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Optimize a Chilled-Water Plant with Magnetic-Bearing Variable Speed Chillers

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ABSTRACT

Variable speed centrifugal chillers perform much more efficiently at part-load ratio of cooling demand as well as partial compression ratio of lift head compared to constant speed centrifugal chillers. Magnetic bearing technology further improves the benefit of variable speed centrifugal chillers. This performance difference requires a reconsideration of most effective plant operation when magnetic bearing centrifugal chillers are incorporated. In this paper, a component-based model is developed for a chilled-water plant with oil-free chillers at the Texas A&M University System RELLIS Campus. The model, calibrated by on-site measured data from the building automation system, is used to evaluate the savings potential for different operation strategies. An optimal chiller staging strategy, based on the optimal efficiency curve, is simulated for the case study plant under constant water flow and variable water flow scenarios respectively. The results show that optimal chiller staging only improves plant performance by 2.1% for the constant water flow system. Implementing optimal staging with variable water flow would increase the energy savings to 13.7% compared to current operation of the case-study plant.

INTRODUCTION

Magnetic bearing centrifugal chillers with variable-speed drives (VSD), also known as oil-free centrifugal chillers, allow the centrifugal compressors to operate without the use of oil for lubrication, which reduces energy losses due to friction and increases the heat transfer efficiency in the chiller, because no oil enters the evaporator or condenser. A variable speed drive on the motor allows the compressor to operate much more efficiently at partial loads than standard compressors. The oil-free system also eliminates the need for oil maintenance, resulting in operations and maintenance savings. When compared to a conventional centrifugal chiller with VSD, Yu and Chan [2015] determined that oil lubrication-based operations suffered from a 3.2% performance decrease in the chiller as well as accelerated performance degradation.

Furthermore, because there is no limitation on the minimum compression ratio for the oil return system, magnetic bearing centrifugal chillers are capable of operating with lower condenser water supply temperatures compared to some conventional compressor systems. This results in an additional improvement in efficiency and higher capacity. Some magnetic bearing chillers are also able to operate continuously with minimum entering condenser water temperature below the leaving chilled-water set point. This is sometimes referred to as inverted or upside down operation. It can operate stably with an entering condenser water temperature 35.0°F (1.7°C), below the leaving chilled-water set point. Because free cooling is not so free anymore, these new chillers lower costs by eliminating the need for water-to-water heat exchangers and their accompanying expenses, such as piping controls and operation and maintenance expenses.

Deng [2018] examines actual performance of Magnetic Bearing Centrifugal Chillers in different buildings and cities through a whole year and compares the results to those of conventional centrifugal chillers. It was found that magnetic bearing centrifugal chillers clearly performed much more efficiently, especially at part-load ratio as well as partial compression ratio. Thus, to fully take advantage of magnetic bearing centrifugal chillers for truly energy efficient operation, one must optimize a magnetic bearing centrifugal chiller's operation based on an annual hourly simulation of cooling demand

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and compression ratio demand rather than just thinking of the nominal rated conditions.

Li [2004] claimed that the key contributor to poor chiller plant performance was the mismatch between the demand and supply sides. Due to improper cooling load calculations or unreasonable safety factors, oversized chiller capacities have become a common phenomenon in HVAC systems. Even a perfectly designed chiller plant could be very significantly oversized in actual operation since the cooling load reaches its peak level for only a small proportion of time in each year (Cheng [2017]). This oversized cooling capacity decreases operational energy performance and wastes energy throughout the cooling season (Woradechjumroen [2014], Djunaedy [2011]) for conventional chillers. However, a magnetic bearing variable speed centrifugal chiller may solve this mismatch issue, because it can operate much more efficiently at partial loads than conventional compressors. Guo [2014], analyzed the energy saving rate of a data center by replacing conventional chillers with magnetic bearing centrifugal chillers. Results showed a 37% (minimum) reduction in energy consumption. However, Guo relied on four discrete design values of the Integrated Part-Load Value (IPLV) to calculate the energy-savings instead of using an actual load profile. The IPLV assumes the entering condenser water temperature is dropping with load. This operating profile allows compression speed requirements for flow and lift head to decline together reducing the need for mechanical unloaders or over compression. If chillers run only at these four discrete operating points this would be appropriate, but this is rarely the case. AHRI 550/590 [2011] advises that IPLV "was derived to provide a representation of the average part-load efficiency for a single chiller only". Most chilled-water plants employ more than one chiller to meet the load. The IPLV rarely depicts an actual chiller's load profile because most chiller plants have more than one chiller and the local weather likely does not match the standard AHRI profile. Carrier [2015] stated that the reality of actual operation is much less elegant than IPLV, resulting in operating points where the use of mechanical unloaders or over compression are more likely to occur. This is because the actual operating points are further from the ideal speed curve than the operating points in the AHRI IPLV load profile.

Hartman [2001] developed operating strategies to enhance performance of all-variable speed chillers in comfort conditioning chiller plant applications. Optimum overall chiller plant performance is achieved when the rate of plant marginal capacity versus marginal power use is the same for each element of the system.

Parker [2012] assessed operation of variable-speed magnetic bearing centrifugal chillers with magnetic bearings. The monitored data show the new magnetic bearing centrifugal chillers are more efficient than the original chillers, especially at lower loads. At high load ratio, the efficiency benefit is measurable, but is not as significant. However, as the load ratio decreases, the efficiency benefit becomes more pronounced. Therefore, this report recommended improving chiller-staging controls. If multiple chillers were available, it may reduce total chiller plant power by operating two chillers at half load rather than a single chiller at full load.

