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Large University Central Chiller Plant Design Considerations

Large campus chilled water plants have unique constraints and need careful evaluation for successful chiller plant master planning. Similar to reviewing a fan curve for centrifugal fans and pump curves for centrifugal pumps, the centrifugal chiller compressor map needs to be evaluated to better apply and understand the operating envelope the chiller is capable of operating in. This newsletter illustrates various ways to better select and apply centrifugal chillers to an application. Several constraints and potential solutions are discussed in this newsletter, as well as various construction and design options that will help deliver an efficient, reliable, and robust chiller plant for the university.

Introduction

When planning a plant expansion or replacing end of life equipment within large university central plants, there are many design issues and recommendations that need to be considered to ensure a highly reliable, efficient, low maintenance central plant design. For the purposes of this newsletter, the design considerations below are specific to chiller plants utilizing multiple chillers that are 1500 tons and above.

Common Issues for Large University Central Plants

Every large central plant has unique considerations that must be addressed due to various constraints specific to the installation. Some of the potential chiller plant constraints, such as plants that are distribution limited or space constrained, or plants that need additional redundancy are discussed with examples of possible solutions to the design constraint.

Example 1: Existing Chilled Water System Is Distribution Limited

Consider an existing university chilled water system that was designed originally for the following criteria:

- Total peak campus load of 15,000 tons
- Design chilled water temperatures of 52°F return chilled water; 40°F chilled water temperature
- Tower water system designed for 85°F supply, 95°F return water, and 78°F wet-bulb design
- Distribution piping is directly connected to each airside load within each campus building (no intermediate heat exchanger at the building level) and the header consists of a looped piping network
- Variable primary pumping system will increase by 5,000 tons (total load of 20,000 tons) in the next 10 years.

The flow rate through the piping for the original design conditions was 30,000 gpm as validated by equation 1 below:

$$q = \dot{m} c_p \Delta T$$

where

q =	total heat transfer, Btu/hr
$\dot{m}$ =	mass flow rate, lb/hr
c_p =	specific heat, Btu/lb-°F
ΔT =	temperature difference, °F

Mass flow rate for the original design case was therefore:

$$\frac{15{,}000\ Tons \times 12{,}000\ \frac{Btu}{hr}/Ton)}{1\ \frac{Btu}{lb - °F} \times (52°F - 40°F)} = 15{,}000{,}000\ lb/hr$$

Specific heat of water from *2013 ASHRAE Fundamentals.*[1]

Converting lb/hr to gallons per minute (gpm) results in a system flow rate of 30,000 gpm (or 15,000 gpm in each loop of the distribution network). Figure 4, Friction Loss for Water in Commercial Steel Pipe (Schedule 40), in *ASHRAE Fundamentals*[2] shows for 24 inch piping the velocity is 15 fps (upper limit as recommended in Table 9, Maximum Water Velocity to Minimize Erosion[3]).

1 American Society of Heating, Refrigerating and Air-Conditioning Engineers, *2013 ASHRAE Handbook: Fundamentals* (Atlanta, GA: American Society of Heating, Refrigeration and Air-Conditioning Engineers, 2013), F33.2.

2 *2013 ASHRAE Handbook: Fundamentals, F22.7.*

3 *2013 ASHRAE Handbook: Fundamentals, F22.5.*

In this example the piping network is the main constraint (no additional flow can be pushed through the piping network and it is already in a looped arrangement). One potential solution is to lower the chilled water set point for all future chillers to enable more tons at the same flow rate. However, the airside loads have to be evaluated to ensure reliable control of discharge air temperature.

The first step in the evaluation is to determine what the revised chilled water set point would need to be to satisfy the new 20,000 ton load with 30,000 gpm of flow.

Using the following simplified heat duty calculation, the leaving chilled water temperature can be found:

$$52°F - \frac{20{,}000\ \mathit{Tons} \times 12{,}000}{500 \times 30{,}000\ \mathit{gpm}} = 36°F$$

As shown the required chilled water supply temperature is 36°F. One strategy to meet this future load would be to begin replacing aging chillers with new chillers capable of operating reliably over a wide operating envelope that can satisfy the lower 36°F supply temperature, as well as operate at the higher reset chilled water temperature during non-peak times to conserve energy.

Chiller selection techniques will be discussed later in this newsletter.

Evaluation of Air Side Systems with 36°F Supply Water

Now that the required chilled water supply temperature is known, it is necessary to consider the impact at the building level. Note: In this example the piping is directly connected to the airside building coils; in a system with individual heat exchangers isolating each building, the controls would simply make sure the building loads receive the original design temperature.

