Scoping a Chilled Water Plant

A Scavenger Hunt to Develop Your Scoping Skills

Presented By:

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- Senior Engineer

A Few Very Useful Relationships

$$Q_{Btu/Hr} = 500 \times Flow_{gpm} \times \left(t_{Entering,°F} - t_{Leaving,°F}\right)$$

Where:

$Q_{Btu/Hr} =$ Load in Btu/hr

$500 =$ Units conversion constant, good for water between 30 and 200°F

$Flow_{gpm} =$ Flow through the heat exchanger in gallons per minute

$t_{Entering,°F} =$ Temperature entering the heat exchanger in °F

$t_{Leaving,°F} =$ Temperature leaaving the heat exchanger in °F

$$Q_{\text{Btu per hour}} = 1.08 \times Flow_{\text{Cubic Feet per Minute}} \times \left(Temperature_{\text{In, °F}} - Temperature_{\text{Out, °F}}\right)$$

Where:

$Q_{\text{Btu per hour}} =$ Sensible energy change in the air stream

$1.08 =$ Unit conversion constant for dry air at 70°F

$Flow_{\text{Cubic Feet per Minute}} =$ The flow rate for the current operating mode based on TAB data

$\left(Temperature_{\text{In, °F}} - Temperature_{\text{Out, °F}}\right) =$ Heat exchanger temperature difference

A Few Very Useful Relationships

$$Q_{Latent} = 0.68 \times Flow \times \left(w_{Entering} - w_{Leaving}\right)$$

Where:

Q_{Latent} = Latent load in Btu/hr

0.68 = Units conversion constant

$Flow$ = Air flow in cubic feet per minute

$\left(w_{Entering} - w_{Leaving}\right)$ = Humidity ratio change across the process in grains of water per pound of dry air. Note that there are 7,000 grains per pound

$$Q_{\text{Btu per hour}} = 4.5 \times Flow_{\text{Cubic Feet per Minute}} \times \left(Enthalpy_{\text{In, Btu per pound}} - Enthalpy_{\text{Out, Btu per pound}}\right)$$

Where:

$Q_{\text{Btu per hour}}$ = Total energy change in the air stream

4.5 = Unit conversion constant for dry air at 70°F

$Flow_{\text{Cubic Feet per Minute}}$ = The flow rate for the current operating mode based on TAB data

$\left(Enthalpy_{\text{In, Btu per pound}} - Enthalpy_{\text{Out, Btu per pound}}\right)$ = Heat exchanger enthalpy difference

A Few Very Useful Relationships

$$hp_{Air} = \left(\frac{Flow \times Static}{6,356} \right)$$

Where:

$hp_{Air} =$ The energy represented by the mass of air in motion at an elevated pressure

$Flow =$ Flow produced by the fan in cfm

$Static =$ Static produced by the fan in inches water column

6,356 = A units conversion constant that will work for air
at the temperatures and pressures typically encountered
in HVAC systems for up to about 2,000 - 3,000 feet in altitude.

A Few Very Useful Relationships

$$hp_{Water} = \left(\frac{Flow \times Head}{3{,}960 \times Efficiency_{Pump}} \right)$$

Where:

$hp_{Water} =$ The energy represented by the mass of water in motion at an elevated pressure

$Flow =$ Flow produced by the pump in gpm

$Head =$ Head produced by the pump in feet water column

3,960 = A units conversion constant that will work for water at the temperatures and pressures typically encountered in HVAC systems.

$Efficiency_{Pump} =$ Pump efficiency, read from the pump curve or estimated from past experience; .40 - .70 for small (under 500 gpm) pumps, .70 - .85 for large pumps

A Few Very Useful Relationships

$$kW = \left(\frac{Flow_{gpm} \times Head_{ft.w.c.}}{3{,}960 \times \eta_{Pump} \times \eta_{Motor} \times \eta_{VSD}} \right) \times 0.746$$

Where:

kW = Input to the system to produce the flow and head.

0.746 = Horsepower to kW conversion constant.

η_{Pump} = Pump efficiency. We usually try to get this number from the pump curve or from the pump's rated brake horse power (bhp), flow and head. Lacking that, we will make a geometrically similar pump selection (same flow rate, head, impeller diameter, and speed) using manufacturer's software and use that efficiency. Lacking that is is reasonable to assume that for a pump rated for 300 gpm or less the efficiency might be in the range of 45-60%. For pumps rated between 300 gpm and 1,500 gpm, efficiencys might range from 60% to 75%. For pumps over 1,500 gpm, efficiencies might range from 75% to as high as 87%. Generally, efficiency will improve with pump size.

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A Few Very Useful Relationships

$$kW = \left(\frac{Flow_{gpm} \times Head_{ft.w.c.}}{3{,}960 \times \eta_{Pump} \times \eta_{Motor} \times \eta_{VSD}} \right) \times 0.746$$

Where:

kW = Input to the system to produce the flow and head.

0.746 = Horsepower to kW conversion constant.

η_{Motor} = Motor efficiency. We usually try to get the motor performance curve and select the efficiency from the curve for the bhp that the pump impeller is extracting from it. If we can't get the motor curve, we use a similar motor selected from MotorMaster^{TM} International. In all cases we adjust the efficiency for the motor operating point vs. using the motor's rated nameplate efficiency. Lacking anything else, it is reasonable to assume that the motor efficiency will improve by 1-2% over the nameplate efficiency when the pump is at 65-85% of its rated load, drop back to near nameplate efficiency at around 50% load, and then drop sharply towards 0 at 20-30% of rated load.

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A Few Very Useful Relationships

$$kW = \left(\frac{Flow_{gpm} \times Head_{ft.w.c.}}{3{,}960 \times \eta_{Pump} \times \eta_{Motor} \times \eta_{VSD}} \right) \times 0.746$$

Where:

kW	=	Input to the system to produce the flow and head.
0.746	=	Horsepower to kW conversion constant.
η_{VSD}	=	Variable speed drive efficiency. Where possible, we try to get the manufacturer's data for this. But this data is difficult to obtain and not consistent in its development. Lacking manufacture specific data, we use generic data as published by the Department of Energy on their Industrial Best Practices web site. Lacking any other source, it is reasonable to assume there will be at least 4-6% loss in the drive with it at full speed with a gradual decay to 80% efficiency at about 20% load.

