta_{\text{m}} \eta_{\text{pol}}} \quad (8)$$

The chiller efficiency is presented based on the coefficient of performance relationship and can be calculated from:

$$\text{COP} = \frac{Q_{\text{evap}}}{W_{\text{comp}}} \quad (9)$$

Another industry-wide used efficiency measure is the efficiency ratio which defined by:

$$\text{ER} = \frac{W_{\text{comp}} (\text{kW})}{Q_{\text{evap}} (\text{Tons})} \quad (10)$$

The cooling tower removes heat from the circulating condenser water into the air stream. Most of the heat transfer occurs within the cooling tower due to latent heat (phase change) and only small percentage of the total heat transfer is due to sensible heat (change in temperature). The relative percentage depends on the ambient conditions and on the return condenser water temperature.

The heat transfer to the air stream can be described by:

$$Q_{\text{ctair}} = \dot{m}_{\text{ctair}} \left[C_{\text{pair}} (T_{\text{air2}} - T_{\text{air1}}) + (\omega_2 h_{v2} - \omega_1 h_{v1}) \right] \quad (11)$$

and is approximately equal to the heat removed from the circulating condenser water:

$$Q_{\text{cond}} = C_{\text{pctw}} \dot{m}_{\text{ctw}} (T_{\text{ctwr}} - T_{\text{ctws}}) \quad (12)$$

The amount of water evaporated and carried out with the air stream can be calculated from:

$$\dot{m}(\text{evaporated}) = \omega_2 - \omega_1 \quad (13)$$

The amount of water needed to make up the amount of water evaporated is found by:

$$\dot{m}_{\text{mkw}} = \frac{\dot{m}_{\text{ctair}} \dot{m}_{\text{evaporated}}}{\rho_{\text{ctw}}} \quad (14)$$

The difference between the entering condenser water temperature and the ambient wet bulb temperature is called the approach:

$$\text{Approach} = (T_{\text{ctws}} - T_{\text{wb}}) \quad (15)$$

The range is the difference between the entering and leaving condenser water temperature:

$$\text{Range} = (T_{\text{ctwr}} - T_{\text{ctws}}) = \frac{Q_{\text{ct}}}{(C_{\text{pctw}} \dot{m}_{\text{ctw}})} \quad (16)$$

Chiller power consumption is primarily a function of the heat load and the temperature difference between the condenser leaving water temperature and the evaporator leaving water temperature. The following relationship is presented as an estimate to the chiller power consumption [2]:

$$\text{CPR} = a_0 + a_1 x + a_2 x^2 + a_3 y + a_4 y^2 + a_5 xy \quad (17)$$

where the coefficients a_0 to a_5 are determined experimentally using least-squares fit to a chiller performance data. CPR is the compressor power ratio and is defined by the following equation:

$$\text{CPR} = \frac{W_{\text{comp}}}{W_{\text{comp-design}}} \quad (18)$$

and x is the evaporator heat load ratio:

$$x = \frac{Q_{\text{evap}}}{Q_{\text{evap-design}}} \quad (19)$$

and y is the temperature difference ratio and it is obtained from:

$$y = \frac{(T_{\text{ctwr}} - T_{\text{chws}})}{(T_{\text{ctwr}} - T_{\text{chws}})_{\text{design}}} \quad (20)$$

The effect of the condenser and the evaporator pressure drop is taken into account in Eq. [17] since the variable x and y are based on real experimental values. The model runs check to make sure all user inputs are within practical range (constraints) and displays warning or error messages if any of the user inputs or calculated values based on these inputs is outside the practical range. For example the relative humidity lower and higher limits imposed are 10% and 95% respectively, the ambient temperature limit is 50 °C, the calculated heat load should be less than the chiller full design load (maximum load) and the approach temperature should be equal or greater than 2 °C. The model flow chart is presented in Fig. 2.