Figure 1 shows the coil performance with the original design conditions. Figure 2 shows the revised coil performance. As shown the 6-row coil in the large airside systems modulates the flow lower to maintain the required performance.

In operation the variable air volume fan would reduce airflow slightly to match room load and all space loads will be satisfied.

Figure 1 – Original Coil Selection with 40°F CHWS

Cooling Application's Balance Criteria: Fluid Flow

Coil Model	**Carrier 28MD**	
Row / FPI / Circ	**6 / 8 / FL**	
Fin Type	**Sine Wave**	
Face Area Type	**Large**	
Coil Face Area	**49.83**	sqft
Face Velocity	**501.7**	fpm
Fin-Casing Material	**Al-Galv.**	
Tube Diameter	**0.625**	in
Tube spacing: Stf x Str	**1.5 x 1.3**	in
Tube Wall Thickness	**0.020**	in
Actual Airflow	**25000**	CFM
Site Altitude	**0**	ft
Total Cooling Capacity	**1266.00**	MBH
Sensible Cooling Capacity	**675.90**	MBH
Fluid Flow Rate	**210.8**	gpm
Fluid Pressure Drop	**19.7**	ft wg
Fluid Velocity	**4.9**	ft/s
Entering Fluid Temperature	**40.00**	F
Leaving Fluid Temperature	**52.00**	F
Fluid Temperature Rise	**12.0**	F
Entering Air Dry Bulb	**75.00**	F
Entering Air Wet Bulb	**67.00**	F
Entering Air Enthalpy	**31.47**	BTU/lb
Leaving Air Dry Bulb	**50.00**	F
Leaving Air Wet Bulb	**49.80**	F
Leaving Air Enthalpy	**20.2**	BTU/lb
Air Friction	**0.82**	in wg
Brine	**FW**	
Brine Concentration	**0**	%
Fouling Factor	**0.0**	(hr-sqft-F)/BTU

NOTE: Certified in accordance with the AHRI Forced-Circulation Air-Cooling and Air-Heating Coils Certification Program which is based on AHRI Standard 410 within the Range of Standard Rating Conditions listed in Table 1 of the Standard. Certified units may be found in the AHRI Directory (www.ahridirectory.org).

It may be possible to overcome distribution limited chilled water systems and satisfy space loads with a lower chilled water set point. A few cautionary notes are provided below:

- Figures 1 and 2 show the chilled water flow rate dropped from 210 gpm to 166 gpm. If the original valve was already oversized, there may be controllability issues at the lower flow rate and reduced load. Detailed evaluation of existing valve characteristics needs to be performed.
- Larger coils in air handlers typically perform the same with lower temperatures; however, in systems with a high number of small 1 or 2 row coils, the loads will be met but the delta T may be lower than 16°F.
- An evaluation of piping insulation values is required to ensure the exposed piping insulation is sufficient to prevent condensation.

A comprehensive plan will be needed to schedule chiller replacement and to ensure sufficient replacement chillers capable of producing the colder water without surge are in place in line with the load growth schedule.

Example 2: Existing Cooling Tower System Is Space Constrained

This example uses the same university system as described in Example 1; however, in this example we will examine the situation in which the condenser water system is constrained. Assume the existing tower system is a 5-cell cooling tower arrangement, designed for 15,000 tons of chiller capacity, cooling 45,000 gpm of water from 85 to 95°F.

The original tower selection information is shown in Table 1.

Figure 2 – Revised Coil Performance with 36°F CHWS

Cooling Application's Balance Criteria: Fluid Flow

Coil Model	**Carrier 28MD**	
Row / FPI / Circ	**6 / 8 / FL**	
Fin Type	**Sine Wave**	
Face Area Type	**Large**	
Coil Face Area	**49.83**	sqft
Face Velocity	**501.7**	fpm
Fin-Casing Material	**Al-Galv.**	
Tube Diameter	**0.625**	in
Tube spacing: Stf x Str	**1.5 x 1.3**	in
Tube Wall Thickness	**0.020**	in
Actual Airflow	**25000**	CFM
Site Altitude	**0**	ft
Total Cooling Capacity	**1331.10**	MBH
Sensible Cooling Capacity	**713.60**	MBH
Fluid Flow Rate	**166.1**	gpm
Fluid Pressure Drop	**12.6**	ft wg
Fluid Velocity	**3.8**	ft/s
Entering Fluid Temperature	**36.00**	F
Leaving Fluid Temperature	**52.00**	F
Fluid Temperature Rise	**16.0**	F
Entering Air Dry Bulb	**75.00**	F
Entering Air Wet Bulb	**67.00**	F
Entering Air Enthalpy	**31.47**	BTU/lb
Leaving Air Dry Bulb	**48.60**	F
Leaving Air Wet Bulb	**48.60**	F
Leaving Air Enthalpy	**19.6**	BTU/lb
Air Friction	**0.81**	in wg
Brine	**FW**	
Brine Concentration	**0**	%
Fouling Factor	**0.0**	(hr-sqft-F)/BTU

NOTE: Certified in accordance with the AHRI Forced-Circulation Air-Cooling and Air-Heating Coils Certification Program which is based on AHRI Standard 410 within the Range of Standard Rating Conditions listed in Table 1 of the Standard. Certified units may be found in the AHRI Directory (www.ahridirectory.org).

