UMassMemorial
Medical Center

UMASS MEMORIAL MEDICAL CENTER

POWER PLANT

55 LAKE AVENUE NORTH

WORCESTER, MA

PREPARED FOR

DCAMM

Prepared by

B2Q Associates, Inc.

Andover, MA

B2Q

Revision Date

12/21/2015

TECHNICAL ASSISTANCE STUDY

TECHNICAL ASSISTANCE STUDY - WATERSIDE ECONOMIZER UMASS MEMORIAL MEDICAL CENTER - POWER PLANT

INTRODUCTION

This technical assistance study was conducted by B2Q Associates, Inc. in collaboration with the Massachusetts Department of Capital Asset Management and Maintenance (DCAMM) and the University Of Massachusetts Medical Center (UMass). This study is part of DCAMM's Accelerated Energy Program (AEP), whose focus is to assess the feasibility of a "deep energy retrofit" which includes a goal to reduce purchased energy by 20%-30% and then identify renewable energy systems where financially and technically feasible. This study focuses on winter free cooling opportunities in the central power plant and is intended to augment the focused ASHRAE Level 2 study already conducted for six of the campus' buildings and chilled water pumping systems. The intent is to present additional opportunities which could be part of an integrated package of energy efficiency measures and renewable energy systems that will maximize net present value of the buildings over 10-20 years.

The intent of this report is to present the results associated with our investigation and analysis of two options for winter free cooling associated with the campus' central chilled water plant. The report includes an existing condition description and assessment, description of the measures evaluated, discussion of energy modeling methodology, opinions of probable cost, and recommendations for next steps.

B2Q worked with UMass Medical Power Plant Staff and York Factory Representatives from January 2015 through the writing of this report in support of the following efforts:

1. Obtained historical chilled water temperature and flow records in order to develop an annual chilled water load profile for the campus. This load profile was used as the basis for all energy modeling associated with winter free cooling opportunities.
2. Obtained historical trend data from the plant's SCADA control system to establish a baseline model of electrical power, steam, and chilled water production during normal winter operation.
3. Gather available documentation for all major equipment in the chilled water plant including mechanical submittals, design and off-design performance data, piping & instrumentation diagrams (P&IDs), process flow diagrams (PFDs), and sequences of operation.
4. Walk through the power plant to view major equipment, gather nameplate data, inspect the configuration of piping, valves, and cooling towers, and interview plant operators to gain an understanding of the systems and document control strategies. Potential locations for a future plate and frame heat exchanger were also evaluated during visits to the site.
5. Develop a baseline model of the chilled water plant to simulate hourly chiller, pumping, and cooling tower energy consumption during the winter months.
6. Develop proposed case models of the plant to simulate equipment performance and calculate energy savings associated with the implementation of two separate winter

free cooling options: a plate and frame heat exchanger and chiller “thermo-siphoning“ (or refrigerant migration).

7. Develop budgetary Opinions of Probable Construction Costs for the measures identified and calculate simple payback estimates. These estimates presented in the draft report do not include the cost reduction impact of potential utility incentives. These will be accounted for in the final report after utility review.

Contacts

UMass Medical

Joseph Collins	Director of Energy Resources	(508) 856-2220	joseph.collins@umassmed.edu
Steven Blair	Assistant Director, Power Plant	(508) 856-2153	steven.blair@umassmed.edu
Mark Armington	Director Facilities	(508) 856-5202	mark.armington@umassmed.edu
David Macneil	Sr. Mechanical Project Manager	(508) 856-4776	david.macneil@umassmed.edu
Andrew Doe	Mechanical Engineer	(508) 856-2498	andrew.doe@umassmed.edu

DCAMM

Tony Ransom	Program Manager	(617) 727-4030 x31561	edward.ransom@state.ma.us
John Crisley	Project Manager	(617) 727-4030 x31561	john.crisley@state.ma.us
Ray Soohoo	Energy Planner	(617) 727-4030 x31509	raymond.soohoo@state.ma.us

B2Q Associates

Paul Banks	Principal	(978) 447-5601	pbanks@b2qassociates.com
Michael Margareci	Vice President	(978) 447-5602	mmargareci@b2qassociates.com
Kevin Keena	Sr. Project Manager	(978) 447-5603	kkeena@b2qassociates.com
Sam Deptula	Engineer	(978) 447-5611	sdeptula@b2qassociates.com

APPROACH & MODELING METHODOLOGY

Hourly baseline and proposed case models were developed for the two winter free cooling scenarios evaluated as part of this study using custom Microsoft Excel spreadsheets. The spreadsheet models make use of multi-variate performance curves for the plant's major equipment including chillers and cooling towers. Curves were either developed using model-specific performance data supplied by the equipment manufacturer or library curves from the eQuest building analysis program were used when model-specific curves were unavailable. For example, eQuest library curves were used to calculate hourly cooling tower capacity as a function of approach, range, and airflow. In cases where library curves were used, corrections were made for each piece of equipment's unique design criteria.

Spreadsheet models were created using information collected over the course of multiple site visits, through discussions with facilities staff, original equipment submittals, nameplate and physical operating information, and data supplied from equipment manufacturers. Typical Meteorological Year 3 (TMY3) weather data from the Worcester, MA municipal airport weather station was used in the analysis to simulate hourly ambient conditions for a "typical" year. TMY3 data is derived by National Renewable Energy Laboratory (NREL) from historical meteorological data available from 1991 - 2005. The data is specifically selected so that it presents the range of weather phenomena for the location in question, while still giving annual averages that are consistent with the long-term averages.

In addition to the electric chiller savings and pumping/cooling tower fan electricity penalties associated with the implementation of a winter free cooling strategy, the simulation models also consider the potential impacts on the power plant's operation. Interval data from the power plant's SCADA system was obtained from January - February 2015 to develop existing and proposed case load profiles for the gas turbine generator and two operational steam turbines, as well as the campus electric demand. Refer to the section titled "Power Plant Modeling" on Page 17 for more details.

EXECUTIVE SUMMARY TABLE

Measure	Electric Energy Savings	Equivalent Gas Energy Savings	Equivalent Gas Cost Savings	Total Cost Savings	Retrofit Cost	Potential Utility Incentive	Net Project Cost	Simple Payback Period
	kWh	*therms*	*$*	*$*	*$*	*$*	*$*	*yrs*
ECM-23A: Retrofit 3 Chillers with Thermosiphon	985,091	72,514	$61,637	$98,509	$1,424,100	$295,527	$1,128,573	18.3
ECM-23B: Retrofit 2 Chillers with Thermosiphon	938,848	69,206	$58,825	$93,885	$996,100	$281,654	$714,446	7.6
ECM-23C: Retrofit 1 Chiller with Thermosiphon	509,048	38,013	$32,311	$50,905	$543,000	$152,714	$390,286	7.7
Plate & Frame Heat Exchanger	764,268	57,411	$48,800	$76,427	$1,115,232	$229,280	$885,952	11.6

Notes:

1. The “Equivalent Gas Energy Savings” estimates the magnitude of natural gas saved by reducing chiller plant electric demand during periods when waterside economizer is active. Natural gas savings consider the impacts of electric demand reduction on the power plant’s combustion gas turbine (G4) and heat recovery steam generator, 1,100 psig steam boilers, 250 psig steam boilers, and 5,000 kW back-pressure turbine (G3). Refer to the “Power Plant Modeling” Section on Page 17 for more details on the natural gas savings methodology used in the study.
2. “Equivalent Gas Cost Savings” assume a natural gas rate of $0.85/therm.
3. Simple Payback Periods are based on the “Equivalent Natural Gas Cost Savings”

FACILITY & PROCESS DESCRIPTION

The power plant facility was originally constructed in the 1970s to provide electrical power, steam, and chilled water to the main campus buildings. The plant underwent significant expansions in 2001 and again in 2012, the latter of which included the installation of a new gas turbine, heat recovery steam generator (HRSG), and electric chiller to efficiently meet the growing energy needs of the campus. The sections below describe the major equipment and control strategy for the central plant.

Power Production

Electrical power is produced using a single 7,500 kW *Solar Taurus* gas turbine generator (G4) and up to three steam turbine generators. The gas turbine is equipped with duct burners to produce 60,000 lbs/hr of superheated steam at 1,100 psig and 850°F. The gas turbine's minimum turndown is approximately 5,000 kW, based on information provided by the plant operator.

The single back-pressure ("topping") turbine (G3) is rated for 5,000 kW and discharges saturated steam at approximately 250 psig. Steam can be sent to G3 from the gas turbine's HRSG, as well as Boilers #3 and #4, which are each rated at 115,000 lbs/hr. G3 is equipped with 4"and 6" bypass pressure-reducing valves (PRVs) that can be used during periods of high steam load but low electric load on the campus. Steam is bypassed around G3 to reduce its electric output without impacting steam flow to G1/G2 and the campus steam distribution system. During the peak summer months, 250 psig steam is distributed to one or more of the three 2,500 ton steam turbine-driven chillers in the plant.

Saturated 250 psig steam is also sent from G3 to one of the two extraction turbines (G1 / G2), which are each rated for 2,500 kW and discharge to a surface condenser. Steam is extracted at 50 psig and distributed to the various buildings on campus for space and process heating. 50 psig steam is also distributed to Chiller 4, a turbine driven unit rated at 5,000 tons.

250 psig steam can also be generated by Boilers #1 and #2, which are each rated at 80,000 lbs/hr and are used to supplement G-4 and the 1,100 psig boilers during periods of high steam demand.

Heat is rejected from the surface condensers associated with G1 and G2 via a process condenser water loop. The return water of the process loop is separate from the chiller condenser water return, but the two loops share the same towers and can draw supply water from the same tower sumps.

The table below summarizes the power plant's boiler, gas turbine, and steam turbine nameplate capacities.

Table 1: Combustion turbine, steam turbine, and boiler nameplate data

Tag	Type	Nameplate Capacity / Units	Notes
-	-	-	-
G1	Condensing Steam Turbine	2,500 kW	Extraction at 50 psig
G2	Condensing Steam Turbine	2,500 kW	Extraction at 50 psig
G3	Back-Pressure (Topping) Turbine	5,000 kW	250 psig Discharge
G4	Combustion Turbine Generator	7,500 kW / 60,000 lbs/hr	Solar Taurus w/Supp. Firing
B1	250 psig Steam Boiler	80,000 lbs/hr	Saturated
B2	250 psig Steam Boiler	80,000 lbs/hr	Saturated
B3	1100 psig Steam Boiler	115,000 lbs/hr	Superheated to 850°F
B4	1100 psig Steam Boiler	115,000 lbs/hr	Superheated to 850°F

Chiller Plant

Chillers

The chiller plant consists of a total of 5 centrifugal chillers, including 4 steam turbine-driven machines and 1 open drive electric. All chillers are York Model OM Titan and are designed for 42°F chilled water supply temperature and 85°F condenser water supply temperature. Each of the 3 2,500 ton steam turbine driven chillers operate at 250 psig inlet pressure; the 5,000 ton machine operates at 50 psig inlet. The table below summarizes key design parameters for the chillers.

Table 2: Chiller nameplate data

Tag	Make	Model	Rated Capacity	Drive Type	Entering Steam Pressure	Design LCHWT	Design ECWT
-	-	-	*tons*	-	*psig*	*°F*	*°F*
CH-1	York	Titan OM	2,470	Steam Turbine	250	42	85
CH-2	York	Titan OM	2,470	Steam Turbine	250	42	85
CH-3	York	Titan OM	2,470	Steam Turbine	250	42	85
CH-4	York	Titan OM	5,000	Steam Turbine	50	42	85
CH-5	York	Titan OM	4,000	Electric, Open, VFD	-	42	85

During the fall, winter, and spring (approximately between November and March), the 4,000 ton electric chiller operates as the lead means of cooling for the plant. During the summer months, the 5,000 ton York steam turbine driven chiller is brought on to meet the greater campus chilled water load. A 2,500 ton chiller is used during peak periods.

The chiller plant consists of 6 cooling tower cells manufactured by Marley, including the plant's 3 original cells, a 4th cell installed as part of the 2001 expansion, and cells 5 and 6, which were installed during the 2012 expansion. The total tower design heat rejection capacity is approximately 39,300 tons. The table below summarizes each tower cell's design parameters.