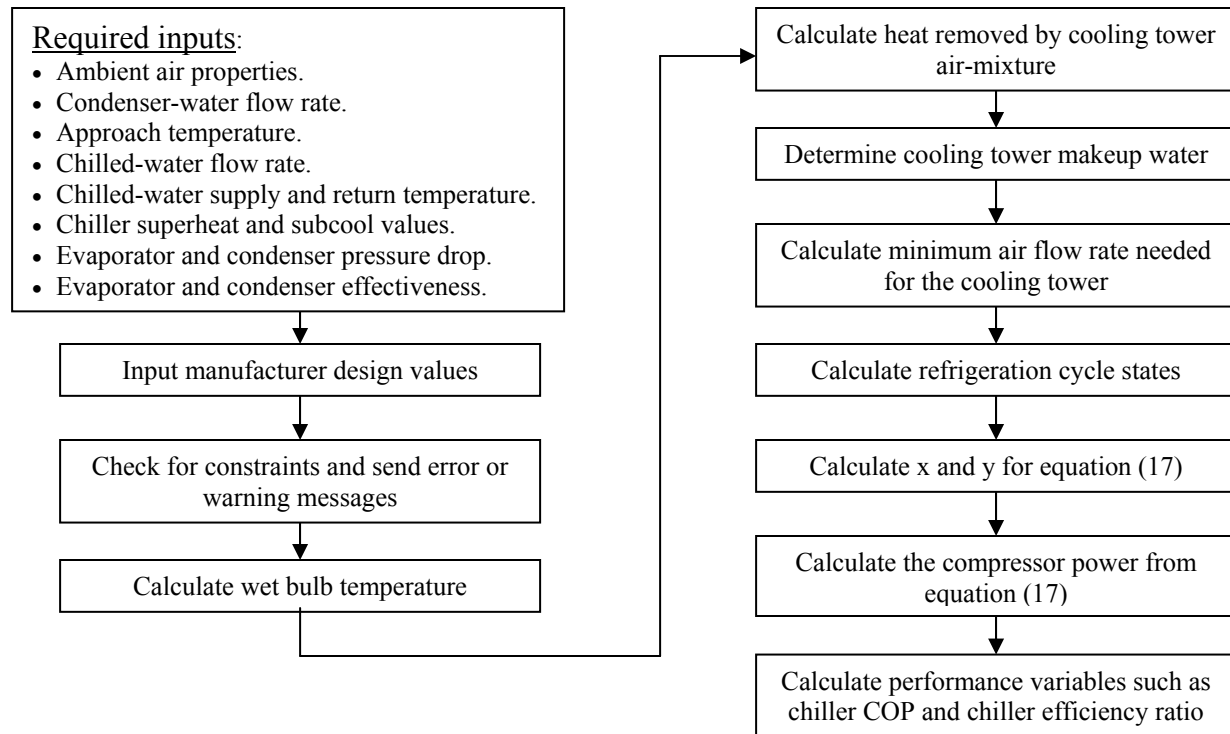


FIG. 2
Cooling system model flow chart

Experimental Setup

Chilled-Water Plant

The chilled-water plant in this study refers to two centralized centrifugal chillers (one constant speed chiller and the other is a variable speed chiller) each with a primary and a secondary chilled water loop feeding the HP campus which includes a 704 kW (200 Ton) capacity data center. The two centrifugal chillers are meant to maintain the proper environment conditions for the data center and campus and the typical loading for October-December period is expected to be between 30%-60%. Two constant speed pumps are used in the primary loop with one acting as a redundant pump. The secondary loop employs two variable speed pumps with one being a redundant pump. The chilled-water network is linked with a newer variable speed chiller set as the primary chiller and an older constant speed chiller set as a backup chiller with each chiller having its own primary pumps. The operator can override the settings and run either chiller as a primary chiller or both. Each chiller condenser is water-cooled by water supplied from one of two cooling towers operating with variable speed blowers.

The building management system (BMS) is used to monitor and collect the data needed for this experiment. The two chillers in this study are connected to the BMS where trends and reports can be generated and reviewed. The reports set to gather data needed for the current analysis include:

- evaporator water temperature (return and supply)
- condenser water temperature (return and supply)
- refrigerant pressures
- ambient conditions

In addition, the power consumed by the chiller's compressor was measured and recorded using DENT instrument ELITEpro power meter. This meter measures power, current, voltage and power factor for 3-phase 4-wire supply. It is equipped with extended internal memory for up to 100,000 records so it can be used to measure chiller power over several days at a time depending on the collection frequency specified.

Chiller

Both chillers are water-cooled centrifugal chillers and are connected to the building control unit. The first chiller is a 2286 kW (650 Ton) constant speed chiller commissioned in 1997. The second is a 2110 kW (600 Ton) variable speed chiller commissioned in 2005. The chillers performance is monitored around the clock and alarms are set to inform the operators of any issues that might need their immediate attention. Some of the points monitored are: condenser pressure and temperature, evaporator pressure and temperature, chilled water supply temperature, chilled water return temperature, chilled water flow rate, condenser water inlet temperature, condenser water existing temperature and the condenser water flow rate. The chillers are equipped with an automatic shutdown for the following cases: low chilled water temperature, low evaporator refrigerant temperature, high condenser refrigerant temperature, low pressure ratio, low oil pressure, high oil temperature, high motor current and high motor temperature.

Compressor

The centrifugal compressor raises the pressure of the refrigerant by converting the kinetic energy of the refrigerant into pressure. The constant speed chiller uses a capacity control guide vanes that handles lower loads by gradually closing the guide vanes on the compressor inlet while the motor is still turning at full speed. Fig. 3 shows a typical performance curve for constant speed centrifugal chillers.

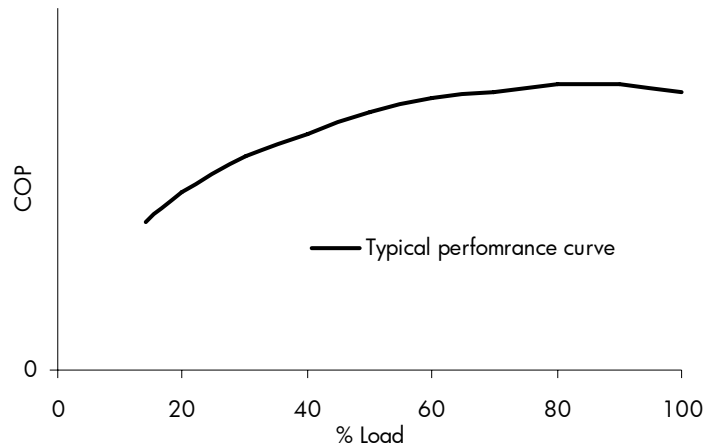


FIG. 3
Typical performance curve for constant speed centrifugal chillers

The variable speed chiller uses, in addition to the guide vanes, a variable frequency drive motor to control the chiller capacity and optimize the compressor power consumption. The variable speed centrifugal chiller tends to be more efficient at part loading. Fig. 4 shows a typical performance curve for variable speed centrifugal chillers.