Table 1 — Original Tower Performance

Conditions			
Tower Water Flow	45000 gpm	Air Density In	0.07094 lb/ft^3
Hot Water Temperature	95.00 °F	Air Density Out	0.07085 lb/ft^3
Range	10.00 °F	Humidity Ratio In	0.01712
Cold Water Temperature	85.00 °F	Humidity Ratio Out	0.03124
Approach	7.00 °F	Wet-Bulb Temp. Out	90.05 °F
Wet-Bulb Temperature	78.00 °F	Estimated Evaporation	460 gpm
Relative Humidity	50.0 %	Total Heat Rejection	224210000 Btu/h
Capacity	103.0 %		

The original selection had limited spare capacity (3%). The existing plant has 5 chillers all rated at 3000 tons, 85 to 95°F. Under the replacement plan, consideration is being given to replace two older 3000 ton chillers and adding two new 2500 ton chillers to expand the plant capacity to 20,000 tons. See Tables 2 and 3.

Table 2 — Performance of New 2,500 Ton Chillers with 85 to 105°F and 3-Pass Condenser

Cooler	
Entering Temp.	52.00 °F
Leaving Temp.	40.00 °F
Flow Rate	4989.1 gpm
Pressure Drop	26.2 ft wg
Condenser	
Leaving Temp.	105.00 °F
Entering Temp.	85.00 °F
Flow Rate	3590.6 gpm
Pressure Drop	12.0 ft wg

Table 3 — Performance of Replacement 3,000 Ton Chillers with 85 to 105°F and 3-Pass Condenser

Cooler	
Entering Temp.	52.00 °F
Leaving Temp.	40.00 °F
Flow Rate	5986.9 gpm
Pressure Drop	36.3 ft wg
Condenser	
Leaving Temp.	105.00 °F
Entering Temp.	85.00 °F
Flow Rate	3590.6 gpm
Pressure Drop	12.0 ft wg

Selecting the new and replacement chillers to operate with 85 to 105°F condenser water temperatures (and using a 3-pass condenser) allows for a blended average of 98.7°F return condenser water temperature and reduces the total tower flow rate from 45,000 gpm to 42,834 gpm. Tower curves are shown in Figure 3 (top curve is a 13.7°F range) showing that the existing tower at reduced flow rate and higher range can produce 86°F tower water.

If the remaining three towers are capable of operating with 1°F higher ECWT, this may be a viable approach to increasing plant capacity in a scenario where expansion is not possible.

As shown in this example, it is possible to expand plant capacity without adding cooling tower capacity; however, this approach must be thoroughly evaluated against other options because of the higher energy use of the chilled water system due to the higher lift requirements. In situations where it is physically not possible to add cooling tower cells this solution can be feasible.

Example 3: Planning for Cooling Tower Cell Failure

This example will evaluate the impact on chiller operation during a component failure. The typical approach to achieving N+1 redundancy in the chiller plant design is to add one additional cooling tower cell. For universities that have already undergone plant expansion and don't have the luxury of additional space or for projects that are budget constrained, adding an additional cell may not be feasible. A possible way to alleviate the need for the additional cooling tower cell is to calculate the revised condenser loop conditions during the upset condition and select all replacement chillers to operate with this wide operating envelope.

Consider the following university chiller plant design situation:

- Total peak campus load of 9,000 tons
- Design chilled water temperatures of
  52°F return chilled water;
  42°F chilled water temperature
- Tower water system designed for 85°F supply, 95°F return water, and 78°F wet-bulb design
- Variable primary pumping system
- Three 3000 ton chillers, three chilled water pumps, three tower cells, three condenser water pumps

Cooling tower performance with these design conditions is shown in Table 4.

As shown, the total tower flow is 27,000 gpm divided over three cells. During an upset condition (one tower cell down) the revised condenser loop conditions can be calculated.

Figure 4 shows the system schematic with the single tower fan failure; Figure 5 is a plot of condenser water temperature versus time after fan failure. In this example the total system volume contained in the tower sump, condenser piping, and chiller bundles is 41,000 gallons, resulting in a loop cycle time of 1.52 minutes.