Table 3: Cooling tower design data

Tag	Make	Model	Rated Capacity	Design Wetbulb	Design Approach	Design Range	Design CWST	Fan Power
-	-	-	*gpm*	*°F*	*°F*	*°F*	*°F*	*bhp*
CT-1	Marley	6611-03	9,450	75	10	18	85	150
CT-2	Marley	6611-03	9,450	75	10	18	85	150
CT-3	Marley	6611-03	9,450	75	10	18	85	150
CT-4	Marley	6610-0-1	8,065	75	10	18.5	85	150
CT-5	SPX	F466-4.0-2	9,450	75	10	15	85	192.5
CT-6	SPX	F466-4.0-2	9,450	75	10	15	85	192.5

Condenser water is pumped to the cooling towers via two separate return loops: one is used for chiller condenser heat rejection and the other for component cooling and surface condenser heat rejection. Return water from either loop can be sent to any tower cell through the use of manual isolation valves. Supply water is drawn from a common tower sump; however, a valve can be used to isolate tower cells 1 - 3 from 4 - 6 if necessary.

ANNUAL CHILLED WATER LOAD PROFILE

In order to create a baseline model of the chiller plant, an hourly campus chilled water load profile was developed using daily logs of minimum and maximum flow, supply temperature, and return temperature. The average flow and average temperature measurements were used to calculate the average daily chilled water load using the following equation:

$$\dot{E} = \dot{V} * \frac{60}{7.4805} * \rho * \ c_p * \ \Delta T$$

where:

$\dot{E}$ = Rate of Cooling [Btu/hour]
$\dot{V}$ = Volumetric Flow Rate [gallons/minute]
7.4805 = Volume Conversion [gallons/cubic foot]
ρ = Density of water [62.42 lbm/cubic foot]
c_p = Average specific heat of water [1.003 Btu/lbm-°F]
ΔT = Average difference between return and supply chilled water temperature [°F]

In order to convert the daily load profile to an hourly profile, the eQuest building models developed and calibrated as part of a previous study were used. Using eQuest, the hourly chilled water loads from each of the six building models available were exported to a spreadsheet file and the coincident loads combined. Four separate average daily load profiles were generated using filter criteria to separate weekdays and weekends, as well as periods when the outdoor air wet bulb temperature was below and above 35°F. This threshold temperature was selected so that specific weekday and weekend load profiles could be developed for the periods when winter free cooling would be active.

Table 4 on Page 12 summarizes the four average load profiles developed using data obtained from the building eQuest models. Note that the average profiles have been normalized so that each hourly value represents a percentage of the average chilled water load during the day. For example, a value of 76.4% at 12:00 AM would indicate that the load during the twelve o'clock hour is 76.4% of the entire day's average load.

This method was used so that the values listed in Table 4 could be directly multiplied by the average daily chilled water load calculated previously to obtain an estimated chilled water load for each hour of the day throughout the year.

Table 4: Summary of four chilled water load profiles developed using eQuest building simulation results

Hour	< 35°F OA Wetbulb		>= 35°F OA Wetbulb	
HH:MM	*Weekday*	*Weekend*	*Weekday*	*Weekend*
12:00 AM	76.4%	92.8%	76.7%	86.0%
1:00 AM	76.0%	93.1%	76.1%	86.8%
2:00 AM	76.0%	94.2%	75.8%	86.6%
3:00 AM	76.6%	95.6%	76.8%	88.6%
4:00 AM	86.9%	97.1%	89.3%	90.0%
5:00 AM	107.4%	97.7%	96.5%	94.3%
6:00 AM	112.1%	99.8%	106.0%	101.0%
7:00 AM	116.0%	98.9%	112.0%	104.0%
8:00 AM	117.0%	100.8%	115.8%	107.0%
9:00 AM	118.0%	102.6%	118.0%	109.1%
10:00 AM	119.3%	105.0%	117.1%	112.0%
11:00 AM	120.1%	110.1%	116.6%	113.7%
12:00 PM	118.2%	106.7%	119.2%	113.9%
1:00 PM	119.3%	105.7%	121.4%	114.6%
2:00 PM	119.2%	107.6%	122.6%	116.4%
3:00 PM	118.8%	106.6%	123.9%	114.7%
4:00 PM	116.9%	106.2%	117.8%	112.4%
5:00 PM	113.8%	103.3%	114.0%	104.4%
6:00 PM	105.8%	99.6%	96.6%	98.1%
7:00 PM	79.4%	97.8%	87.1%	93.6%
8:00 PM	77.1%	95.1%	83.1%	92.5%
9:00 PM	76.6%	95.0%	80.1%	88.3%
10:00 PM	76.6%	94.4%	79.2%	87.6%
11:00 PM	76.5%	94.1%	78.2%	84.6%

The chart below illustrates the annual chilled water load profile developed using daily data obtained from the plant's SCADA system and hourly data derived from the eQuest building models.

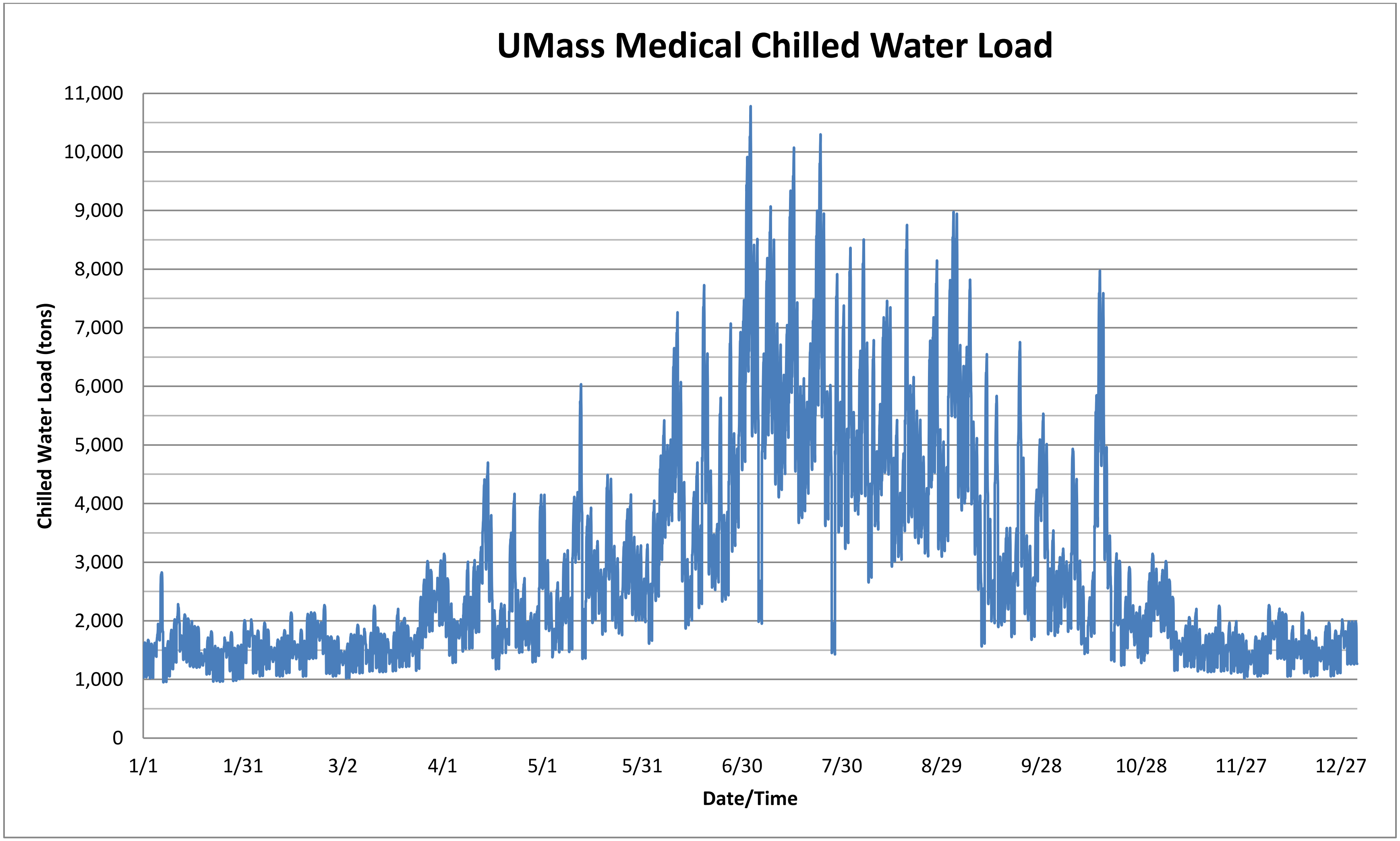


Figure 1: Annual hourly chilled water load profile for UMass Medical Center

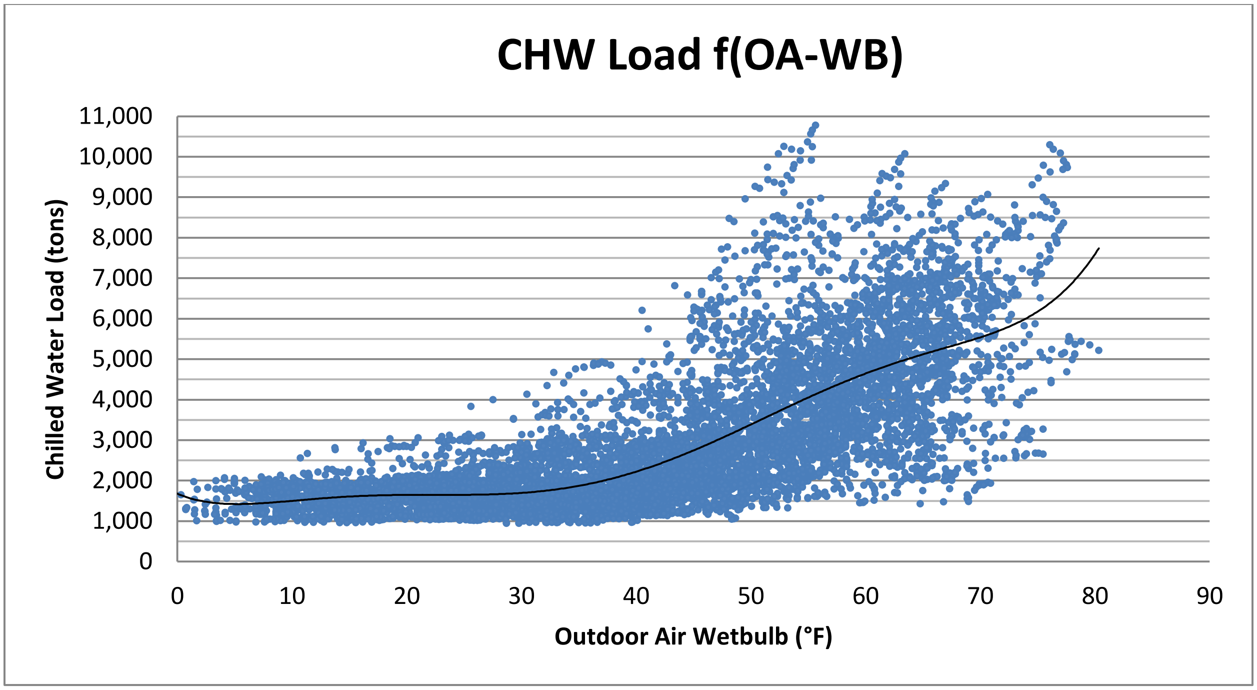


Figure 2: Scatterplot of hourly chilled water load versus outdoor air wet-bulb temperature. The chart indicates that the chilled water load averages approximately 1,500 - 1,600 tons when the outdoor air wet-bulb temperature is below 30°F. The maximum chilled water load in these conditions is approximately 3,000 tons.