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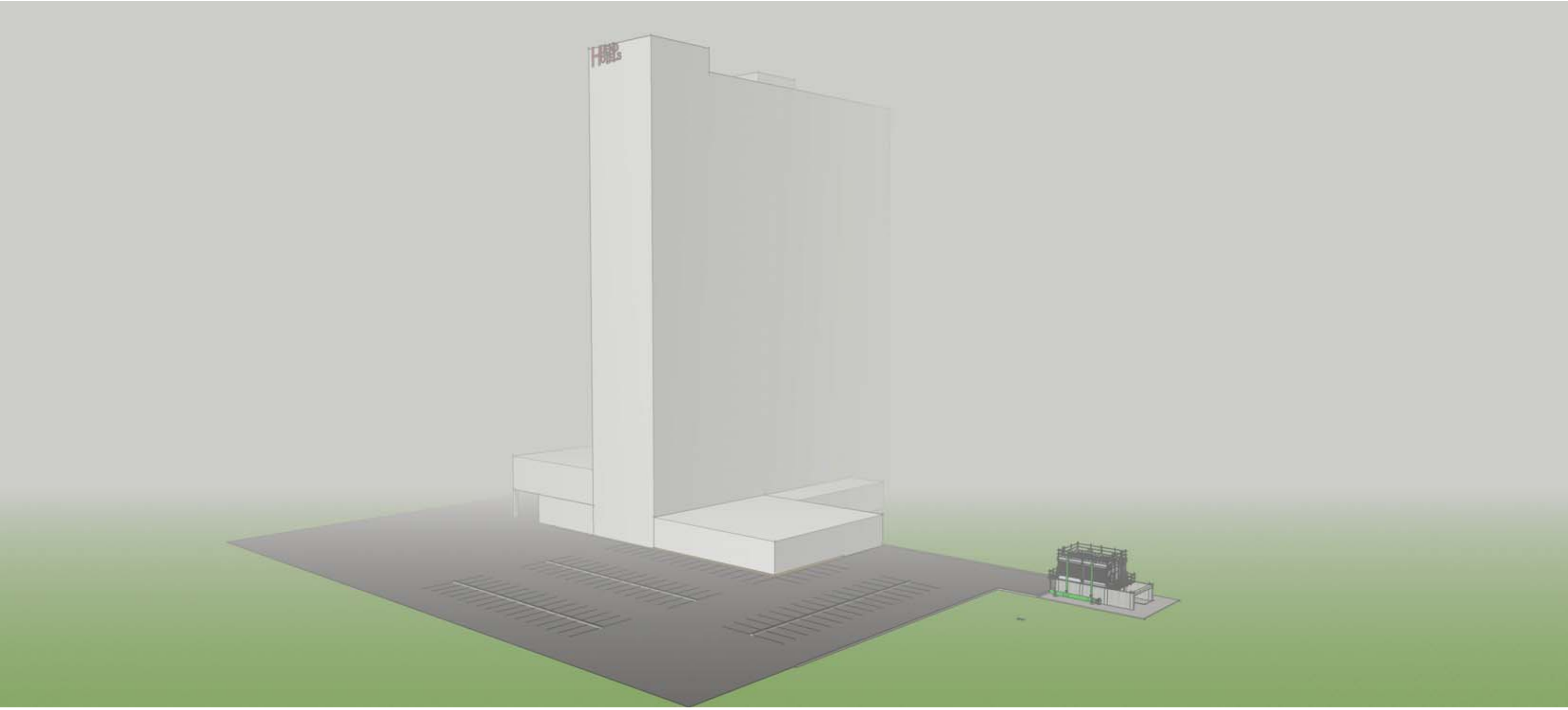
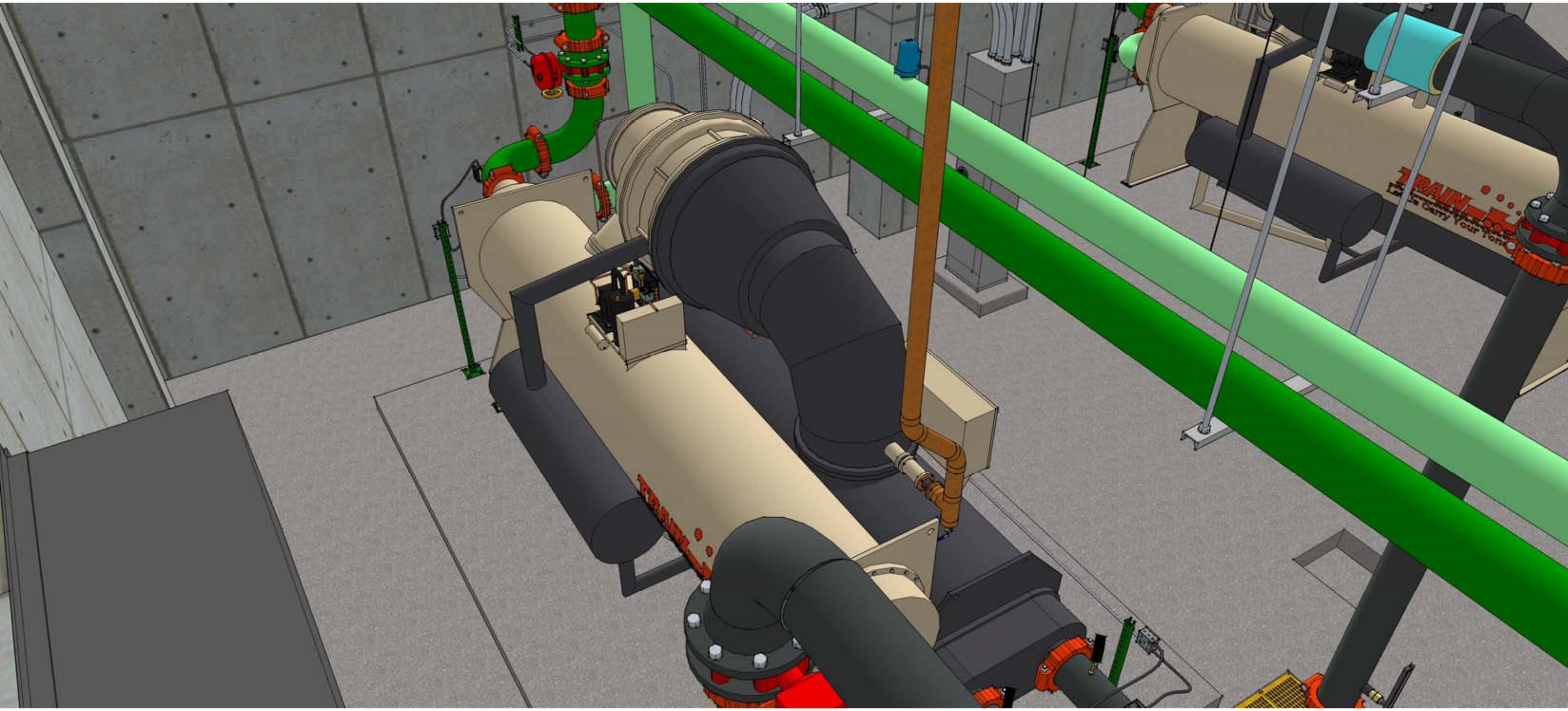
Heading Out to the Field