Figure 1 shows a performance comparison of constant, variable speed and magnetic bearing variable speed centrifugal chillers at different load conditions. Constant speed chillers have a relatively flat performance curve across a range

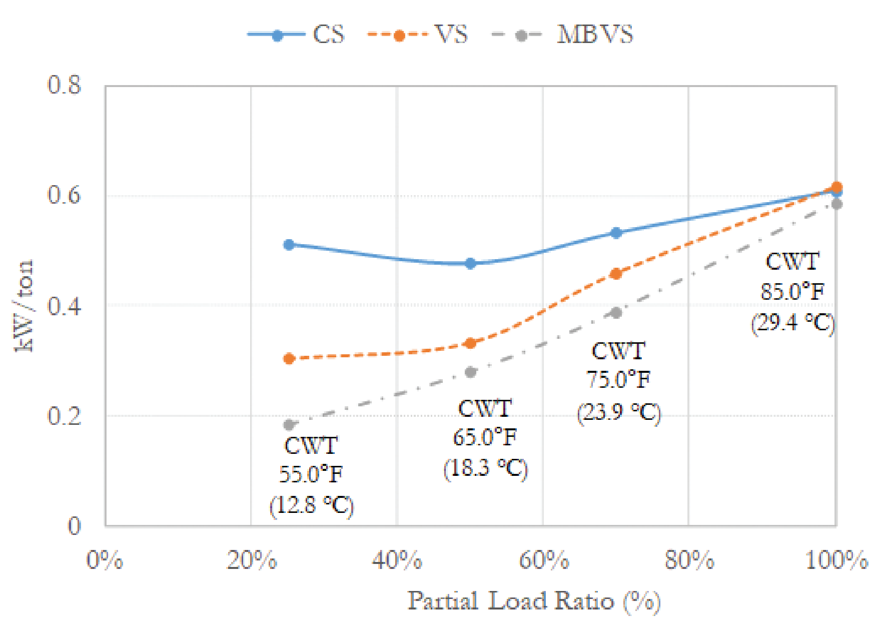

Figure 1 *Performance comparison of constant-speed, variable-speed, and magnetic-bearing variable-speed centrifugal chillers.*

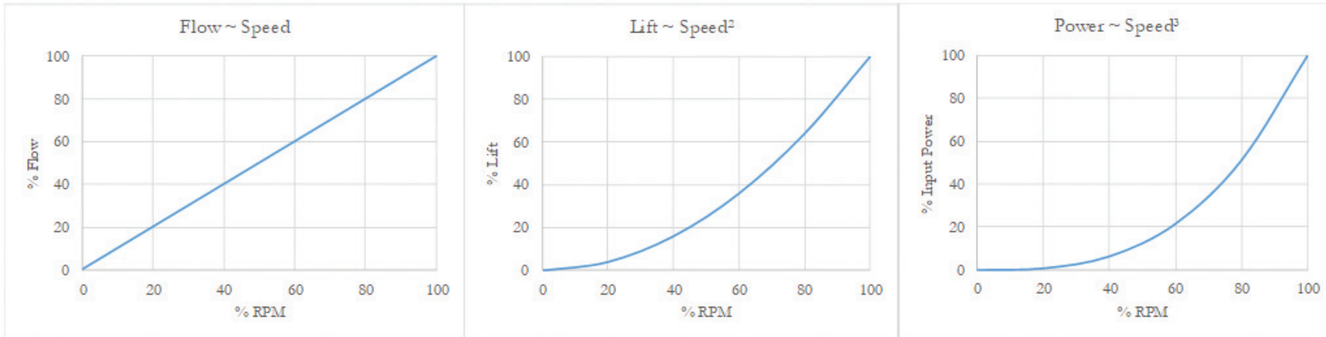

Figure 2 *Ideal fan laws.*

of load conditions; therefore, the focus of constant speed chiller plant operation has been minimizing the amount of on-line equipment. This permits chillers to operate as close as possible to their full capacity to reduce the percent of auxiliary electricity consumption from water pumps and cooling tower fans that are usually sequenced with chillers. However, Figure 1 also shows that Variable Speed (VS) chillers and Magnetic Bearing Variable Speed (MBVS) chillers operate most efficiently well below full load. The efficiency improvement of MBVS chillers can be very substantial at low condenser water temperatures. Part of the improved part-load efficiency curve for MBVS chillers is the change in motor type, with higher motor efficiency at part load and allowing the elimination of gear losses, not only the reduction of frictional losses in bearings. One should also note that the condenser water temperatures listed in Figure 1 are per AHRI 550/590 condenser relief assumptions. They are for rating purpose only and actual condenser water temperatures will not always vary with load as shown.

These performance differences require reconsideration of the most efficient operation strategy for a plant using magnetic bearing variable speed centrifugal chillers. In this paper, a component-based model is developed for a chilled-water plant with magnetic bearing centrifugal chillers at the Texas A&M University System RELLIS Campus, in Bryan. The models, calibrated by on-site measured data from the building automation system, are used to evaluate the savings potential for different operation strategies.