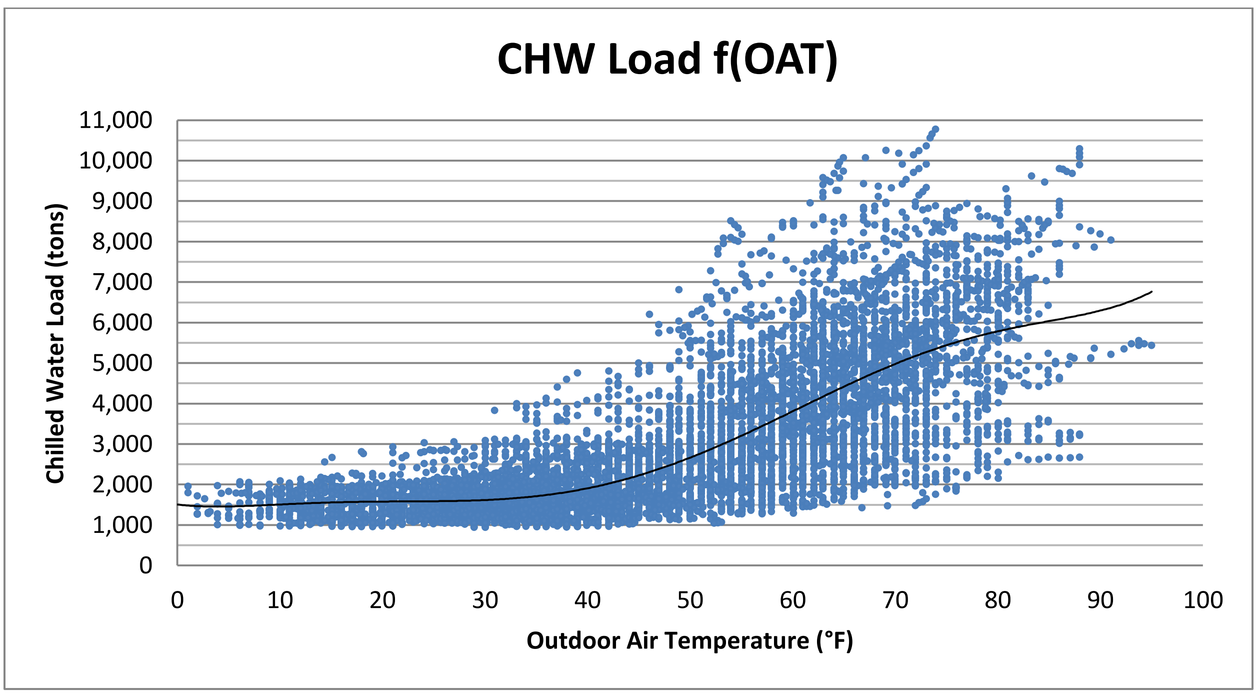


Figure 3: Scatterplot of hourly chilled water load versus outdoor air temperature. The chart indicates that the chilled water load averages approximately 1,500 - 1,600 tons when the outdoor air dry bulb temperature is below 30°F.

ENERGY MODELING METHODOLOGY

BASELINE CHILLER PLANT MODELING

An existing case hourly spreadsheet model of the chilled water plant was developed to simulate energy consumption during periods when a plate & frame heat exchanger or thermo-siphon sequence could be enabled for “free” winter cooling.

During periods when either free cooling sequence is disabled in the proposed case, the chiller plant’s energy consumption will remain generally unchanged. For this reason, performance and energy consumption of the plant’s equipment was not modeled during these periods.

The baseline model is divided into several sections, each calculating performance for a different group of equipment including: chilled water pumping, condenser water pumping, cooling towers, and chillers. The table below summarizes the key parameters that are used as inputs to the baseline model. This information was gathered from multiple sources including equipment nameplates, drawings, submittals, manufacturer performance data, and historical trend data.

System/Equipment	Parameter	Value	Units
-	-	-	-
Chiller 5	Chiller 5 Design Capacity	4000	tons
	Chiller Type	Cent/Open	-
	Drive Type	VFD	-
	Design Chiller Efficiency	0.1777	EIR[1]
	Chiller Rated LCHWT	42.0	°F
	Chiller Rated ECWT	85.0	°F
	Design Evaporator Flow	6,400	gpm
	Design Condenser Flow	8,000	gpm
Cooling Tower Cells #1 & #2 (each)	Design Capacity	7,088	tons HR
	Design Flow	9,450	gpm
	Design Wet-bulb Temperature	75	°F
	Design Approach	10	°F
	Design Range	18	°F
	Approach Coefficient[2]	0.5	-
	Fan Power	150	bhp
	Fan Drive Type	VFD	-
	Winter CWST Set-point	65	°F

[1] EIR = Energy Input Ratio; Defined to be the ratio of the electric energy input (Btu/hr) to the rated capacity (Btu/hr) of the chiller (i.e., the reciprocal of COP) at the Air-Conditioning, Heating, and Refrigeration Institute (AHRI, formally known as ARI) rated condtion.

[2] Performance characteristic indicating how the cooling tower approach changes as a function of range and ambient wet bulb conditions. A value greater than 0 indicates the cooling tower approach increases at wet bulb conditions below design.

System/Equipment	Parameter	Value	Units
-	-	-	-
Chilled Water Pumps (each)	Design Flow	6400	gpm
	Design Head	137.5	ft wg
Condenser Water Pumps (each)	Design Flow	7500	gpm
	Design Head	110	ft wg

For each hour of the year, equipment performance was calculated using Worcester, MA TMY3 ambient weather conditions, the campus chilled water load, the input parameters above, and the plant's sequences of operation. In addition, performance curves for cooling towers and chillers available through the eQuest building simulation program's library were used where necessary to calculate part load and off-design energy use. The library chiller performance curves were modified to reflect the actual part load and off-design performance of Chiller #5, as determined based on performance runs obtained from York. A chart showing performance curves at various entering condenser water temperatures is included in the Appendix.

Performance curves are one of three types: quadratic, cubic, or bi-quadratic. Quadratic and cubic curves calculate an independent variable's value based on a single dependent variable whereas bi-quadratic curves determine the independent variable's value based on two unique dependent variables.

Equipment/ System	Independent Variable	Dependent Variables	Curve Type	Curve Coefficients					
-	-	-		*a*	*b*	*c*	*d*	*e*	*f*
Chiller 5	EIR	PLR, DT	Bi-Quadratic	0.14703	-0.0035	1.01161	-0.0036	0.00027	-0.0116
	EIR	CHWST, ECWT	Bi-Quadratic	1.42868	-0.0823	0.0003	0.03622	-0.0003	0.00044
	Capacity	CHWST, ECWT	Bi-Quadratic	-0.3892	-0.022	-0.0003	0.04975	-0.0005	0.00067
Cooling Tower Cells #1 & #2	Capacity	Approach, WB	Bi-Quadratic	0.50061	0.00588	0.00022	-0.0191	0.00022	0.00106
	Capacity	Range, WB	Bi-Quadratic	0.08352	0.11247	-0.0014	3.4E-05	3.1E-05	-0.0003
	Capacity	Airflow	Quadratic	0.04977	1.04698	-0.0965	-	-	-
	Fan Power	Speed	Cubic	0.33163	-0.8857	0.60557	0.94848	-	-
Pumps	Power	Flow	Quadratic	0.36977	0.84038	-0.2101	-	-	-

POWER PLANT MODELING

If any of the free winter cooling measures proposed in this study are implemented, the overall electric demand on the campus' power plant may be reduced by as much as 700 kW, although the average electric demand savings typically 500 kW or less. Since the campus imports a relatively fixed minimum amount of electricity of 500 kW, any electric demand reduction will require the output of the combustion turbine generator (G4) to be reduced by a corresponding amount. Since reducing the electric output of G4 also reduces the turbine's mass flow and HRSG steam output, the lost steam flow would be made up by the 1,100 psig steam boiler (Boiler #3 or #4).

Based on discussions with the power plant director and operators, the minimum allowable turndown for G4 is approximately 4,800 - 5,000 kW. A review of historical data available from the power plant's SCADA system between January - February 2015 indicates that there are already periods when this minimum threshold is reached. When it occurs, one of two actions are taken: 1) the combustion turbine generator is shut down, additional electricity is purchased, and high pressure steam output from the boiler(s) is increased, or 2) 1,100 psig superheated steam is bypassed around the 5,000 kW back-pressure turbine (G3) using one of two pressure reducing valves. In scenario 2, the gas turbine remains on at minimum output but the plant's electric output is reduced by decreasing output from G3 while maintaining the same total steam flow to the extraction turbine(s) and the medium pressure campus distribution. Both scenarios are undesirable due to the greater cost of purchasing electricity compared to on-site co-generation and the inefficiency of reducing the enthalpy of the 1,100 psig high pressure steam without it performing useful work in the back pressure turbine.

If a waterside economizer strategy is implemented in the chiller plant, there will be a greater number of hours when the electric load is below the combustion turbine's minimum turndown. In these scenarios, it may be more efficient to keep G4 running at minimum, reduce steam output from the 1,100 psig boilers, and bring on a 250 psig boiler (Boiler #1 or #2) to make up the necessary steam. By reducing 1,100 psig steam flow to the back-pressure turbine G3, the electric output of the plant can be reduced when the campus electric demand is low and waterside economizer is active. Energy savings during these periods will be realized by generating steam at a lower enthalpy (1202.38 Btu/lb at 250 psig saturated compared to 1413.75 Btu/lb at 1,100 psig and 850°F superheat temperature).

ENERGY CONSERVATION MEASURES

The following sections describe the base case, proposed case, and energy savings methodology for each of the two waterside economizer technologies analyzed as part of this study. “Retrofit Chillers Thermo-siphon” is divided into three options, each having unique potential energy savings and implementation costs. The options include the retrofit of 1, 2, or 3 steam turbine-driven chillers with thermo-siphon capability. ECM-23B is the measure option selected for inclusion in DCAMM’s AEP audit template as it offers the best combination of return on investment and operational flexibility for UMass Medical Center.

ECM-23.00: Retrofit Chillers with Thermo-Siphon

This measure is proposed as three separate options, which includes the retrofit of 1, 2, or 3 chillers with thermo-siphon capability. The potential energy savings and estimated project economics for each scenario is included separately in the following sections and can be compared using the Executive Summary Table on Page 7.

ECM-23A Retrofit Three Chillers with Thermo-siphon

Measure Economics Summary

ECM #	23A	Retrofit Three Chillers with Thermo-Siphon					
Electric Energy Savings	Electric Cost Savings	Equivalent Gas Energy Savings	Equivalent Gas Cost Savings	Estimated Retrofit Cost	Potential Utility Incentive	Net Project Cost	Simple Payback Period
kWh	$	therms	$	$	$	$	years
985,091	$98,509	72,514	$61,637	$1,424,100	$295,527	$1,128,573	18.3

Base Case

Between late October and early April, the plant's 4,000 ton electric chiller #5 is used exclusively to meet the campus chilled water load and all four steam turbine-driven chillers generally remain off. The table below summarizes conditions that are typical of "winter" operation.

Parameter	Value	Units
-	-	-
Chilled Water Supply Temperature	42	°F
Chilled Water Flow	6,000	gpm
Chiller Condenser Water Supply Temperature	~68	°F
Cooling Tower Sump Temperature	~65	°F
Condenser Water Flow (through chiller)	6,000	gpm
Process Condenser Water Flow (estimated)	4,800	gpm
Quantity of Tower Cells with Flow	2	-
Quantity of Tower Cells Used for Chiller CW	1	-
Quantity of Tower Cells Used for Component Cooling	2	-

As listed in the table above, two cooling tower cells are typically used during the winter to reject heat from the plant's two condenser water loops. Tower cells #1 and #2 are used instead of the plant's two newest counter flow cells, #5 and #6 to reduce wear on this equipment. It was observed during visits to the site that return condenser water flow from chiller #5 and the heat recovery steam generator components is directed to Tower cell #1 only, while return water from the main process component cooling loop is directed to both Tower cells (#1 and #2). As a result, the condenser water flow to cell #1's hot deck is more than three times that of cell #2.

In the existing case configuration, the minimum condenser water supply set-point is limited to no lower than 65°F; temperatures below this level can cause unwanted sub-cooling of steam condensate leaving turbine surface condensers.

Proposed Case

Overview

This measure's focus is to retrofit three of the existing steam turbine-driven chillers with thermo-siphon free-cooling capability. Thermo-siphon operation is based on the principle that refrigerant migrates to the area of lowest temperature. When condenser water is available at temperatures lower than the required leaving chilled water temperature, the differential between the chiller's evaporator and condenser can be used to complete the refrigeration cycle without work input from the compressor.