As illustrated in Figures 4 and 5, the upset condition of 92.2 to 101.8°F condenser conditions is reached in approximately 15 minutes of operation. Selecting the chillers capable of operating at this condition and designed to operate at 85 to 95°F eliminates the need for the additional cooling tower cell.

Figure 3 – Cooling Tower Performance Curve at Revised Conditions

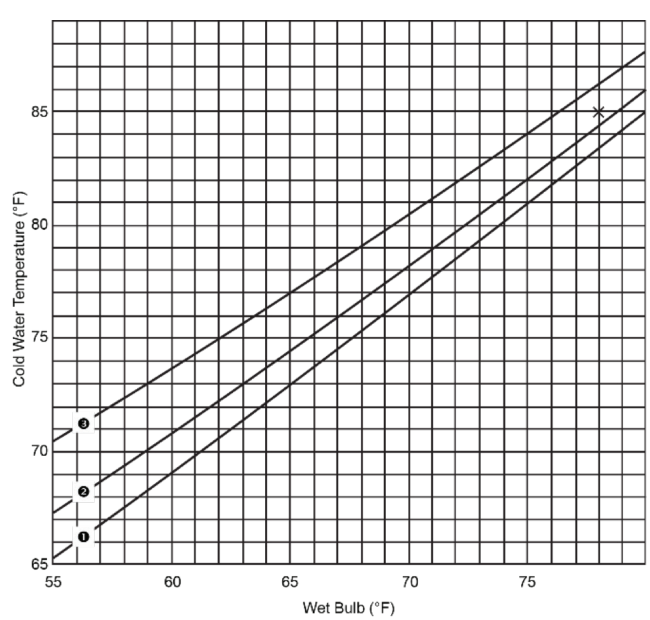
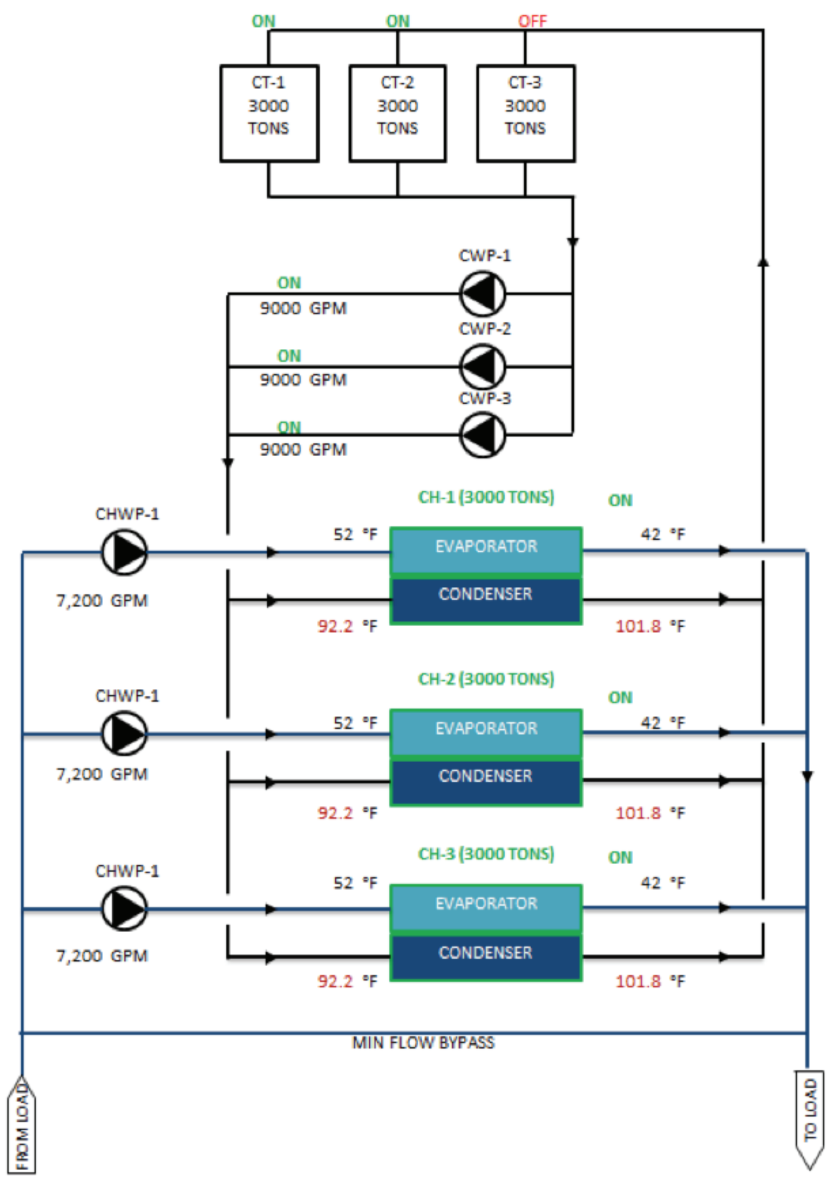