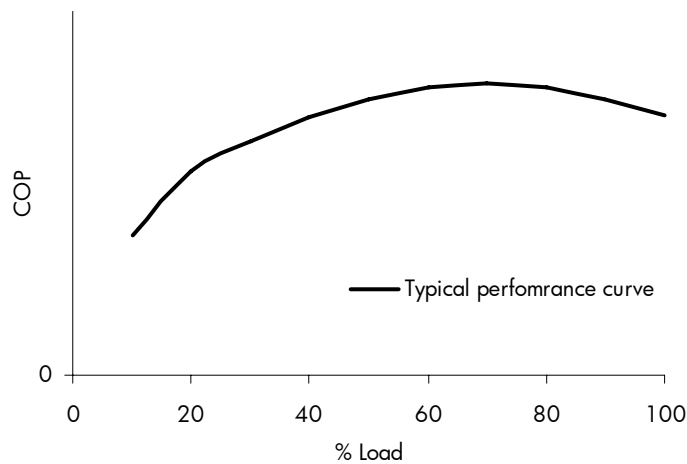


FIG. 4
Typical performance curve for variable speed centrifugal chillers

For both chillers, the motors are refrigerant cooled and the compressors are hermetically sealed. Chiller motors are typically over 95% efficient, and thus a little less than 5% of the motor power is added as heat to the refrigerant

Condenser/ Evaporator/Expansion Device

The heat exchanger in the evaporator section is made of carbon steel shell and externally finned copper tube. The evaporator uses a flooded type heat exchanger which is very efficient. The flooded evaporators have the chilled water in the tubes and the refrigerant in the shell. Tubes are mechanically expanded into tube sheets. The condenser is also made of carbon steel shell and externally finned copper tube with baffle to prevent direct impingement of compressor discharge gas upon the tubes and reduce the pressure drop.

An expansion device acts as a pressure reducing means and modulates the refrigerant flow rate over the range of capacity and temperature.

Cooling Tower

There two 600 Ton (cooling tower ton = 15,000 Btu/hr) cooling towers to cool the condenser water. The two cooling towers are both open-tower design. The design condition at full fan power are as follows: 10 °C (18F) range, the water flow rate is 1050 gpm, the return water temperature from the condenser is 35 °C (95F) and the wet bulb temperature is 19.7 °C (67.5F). Each cooling tower is equipped with a 50-hp variable speed fan.

Experimental Results and Discussion

The data collection from two centralized centrifugal chillers at Hewlett-Packard Laboratories is an ongoing activity. The experimental data shown here was collected during two month period from mid-October to mid-December, 2005. The part-load performance of both centrifugal chillers were recorded and compared. The variables collected were: supply and return chilled water temperature and flow rate, supply and return condenser water temperature and flow rate, chiller refrigerant suction and discharge pressures, compressor power and power factor. The compressor power is very sensitive to the condensing water temperature. In these experiments the condenser water temperature varied between 14.4 °C and 22.8 °C (58F-73F) for the constant speed chiller and between 15.6 °C and 20 °C (60F-68F) for the variable speed chiller due to the variation in the ambient air conditions.

The experimental data was monitored and recorded through the BMS over two month period noted above and saved in a report format. The sampling interval for the trending point is set to one minute interval. The objective was to capture data when the chillers were at various part-load conditions. Most of the data gathered was between 20% and 70% with very few data points above 70% loading which reflects the typical loading conditions during the time of year (mid-October to mid-December) the experiments were conducted. For the ambient temperatures, the average low for the month of October is 10 °C (50F) and the average high is 23 °C (73F), the average low for the month of November is 7 °C (44F) and the average high is 18 °C (64F). For

the month of December the average low is 4 °C (39F) and the average high is 14 °C (58F). The condenser water temperature for these experiments ranged anywhere between 15.6 °C (60F) and 21 °C (70F).

The data points are collected for a one minute sampling rate. The experimental data shows the constant speed centrifugal chiller exhibits an increasing efficiency with increasing heat load. The constant speed centrifugal chiller performance varies slightly above 70% loading for a given condenser water temperature. The performance is also less affected by condenser water temperature variation than a variable speed chiller and this is partly due to the fact the constant speed compressor suffer losses due to friction and drag at lower loads as the guide vanes close down to limit the amount of refrigerant flow through the compressor. The variable speed chiller performance is affected much more by load variations and condenser water temperature variations since its compressor controls the speed of the motor using a variable frequency drive which produces a more efficient mechanism of controlling the refrigerant flow. At full load, the chiller variable frequency drive cost the chiller less than 4% of the total power however this cost is compensated at part-loading. It is difficult to obtain an accurate data for loading below 20% on both chillers as the relative error in measurements (± 0.5 °C over temperature difference of 4 °C or 5 °C) becomes large especially for constant water flow rate.