During periods when the ambient wet-bulb is approximately 30°F or less, the condenser water supply temperature set-point can be reset and the cooling tower can be used to make condenser water at between 38-40°F, which is sufficient to supply chilled water at 42°F to the campus. The proposed system would include retrofitting Chiller #4, Chiller #3, and Chiller #2 (or Chiller #1) with thermo-siphon capability, which would offer approximately 3,200 tons of cooling capacity at a 38°F condenser water supply temperature and 42°F chilled water supply temperature. Chillers would be staged on in free cooling mode as necessary to meet the chilled water load. The charts on the following pages show each chiller's capacity as a function of temperature differential (chilled water supply temperature - condenser water supply temperature)

Upon changeover to free cooling, new shutoff valves in each chiller's liquid and gas lines are opened and steam flow to all compressor turbines would remain off. Liquid refrigerant would drain by gravity from the storage tank into the evaporator, flooding the tube bundle. Since the refrigerant temperature and pressure will be higher in the evaporator than in the condenser, due to the water temperature difference, the refrigerant gas boiled off in the evaporator will flow to the condenser (via a new bypass line around the compressor). The gas then condenses and flows by gravity back to the evaporator. This automatic refrigeration cycle is sustained as long as a temperature difference exists between the condenser water and evaporator water. The load capacity of each chiller is directly proportional to the nameplate capacity of the chiller and this temperature differential.

While the thermo-siphon sequence is active, the electric chiller would be shut down and the chilled water load would be met primarily using pumping and tower fan energy.

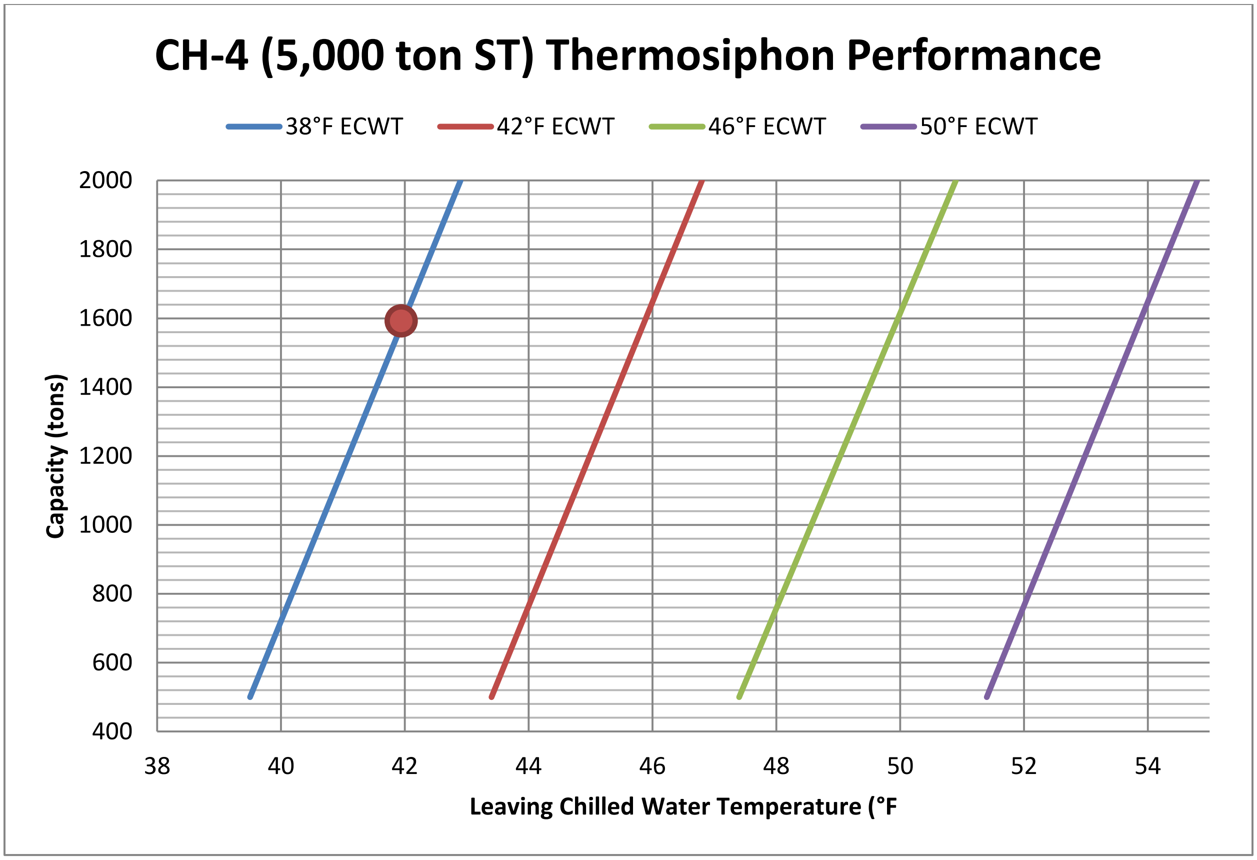


Figure 4: Chiller 4 (5,000 ton steam turbine-driven chiller) performance in thermo-siphon mode. Cooling capacity is illustrated as a function of leaving chilled water temperature at varying entering condenser water temperatures. The chart indicates that at a 42°F leaving chilled water set-point, Chiller 4 has a capacity of 1,600 tons when the entering condenser water temperature is 38°F.

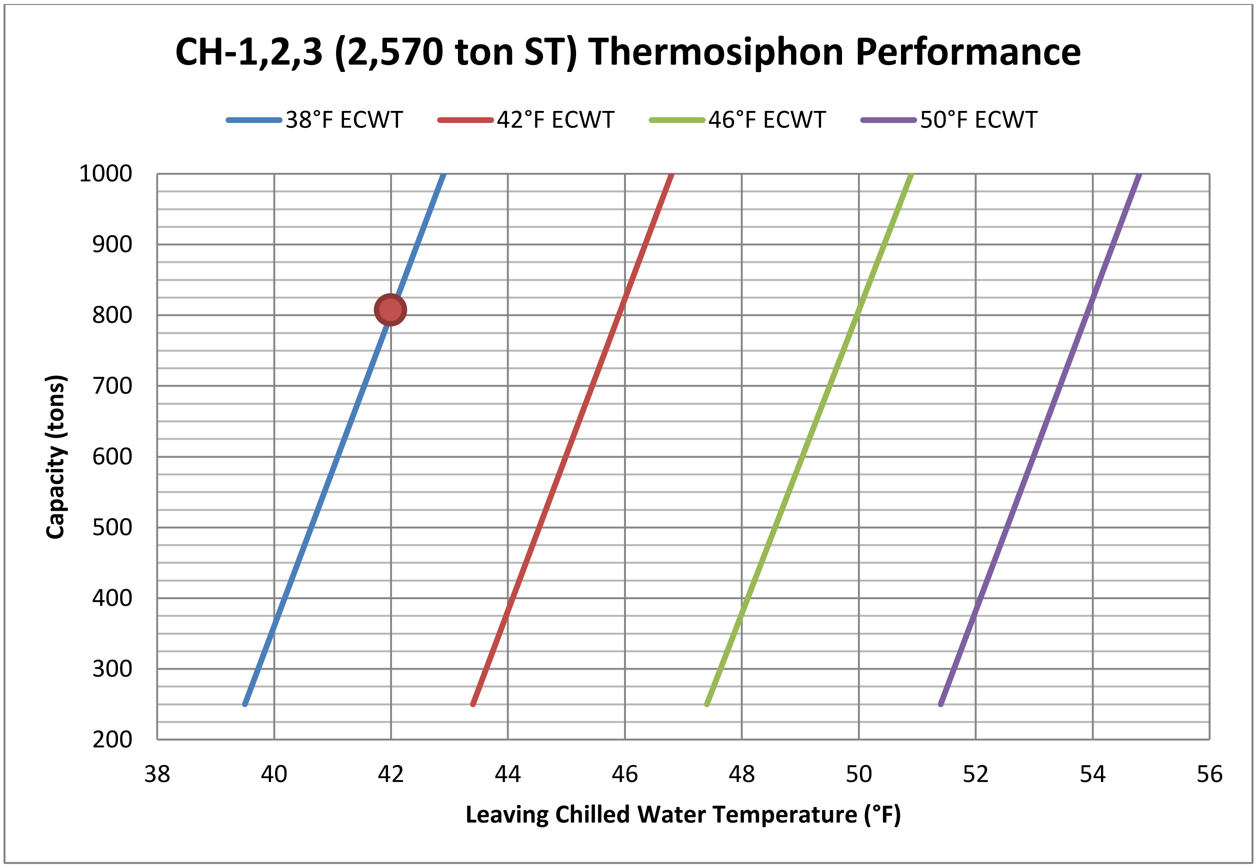


Figure 5: Chiller 1,2,3 (2,500 ton steam turbine-driven chiller) performance in thermo-siphon mode. Cooling capacity is illustrated as a function of leaving chilled water temperature at varying entering condenser water temperatures. The chart indicates that at a 42°F leaving chilled water set-point, Chillers 1,2,3 each have a capacity of 800 tons when the entering condenser water temperature is 38°F.

Cooling Tower Operation

Since many of the power plant's process cooling loads have minimum condenser water temperature requirements of approximately 65°F, it is necessary to direct the process and chiller condenser water loop flows to separate tower cells while the free cooling sequence is active. It is also necessary to separate the cooling tower sumps, since all six cells share a common sump.

In order to isolate the two loops, up to four tower cells will be needed during thermo-siphon operation while all three chillers are operating: Tower cells #1, #2, #5, and #6. Return water from the process loop will be sent to the hot decks of cells #5 (and/or #6) by opening the corresponding isolation valves. Return condenser water from the active chillers will be sent to cell #1 (and/or #2) by opening their isolation valves. Each of the eight valves (2 valves per tower cell) will require new electric actuators so that tower isolation and staging can be controlled automatically. New actuators have not been carried for cells #3 or #4.

In order to isolate the 'warm' (~65°F) and 'cold' (~38°F) tower sumps while the plate and frame is active, the existing manual butterfly isolation valve that can separate the sumps of cells #1 and #2 from the others will be retrofitted with a high-torque electric actuator. The plate and frame heat exchanger will draw low temperature supply water from the basins of cells #1 and #2, while all process cooling water will be supplied from the warmer basin shared between cells #3, #4, #5, and #6.

Pump Operation

In the proposed case, condenser water pumps will be staged on according to the number of chillers running in free cooling mode. The maximum flow to tower cells #1 and #2 while the thermo-siphon sequence is active on all three chillers will be 18,900 gpm, the combined design capacity of the two cells.

Sequence of Operation

The following general sequence of operation is proposed for chiller thermo-siphon operation. In addition, the sequence should be limited to a threshold number of starts per day and should only be enabled when the ambient wet-bulb temperature has been below 30°F (adjustable based on chilled water load) for one hour continuously. This temperature set-point is determined based on the following full load criteria:

- Condenser water flow of 9,450 gpm per tower cell, 18,900 gpm total
- A target leaving chilled water set-point of 42°F
- Approximate 8°F tower approach at 30°F ambient wet bulb
- 1,500 tons maximum heat rejection per cell, 3,000 ton total capacity.

Starting Thermo-siphon Sequence

1. Start Chiller #4 condenser water pump and open condenser water isolation valve on cooling tower cell #1. Close the isolation butterfly valve in the cooling tower sump to isolate cells #1 and #2 from #3 - #6.
2. Precool the condenser water loop serving Chiller #4 by setting the fan speed on cell(s) #1 (and #2 if active) to 100%.
3. After the condenser water supply temperature has reached approximately 39°F, start the thermo-siphon sequence by opening the necessary refrigerant flow control valves on Chiller #4.
4. Bring on additional chillers (#3, #2) and modulate the condenser water supply temperature as necessary to meet the leaving chilled water temperature set-point.
5. Shut down Chiller #5, allowing chilled water and condenser water flow to continue to circulate for a predetermined amount of time before closing both isolation valves.

Stopping Waterside Economizer Sequence

If the ambient wet-bulb temperature is greater than set-point or the leaving chilled water temperature is 1°F or more above set-point for a predetermined amount of time, the thermo-siphon controls shall initiate a shutdown sequence.

1. Open chiller flow isolation valves and start Chiller #5
2. Once the chiller is maintaining the chilled water supply temperature set-point, isolate the plate and frame by closing its isolation valves.
3. Shut down cooling tower fans on Cells #1 and/or #2.

Energy Savings Methodology

Energy savings associated with this measure were calculated using an hourly spreadsheet model similar to the baseline model discussed in the "Baseline Model Methodology" section. The table below summarizes key input parameter differences between the baseline and proposed case models.