CENTRIFUGAL CHILLERS PERFORMANCE CURVE

All centrifugal compressors (regardless of bearing type, refrigerant type, motor type, oiled or oil free) must generally adhere to the ideal fan laws, illustrated in Figure 2 [Carrier, 2015].

a. First law: Mass flow (capacity) is linear with speed.
b. Second law: Lift (ability to generate head) increases with the square of speed.
c. Third law: Power increases with the cube of speed. (It may be less than cubic due to the proximity of guide vanes to the impeller inlet.)

In simple terms, the ability to generate head in a centrifugal compressor is related to the square of speed. At 80% speed, a centrifugal will generate 80%×80%=64% of its design head. If the operating condition requires more than 64% of design head, the centrifugal compressor will surge. Centrifugal compressors move gas through an open path between the low-pressure evaporator to the higher-pressure condenser. If the pressure difference (head) is too large, flow reversal will occur (surge).

To avoid surge, the centrifugal compressor must either speed up or engage hot gas bypass (also called load balance valves or recirculation flow on many magnetic bearing machines). So, while we may be inclined to believe that compressor speed is typically based on capacity, it is actually often based on the head required. For example, if a chiller is operating at 50% load and 75.0°F (23.9°C) entering condenser water, the first law would suggest we could operate at 50% speed, but the 2nd law requires us to run at 84% speed.

$$\%\text{Lift} = \frac{\text{CWET}_{actual} + \Delta T_{actual} - \text{CHWLT}_{actual}}{\text{CWET}_{design} + \Delta T_{design} - \text{CHWLT}_{design}}$$

$$\%\text{Lift} = \frac{75°\text{F} + 5°\text{F} - 44°\text{F}}{85°\text{F} + 10°\text{F} - 44°\text{F}} = \frac{36°\text{F}}{51°\text{F}} = 71\% \tag{1}$$

$$\%\text{Speed} = \sqrt{71\%} = 84\%$$

This raises an interesting question: If the centrifugal compressor must operate at 84% speed to develop sufficient head; how do we limit the capacity to 50%? This requires an unloading device such as inlet guide vanes, discharge flow restrictions, load balance valves (hot gas bypass) or some other means of diverting or restricting flow. The net result is the compressor is compressing more refrigerant or head than is required to meet the load, which represents an increase in power per unit of cooling at the operating point.

The ideal fan laws define a minimum speed requirement for flow and a separate minimum speed requirement for the generation of head in a centrifugal compressor. Combining these two minimum speed requirements, it becomes apparent that the ideal speed for a centrifugal compressor occurs when the speed required for mass flow exactly matches the speed required to generate head (as shown in Figure 3). Above this speed, the high head forces the centrifugal compressor to operate at higher speeds than needed for capacity and mechanical unloaders are employed. Below this speed, the speed for mass flow exceeds the head requirements resulting in over-compression. In either case, the performance of the centrifugal chiller will decrease. The further the operating point is from the ideal speed curve, the greater the impact will be. Figure 4 illustrates the optimal efficiency curve for the case study of one particular model of MBVS chiller with a single compressor.

The optimal efficiency curve of a variable speed chiller when operating with a fixed chilled-water supply temperature is a simple concept. It is the locus of points of highest chiller operating efficiency at various condenser water temperature and load conditions. In other words, the speed required for mass flow exactly matches the speed required to generate head. Notice that the best efficiency (lowest kW/ton) for the oil-free variable speed centrifugal chiller when the entering condenser water temperature is 85.0°F (29.4°C) is achieved at about 58% load. This is the point on the optimal efficiency curve for that condensing water temperature. The optimal efficiency curve is developed by connecting the points of highest efficiency for each condensing water temperature. In Figure 4, the performance curves are for constant condenser water and chilled-water flows. For applications where chilled-water temperature is variable or condenser water flow is varied, a new optimal efficiency curve can be constructed by using lift temperature head, which is the difference between leaving chilled-water temperature and leaving condenser water temperature. The optimal load ratio is the fraction of the design maximum capacity at which the chiller will be operating on its optimal efficiency curve at the current lift head conditions. One needs to notice that the "ideal centrifugal speed curve" (Figure 3) and the "optimal efficiency curve" (Figure 4) are unique to a specific chiller model and that such