Figures 5 and 6 show the experimental chiller performance in terms the average COP as a function of percent heat load. The average values in this case represent the average COP of the same load for the range of the condenser supply water temperature in the current experiments. The figures are representative samples of the matrix of the total results. Fig. 5 shows the calculated average performance for the constant speed chiller. The chiller performance increases as the heat load increases. The condenser water temperature recorded for the constant speed chiller ranged between 14.4 °C and 22.8 °C (58F-73F).

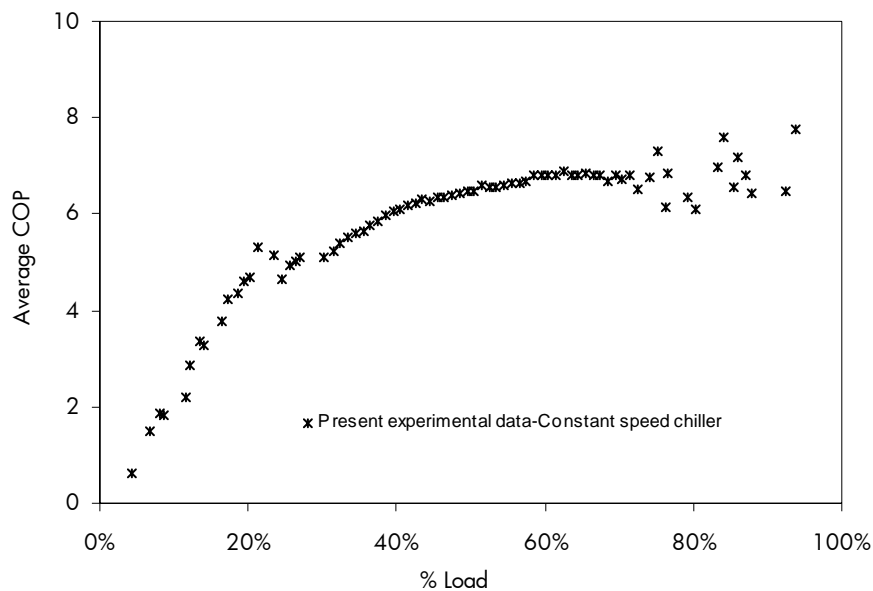


FIG. 5
Experimental coefficient of performance (COP) for the constant speed centrifugal chiller

Figure 6 shows the variable speed centrifugal chiller experimental performance. The current experiments show the performance to peak between 50%-70% loading for the given ambient conditions. The condenser water temperature recorded for the constant speed chiller ranged between 15.6 °C and 20 °C (60F-68F).

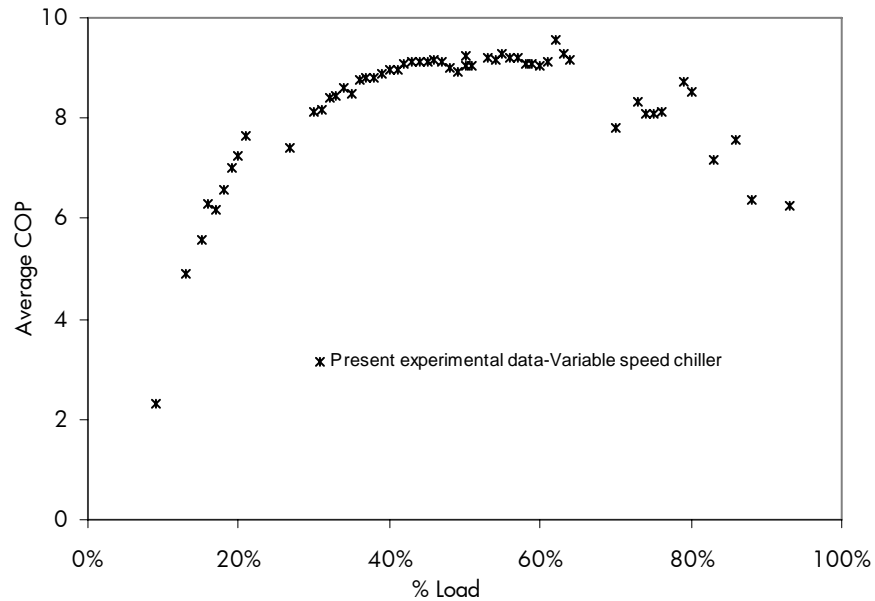


FIG. 6
Experimental coefficient of performance (COP) for the variable speed centrifugal chiller

Figures 7 and 8 represent a chiller manufacturer performance graphs. These performance graphs are generated from an original manufacturer performance map obtained by the authors [8]. The performance data is for centrifugal chillers at various condenser temperatures, 12.8 °C -26.7 °C (55F-80F). Fig. 7 gives the curves for the constant speed centrifugal chillers where the curves show the centrifugal chiller performance to increase as the chiller load increases and peaks when the chiller is fully loaded to its design capacity. Fig. 8 gives the curves for the variable speed chiller however for this case the curves show the chiller performance to peak at chiller load less than the full design load. The peak appears between 70%-90% load for condenser water temperature of 26.7 °C (80F) and appears between 40%-60% load for condenser temperature of 12.8 °C (55F).