Parameter	Base Model	Proposed Model	Units
-	*Value*	*Value*	-
Chilled Water Supply Temperature	42	42	°F
Chilled Water Flow	6,000	6,000+	gpm
Chiller Condenser Water Supply Temperature	~68	38-39	°F
Cooling Tower Sump Temperature (Chiller/Process)	65/65	38-39/65	°F
Condenser Water Flow (through chiller(s))	6,000	8,000+	gpm
Process Condenser Water Flow (estimated)	4,800	4,800	gpm
Quantity of Tower Cells with Flow	2	2-4	-
Quantity of Tower Cells Used for Chillers	1	1-2	-
Quantity of Tower Cells Used for Process Cooling	2	1-2	-

As shown in the table above, the proposed case condenser water flow while in thermo-siphon will vary depending on the number of chillers used to meet the load. For example, when the chilled water load is approximately 1,600 tons or less, Chiller #4 can be used with a corresponding condenser water flow of 8000 gpm. When Chiller #3 or #2 is added, an additional 7,500 gpm of condenser water flow will be required and a second cooling tower cell will be used. When all three chillers are running in thermo-siphon mode, 18,900 gpm of flow will be delivered, which is equal to the combined flow capacity of tower cells #1 and #2.

Proposed case tower fan speed and energy consumption were modeled using the same capacity curves described in the baseline model methodology. The proposed case condenser water flow, approach temperature, and range were used to calculate the new tower fan speed. Hourly approach and range were calculated using the equations in the table below. For each hour of the year, the model calculates whether or not the thermo-siphon sequence is active based on the ambient wet-bulb temperature and chilled water load. If the minimum tower approach temperature (as calculated using the formula below) plus the hourly wet-bulb temperature is less than or equal to the target condenser water supply set-point (38-39°F), then thermo-siphon would be active. During the first hour of operation, the energy model assumed the average tower fan speed for active cell(s) would be 100% in order to pull the condenser water supply temperature down to the set-point necessary for free cooling operation.

Parameter	Calculation	Units
-	-	-
Cooling Tower Range (ΔT)	= E [Btu/hr] / (500 * Q [gpm])	°F
Approach Temperature	= T,CWST - T,wetbulb	°F
Minimum Approach Temperature	= m * (T,design wetbulb - T,hourly wetbulb) + T,design approach	°F

where:

m = Approach Coefficient (-0.2)

As described in the baseline model methodology, the hourly cooling tower cell airflow ratio and fan VFD speed were calculated using performance curves obtained from the eQuest simulation program's library and adjusted for model-specific tower design criteria. The following equation for cooling tower capacity was solved for the airflow ratio variable:

$$Tower\ Capacity = Tower\ Load * \frac{Cap\ f(App, WB)}{Cap\ f(Range, WB)} * Cap\ f(Airflow)$$

where:

$Tower\ Capacity$ = Hourly Heat Rejection Capacity of the Tower Cell [Btu/hour]
$Tower\ Load$ = Hourly Heat Rejection Load on the Tower Cell [Btu/hr]
$Cap\ f(Airflow)$ = Cubic performance curve with dependent variable is cell airflow ratio

During periods when the calculated airflow ratio required was less than the minimum speed of the tower fan VFD (20%), a cycling factor was calculated to model the hourly fan energy consumption. The calculation used was:

$$Hourly\ Fan\ VFD\ Cycle\ Factor = \frac{Calculated\ Airflow\ Ratio}{Minimum\ Fan\ VFD\ Speed}$$

where:

All variables are ratios between 0 and 1

The charts on the following page show the calculated proposed case tower fan VFD speed and cycle ratio for Cells #1 and #2 during each hour of thermo-siphon operation. Note that periods when the fan speed is indicated as 0% represents periods when the free cooling sequence is disabled.

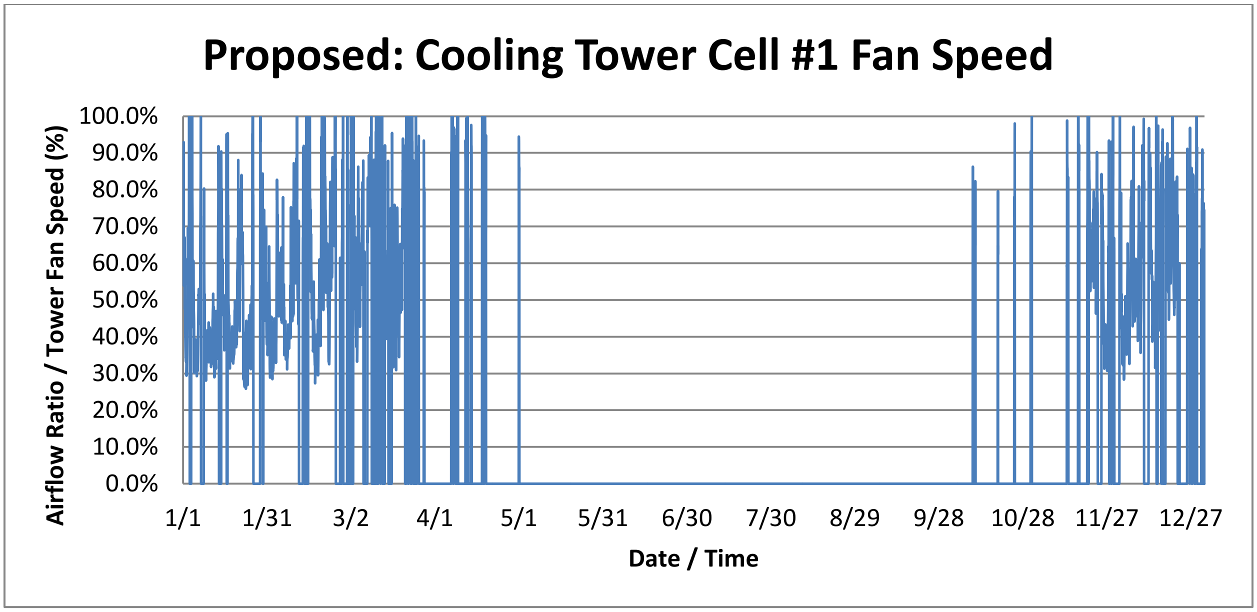


Figure 6: Profile of Cooling Tower Cell #1 Airflow Ratio / VFD Speed in proposed case thermo-siphon model. The fan speed varies with both the tower range (load) and ambient conditions. The fan speed is typically at or near maximum when the outdoor air wet-bulb temperature approaches the maximum set-point. Cell #1 is defined as the lead tower cell in the proposed case model.

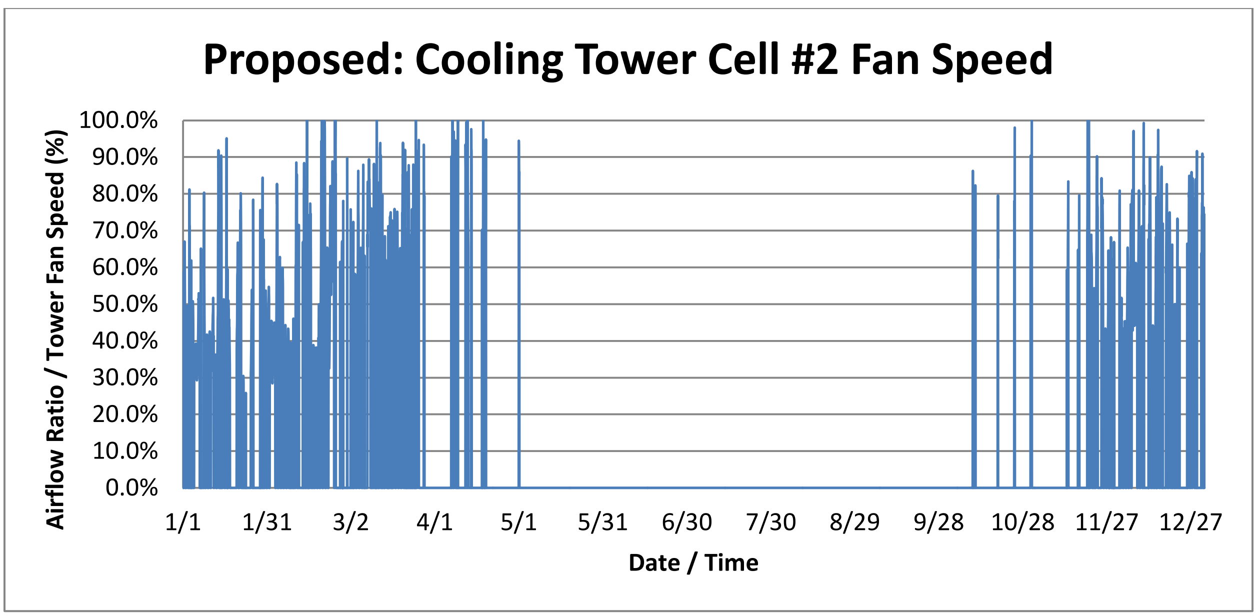


Figure 7: Profile of Cooling Tower Cell #2 Airflow Ratio / VFD Speed in proposed case thermo-siphon model. The fan speed varies with both the tower range (load) and ambient conditions. The fan speed is typically at or near maximum when the outdoor air wet-bulb temperature approaches the maximum set-point. Cell #2 is defined as the lag tower cell in the proposed case model and is brought on when more than one chiller is active.

The chart below shows the calculated proposed case tower fan VFD speed and cycle ratio for Cell #6 during each hour of plate and frame operation. This cell will reject heat only from the process condenser water loop while the plate & frame heat exchanger is active.

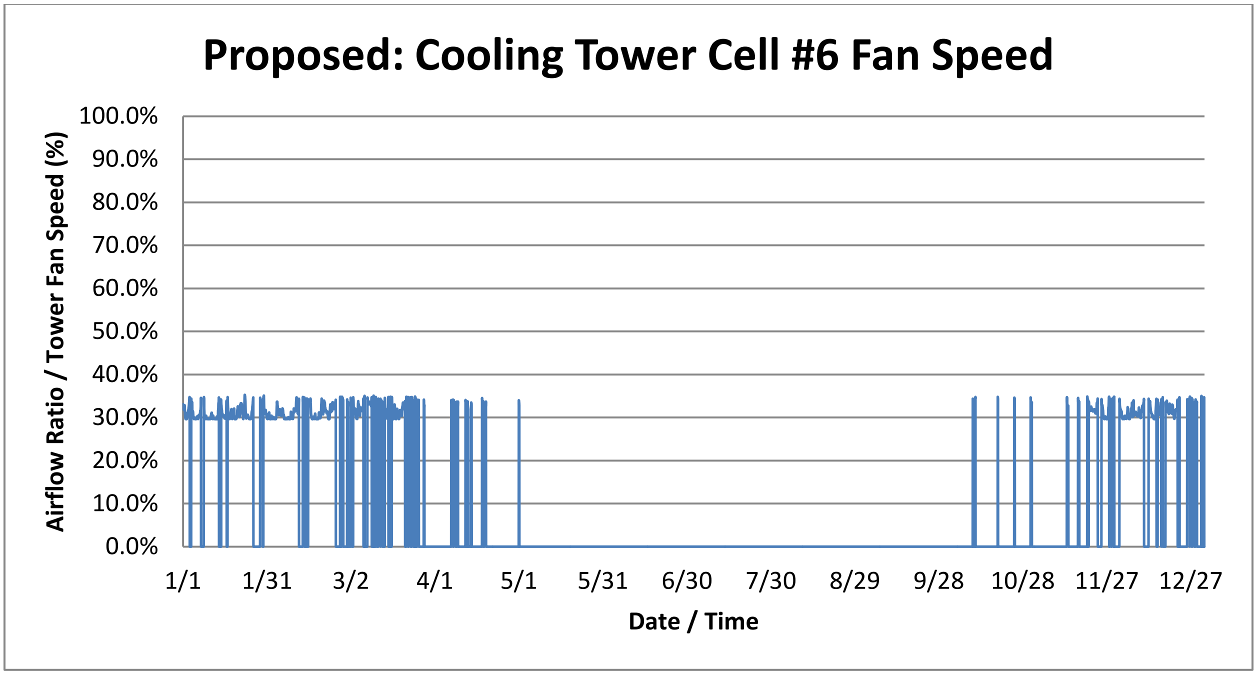


Figure 8: Proposed case tower fan airflow ratio / VFD speed for Cell #6, which rejects heat from the process condenser water loop while the plate & frame heat exchanger is active. The model results indicate that one cell can be used to reject the loop's heat compared to the two cells that are used during the winter in the existing case.

COST ESTIMATE

The cost estimate for this measure is shown in the table on the following page. The estimate includes a complete controls upgrade for each of the chillers in the measure, which incorporates a new control panel and replacement of all sensors and end devices, including the steam turbine governor.