Setting the Scene



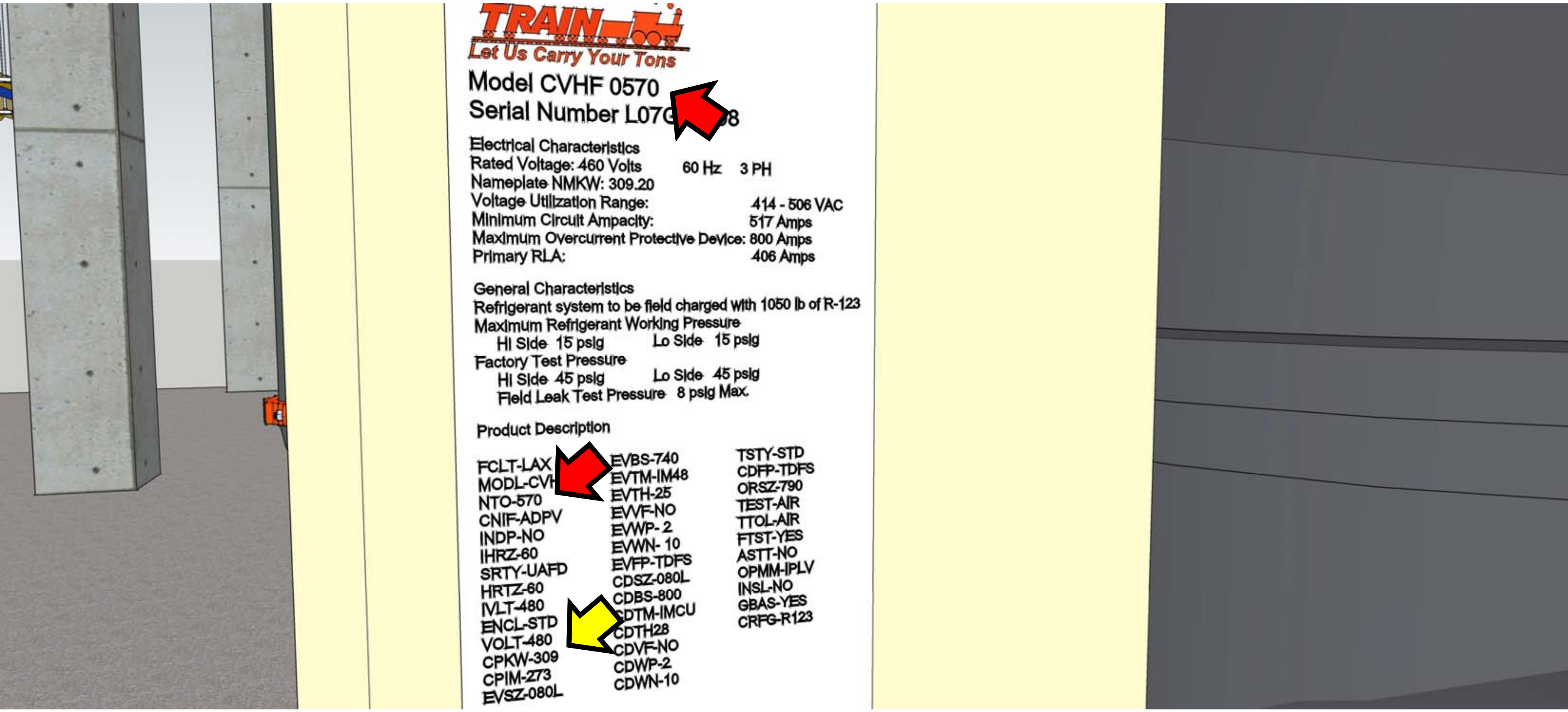
Getting Around in SketchUp



Situational Awareness

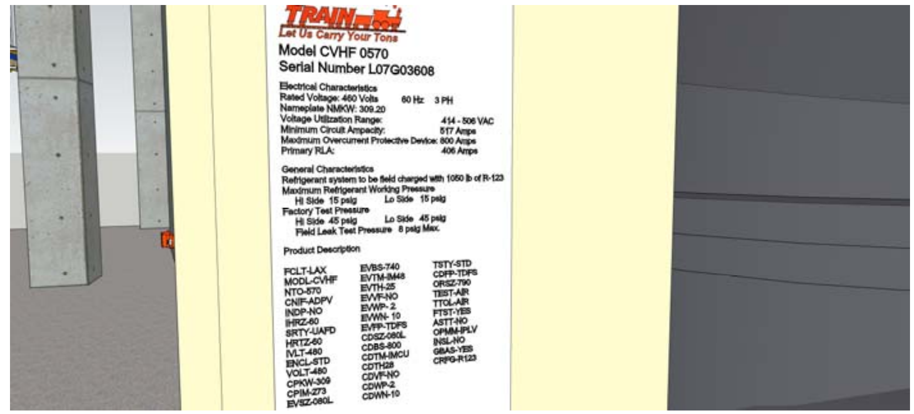
What is the Approximate Load on the Plant?

What is the Approximate Load on the Plant? “Plan A”



What is the Approximate Load on the Plant? “Plan A”

$$kW_{FullLoad} = 309$$
$$Tons_{FullLoad} = 570$$
$$(kW / Ton)_{FullLoad} = 0.54$$



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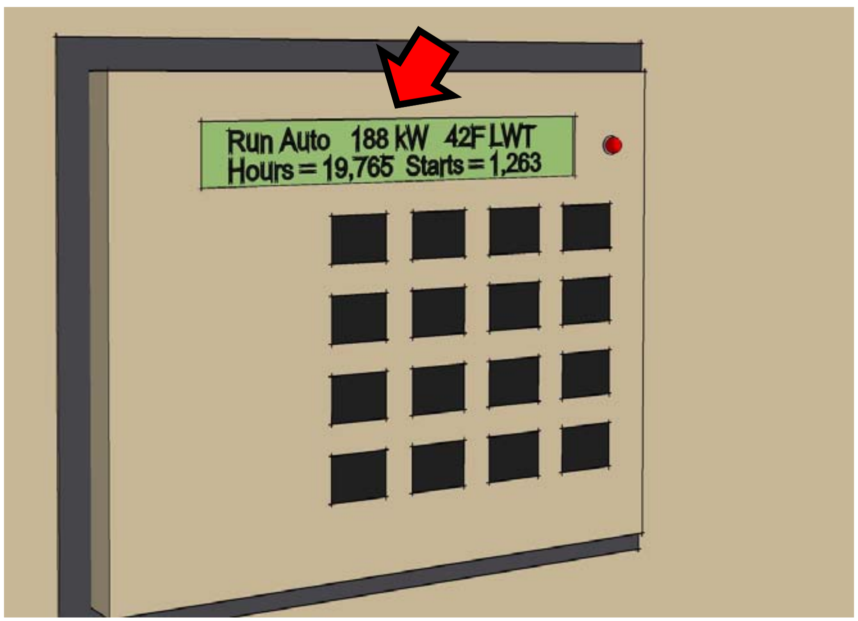