Figures 9 and 10 show the current experimental curves superimposed over the manufacturer performance curves [8]. Fig. 9 shows the current constant speed chiller data to compare well with the manufacturer data for the 80F (26.7 °C) condenser temperature.

Figure 10 compares the current experimental data for the variable speed chiller with the manufacturer performance data [8]. This plot shows the current experimental data to compare well with the manufacturer performance data for 70F (21.1 °C) condenser temperature up to chiller load of 60%. For chiller load higher than 60%, the current data tend to decrease with a higher slope than the manufacturer data. This can be explained by the fact the manufacturer

obtained their data through tightly controlled test where all chiller variables are maintained constant while varying the condenser water temperatures. The experimental data is for an average value of the coefficient of performance for data collected under practical conditions where the condenser water temperature varied between 15.6 °C and 20 °C (60F-68F).

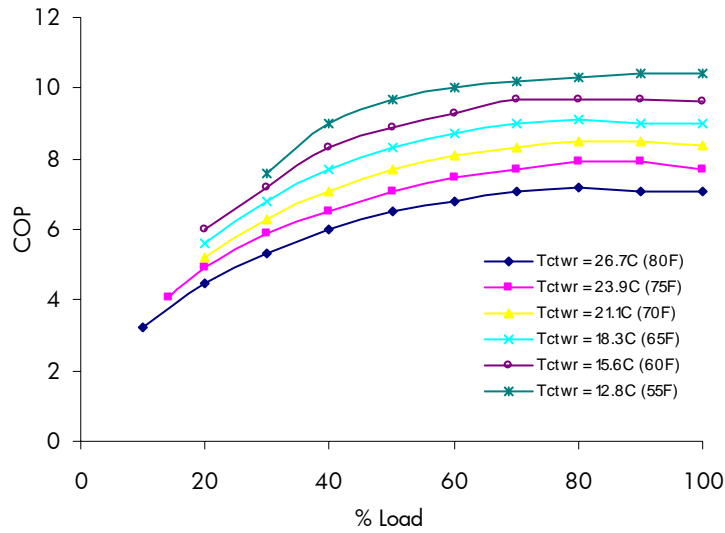


FIG. 7

Manufacturer performance data at various condenser water temperatures for constant speed chillers

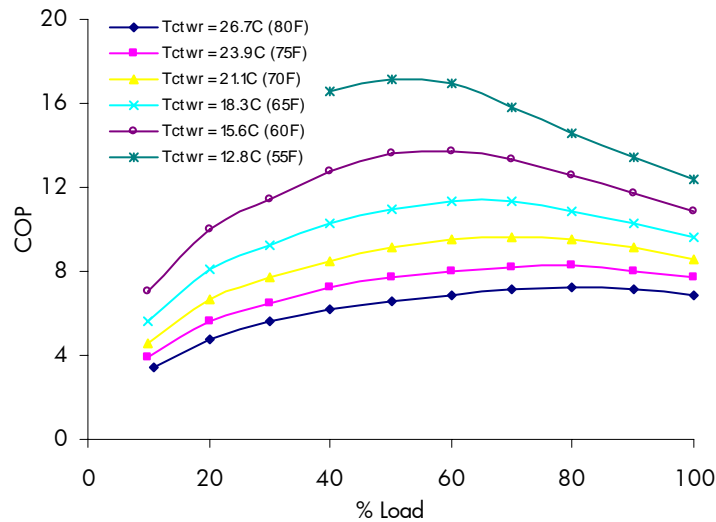


FIG. 8

Manufacturer performance data [8] at various condenser water temperatures for constant speed chillers

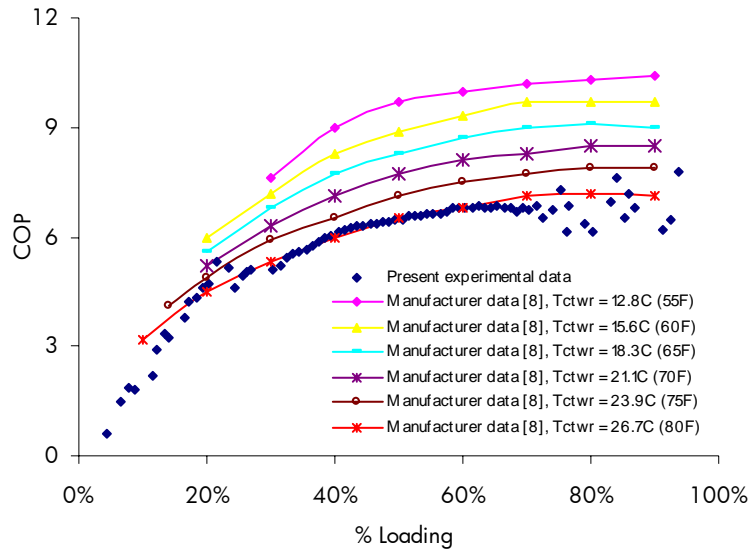


FIG. 9

Current experimental data compared to the manufacturer performance data [8] for constant speed chillers