Opinion of Probable Construction Cost												
ECM-2-A: Retrofit Three Chillers with Thermosiphon												

B2Q Associates, Inc.
100 Burtt Rd. Ste. 212
Andover, MA 01810
(978) 208 - 0609

Customer: UMASS MEDICAL
Address: Power Plant

Date: 4/15/2015
Estimated By: SD
Checked By:

General			Materials				Labor					
Number	Source	Item	Type	Quantity	Unit Cost	Total Cost	Unit Rate	Workers	Hours Each	Labor Cost	Total Cost
1	4	Chiller 2,3 Thermosiphon Retrofit	ea	2	$60,000	$120,000	$150			$0	$120,000
2	4	Chiller 4 Thermosiphon Retrofit	ea	1	$60,000	$60,000	$150			$0	$60,000
3	1	CT Control Valve Actuators	ea	14	$2,500	$35,000	$150	2	8	$33,600	$68,600
4	1	CT Sump Isolation Valve Actuator	ea	1	$5,000	$5,000	$150	2	16	$4,800	$9,800
5	2	Chiller 2,3 Control Panel Replacement	ea	3	$250,000	$750,000	$150			$0	$750,000
7	3	Condenser water loop isolation valves	ea	1	$10,000	$10,000	$150	2	24	$7,200	$17,200
8	3	SCADA Control Points & Programming	ea	14	$1,500	$21,000	$150	1	8	$16,800	$37,800
9	3	Staging Sequence Programming	ea	1		$0	$150	2	100	$30,000	$30,000
10	3	Contractor Commissioning	ea	1		$0	$150	1	28	$4,200	$4,200
11	3	As-Built	ea	1		$0	$150	1	28	$4,200	$4,200
						$0				$0	$0
						$0				$0	$0

Sources	
1	Means
2	Vendor Quote
3	Other
4	Vendor Allowance

Subtotal		$1,101,800
Contingency	10%	$110,200
Engineering	5%	$60,600
Construction Administration	5%	$60,600
Commissioning	2.5%	$30,300
Construction Observation	2.5%	$30,300
Project Closeout & Expenses	2.5%	$30,300
Total		$1,424,100

ECM-23B Retrofit Two Chillers with Thermo-siphon

Measure Economics Summary

ECM #	23B	Retrofit Two Chillers with Thermo-Siphon					
Electric Energy Savings	Electric Cost Savings	Equivalent Gas Energy Savings	Equivalent Gas Cost Savings	Estimated Retrofit Cost	Potential Utility Incentive	Net Project Cost	Simple Payback Period
kWh	$	therms	$	$	$	$	years
938,848	$93,885	69,206	$58,825	$996,100	$281,654	$714,446	12.1

Base Case

Refer to the base case description for ECM-23A on Page 19. The same base case is used for ECM-23B.

Proposed Case

The proposed case for this measure is similar to the proposed case for ECM-23A, except that only two chillers would be retrofit with thermo-siphon capability (Chiller #4 and Chiller #3, #2, or #1). At 42°F leaving chilled water temperature and 38°F entering condenser water temperature, the two chillers would have a capacity of 2,400 tons, which can meet the campus chilled water load for the majority of winter hours. The same sequence of operation would apply, except when both available chillers are operating in thermo-siphon and the chilled water load exceeds available capacity. In this situation, Chiller #5 would be started and the 2,500 ton chiller operating in thermo-siphon shut down to maintain a minimum load on Chiller #5 of at least 20% (800 tons). During this period, one of the two active cooling tower cells supplying 38°F condenser water could also be shut down due to the reduction in condenser water flow.

Energy Savings Methodology

Energy savings for this measure were calculated using the same methods used for ECM-23A. During hours when the campus chilled water load exceeds 2,400 tons, Chiller #3 was modeled to shut down and the electric chiller was modeled to turn on with a minimum load 800 tons. The same performance curves used in the base case model were used to calculate hourly proposed electric chiller demand as a function of part load ratio, entering condenser water temperature, and leaving chilled water temperature. At all times while the thermo-siphon sequence is active based on ambient wet-bulb conditions, Chiller #4 remains on in the proposed case model with a maximum capacity of 1,600 tons.

Cost Estimate

The cost estimate for this measure is shown in the table on the following page. The estimate includes a complete controls upgrade for each of the chillers in the measure, which incorporates a new control panel and replacement of all sensors and end devices, including the steam turbine governor.

Opinion of Probable Construction Cost

ECM-2-B: Retrofit Two Chillers with Thermosiphon

B2Q Associates, Inc.
100 Burtt Rd. Ste. 212
Andover, MA 01810
(978) 208 - 0609

Customer: UMASS MEDICAL
Address: Power Plant

Date: 4/15/2015
Estimated By: SD
Checked By:

General			Materials				Labor				
Number	Source	Item	Type	Quantity	Unit Cost	Total Cost	Unit Rate	Workers	Hours Each	Labor Cost	Total Cost
1	4	Chiller 2 Thermosiphon Retrofit	ea	1	$60,000	$60,000	$150			$0	$60,000
2	4	Chiller 4 Thermosiphon Retrofit	ea	1	$60,000	$60,000	$150			$0	$60,000
3	1	CT Control Valve Actuators	ea	12	$2,500	$30,000	$150	2	8	$28,800	$58,800
4	1	CT Sump Isolation Valve Actuator	ea	1	$5,000	$5,000	$150	2	16	$4,800	$9,800
5	2	Chiller Control Panel Replacement	ea	2	$250,000	$500,000	$150			$0	$500,000
6	3	Condenser water loop isolation valves	ea	1	$10,000	$10,000	$150	2	32	$9,600	$19,600
7	3	SCADA Control Points & Programming	ea	12	$1,500	$18,000	$150	1	8	$14,400	$32,400
8	3	Staging Sequence Programming	ea	1		$0	$150	2	80	$24,000	$24,000
9	3	Contractor Commissioning	ea	1		$0	$150	1	20	$3,000	$3,000
10	3	As-Built	ea	1		$0	$150	1	20	$3,000	$3,000
						$0				$0	$0
						$0				$0	$0

Sources	
1	Means
2	Vendor Quote
3	Other
4	Vendor Allowance

	Subtotal	$770,600
Contingency	10%	$77,100
Engineering	5%	$42,400
Construction Administration	5%	$42,400
Commissioning	2.5%	$21,200
Construction Observation	2.5%	$21,200
Project Closeout & Expenses	2.5%	$21,200
	Total	$996,100

ECM-23C Retrofit One Chiller with Thermo-siphon

Measure Economics Summary

ECM #	23C	Retrofit One Chiller with Thermo-siphon					
Electric Energy Savings	Electric Cost Savings	Equivalent Gas Energy Savings	Equivalent Gas Cost Savings	Estimated Retrofit Cost	Potential Utility Incentive	Net Project Cost	Simple Payback Period
kWh	**$**	**therms**	**$**	**$**	**$**	**$**	**years**
509,048	$50,905	38,013	$32,311	$543,000	$152,714	$390,286	12.1

Base Case

Refer to the base case description for ECM-23A on Page 19. The same base case is used for ECM-23C.

Proposed Case

The proposed case for this measure is similar to the proposed case for ECM-23B, except that only Chiller #4 would be retrofit with thermo-siphon capability and its controls upgraded. At 42°F leaving chilled water temperature and 38°F entering condenser water temperature, the machine has a capacity of 1,600 tons, which can meet the campus chilled water load for approximately 1,300 hours per year. The same sequence of operation would apply, except when the chilled water load exceeds 1,600 tons and ambient conditions allow continued use of thermo-siphon operation. In this situation, Chiller #5 would be started and the load would be balanced between the two chillers to maintain at least 800 tons on Chiller #5.

Energy Savings Methodology

Energy savings for this measure were calculated using the same methods used for ECM-23B. During hours when the campus chilled water load exceeds 1,600 tons, Chiller #5 was modeled to turn on with a minimum load of 800 tons. The same performance curves used in the base case model were used to calculate hourly proposed electric chiller demand as a function of part load ratio, entering condenser water temperature, and leaving chilled water temperature. At all times while the thermo-siphon sequence is active based on ambient wet-bulb conditions, Chiller #4 remains on in the proposed case model with a maximum capacity of 1,600 tons.

Cost Estimate

The cost estimate for this measure is shown in the table on the following page. The estimate includes a complete controls upgrade for Chiller #4, which incorporates a new control panel and replacement of all sensors and end devices, including the steam turbine governor.

Opinion of Probable Construction Cost

ECM-2-C: Retrofit One Chiller with Thermosiphon

B2Q Associates, Inc.
100 Burtt Rd. Ste. 212
Andover, MA 01810
(978) 208 - 0609

Customer: UMASS MEDICAL
Address: Power Plant

Date: 4/15/2015
Estimated By: SD
Checked By:

General			Materials				Labor				
Number	Source	Item	Type	Quantity	Unit Cost	Total Cost	Unit Rate	Workers	Hours Each	Labor Cost	Total Cost
1	4	Chiller 4 Thermosiphon Retrofit	ea	1	$60,000	$60,000	$150			$0	$60,000
2	1	CT Control Valve Actuators	ea	8	$2,500	$20,000	$150	2	8	$19,200	$39,200
3	1	CT Sump Isolation Valve Actuator	ea	1	$5,000	$5,000	$150	2	16	$4,800	$9,800
4	2	Chiller Control Panel Replacement	ea	1	$250,000	$250,000	$150			$0	$250,000
5	3	Condenser water loop isolation valves	ea	1	$10,000	$10,000	$150	2	32	$9,600	$19,600
6	3	SCADA Control Points & Programming	ea	8	$1,500	$12,000	$150	1	8	$9,600	$21,600
7	3	Staging Sequence Programming	ea	1		$0	$150	2	50	$15,000	$15,000
8	3	Contractor Commissioning	ea	1		$0	$150	1	16	$2,400	$2,400
9	3	As-Built	ea	1		$0	$150	1	16	$2,400	$2,400
						$0				$0	$0
						$0				$0	$0

Sources	
1	Means
2	Vendor Quote
3	Other
4	Vendor Allowance

Subtotal		$420,000
Contingency	10%	$42,000
Engineering	5%	$23,100
Construction Administration	5%	$23,100
Commissioning	2.5%	$11,600
Construction Observation	2.5%	$11,600
Project Closeout & Expenses	2.5%	$11,600
Total		$543,000

WATERSIDE ECONOMIZER (PLATE & FRAME HEAT EXCHANGER)

MEASURE ECONOMICS SUMMARY

ECM #	23D	Waterside Economizer (Plate & Frame Heat Exchanger)					
Electric Energy Savings	Electric Cost Savings	Equivalent Gas Energy Savings	Equivalent Gas Cost Savings	Estimated Retrofit Cost	Potential Utility Incentive	Net Project Cost	Simple Payback Period
kWh	$	therms	$	$	$	$	years
764,268	$76,427	57,411	$48,800	$1,115,232	$229,280	$885,952	18.2

BASE CASE

Between late October and early April, the plant's 4,000 ton electric chiller #5 is used exclusively to meet the campus chilled water load. The table below summarizes conditions that are typical of "winter" operation.

Parameter	Value	Units
-	-	-
Chilled Water Supply Temperature	42	°F
Chilled Water Flow	6,000	gpm
Chiller Condenser Water Supply Temperature	~68	°F
Cooling Tower Sump Temperature	~65	°F
Condenser Water Flow (through chiller)	6,000	gpm
Process Condenser Water Flow (estimated)	4,800	gpm
Quantity of Tower Cells with Flow	2	-
Quantity of Tower Cells Used for Chiller CW	1	-
Quantity of Tower Cells Used for Component Cooling	2	-

As listed in the table above, two cooling tower cells are typically used during the winter to reject heat from the plant's two condenser water loops. Tower cells #1 and #2 are used instead of the plant's two newest counter flow cells, #5 and #6 to reduce wear on this equipment. It was observed during visits to the site that return condenser water flow from chiller #5 and the heat recovery steam generator components is directed to Tower cell #1 only, while return water from the main process component cooling loop is directed to both Tower cells (#1 and #2). As a result, the condenser water flow to cell #1's hot deck is more than three times that of cell #2.

In the existing case configuration, the minimum condenser water supply set-point is limited to no lower than 65°F; temperatures below this level can cause unwanted sub-cooling of steam condensate leaving turbine surface condensers.