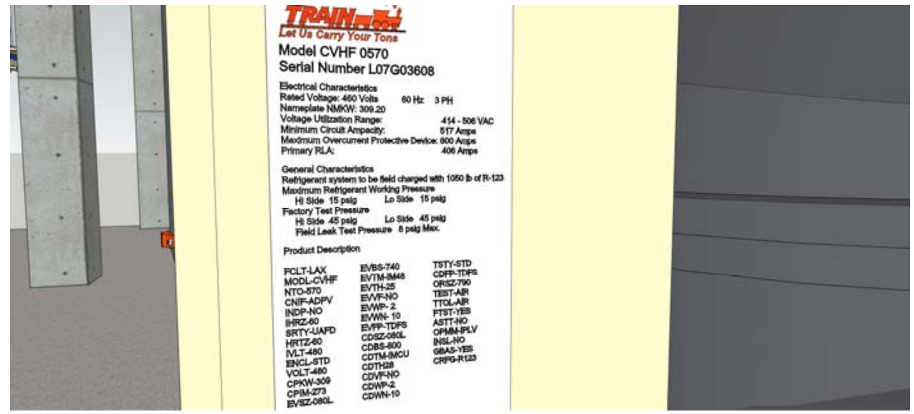

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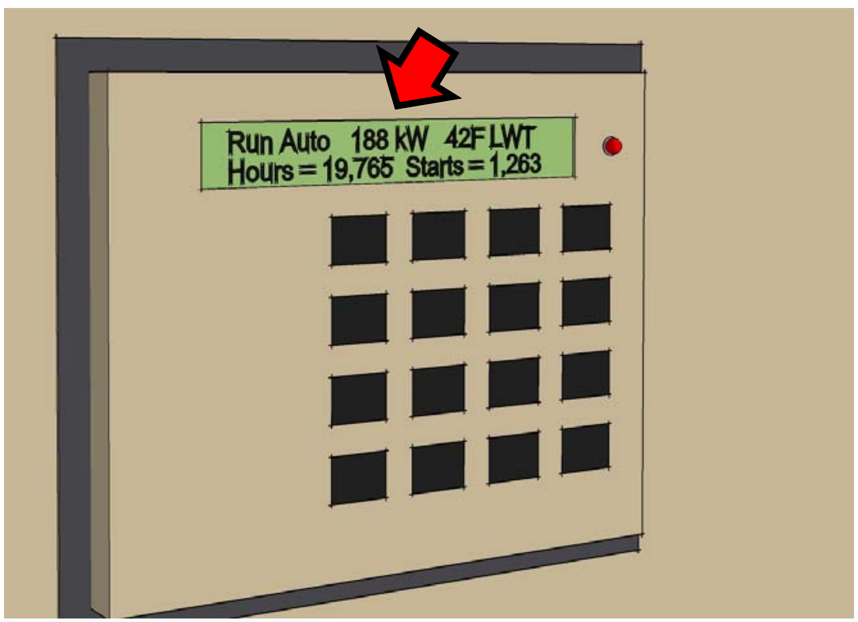

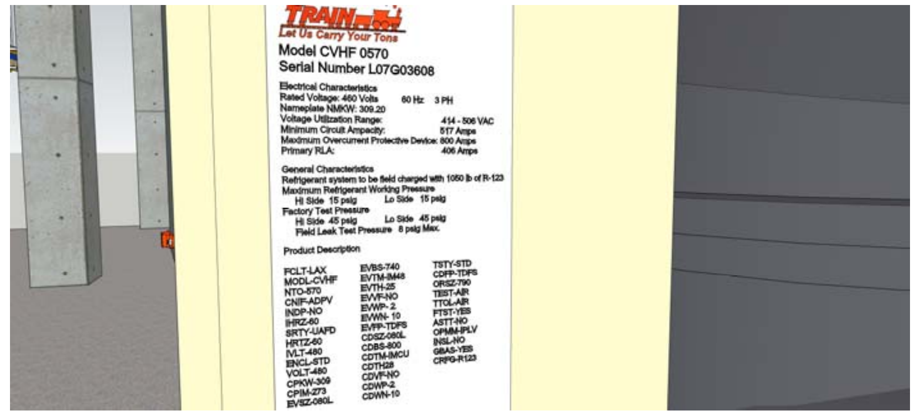

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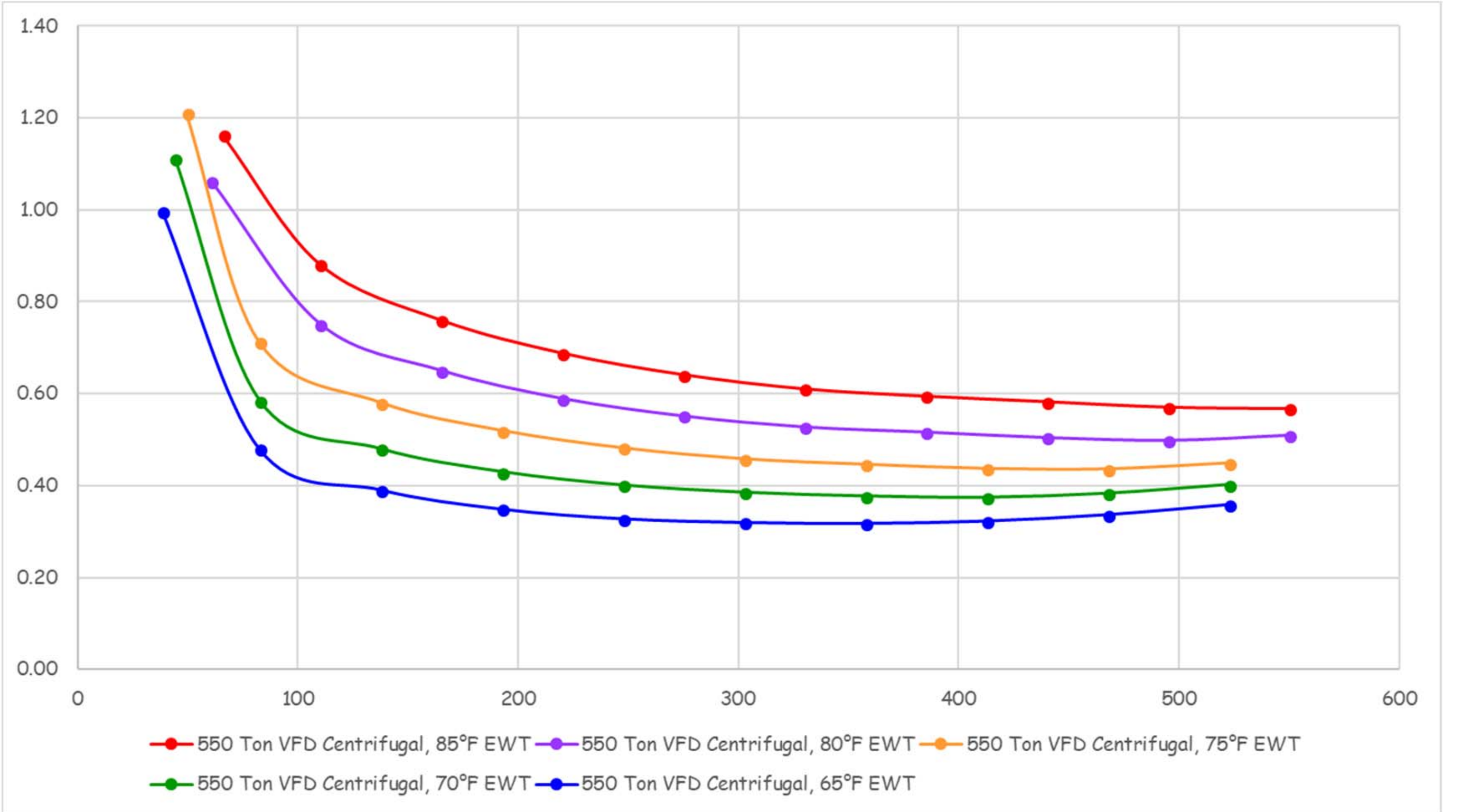
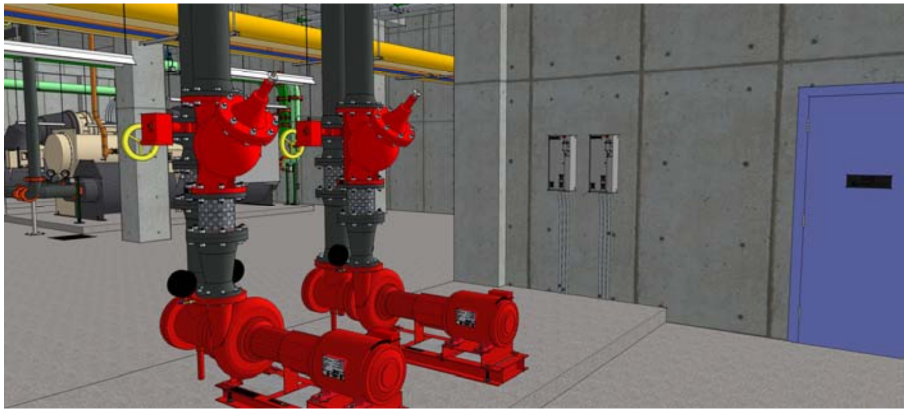
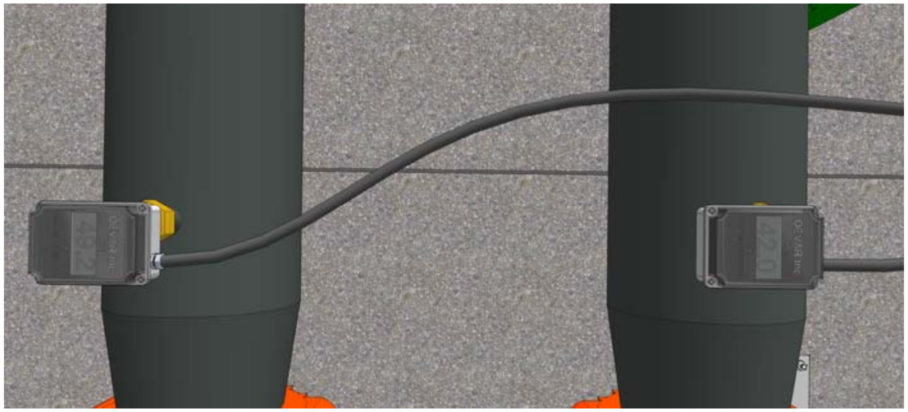