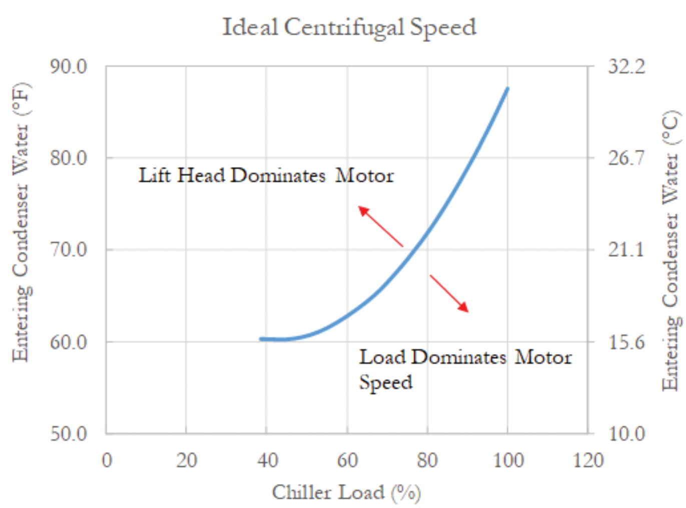

Figure 3 *Ideal centrifugal speed.*

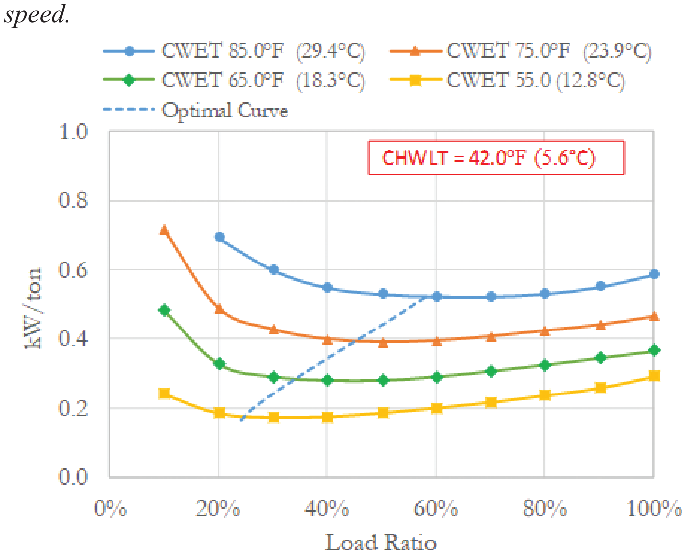

Figure 4 *Chiller performance and optimal efficiency curves.*

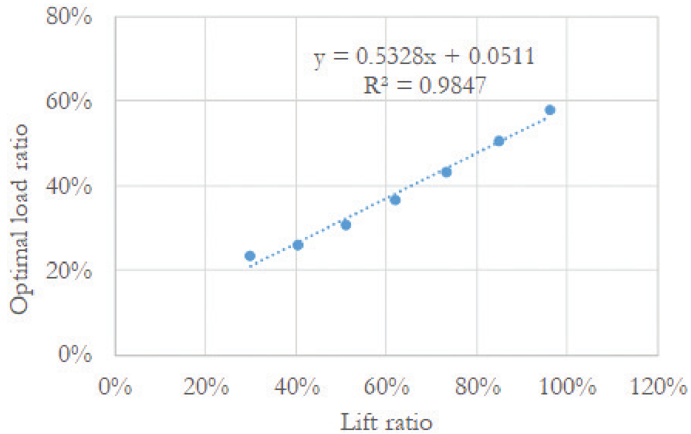

Figure 5 *Optimal load ratio model of case study chiller.*

curves will vary from model to model due to chiller performance changes.

The operation of a chilled-water plant should be close to the optimal efficiency curve to improve chiller efficiency. However, higher chiller efficiency will not guarantee optimum overall chiller plant performance. The operation of chilled-water pumps, condenser pumps and tower fans that serve each on-line chiller also affects the overall plant performance.

DESIGN INFORMATION

The case-study plant consists of four magnetic-bearing variable-speed centrifugal chillers, four variable-speed condenser water pumps, four variable speed chilled-water pumps and four induced draft type cooling towers with variable speed fans. One chilled-water pump, one condenser water pump and one cooling tower is energized for each chiller staged on. This control sequence is not optimum; Taylor [2012] suggested a better operation strategy to run as many tower cells as allowed by the minimum flow. The speed of chilled-water pumps and a bypass valve are modulated to maintain the chilled-water flow set point. The fan speed is modulated to maintain the tower cooling water leaving temperature (CWLT) set point. The condenser water (CW) flow rate can be adjusted by changing CW pump speed. The chiller is controlled to maintain a constant chilled-water leaving temperature.

The design information for the chillers, pumps and cooling towers is presented in Table 1, Table 2, and Table 3, respectively.

MODELS AND ASSUMPTIONS

A variable speed chiller model is developed based on manufacturer's performance data. The coefficient of determination r2 is 0.9928.

$$\frac{P_{CHLR}}{P_{design}} = \begin{pmatrix} -0.54838X + 0.4081xX^2 - 0.14978Y \\ +0.03426Y^2 + 0.4006XY + 0.0426 \end{pmatrix} \quad (2)$$

where

X = chiller partial load ratio ($0.10 \le X \le 1.05$)

Y = chiller lift ratio (CWLT – CHWLT)/Design Lift ($0.20 \le Y \le 1.05$)

Although the water flow rates are not input for this model, the effect of variable water flow can be captured by the CWLT and CHWLT. For example, when condenser water flow falls, the CWLT will rise, resulting in higher chiller energy consumption. Taylor [2011] found that the variable condenser water flow might make plant efficiency worse if not properly controlled because of this effect. Therefore, CWLT is selected as an input to simulate this effect. The minimum flow ratios of chilled water and condenser water are 45% and 60% respectively.