PROPOSED CASE

OVERVIEW

This measure's focus is to implement a water-side economizer in the central chilled water plant to make 42°F chilled water directly using the cooling towers when ambient conditions permit. This would be accomplished by installing a new plate & frame heat exchanger sized for a 3,000 ton capacity and the necessary piping, valves, pumps, and controls to automatically switch between mechanical and free-cooling modes. During periods when the ambient wet-bulb temperature is approximately 26°F or less, the condenser water supply temperature set-point can be reset and the cooling towers can be used to make water at 39.5°F, which is sufficient to supply chilled water at 42°F to the campus. While this "economizer" sequence is active, the electric chiller would be shut down and the chilled water load would be met using only pumping and tower fan energy.

Based on information gathered during site visits and piping drawings, it is not practical to isolate the two newer cooling tower cells for use in the free cooling sequence, due to the existing condenser water piping configuration. It was also determined that the older cooling tower cells (#1, 2, 3) require at least 75% of design flow when in free cooling and the cells' performance was found to be a limiting factor in the application of the measure. As a result, plate and frame operation would be limited to periods when the ambient wet bulb temperature is approximately 26°F or less, which corresponds to 1,343 hours per year on average. More run hours could be gained by using a larger plate with a greater flow capacity or by raising the chilled water supply temperature during the winter; however, these options may not be feasible due to the increased cost and physical size of a significantly larger heat exchanger and the requirements on the power plant to maintain a 42°F chilled water supply temperature.

COOLING TOWER OPERATION

Since many of the power plant's process cooling loads have minimum condenser water temperature requirements of approximately 65°F, it is necessary to direct the process and plate & frame condenser water loop flows to separate tower cells while the free cooling sequence is active. It is also necessary to separate the cooling tower sumps, since all six cells share a common sump.

In order to isolate the two loops, up to three tower cells will be needed during plate and frame operation: Tower cells #1 (or #2/#3), #5, and #6. Return water from the process loop will be sent to the hot decks of cells #5 (and/or #6) by opening the corresponding isolation valves. Return condenser water from the plate & frame heat exchanger will be sent to cell #1 (or #2) by opening their isolation valves. Each of the eight valves (2 valves per tower cell) will require new electric actuators so that tower isolation and staging can be controlled automatically. New actuators have not been carried for cells #3 or #4.

In order to isolate the 'warm' (~65°F) and 'cold' (~39.5°F) tower sumps while the plate and frame is active, the existing manual butterfly isolation valve that can separate the sumps of cells #1-3 from the others will be retrofitted with a high-torque electric actuator. The plate and frame heat exchanger will draw low temperature supply water from the basins of cells #1 and #2, while all process cooling water will be supplied from the warmer basin shared between cells #4, #5, and #6.

SEQUENCE OF OPERATION

The following general sequence of operation is proposed for plate and frame operation. In addition, the free cooling sequence should be limited to 1 start per day and should only be enabled when the ambient wet-bulb temperature has been below 26°F for one hour continuously. This temperature set-point is determined based on condenser water flow of 7,230 gpm, a target leaving chilled water set-point of 42°F, a 2.5°F design heat exchanger approach, and an approximate 13.5°F tower approach at 26°F ambient wet bulb.

Starting Waterside Economizer Sequence

1. Divert all condenser water flow associated with the power plant process and Chiller #5 to tower cells #5 and #6 if not already in this state by opening the appropriate tower hot deck isolation valves. Close the isolation butterfly valve in the cooling tower sump to isolate cells #1 and #2 from #3 - #6.
2. Start the condenser water pump for the plate & frame heat exchanger and precool the condenser water loop by setting the fan speed on cells #1 (or #2) to 100%.
3. After the condenser water supply temperature has reached approximately 39.5°F, start the plate and frame heat exchanger by opening the chilled water flow control valves.
4. Modulate the condenser water supply temperature as necessary to maintain the chilled water supply temperature set-point.
5. Shut down Chiller #5, allowing chilled water and condenser water flow to continue to circulate for a predetermined amount of time before closing both isolation valves.

Stopping Waterside Economizer Sequence

If the ambient wet-bulb temperature is greater than set-point or the heat exchanger leaving chilled water temperature is 1°F or more above set-point for a predetermined amount of time, the waterside economizer shall initiate a shutdown sequence.

1. Open chiller flow isolation valves, start associated condenser water and chilled water pumps, and start Chiller #5
2. Once the chiller is maintaining the chilled water supply temperature set-point, isolate the plate and frame by closing its isolation valves.
3. Shut down cooling tower fans until the condenser water supply temperature reaches set-point.

EQUIPMENT LOCATIONS

Due to space limitations within the existing power plant building's footprint, the plate and frame heat exchanger will need to be located outside in an enclosure adjacent to the building.

The proposed location for the heat exchanger is on the south side of the building between the loading dock used to accept Ammonia shipments and the acoustic enclosure containing the combustion turbine's natural gas compressor. The dimensions of the proposed heat exchanger frame are approximately 17 ft long by 3 ft wide by 10 ft tall. Refer to the Appendix for the heat exchanger's specification sheet and dimensions.

ENERGY SAVINGS METHODOLOGY

Energy savings associated with this measure were calculated using an hourly spreadsheet model similar to the baseline model discussed in the previous section "Baseline Model Methodology." The table below summarizes key input parameter differences between the baseline and proposed case models.

Parameter	Base Model	Proposed Model	Units
-	*Value*	*Value*	-
Chilled Water Supply Temperature	42	42	°F
Chilled Water Flow	6,000	6,000	gpm
Chiller/Plate Condenser Water Supply Temperature	~68	39.5	°F
Cooling Tower Sump Temperature (Plate/Process)	na/65	39.5/65	°F
Condenser Water Flow (through chiller/plate)	6,000	9,450	gpm
Process Condenser Water Flow (estimated)	4,800	4,800	gpm
Quantity of Tower Cells with Flow	2	2	-
Quantity of Tower Cells Used for Chiller/Plate CW	1	1	-
Quantity of Tower Cells Used for Process Cooling	2	1-2	-

As shown in the table above, the proposed case condenser water flow for the plate & frame heat exchanger is 7,230 gpm, which is 76% of design flow for a single cooling tower cell (Cell #1, #2, #3, #5, and #5). Maintaining condenser water flow at this level will maintain a sufficiently low temperature gradient within the tower and reduce the possibility of water reaching the freezing point at the bottom of the cell's air inlet face.

Proposed case tower fan speed and energy consumption were modeled using the same capacity curves described in the baseline model methodology. The proposed case condenser water flow, approach temperature, and range were used to calculate the new tower fan speed. Hourly approach and range were calculated using the equations in the table below. For each hour of the year, the model calculates whether or not the plate and frame heat exchanger is active based on the ambient wet-bulb temperature. If the minimum tower approach temperature (as calculated using the formula below) plus the hourly wet-bulb temperature is less than or equal to the target condenser water supply set-point (39.5°F), then the plate would be active. During the first hour of plate operation, the energy model assumed the average tower fan speed for active cell(s) would be 100% in order to pull the condenser water supply temperature down to the set-point necessary for plate operation.

Parameter	Calculation	Units
-	-	-
Cooling Tower Range (ΔT)	= E [Btu/hr] / (500 * Q [gpm])	°F
Approach Temperature	= T,CWST - T,wetbulb	°F
Minimum Approach Temperature	= m * (T,design wetbulb - T,hourly wetbulb) + T,design approach	°F

where:

m = Approach Coefficient (0.5)

As described in the baseline model methodology, the hourly cooling tower cell airflow ratio and fan VFD speed were calculated using performance curves obtained from the eQuest simulation program's library and adjusted for model-specific tower design criteria. The following equation for cooling tower capacity was solved for the airflow ratio variable to determine the hourly airflow ratio required by each cell.

$$Tower\ Capacity = Tower\ Load * \frac{Cap\ f(App, WB)}{Cap\ f(Range, WB)} * Cap\ f(Airflow)$$

where:

$Tower\ Capacity$ = Hourly Heat Rejection Capacity of the Tower Cell [Btu/hour]
$Tower\ Load$ = Hourly Heat Rejection Load on the Tower Cell [Btu/hr]
$Cap\ f(Airflow)$ = Cubic performance curve with dependent variable is cell airflow ratio

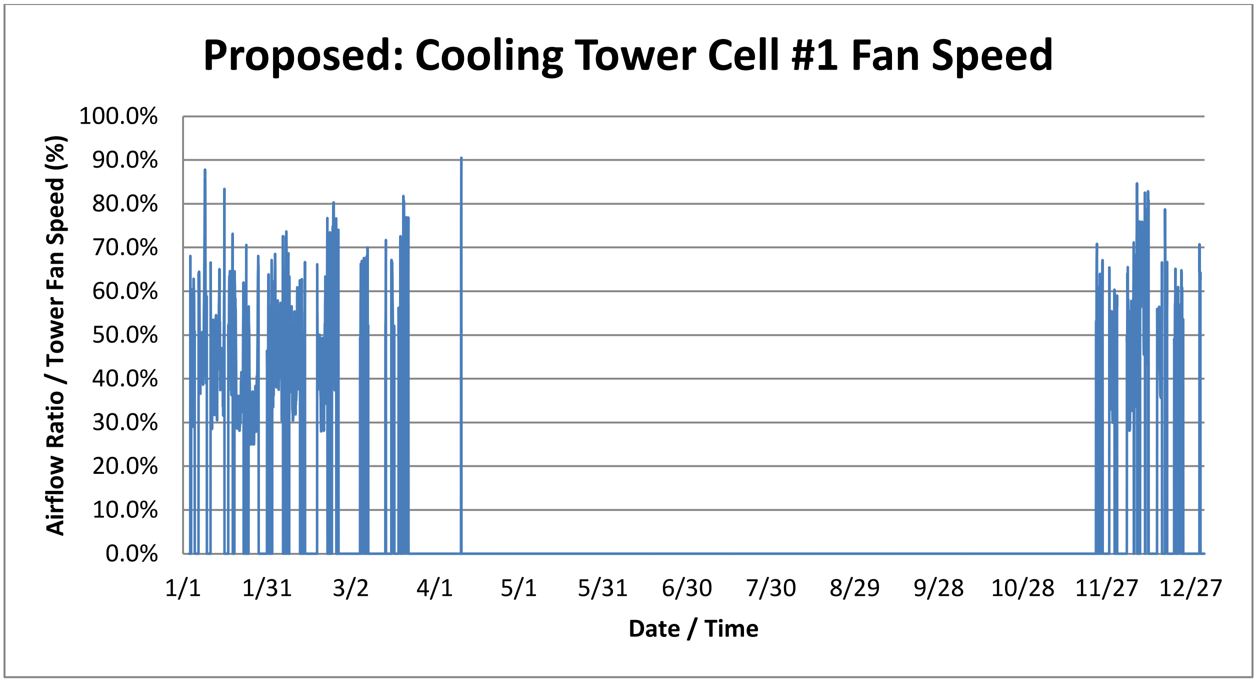
During periods when the calculated airflow ratio required was less than the minimum speed of the tower fan VFD (20%), a cycling factor was calculated to model the hourly fan energy consumption. The calculation used was:

$$Hourly\ Fan\ VFD\ Cycle\ Factor = \frac{Calculated\ Airflow\ Ratio}{Minimum\ Fan\ VFD\ Speed}$$

where:

All variables are ratios between 0 and 1

The chart below shows the calculated proposed case tower fan VFD speed and cycle ratio for Cell #1 during each hour of plate and frame operation. Note that periods when the fan speed is indicated as 0% represents periods when the waterside economizer sequence is disabled.



The charts below show the calculated proposed case tower fan VFD speed and cycle ratio for Cell #6 during each hour of plate and frame operation. This cell will reject heat only from the process condenser water loop while the plate & frame heat exchanger is active.