What is the Approximate Load on the Plant? “Plan B”

$$Q_{Btu/Hr} = 500 \times Flow_{gpm} \times \left(t_{Entering,°F} - t_{Leaving,°F}\right)$$



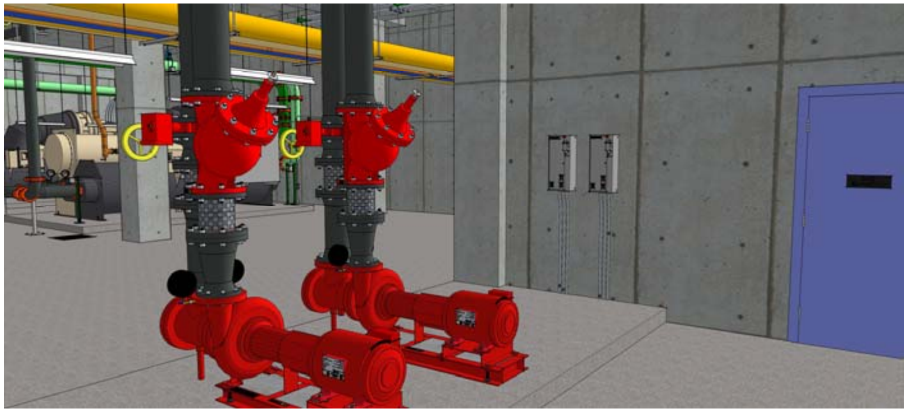


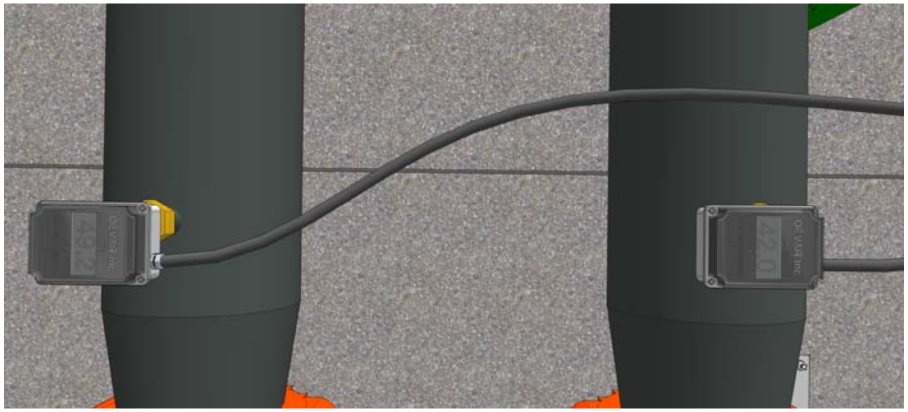
What is the Approximate Load on the Plant? “Plan B”

$$Q_{Btu/Hr} = 500 \times Flow_{gpm} \times \left(t_{Entering,°F} - t_{Leaving,°F}\right)$$

$$Q_{Btu/Hr} = 500 \times 1{,}100 \times \left(49.2 - 42.0\right)$$

$$Q_{Btu/Hr} = 3{,}960{,}000 \text{ Btu/hr}$$





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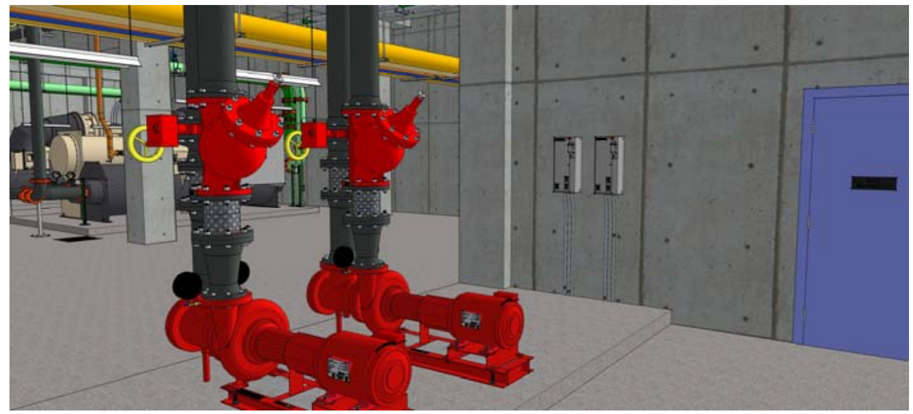
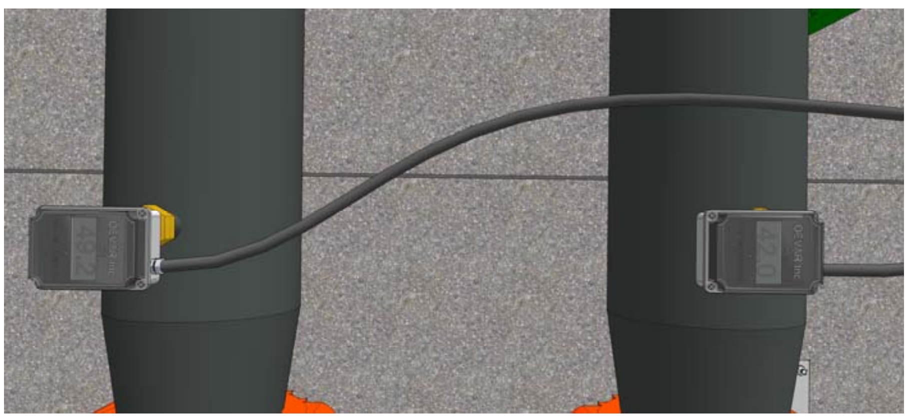
$$Q_{Btu/Hr} = 500 \times 1{,}100 \times \left(49.2 - 42.0\right)$$