COOLING TOWER MODELING

The mass and heat transfer process in a cooling tower is complicated. The effectiveness-NTU model is a popular model in cooling tower simulations, but iterations are required to obtain a converged solution (Braun, 1989). The overall number of transfer units (NTU) can be correlated with the following form:

$$\mathrm{NTU} = c\left(\frac{\dot{m}_w}{\dot{m}_a}\right)^{1+n} \quad (3)$$

where

$\dot{m}_w$ = water mass flow

$\dot{m}_a$ = air mass flow

The value of c is between 1.0 and 3.0 for towers, and n ranges between –0.4 and –0.8 (Kreider et al., 2002). The cooling tower CWLT at the given airflow rate can be calculated by following equation:

Table 1. The Chiller Design Information

CHLRs	Capacity		CHW Flow		CW Flow		Efficiency		Power Input	Number	Compressor
	Ton	kW	GPM	L/S	GPM	L/S	kW/ton	COP	kW		
CHLR	800	2813	1,600	101	2,400	151	0.5868	5.993	469.4	4	Single

Table 2. The Cooling Tower Design Information

CTs	Design water flow		Min. Water Flow ratio	Entering Water Temp.		Leaving Water Temp.		Fan Motor	WB		Number
	GPM	L/S	%	°F	°C	°F	°C	kW	°F	°C	
CT	2400	151	60	95.0	35.0	85.0	29.4	37.25	80.0	26.7	4

Table 3. The Pump Design Information

Pumps	Design Water Flow		Head		Motor		Eff.	Number
	GPM	L/S	ft	m	hp	kW	%	
CHWP	1600	101	75	23	40	30	84	4
CWP	2400	151	80	24	75	56	80	4

Table 4. Simple Scenario of Chiller Staging for a Two-Chiller Plant

	Plant Load Tons, kW	Plant Load Ratio	Chiller Load Ratio	
			Two-Chiller Operation	One-Chiller Operation
Load 1	800.0 (2813)	50%	50%	100%
Load 2	640.0 (2251)	40%	40%	80%
Load 3	480.0 (1688)	30%	30%	60%
Load 4	320.0 (1125)	20%	20%	40%

$$T_{w,o} = T_{ref} + \left[\frac{\dot{m}_{w,i}(T_{w,i} - T_{ref})c_{pw} - \dot{m}_a(h_{a,o} - h_{a,i})}{\dot{m}_{w,o}c_{pw}}\right] \quad (4)$$

The cooling tower coefficients c (3.00) and n (–0.66) are identified using the manufacturer's design data. The root mean square error (RSME) and normalized root mean square deviation (NRMSD) are 1.56°F (0.87°C) and 2.2% respectively.

PUMP AND FAN MODELING

Because the chilled-water pump and condenser water pump speeds were operating at constant values of 82% and 92% respectively, trend data cannot be used to create a regression model. The pump models are developed based on the pumps' design information and metered power at fixed speed.

CHW Pump power kW model:

$$P_{CHWP} = 40 \times \text{Speed}^{2.75} \times 0.745 \quad (5)$$

CW Pump power kW model:

$$P_{CWP} = 60 \times \text{Speed}^{1.1} \times 0.745 \quad (6)$$

where the speed is normalized to 100% of rated design speed.

The CW pump model is not a cubic relationship because the condenser water loop is an open loop with a fixed tower height head. When the condenser water flow is reduced, the head of the condenser pump is not reduced as the power of the flow ratio. Therefore, the condenser water pump model deviates from the ideal fan law, where power varies as the cube of speed.

COOLING TOWER FAN MODEL

The cooling tower fan power and speed trend data are available to develop a fan power model as shown in Figure 6.

$$P_{fan} = 31.382 R_{airflow}^3 + 5.6256 R_{airflow}^2 - 1.4433 R_{airflow} \quad (7)$$

where

$$R_{airflow} = \frac{\text{Airflow}}{\text{Design Airflow}}$$

and

$$R_{airflow} \geq 20\%$$

CALIBRATION OF MODELS

The CHW trend data are available from August through December 2018, while the power for each of the system components (chillers, pumps and cooling tower fans) are only available from September through December 2018. Therefore, the trend data from September through December 2018 are used to calibrate the models. Inputs of simulation models are measured chilled-water flow, chilled-water supply and return temperature, condenser water flow, cooling tower fans speed and wet bulb temperature. The cooling tower model calculates cooling water leaving temperature from the tower for given hot water temperature, water flow, airflow and wet bulb temperature. The total power of the chiller compressor, condenser water pump, chilled-water pump and cooling tower fan motors are calculated by models.

$$P_{tot} = P_{CHLR} + P_{CHWP} + P_{CWP} + P_{fan} \quad (8)$$

A comparison of measured and predicted power for each component is shown in Figure 7. The error between predicted and measured total energy consumption of the whole plant over the four-month period is –0.2%. The root mean square error (RSME) and normalized root mean square deviation (NRMSD) are 267.5 kWh/day and 5.9% respectively.

CHILLER STAGE ANALYSIS

Before conducting an analysis of whole plant operation, the simulation models are applied to a simple scenario of two chillers staging. The design capacity of each chiller is 800.0 tons (2813kW) and the minimum load ratio is 20% (160 ton,