Table 4 — Cooling Tower Performance with All Cells in Operation

Conditions			
Tower Water Flow	45000 gpm	Air Density In	0.07094 lb/ft^3
Hot Water Temperature	95.00 °F	Air Density Out	0.07085 lb/ft^3
Range	10.00 °F	Humidity Ratio In	0.01712
Cold Water Temperature	85.00 °F	Humidity Ratio Out	0.03124
Approach	7.00 °F	Wet-Bulb Temp. Out	90.05 °F
Wet-Bulb Temperature	78.00 °F	Estimated Evaporation	460 gpm
Relative Humidity	50.0 %	Total Heat Rejection	224210000 Btu/h
Capacity	103.0 %		

Figure 4 — Revised Plant Conditions with One Tower Cell Offline



Chiller Selection Recommendations to Address the Above Issues

Before discussing chiller selections it is important to define a few terms:

- **Lift:** Pressure/Temp differential across compressor; defined as Condenser LWT – Evaporator LWT
- **Compressor Surge:** When refrigerant reverses direction or stops flowing through the compressor
- **Stonewall:** Maximum capacity of compressor at given lift
- **Surge Limit:** The maximum lift at which the compressor can operate
- **Rise to Surge:** Amount of spare lift (°F) the compressor is capable of at a given capacity (tons)
- **Turndown to Surge:** Minimum operating capacity of the compressor with constant design entering condenser water temperature

Figure 6 illustrates the key terms. Typically chiller specifications only require the chiller be capable of operating from 100 to 10% load with constant design entering condenser water temperature of 85°F. This is a good starting point but not sufficient if operation is required beyond these conditions. With modern variable speed centrifugal chillers, it is possible to specify a minimum rise to surge value and only incur a small efficiency decline while providing for a much more robust and surge resistant chiller plant design. Recommended language for specifying rise to surge and turndown is shown below:

*Chiller shall have a minimum rise to surge of **x°F from 100% to x% load with constant design entering condenser water temperature.*

The minimum turndown percent and rise to surge (°F) should be determined based on the system design specific to the university system and load profile.

Now the terms are defined and the concept is complete for how to specify a robust chiller plant, it is important to evaluate the impact this specification has on the energy efficiency of the chiller plant.

Consider the efficiency of a 3,000 ton chiller selected to operate with 42°F chilled water and 85 to 95°F condenser water. Performance for this chiller is shown in Table 5.

Figure 5 — Condenser Water Temperature versus Time after Cooling Tower Fan Failure Occurs at T=0 minutes

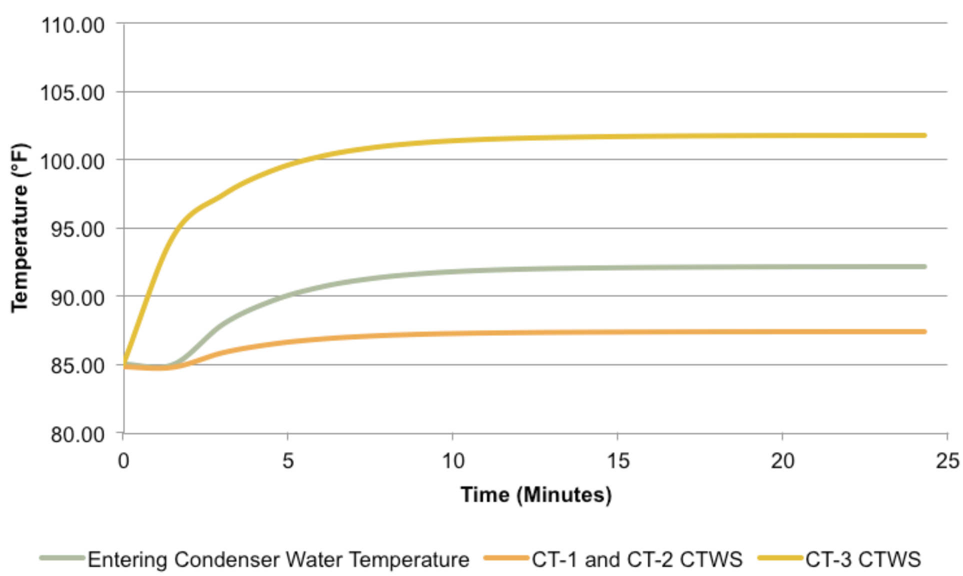


Table 5 — Performance of 3000 Ton Chiller Selected for 85 to 95°F Condenser Water