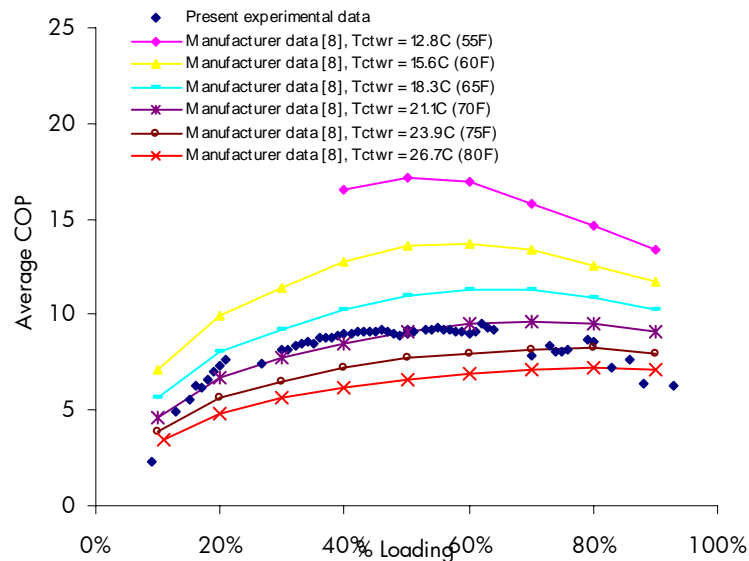


FIG. 10

Current experimental data compared to the manufacturer performance data [8] for variable speed chillers

Current Model Validation

Simulation results of the current cooling system model are compared to the manufacturer performance data [8] for various condenser water temperatures. Figs. 11 and 12 compare the model simulation for variable speed chiller with manufacturer performance values [8] for a condenser temperature of 26.7 °C (80F) and 12.8 °C (55F), respectively. Fig. 11 shows the COP

values obtained from the model simulation compares very well with the performance values. The average deviation between the model results and the performance data [8] values is 5.8% with a maximum deviation of 16% at 10% chiller load while the deviation is less than 5% for the loading larger than 10%. The large deviation at the lower load is due to fact that the error in temperature measurements can appear relatively high at load loads.

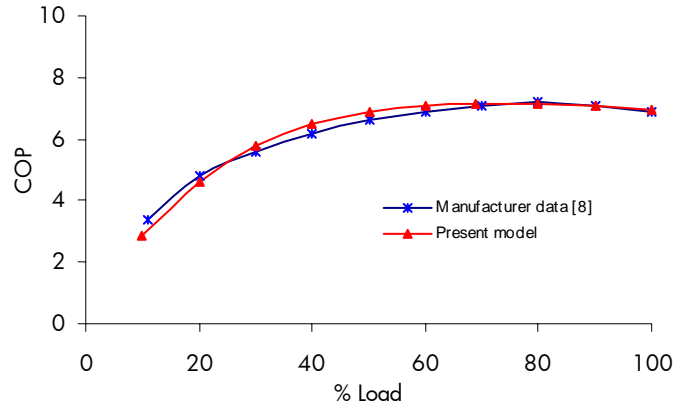


Fig. 11

Current model simulation vs. manufacturer performance data [8] for variable speed chiller at 80F (26.7 °C) condenser water temperature

Figure 12 shows similar comparison for a condenser water temperature of 55F (12.8 °C). The data obtained from the manufacturer for this temperature available only for chiller load 40% or higher. Again the simulation results compared well with the manufacturer performance data [8] with a maximum deviation of 5.3% and an average deviation of 3.3%. The variable speed chiller coefficient of performance peaks at partial load appearing between 60%-90% at condenser water temperature of 26.7 °C (80F) and moving towards smaller partial load as the condenser water temperature decreases. The peak performance appears at load appears between 40%-60% for the manufacturer performance data and between 30%-50% for the current model simulation at a condenser water temperature of 12.7 °C (55F).

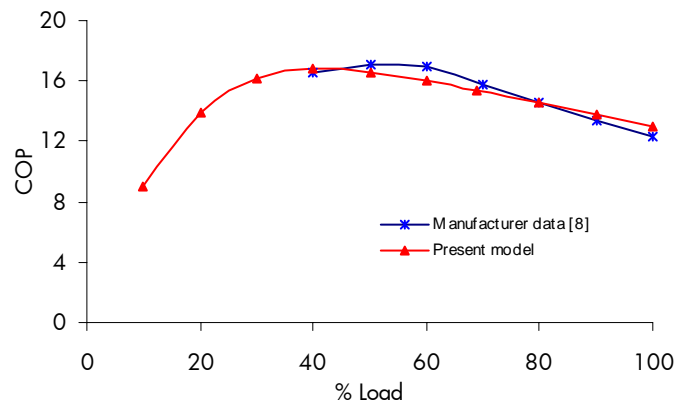


Fig. 12

Current model simulation vs. manufacturer performance data [8] for variable speed chiller at 55F condenser water temperature

Figures 13 and 14 compare the coefficient of performance for the constant speed chillers obtained from model simulation with the manufacturer performance values [8] for a condenser temperature of 26.7 °C (80F) and 15.6 °C (60F), respectively. Fig. 13 shows the performance values obtained from the model simulation to compare very well with the manufacturer data [8]. The average deviation between the model COP values and the performance data values is 3.8% with a maximum deviation of 10.9% at 10% chiller load.