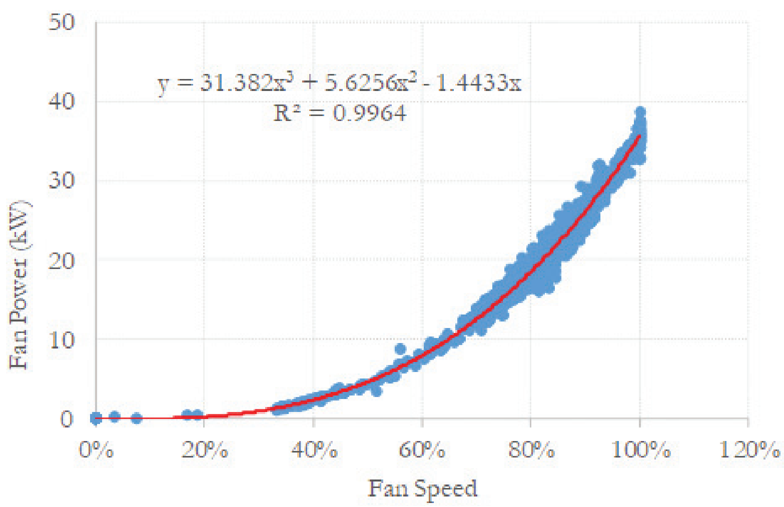

Figure 6 *Cooling tower fan power kW regression model.*

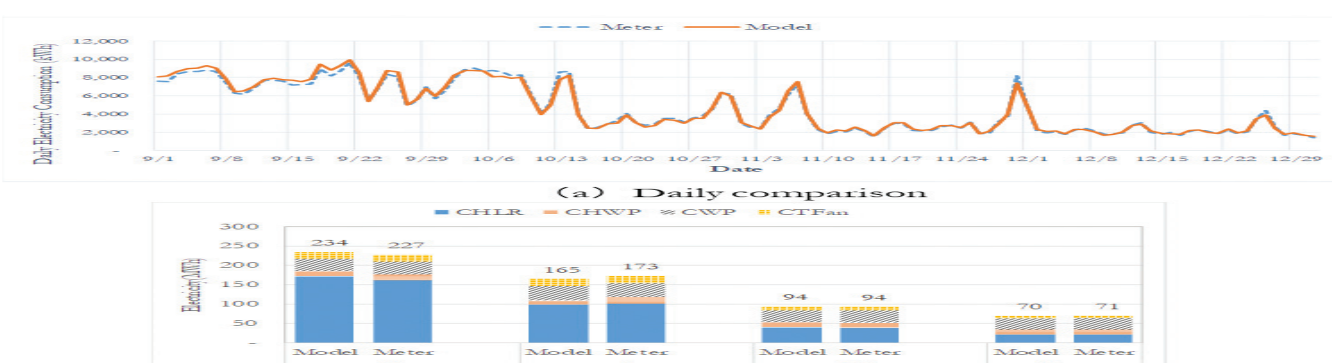

Figure 7 *Comparison of measured and predicted plant power consumption.*

563 kW). If plant-cooling load is between 320.0 tons (1125 kW) (20% plant load) and 800.0 tons (2813 kW) (50% plant load), it can be served by either one or two chillers. Because the oil-free variable speed centrifugal chillers operate most efficiently well below full load, it may reduce chiller power by operating two chillers rather than one. However, it may not reduce plant power consumption, because additional pumps and fans need to run to support another chiller on-line. Figure 8 presents the contradistinction of energy savings percentage for chiller power only versus whole CHW plant power (chiller + auxiliary) at various wet bulb temperature and load ratios.

Figure 8.a and Figure 8.b illustrate the results for a constant water flow (CWF) plant and a variable water flow (VWF) plant respectively. Figure 9 illustrates boundary curves for adding an on-line chiller. If the operating chiller load ratio is higher than the load ratio limit lines shown in the figure, the energy savings will be positive to stage on one more chiller. With the higher wet bulb temperature, the load ratio boundary is higher. For example, if wet bulb temperature is lower than 60.0°F (15.6°C), a chiller can be added on line when the operating chiller load ratio is higher than 40%. However, when wet bulb temperature is 75.0°F (23.9°C), the operating chiller load ratio should be higher than 70% to add a chiller on-line. For a constant water flow system, the boundary load ratio will be even higher. If the wet bulb temperature is higher than 70.0°F (21.1°C), it will be more efficient to run one chiller than two chillers. For a variable water flow system, the minimum load ratio is 60% and the maximum wet bulb temperature is 85.0°F (29.4°C). If wet bulb temperature is higher than 85.0°F (29.4°C), operating one chiller at full load is more efficient than two chillers.

CHILLER PLANT OPTIMIZATION

As discussed above, the optimal chiller staging strategy is affected by many factors, such as chiller load ratio, wet bulb temperature, the speeds of all pumps and fans, etc. An analysis

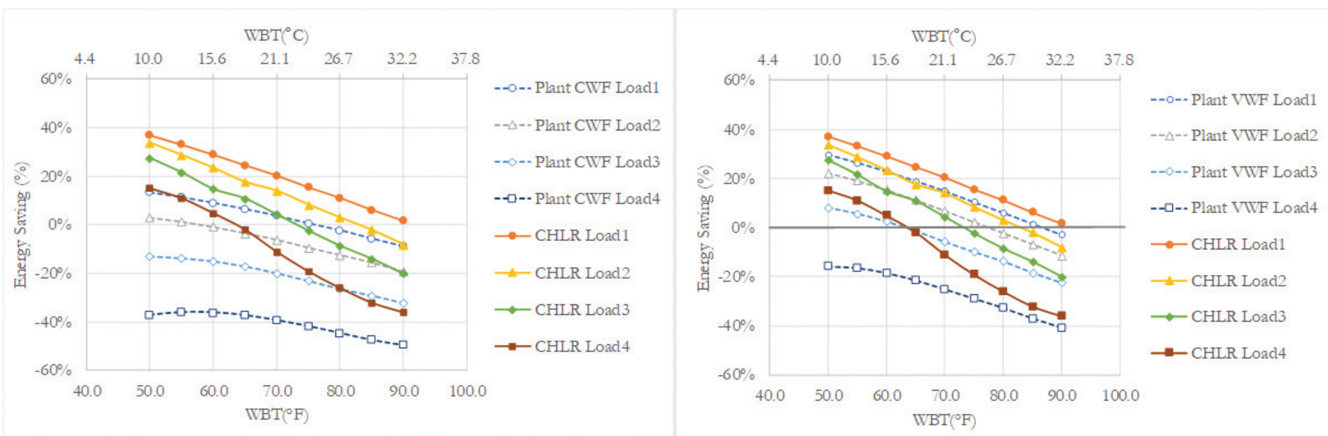

Figure 8 *Energy savings of operating two chillers versus one chiller: (a) constant water flow plant and (b) variable water flow plant.*