Output Type	Full Load
Percent Load	100.00
Chiller Capacity	3000.0 tons
Chiller Input kW	1738.7 kW
Chiller Input Power	0.5796 kW/ton
Chiller COP	6.068
Cooler	
Entering Temp.	56.00 °F
Leaving Temp	42.00 °F
Flow Rate	5136.9 gpm
Pressure Drop	23.3 ft wg
Condenser	
Leaving Temp.	94.31 °F
Entering Temp.	85.00 °F
Flow Rate	9000.0 gpm
Pressure Drop	15.6 ft wg
Motor	
Motor Rated Load	278.5 amps
Motor Locked Rotor	1594 amps
Chiller Rated Line	250 amps
Chiller Inrush	250 amps
Max Fuse/CB	500 amps
Min Circuit Ampacity	312 amps

This chiller selection operates with a 0.5796 kW/ton at design conditions of 85°F entering condenser water temperature. As per typical selections this chiller will not be able to maintain leaving chilled water temperature without surging when the entering condenser water temperature rises above 86.5°F (1.5°F rise to surge at full load).

Consider the second chiller selection shown in Table 6. This chiller was selected with a minimum rise to surge of 5°F from 100 to 15% load (entering condenser water at design of 90°F).

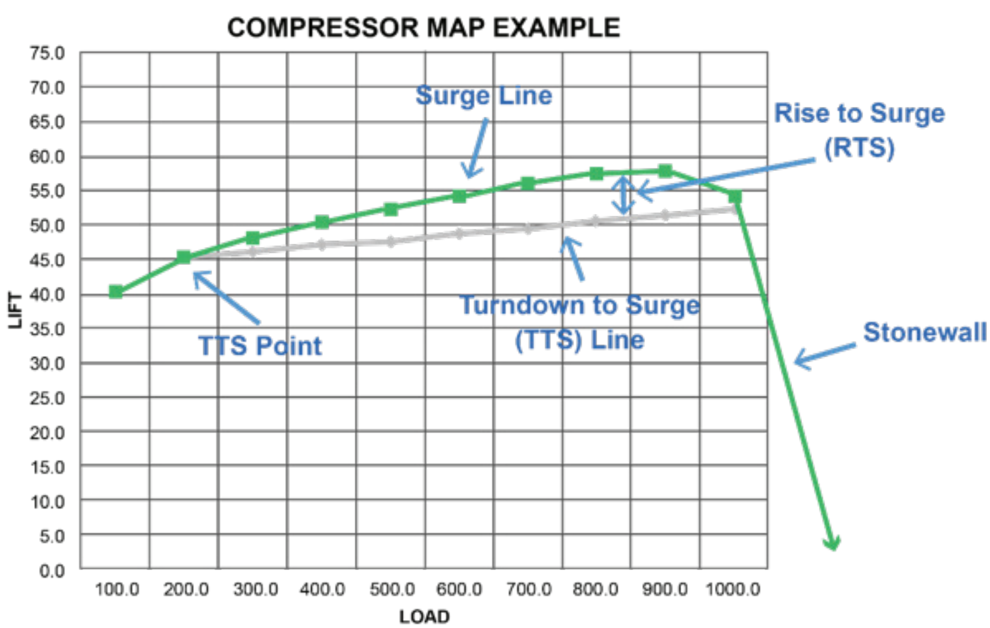
Table 7 shows the same chiller re-rated at the design of 85°F. The total increase in power at full load design is less than 30 kW (0.5888 kW per ton versus original of 0.5796).

This small increase in power consumption now allows for a robust plant design that can operate with a very wide envelope. The chiller selection can operate under the following conditions:

- 100 to 15% load with constant 90°F entering condenser water
- 100 to 15% load with 55°F entering condenser water temperature
- Increased power consumption of less than 30 kW over the first selection that loses control of leaving chiller water temperature with 1.5°F above design entering condenser water
- Price increase is less than 2% in this example (slightly larger motor to handle the higher lift)

One item to note for designing chillers with higher entering condenser water temperature: The electrical systems need to be designed per the electrical load requirements shown in Table 6 (at 90°F entering condenser water).

Figure 6 — Example Compressor Map for Defining Key Terms



As shown in the preceding example there is a trade-off between efficiency and robust design; however, if balanced properly during the design phase, the careful analysis will result in a highly reliable, low maintenance chiller plant that can operate over a wide range of commonly seen ambient and load conditions.