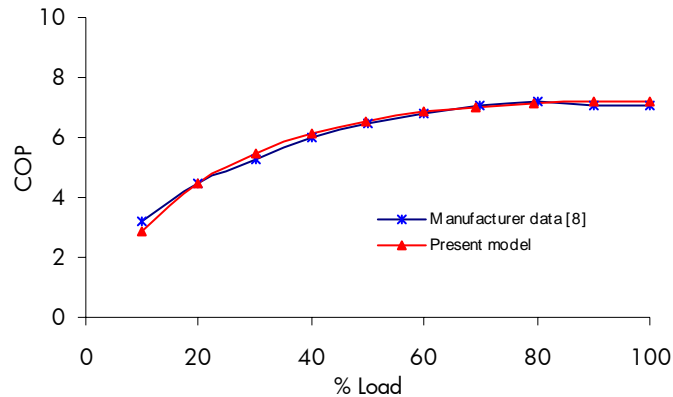


Fig. 13

Current model simulation vs. manufacturer performance data [8] for constant speed chiller at 80F condenser water temperature

Figure 14 shows the same comparison for a condenser water temperature of 15.7 °C (60F). The average deviation between the model COP values and the performance data values is 1.2% with a maximum deviation of 2.4% at 30% chiller load. Both the model simulation and the manufacturer performance data show the constant speed chiller coefficient of performance to increase as the chiller load increases and peaks at full chiller load. The chiller coefficient of performance also increases with decreasing condenser water temperature.

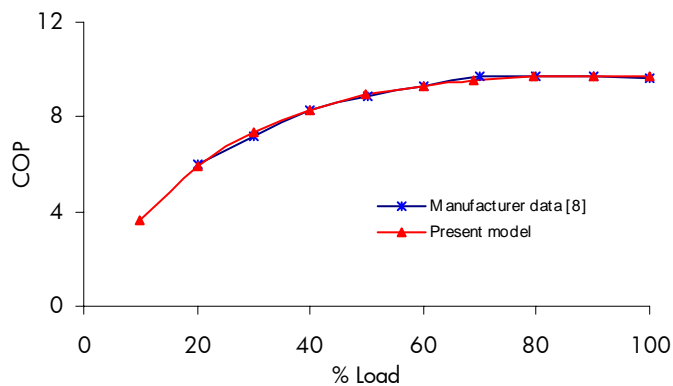


Fig. 14

Current model simulation vs. manufacturer performance data [8] for constant speed chiller at 60F condenser water temperature

Figures 15 and 16 compare the current model simulation results with the experimental results. The model results are for a specific condenser water temperature but the experimental results are for a range of condenser water temperature. Fig. 15 shows three curves of COP values obtained from the model simulation at 15.6 °C (60F), 18.3 °C (65F) and 21.1 °C (70F) condenser water temperature. The model results seem to fall within this range of condenser water temperature which reflects the actual operating range at the time of the data is collected.

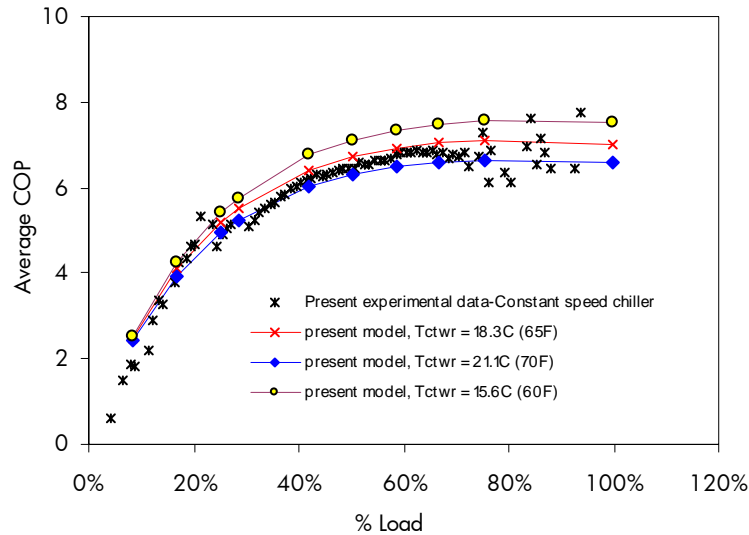


Fig. 15

Current model simulation vs. current experimental results for constant speed chiller

Figure 16 shows three COP plots obtained from the model simulation at 15.6 °C (60F), 18.3 °C (65F) and 21.1 °C (70F) condenser water temperatures. The average COP for the experimental data falls between the 15.6 °C (60F) and the 18.3 °C (65F) curves however the current experimental data curve has a higher slope than the present model estimates.