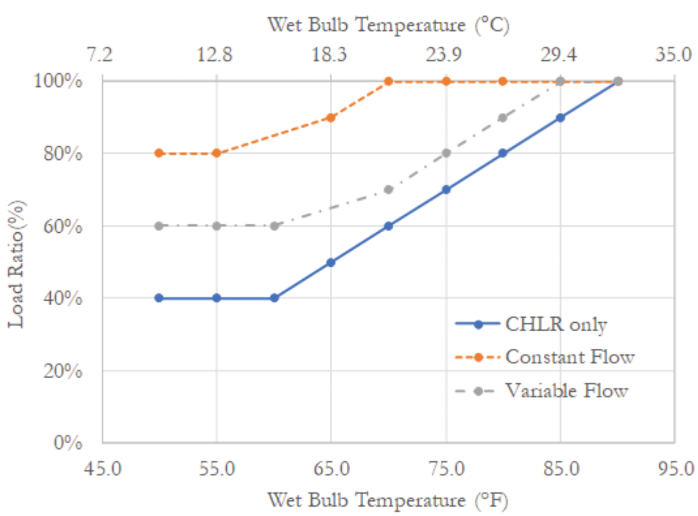

Figure 9 *Load ratio curves for adding a chiller on-line.*

is conducted to calculate the energy savings potential for different chiller plant control strategies serving the cooling load metered by the BAS from August through December 2018. Although the chilled water and condenser water pumps have variable speed drives, the plant has been operating with these pumps set to constant water flow. Therefore, the following three scenarios are simulated to evaluate energy savings potential for the case study plant.

a. Scenario 1: Minimize the number of on-line chillers; Constant water flow
b. Scenario 2: Optimize the number of on-line chillers; Constant water flow
c. Scenario 3: Optimize the number of on-line chillers; Variable water flow

A strong tie exists between the operation of chillers and the heat rejection systems. Hartman [2001] suggested the following rules for operating variable speed chiller plants efficiently:

a. When variable speed chillers are used, optimum performance and simplicity of operation is achieved when all chillers in the plant are variable speed and have identical performance characteristics.
b. The focus must be on operating chillers at equal loading and as near as possible to their optimal efficiency curves, while also coordinating chiller, condenser pump, and tower fan power to minimize the overall plant power consumption at each load condition.

Operating an all-variable speed chiller plant that consists of variable speed chillers, condenser pumps and tower fans at less than full capacity conditions leads to the greatest operating efficiency. However, there are limits to the amount of load reduction that can be accommodated by slowing the equipment.

The optimal efficiency curve is used to establish a sequencing strategy for variable speed chillers, and coordinating variable speed operation of condenser pumps and tower fans with chiller input loadings. The technique operates all equipment as close as possible to the curve of highest operating efficiency. The chilled-water flow and condenser water flow is linearly adjusted with the part-load ratio. The cooling tower fan speed is modulated to achieve the minimum total plant power consumption.

Per Figure 5, the optimal load ratio of the case study chiller model can be approximated as a function of the difference in condenser and chiller water temperatures compared to the design difference of those temperatures.

$$\mathrm{PLR}_{opt} = 0.5328 R_{lift} + 0.0511 \tag{9}$$

where

PLR_{opt} = load ratio at which the chiller will be operating on its optimal efficiency curve at the current condenser conditions.

R_{lift} = chiller lift ratio (CWLT – CHWLT)/Design Lift

The number of on-line chillers, N, can be calculated using the following equations to operate the chilled-water plant as close as possible to the optimal efficiency curve of the operating chillers.

$$\begin{aligned} \mathrm{PLR} &= \mathrm{Max}(\mathrm{PLR}_{opt}, 25\%) \\ n &= \mathrm{Max}\left(\mathrm{Round}\left(\frac{\mathrm{Load}}{\mathrm{PLR} \times \mathrm{Ton}_{design}}\right), N_{max}\right) \\ N_{min} &= \mathrm{Roundup}\left(\frac{\mathrm{Load}}{\mathrm{Ton}_{design}}\right) \\ N &= \mathrm{Max}(n, N_{min}) \end{aligned} \tag{10}$$

where

Load = total plant-cooling load

PLR = part-load ratio of each chiller

Ton_{design} = design cooling capacity of a chiller

N = number of on-line chillers

The energy savings and number of on-line chillers of scenario 1 and scenario 2 are presented in Figure 10 and Figure 11 respectively. Figure 10 shows that the maximum tonnage of a single chiller is 800.0 tons (2813kW) for the baseline operation (scenario 1).