$$Q_{Btu/Hr} = 3{,}960{,}000 \text{ Btu/hr}$$

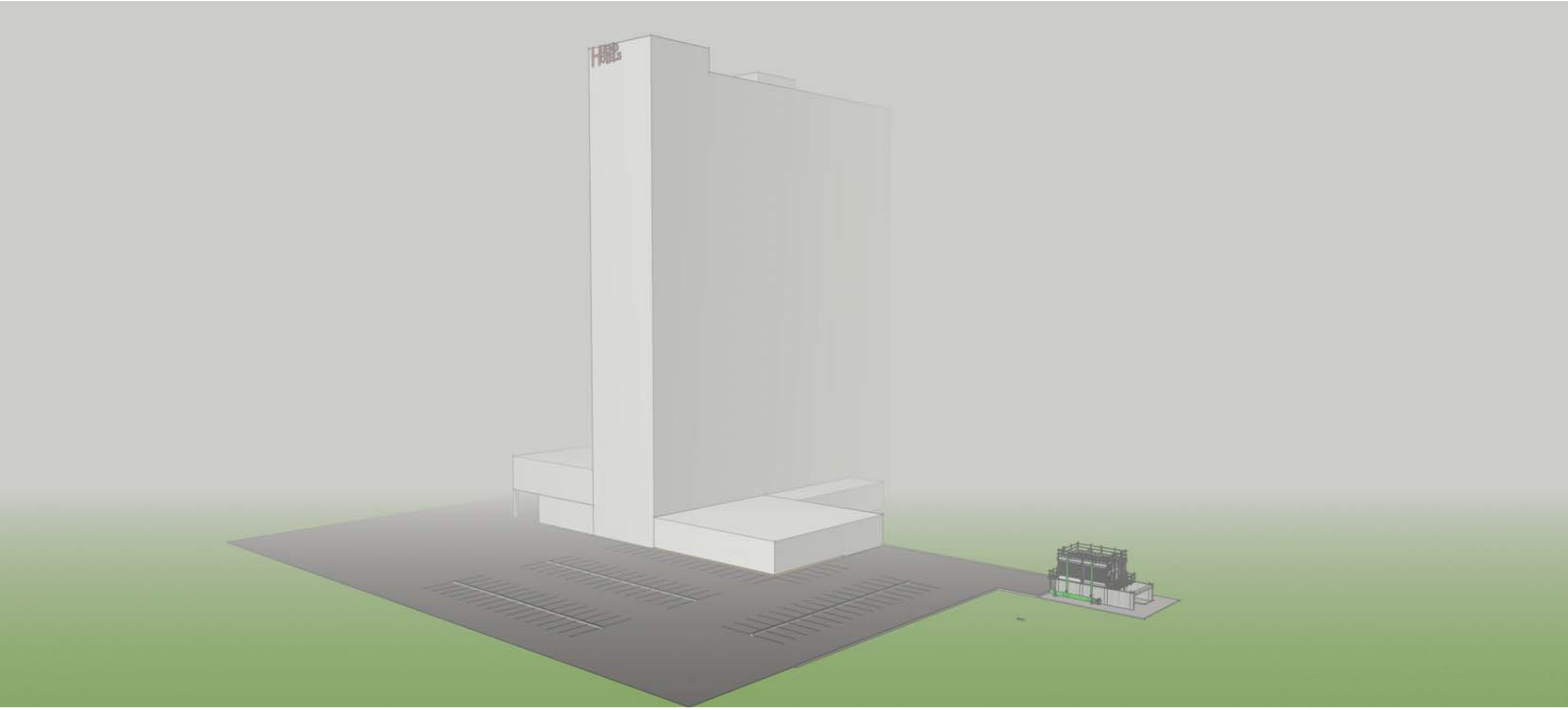
There are 12,000 Btu/hr/ton

$$Q_{Btu/Hr} = 330 \text{ Tons}$$





What Would You Expect the Load to Be on the Plant?



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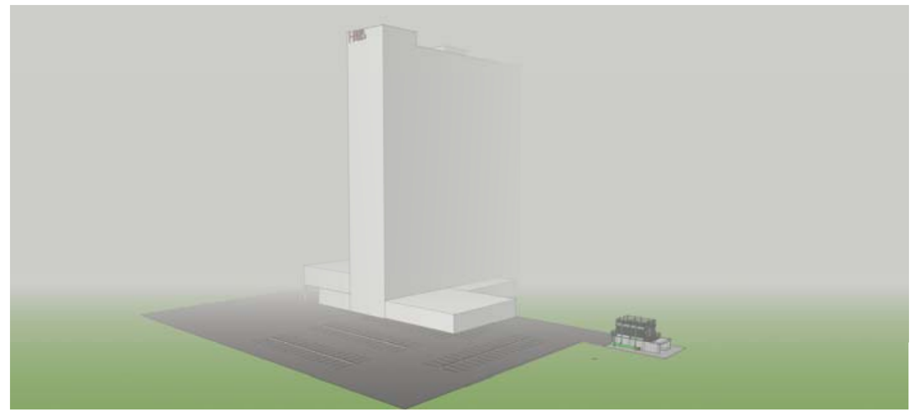

What Would You Expect the Load to Be on the Plant?

COOLING COIL SCHEDULE

Cooling Coil Number	Unit or System Served	Flow, cfm	Maximum Fins per Inch	Rows	Minimum Face Area, sq.ft.	Airside Performance: Entering Air Dry bulb, °F	Airside Performance: Entering Air Wet bulb, °F	Airside Performance: Leaving Air Dry bulb, °F	Airside Performance: Leaving Air Wet bulb, °F	Face Velocity, fpm	Pressure Drop, in.w.c.	Waterside Performance: Entering Water Temperature, °F	Leaving Water Temperature, °F	Flow Rate, gpm	Pressure Drop, ft.w.c.	Tons	Comments
CC-1	AHU1 - Hotel Lobby and Administration	26,000	8	6	52.0	81.0	63.8	51.0	50.5	433	0.63	42.0	56.0	141.0	8.6	82.2	
CC-2	Main Ball Room	20,000	9	6	40.0	86.6	66.1	51.4	50.9	500	0.82	42.0	54.0	148.7	11.0	74.4	
CC-3	Junior Ball Room	15,000	8	6	30.0	80.2	63.5	51.7	51.1	500	0.74	42.0	54.0	88.7	9.2	44.3	
CC-4	Meeting Rooms	15,000	9	6	30.0	90.3	67.6	52.2	51.6	500	0.83	42.0	54.0	120.1	9.1	60.1	
CC-5	Corridor Make-up Air	28,300	8	6	56.6	90.3	67.6	52.8	52.0	500	0.76	42.0	54.0	221.8	6.9	110.9	
CC-6	Corridor Make-up Air	28,300	8	6	56.6	90.3	67.6	52.8	52.0	472	0.70	42.0	54.0	221.8	6.9	110.9	
CC-7	Back of House	10,000	8	6	20.0	81.3	65.0	53.9	53.3	500	0.75	42.0	54.0	57.9	3.1	29.0	
CC-8	Breakfast/Lunch Café	6,500	8	6	13.0	82.7	64.5	50.9	50.4	406	0.56	42.0	54.0	43.9	7.8	21.9	
CC-9	Restaurant and Lounge	11,500	8	6	23.0	82.7	64.5	51.8	51.2	479	0.70	42.0	54.0	73.7	9.3	36.9	
CC-10	Main Kitchen	19,000	8	6	38.0	88.5	67.6	51.5	51.0	396	0.56	42.0	54.0	157.2	9.2	78.6	
VF-1	Electrical Room	8,200	8	3	16.4	83.7	67.6	60.7	57.9	410	0.35	42.0	54.0	42.5	9.8	21.2	
CC-GR01	Typcial North Exposure Guest Room (272 thus)	300	14	3	1.4	72.0	60.0	49.4	49.0	214	0.15	42.0	48.6	2.7	3.5	0.7	
CC-GR02	Typcial East Exposure Guest Room (44 thus)	400	14	3	1.4	72.0	60.0	51.2	50.5	285	0.22	42.0	49.4	2.8	3.7	0.9	
CC-GR03	Typcial South Exposure Guest Room (224 thus)	600	14	3	2.2	72.0	60.0	50.4	49.8	275	0.21	42.0	50.1	4.1	9.7	1.4	
CC-GR04	Typcial West Exposure Guest Room (22 thus)	400	14	3	1.4	72.0	60.0	51.2	50.5	285	0.22	42.0	49.4	2.8	3.7	0.9	
CC-GR05	Typcial Luxury Guest Room (4 thus)	1,000	14	3	3.2	72.0	60.0	51.2	50.4	313	0.26	42.0	50.3	6.3	8.9	2.2	