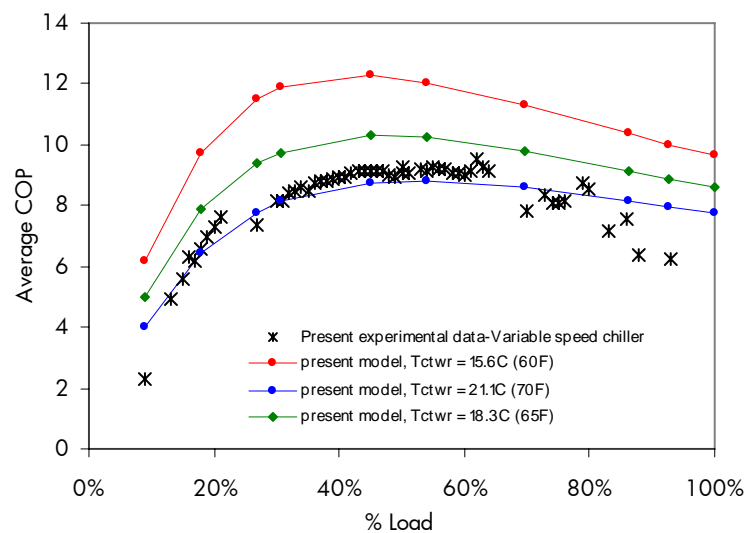


Fig. 16

Current model simulation vs. current experimental results for variable speed chiller

The steady-state uncertainty for the measured and the calculated values are evaluated based on the standard uncertainty analysis. The uncertainty in temperature measurements is ± 0.5 °C, for water flow rate measurement it is less than 5%, for the chiller power measured it is less than 1.0%, for the calculated heat load it is less than 7.8% and less than 7.9% for the calculated chiller coefficient of performance.

Summary and Conclusions

A simplified centralized cooling system model is presented. The model enables the air handling unit to cooling tower analysis when considering the “ensemble” from chips to cooling tower in a data center [1]. The model takes into account the chiller compressor power as a function of the chiller load and the water temperature difference between the evaporator supply and the condenser return. This relationship is described by Eq. [17] and where the variables are developed for constant speed chillers as well as for variable speed chillers using a manufacturer performance data [8]. The model is built to take into account the ambient conditions (temperature and relative humidity) that affect the cooling tower performance which determines the condenser water temperature.

Experimental data from constant speed and variable speed centrifugal chillers are summarized and presented. The data compares well with the simulation results and the manufacturer performance data [8]. The trend for the chiller performance is consistent between the model simulation, manufacturer performance data [8] and the current experimental results. The variable speed chillers performance peaks at partial load and moves from 70%-90% partial load at 26.9 °C (80F) condenser water temperature to a lower partial load of 40%-60% at 12.1 °C (55F) condenser water temperature. The constant speed chillers performance increases with increasing chiller load and peaks at full load. In general, the performance for both chiller increases with decreasing condenser water temperature.

With respect to the ensemble, from chips to cooling tower [1], this paper introduced an EES [2] based tool to enable design and analysis of the heat removal from air handling units to the cooling tower. Based on the typical data center usage models, it may not be appropriate to incorporate a constant speed chiller as its efficiency peaks at full load. Future work will include detailed energy cost, and the total cost of ownership, of operating chiller/cooling tower system over a range of operating conditions.

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Appendix

The EES model developed in this study provides an easy to use graphical user interface (GUI) as shown in Fig. A.1.

CTC Analyzer

Inputs: Cooling Tower

Air Side: Min. air flow (CFM) = ????

Inlet air temperature = [C] Air Flowrate = [CFM]

RH_{inletAir} = [%] RH_{exitAir} = [%]

Water Side:

CT Water Flowrate = [gpm] Approach = [C]

Chiller

Chilled water flow = [gpm] CW Supply temp. = [C]

Superheat = [C] CW Return temp. = [C]

Subcool = [C] Efficiency_{motor} =

Evaporator/Condenser:

Pressure Drop_{evaporator} (%) = Effectiveness_{evap} =

Pressure Drop_{condenser} (%) = Effectiveness_{cond} =

existing air Design compressor power (kW) =

1 = Constant Speed Chiller
2 = Variable Speed Chiller

Design CT return Temp (F) =

cooling tower

inlet air

CT water pump

expansion device

inlet air

chilled water pump

condenser

R123 Refrigerant

Design Load (kW) =

evaporator

compressor

data center

Design CW supply temp (F) =

Outputs

Exit air temperature = ????

Return water temperature_{CT} = ????

Supply condenser water temperature = 26.72 [C] ????

Wetbulb temperature = 20.72 [C] = 69.3 [F]

Range = ????

Heat transferred to the condenser water = ????

The total heat transferred from the waterside has be equal to the heat transferred to the air.

Heat transferred to the air mixture = ????

Cooling Tower load = ????

Compressor Suction Pressure, P₁ = 36.22 [kPa] T₁ = 6.633 [C]

Compressor Exit Pressure, P₂ = 116.5 [kPa] T₂ = ????

Condenser Exit Pressure, P₃ = 104.8 [kPa] T₃ = 26.72 [C]

Evaporator inlet Pressure, P₄ = 40.24 [kPa] T₄ = 4.633 [C]

Pressure Ratio = 3.216

Refrigerant mass flow = ????

W_{compressor} = ????

Q_{evap} = 1572 [kW]

Q_{cond} = ????

Make up water (gal/year) = ????

Chiller COP = ????

Chiller Heat Load = 447.1 [Tons]

Compressor Polytropic Efficiency = ????

Chiller Efficiency Ratio = ????

Fig. A.1
Screen shot of the model GUI.