Table 6 — Performance of 3000 Ton Chiller Selected for 90 to 100°F Condenser Water

Output Type	**Full Load**
Percent Load	100.00
Chiller Capacity	3000.0 tons
Chiller Input kW	1882.7 kW
Chiller Input Power	0.6276 kW/ton
Chiller COP	5.604
Cooler	
Entering Temp.	56.00 °F
Leaving Temp.	42.00 °F
Flow Rate	5136.9 gpm
Pressure Drop	23.3 ft wg
Condenser	
Leaving Temp.	99.42 °F
Entering Temp.	90.00 °F
Flow Rate	9000.0 gpm
Pressure Drop	21.1 ft wg
Motor	
Motor Rated Load	304.5 amps
Motor Locked Rotor	1594 amps
Chiller Rated Line	271 amps
Chiller Inrush	271 amps
Max Fuse/CB	600 amps
Min Circuit Ampacity	338 amps

Table 7 — Same Chiller Rated with 85°F Entering Condenser Water

Output Type	**Full Load**
Percent Load	100.00
Chiller Capacity	3000.0 tons
Chiller Input kW	1766.3 kW
Chiller Input Power	0.5888 kW/ton
Chiller COP	5.973
Cooler	
Entering Temp.	56.00 °F
Leaving Temp.	42.00 °F
Flow Rate	5136.9 gpm
Pressure Drop	23.3 ft wg
Condenser	
Leaving Temp.	94.32 °F
Entering Temp.	85.00 °F
Flow Rate	9000.0 gpm
Pressure Drop	21.1 ft wg
Motor	
Motor Rated Load	296.5 amps
Motor Locked Rotor	1770 amps
Chiller Rated Line	254 amps
Chiller Inrush	254 amps
Max Fuse/CB	500 amps
Min Circuit Ampacity	317 amps

Chiller Options to Consider for Robust, Low Maintenance Operation

The final consideration for chiller plant design is the selection of options and accessories during the specification process.

Option 1: NEMA (National Electrical Manufacturers Association) Rating for Control and Power Panelsy

The typical NEMA rating for standard chiller products is NEMA 1. Figure 7 shows a comparison of NEMA 1 enclosures for protection from falling dust/dirt and for preventing access to hazardous parts. NEMA 1 is typically limited to indoor locations with filtered makeup air. The typical university mechanical room does not meet this requirement due to the lack of filtration on the existing ventilation systems. In this case it might be worthwhile to specify NEMA 12 for the control and power panels. In addition to the NEMA 1 protection requirements, NEMA 12 provides a degree of protection for the following:

- Ingress of solid foreign objects (falling dirt, circulating dust, lint, fibers, and flyings)
- Limited protection due to the ingress of water (dripping and light splashing)

For a relatively small price increase, the electrical enclosures and control panels can be upgraded to a higher rating type based on the types of hazards present at the particular university mechanical room.

Figure 7 — NEMA Indoor Enclosure Types from NEMA 250 [4]

Table 1
[From NEMA 250-2003]
Comparison of Specific Applications of Enclosures
for Indoor Nonhazardous Locations

Provides a Degree of Protection Against the Following Conditions	**Type of Enclosure**									
	1 *	**2 ***	**4**	**4X**	**5**	**6**	**6P**	**12**	**12K**	**13**
Access to hazardous parts	X	X	X	X	X	X	X	X	X	X
Ingress of solid foreign objects (falling dirt)	X	X	X	X	X	X	X	X	X	X
Ingress of water (Dripping and light splashing)	...	X	X	X	X	X	X	X	X	X
Ingress of solid foreign objects (Circulating dust, lint, fibers, and flyings **)	...	...	X	X	...	X	X	X	X	X
Ingress of solid foreign objects (Settling airborne dust, lint, fibers, and flyings **)	...	...	X	X	X	X	X	X	X	X
Ingress of water (Hosedown and splashing water)	...	...	X	X	...	X	X	...	...	...
Oil and coolant seepage	...	...	...	..	...	...	...	X	X	X
Oil or coolant spraying and splashing	...	...	...	...	...	...	...	...	...	X
Corrosive agents	...	...	...	X	...	...	X	...	...	...
Ingress of water (Occasional temporary submersion)	...	...	...	...	...	X	X	...	...	...
Ingress of water (Occasional prolonged submersion)	...	...	...	...	...	...	X	...	...	...

* These enclosures may be ventilated.
** These fibers and flyings are nonhazardous materials and are not considered Class III type ignitable fibers or combustible flyings. For Class III type ignitable fibers or combustible flyings see the National Electrical Code, Article 500.