COOLING TOWER SCHEDULE

Cooling Tower Number	Unit or System Served	Type	Model	Nominal Tons	Flow, gpm	Entering Temperature, °F	Leaving Temperature, °F	Approach Temperature, °F	Rating Wet Bulb Temperature, °F	Airflow, cfm	Fan Speed, rpm	Sound Power, dBA	Lift, ft.w.c.	Efficiency gpm/hp	Heat Rejection Btu/hr	Motor Hp	Motor Speed	Motor Volts	Motor Phase
CT-1	Condenser Water	Induced Draft, Cross Flow	NC8409PAS1-420	550	1,873	95.0	85.0	12.1	72.9	139,200	214	71	12.3	132	9,332,200	15.0	1,200	480	3
CT-2	Condenser Water	Induced Draft, Cross Flow	NC8409PAS1-420	550	1,873	95.0	85.0	12.1	72.9	139,200	214	71	12.3	132	9,332,200	15.0	1,200	480	3

Notes

What Would You Expect the Load to Be on the Plant?

COOLING COIL SCH

Unit or System Served	Flow, cfm	Maximum Fins per Inch	Rows	Minimum Face Area, sq.ft.	Airside Performance			
					Entering Air		Leaving Air	
					Dry bulb, °F	Wet bulb, °F	Dry bulb, °F	Wet bulb
AHU1 - Hotel Lobby and Administration	26,000	8	6	52.0	81.0	63.8	51.0	50.5
Main Ball Room	20,000	9	6	40.0	86.6	66.1	51.4	50.9
Junior Ball Room	15,000	8	6	30.0	80.2	63.5	51.7	51.1
Meeting Rooms	15,000	9	6	30.0	90.3	67.6	52.2	51.6
Corridor Make-up Air	28,300	8	6	56.6	90.3	67.6	52.8	52.0
Corridor Make-up Air	28,300	8	6	56.6	90.3	67.6	52.8	52.0
Back of House	10,000	8	6	20.0	81.3	65.0	53.9	53.3
Breakfast/Lunch Café	6,500	8	6	13.0	82.7	64.5	50.9	50.4
Restaurant and Lounge	11,500	8	6	23.0	82.7	64.5	51.8	51.2
Main Kitchen	19,000	8	6	38.0	88.5	67.6	51.5	51.0
Electrical Room	8,200	8	3	16.4	83.7	67.6	60.7	57.9
Typcial North Exposure Guest Room (272 thus)	300	14	3	1.4	72.0	60.0	49.4	49.0
Typcial East Exposure Guest Room (44 thus)	400	14	3	1.4	72.0	60.0	51.2	50.5
Typcial South Exposure Guest Room (224 thus)	600	14	3	2.2	72.0	60.0	50.4	49.8
Typcial West Exposure Guest Room (22 thus)	400	14	3	1.4	72.0	60.0	51.2	50.5
Typcial Luxury Guest Room (4 thus)	1,000	14	3	3.2	72.0	60.0	51.2	50.4

Are There Potential Opportunities at the Loads Served by the Plant?

☐ Yes

☐ No

Does the Piping Configuration Match the Design Intent System Diagram?

- [ ] Yes
- [ ] No

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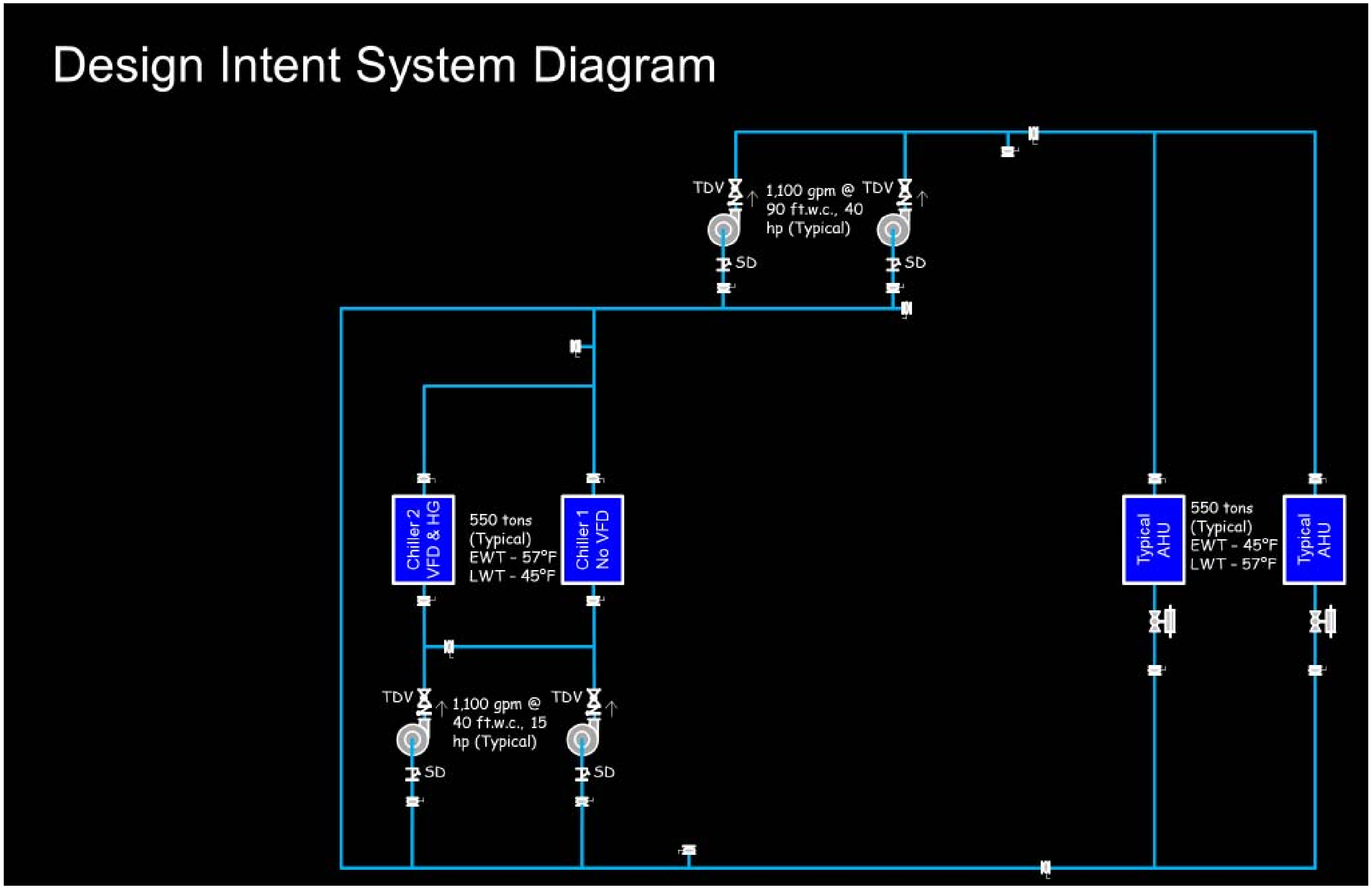