For optimal chiller-staging strategy, the number of on-line chillers is more than the minimal chiller strategy. The reduction of chiller energy consumption (kWh) is about 4.4% over the 5 month period for Scenario 2 relative to Scenario 1. However, the energy consumption of pumps increases for Scenario 2, so the overall electricity energy reduction is only 2.1%. For a variable water flow system, the plant performance is improved from 0.574kW/ton to 0.495 kW/ton. The energy consumption reduction of Scenario 3 is 13.7% for the 5 month time period (August through December). All weather conditions (summer, winter and swing season) are included in this period. Thereore, a similar conclusion can be drawn for the annualized savings percentage. In addition, different savings percentages will be achieved with different weather conditions and load profiles.

CONCLUSIONS

Magnetic bearing centrifugal chillers with variable-speed drives perform much more efficiently at part-load ratio of cooling demand as well as at partial compression ratio of lift head. These performance differences require reconsideration of most effective plant operation when magnetic bearing centrifugal chillers are incorporated. The results of two different chiller staging analyses indicate that the operation of pumps and towers also has a significant impact on overall plant performance, in addition to chiller staging. When outside wet

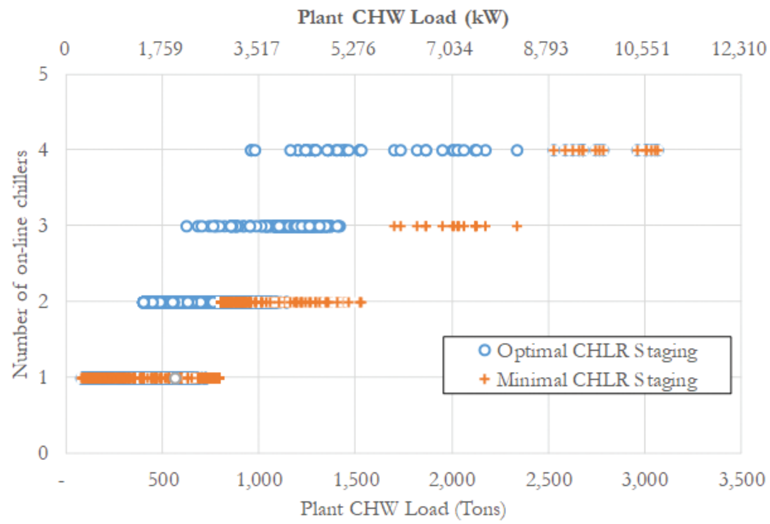

Figure 10 *Number of on-line chillers.*

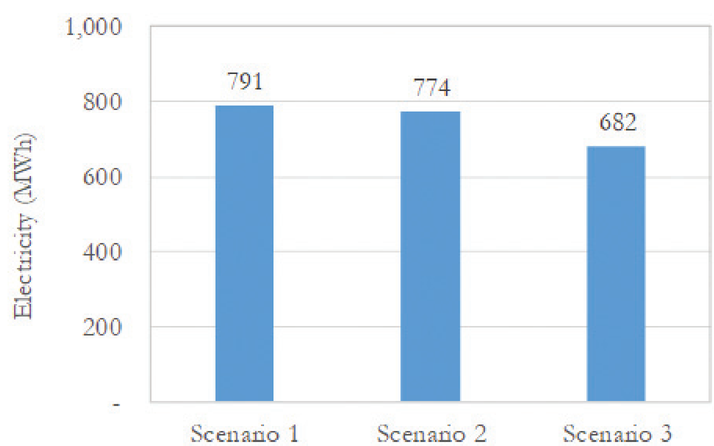

Figure 11 *Plant power consumption of different scenarios for loads of August through December 2018.*

bulb temperature is higher than 70.0°F (21.1°C), it will not be beneficial to run two chillers instead of one for the constant water flow system analyzed. However, for the variable water flow system analyzed, there is a noticeable load zone where implementing optimal chiller staging will save energy.

The optimal efficiency curve is used to establish a sequencing strategy for the case study plant. The technique operates all equipment as close as possible to the curve of highest operating efficiency. A calibrated component-based model has been implemented to calculate the total power of chiller compressor(s), condenser water pump(s), chilled-water pump(s) and fan motors for three different scenarios. The results show that optimal chiller staging only improves plant performance by 2.1% for a constant water flow system. Implementing optimal staging with variable water flow would increase the energy savings to 13.7% compared to current operation of the case-study plant.

NOMENCLATURE

IPLV = integrated part-load value
MBVS = magnetic bearing variable speed
CS = constant speed
VS = variable speed
CWET = condenser water entering temperature
CWLT = condenser water leaving temperature
CHWLT = chilled-water leaving temperature
CW = condenser water
CHW = chilled water
CT = cooling tower
CWP = condenser water pump
CHLR = chiller
PPMP = primary pump
DB = dry bulb temperature, °F (°C)

WB	=	wet bulb temperature, °F (°C)
P	=	power, kW
NTU	=	number of transfer units
c	=	cooling tower model coefficients
c_{pw}	=	water heat capacity, Btu/lbm.°F (W/kg.°C)
$\dot{m}$	=	mass flow, lbm/hr (kg/s)
n	=	cooling tower model index
T	=	temperature, °F (°C)
GPM	=	gallons per minute
PLR	=	part-load ratio
VSD	=	variable-speed drive
x	=	independent variables
y	=	independent variables
V	=	flow rate, gpm

Greek Symbols

ρ	=	density, lbm/ft (kg/m^3)

Subscripts

a	= air
w	= water
o	= outlet
i	= inlet
ref	= reference

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