4 *http://ipi.ir/standard/STANDS/NEMA/250.pdf*

Option 2: Protection of Medium Voltage VFDs (Variable Frequency Drives)

Medium Voltage VFDs are a large component of the purchase price of large chillers (often 50% or more of the purchase price of the chiller). Overlooking the installation requirements of the VFD can lead to premature failure of the VFD.

Some general recommendations are given below for consideration on the siting of the Medium Voltage VFDs:

- Locate the Medium Voltage VFDs in a separate room that is held under a slight positive pressure relative to surrounding spaces (5 to 10% outside air is typically sufficient)
- Provide separate air-handling system to condition the room and use MERV 11 filters or greater

The air change rate in the room should be similarly matched to the airflow volume being discharged from the VFD to limit the recirculation of hot discharge air from the VFD back into the intake.

Since cooling is all sensible, it may be possible to utilize chilled water return water to provide the level of cooling required in the room and limit unnecessary latent cooling. Best practices for airside system design:

- Locate return air duct above the hot discharge path of the VFD (temperature discharge from VFDs can be in excess of 110°F)

- Provide averaging thermistor across the intake of the VFD to control the air system and maintain required intake temperature to the VFD (often it is difficult to locate a single space sensor due to the high air change rate within the room)

- Provide appropriate redundancy to allow for continued operation of the chiller system in the event of a component failure; this is particularly important if all chiller VFDs are in a single room

- If the supply or return ductwork passes through a 1½ hour or higher fire rated wall, make sure multiple fire dampers are used to ensure inadvertent closure of a single fire damper does not interrupt operations

Option 3: Consider Rigid or Flexible Conduit for All Control Cables

Standard industry construction for cabling is to clip the instrument cabling to the exterior of the chiller. For longevity of life and protection of the cabling in the industrial campus plants, it may be advantageous to require the chiller manufacturer install the wiring in rigid or flexible conduit to provide an additional level of protection from inadvertent damage while working on the chillers.

Conclusion

As discussed in this newsletter, every large university central plant has unique considerations that must be addressed due to various site constraints.

Detailed evaluation is required for proper application of large centrifugal chillers that will be required to operate over a wide operating envelope.

Careful evaluation of compressor design, construction options, and specification requirements is required to ensure a robust central plant design that results in a balance between efficiency and robust plant design.

Simplified Load Calculations

A commonly used formula for calculating load based on flow rate is shown below:

$$q = 500 \times gpm \times \Delta T$$

Where

q =	*Total Heat Transfer, Btu/hr*
gpm =	*System flow rate in gallons per minute*
ΔT =	temperature difference, °F

This simplified version is accurate for water and is an adaptation of the below equation:

$$q = \dot{m} c_p \Delta T$$

where,

q =	total heat transfer, Btu/hr
m =	mass flow rate, lb/hr
cp =	specific heat, Btu/lb-°F
ΔT =	temperature difference, °F

For water the density is 62.4 lb/ft^3 and water has a value of 7.48 gal per cubic feet and the specific heat of water is 1 Btu/lb-F. The corresponding constant 500 is calculated from the following:

$$\frac{62.4\frac{lb}{ft^3}\times 60\frac{min}{hr}\times 1\frac{Btu}{lb-F}}{7.48\frac{gal}{ft^3}} = 500\frac{\min Btu}{hr\ F\ gal}$$

When the transport liquid is at non-standard temperatures or contains a brine solution it is important to fall back on the original mass flow rate, specific heat calculation for accuracy.

Condenser Heat Rejection

A commonly used reference for condenser water is:

1 ton of refrigeration (12,000 Btu/hr) equates to 15,000 Btu/hr of condenser heat rejection.

This would equate to a 10°F delta T on the condenser. This is an old rule of thumb and should be avoided. The heat rejected to the tower loop can be easily calculated based on the efficiency of the chiller.

Consider the chiller performance in Table 7. The efficiency is 0.5888 kW/ton.

$$0.5888\frac{kW}{ton}\times 3412\frac{Btu}{hr-kW} = 2{,}008\frac{Btu}{hr-ton}$$

Adding in the 12,000 Btu/hr of cooling results in a total of 14,008 Btu/hr of rejection to the tower loop. So in the case of Table 7 at 3000 tons, 3 gpm/ton, it is possible to calculate to tower heat rejection and delta T.

$$3000\times 14008\frac{Btu}{hr-ton} = 500\times 9000(x-85)$$

Solving for x:

$$x = \frac{14008 \times 3000}{500\times 9000} + 85 = 94.33F$$

When is 15000 Btu/hr-ton accurate?

$$\frac{3000}{3412} = 0.8792\ kW/ton$$

Concluding: Unless your chiller efficiency is 0.8792 kW/ton, using the 15,000 Btu rule of thumb is not accurate.

turn to the experts