Could the Difference in Piping Configuration Be Important?

☐ Yes

☐ No

What is the Position of the CHWP-2 Hand-Off-Auto (HOA) Selector Switch?

Are There Scheduling Opportunities on This Site?

☐ Yes

☐ No

What is the Operating Status of the VFDs Serving the Distribution Pumps?

Are there Set Point Optimization Opportunities on This Site?

☐ Yes

☐ No

What is the Position of the Throttling Valve on the Chiller Evaporator Pumps?

What is the Position of the Throttling Valve on the Distribution Pumps?

Are There Potential Chilled Water Pump Optimization Opportunities in This Plant?

- ☐ Yes
- ☐ No

Is It Possible to Estimate the Potential Savings There in the Field?

- ☐ Yes
- ☐ No

Is the Chilled Water System a Closed or Open System?

What is the Status of the Make-up Water Valve for the Chilled Water System?

What is the Status of the Make-up Water Valve for the Chilled Water System?

- [ ] Open
- [ ] Closed

Is There a Non-energy Benefit (NEB) Associated with the Make-up State?

- [ ] Yes
- [ ] No

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- [ ] Yes
- [ ] No

Are the Chillers Identical?

- [ ] Yes
- [ ] No

Would You Expect the Chiller Run Hours to be Similar or Different?

- ☐ Similar
- ☐ Different

Is There a Difference In the Number of Starts Each Chiller Has?

- ☐ Yes
- ☐ No

Do You Think There Are Opportunities Related to Optimizing Chiller Staging?

- ☐ Yes
- ☐ No

Does the Piping Configuration Play Into This?

- ☐ Yes
- ☐ No

Have All of the Pumps Been Aligned After Installation?

☐ Yes

☐ No

Are There Opportunities Associated with Aligning the Pumps?

☐ Yes

☐ No

What Would You Expect the CW System Δt To Be Under Current Conditions?

Does the Actual Condenser Water System Δt Match Your Expectations?

- ☐ Yes
- ☐ No

Are There Potential CW Pump Optimization Opportunities in This Plant?

- ☐ Yes
- ☐ No

The Bottom Line So Far

Central Chilled Water Plant Preliminary Findings List

	Finding or Opportunity	Obvious Indicator	Associated Scenes	Savings: Energy Savings	Savings: Other Resource Savings	Non-resource Benefit: Cleaner	Non-resource Benefit: Safer	Non-resource Benefit: Comfort	Non-resource Benefit: Productivity	Non-resource Benefit: Performance	Potential low cost/no cost
1.	The discharge valves are throttled on the evaporator pumps.	The triple duty valves are not fully open (around 40%).	5, 35, 40-42	X							X
2.	The discharge valves are throttled on the distribution pumps.	The triple duty valves are not fully open (around 40%).	3, 36	X							X
3.	The shot feeder valves are open.	The shot feeder is piped across one of the distribution pumps, thus has a large differential pressure across it and the service valves are open.	19, 20	X							X
4.	Evaporator pump CHWP2 is in Hand	The position of the Hand-Off-Auto (HOA) selector switch.	33	X	X						X
5.	Both distribution pumps are running under a condition when one pump could handle the load.	The estimated load condition on the chiller along with pump physics knowledge	9 - 15, 34, 37, 43	X	X						X
6.	It may be possible to leverage the 2/3 rule and reduce distribution pump head at part load instead of maintaining a fixed pressure at the headers.	The current operating pressure relative to nameplate and an estimate of the part load pressure drop to a remote mechanical room.	19, 34, 38	X	X						X
7.	CHWP-3 VFD is in Hand and set for 100%, dead heading the other pump.	Indications on the VFD control panel, knowledge of pump physics, puddle of water under the pump seal.	19, 34, 38	X	X						X
8.	CHWP-3 and 4 appear to have not been aligned in the field and have not had their bases grouted.	Factory paint still on the hold down bolts and shims, no grout in the frame (compare these pumps to CHWP-1 and 2).	37, 38, 31, 32		X						X
9.	The make up line to the CHW system is open. A closed system should not require continuous make-up and keeping the valve open risks water damage if a pipe failed.		23-Jan		X		X				X
10.	The piping configuration does not match the design intent system diagram.	Field verifying the system diagram after noticing a potential conflict between the schematic and the piping plan on the contract documents.	18, 25, 26	X							
11.	The piping configuration as installed does not allow the full benefit of the chiller selection and staging strategy envisioned by the designer. This is related to item 9 above.	CH-2 has hot gas bypass and a VFD, which implies it has better turn-down capability and a part load kW/ton sweet spot, making it the machine intended for low load by itself and part load when CH-1 is on at full load.	9-15, 18, 29, 30	X	X						
12.	The chillers appear to be staged for equal run time instead of for the apparent design intent (see item 10 above).	The number of run hours for both machines are roughly equivalent but CH-1 has many, many more starts because when it is the lead machine at low load, it short cycles since it has no hot gas bypass or VFD.	12, 15	X	X						X
13.	The condenser water pump serving chiller 1 may be running out its curve with out the pump associated with chiller 2 in operation.	Using Q=500xgpmxΔt and a heat balance based on the chilled water load, the compressor kW and the observed CW Δt suggests the condenser flow is in the 2,300 - 2,400 gpm range vs. the 1,650 design	8, 9, 12, 18	X							X
14.	The plant has in the range of 300 tons of load on it when other indicators would suggest that there should be little if any load.	Load estimated from the evaporator flow and Δt or the chiller nameplate data; requirements estimated based on the ambient conditions and the coil schedule on the plans and logic. Leads to opportunities at the loads.	9. 12. 18, 30, 31	X	X					X	X
15.	The current distribution pump selections seem to be farily low efficiency compared to what might be possible. Estimating the water horse power from the nameplate data comes out at 27, which is significantly below the 40 hp motor.	Knowledge of pump physics and some application experience.	37, 38	X						X	