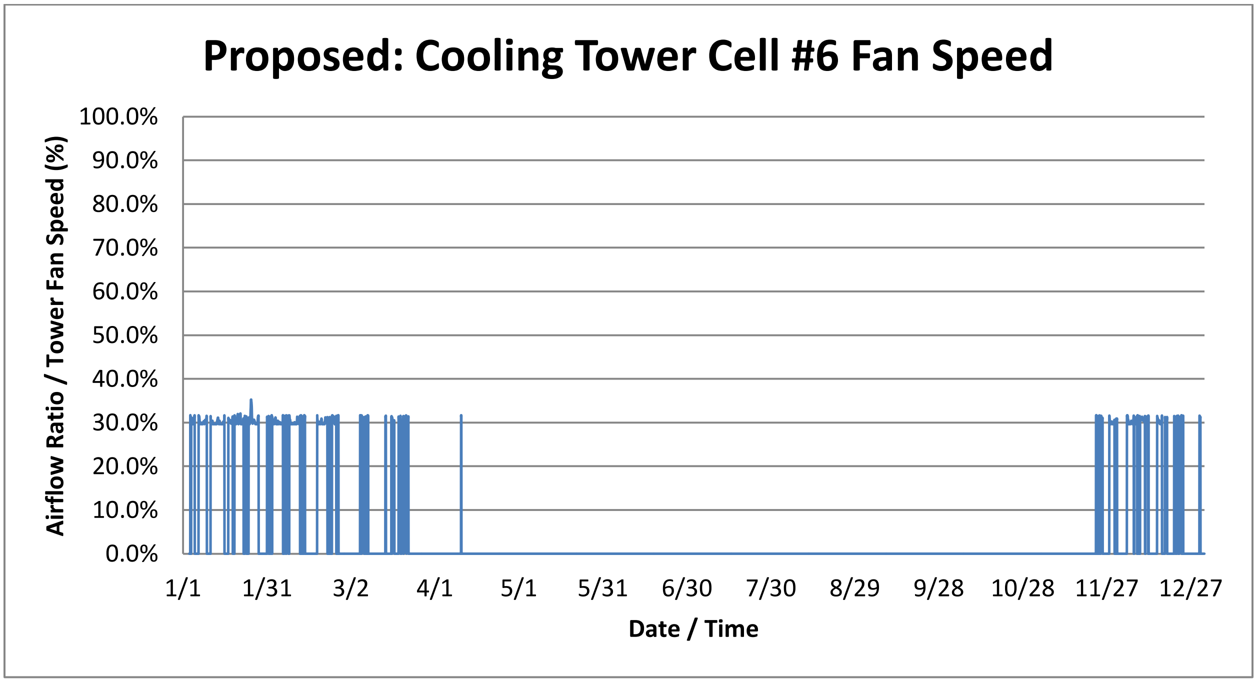


Figure 9: Proposed case tower fan airflow ratio / VFD speed for Cell #6, which rejects heat from the process condenser water loop while the plate & frame heat exchanger is active. The model results indicate that one cell can be used to reject the loop's heat compared to the two cells that are used during the winter in the existing case.

COST ESTIMATE

The cost estimate for this measure is shown in the table on the following page. Refer to the Appendix for the plate and frame heat exchanger selection and vendor quote.

Opinion of Probable Construction Cost

ECM-1: Plate & Frame Heat Exchanger

B2Q Associates, Inc. 100 Burtt Rd. Ste. 212 Andover, MA 01810 (978) 208 - 0609	Customer: UMASS MEDICAL Address: Power Plant	Date: 4/15/2015 Estimated By: SD Checked By:

General			Materials				Labor				
Number	Source	Item	Type	Quantity	Unit Cost	Total Cost	Unit Rate	Workers	Hours Each	Labor Cost	Total Cost
1	2	3,000 ton Plate & Frame Heat Exchanger	ea	1	$154,468	$154,468	$150	4	60	$36,000	$190,468
2	3	Concrete Pad & HX Enclosure	ea	1	$21,000	$21,000	$150	2	12	$3,600	$24,600
3	1	Chiller Isolation Control Valves	ea	2	$3,300	$6,600	$150	2	6	$3,600	$10,200
4	1	CT Control Valve Actuators	ea	8	$2,500	$20,000	$150	2	8	$19,200	$39,200
5	1	CT Sump Isolation Valve Actuator	ea	1	$5,000	$5,000	$150	2	16	$4,800	$9,800
6	2	Condenser Water Pump	ea	1	$36,500	$36,500	$150	2	28	$8,400	$44,900
7	2	Condenser Water Pump Motor (300 hp)	ea	1	$18,830	$18,830	$150	2	12	$3,600	$22,430
8	2	Aegis Split Ring Shaft Grounding Ring	ea	1	$529	$529	$150	1	2	$300	$829
9	2	Chilled Water Pump	ea	1	$31,600	$31,600	$150	2	28	$8,400	$40,000
10	2	Chilled Water Pump Motor (300 hp)	ea	1	$18,830	$18,830	$150	2	12	$3,600	$22,430
11	3	SCADA Control Points	ea	26	$1,500	$39,000	$150	1	8	$31,200	$70,200
12	1	24" Schedule 40 Piping (incl hangers)	lf	100	$301	$30,125	$150	3	1	$45,000	$75,125
13	3	Misc Piping & Hanger Allowance	ea	1	$40,000	$40,000	$150	4	40	$24,000	$64,000
14	1	2" Thickness Insulation (24" Pipe)	ea	100	$23	$2,250	$150	1	0.5	$7,500	$9,750
15	3	Staging Sequence Programming	ea	1		$0	$150	2	100	$30,000	$30,000
16	3	Contractor Commissioning	ea	1		$0	$150	1	32	$4,800	$4,800
17	3	As-Built	ea	1		$0	$150	1	32	$4,800	$4,800

Sources	
1	Means
2	Vendor Quote
3	Other
4	Vendor Allowance

	Subtotal	$663,532
Contingency	20%	$132,800
Engineering	15%	$119,500
Construction Administration	5%	$39,900
Commissioning	5%	$39,900
Construction Observation	10%	$79,700
Project Closeout & Expenses	5%	$39,900
	Total	$1,115,232

APPENDIX

CHILLER 5 PERFORMANCE CURVES

The chart below illustrates chiller compressor motor input power as a function of part load ratio and entering condenser water temperature. All performance is specified at 42°F leaving chilled water temperature, 6,400 gpm evaporator water flow, and 8,000 gpm condenser water flow. The table on the following page lists the chiller's performance data in tabular format. The values are in the form of EIR (Energy Input Ratio), and have been normalized to ARI Standard Conditions (44°F leaving chilled water temperature, 85°F entering condenser water temperature).

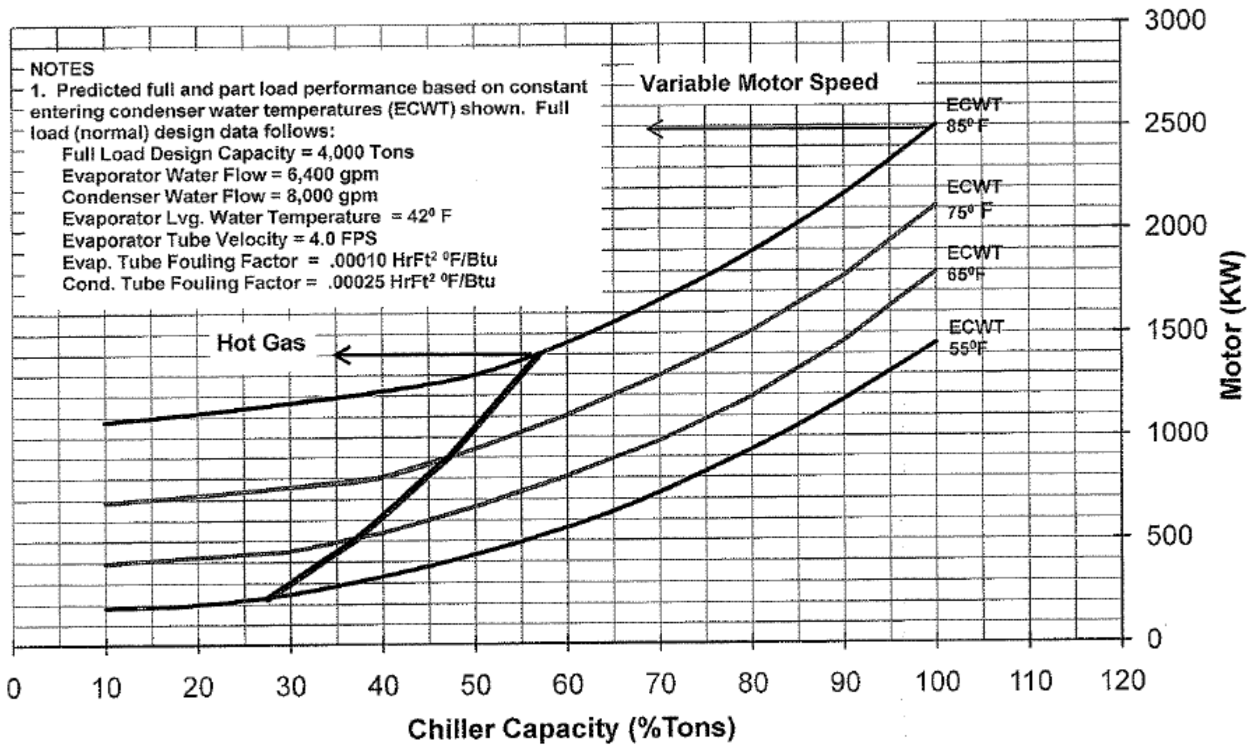


Figure 10: Chart of Chiller 5 Performance Data

Table 5: Chiller 5 part load and off-design performance data, normalized to ARI Standard Conditions.

EIR Ratio (Normalized to ARI Conditions)				
Load Ratio	55	65	75	85
10%	0.7246	1.6111	2.7876	4.3560
15%	0.5007	1.1032	1.8988	2.9559
20%	0.3974	0.8577	1.4629	2.2557
25%	0.3558	0.7036	1.1946	1.8424
30%	0.3281	0.6065	1.0214	1.5669
35%	0.3277	0.5713	0.8977	1.3750
40%	0.3318	0.5406	0.8176	1.2354
45%	0.3425	0.5395	0.7819	1.1267
50%	0.3476	0.5351	0.7602	1.0500
55%	0.3611	0.5378	0.7486	1.0028
60%	0.3753	0.5457	0.7418	0.9748
65%	0.3951	0.5550	0.7439	0.9616
70%	0.4193	0.5630	0.7457	0.9551
75%	0.4472	0.5790	0.7518	0.9496
80%	0.4716	0.5973	0.7572	0.9511
85%	0.4971	0.6255	0.7759	0.9584
90%	0.5236	0.6505	0.7926	0.9687
95%	0.5562	0.6908	0.8218	0.9852
100%	0.5839	0.7052	0.8229	1.0000

Chiller Design Operating Load Conditions		
DESIGN OPERATING CONDITIONS		ENGLISH
Chiller:	Design Load	4000 Tons
	Efficiency (w/o liquid Inj.)	0.625 kW/Ton
	Sound level (acoustically treated)	88.0 dBA @ 100% load
Evaporator:	Water Ent. & Lvg. Temperatures	57 °F to 42 °F
	Water Flow	6,400 gpm
	Water Diff. Press. (nozzle – nozzle)	14.1 ft. (water)
	Tubeside Fouling Factor	0.00025 ft²-h-°F/Btu
	Refrigerant 134a Temperature	40.8 °F
	Surface area	11,942 ft²
	Refrigerant 134a Pressure	50.5 psia
Condenser:	Water Ent. & Lvg. Temperatures	85 °F to 99 °F
	Water Flow	8,000 gpm
	Water Diff. Press. (nozzle – nozzle)	17.8 ft. (water)
	Tubeside Fouling Factor	0.0001 ft²-h-°F/Btu
	Refrigerant 134a Temperature	99.5 °F
	Surface area	56,202 ft²
	Refrigerant 134a Pressure	137.8 psia
	Refrigerant 134a Subcooled Temp.	86.53 °F
Compressor:	Suction & Discharge Temperatures	40.4 °F to 99.5 °F
	Full Load Rotational Speed	3434 rpm
	Shaft Power w/o liquid injection	3125 HP
	Shaft Power with liquid injection	3195 HP
Motor:	Full Load Input Power	2501 KW
	Full Load Amps	380 FLA
	Max. Input Power based on NP data	2802 kW
	Max. Full Load Amps based on NP data	408 FLA
	Full Load Rotational Speed	3585 rpm
	Electrical Requirements	4,160 volts 3 ph 60 hz
Controls:	Electrical Load	5 kVA
	Electrical Power	115 volts 1 ph 60 hz
	Air Consumption	4 - 5 scfm
	Air Pressure Requirement	90 psig
Intercooler	Refrigerant 134a Temperature	71.3 °F
	Refrigerant 134a Pressure	87.7 psia

Figure 11: Table of Design performance criteria for Chiller #5

CHILLER 1,2,3 DESIGN PERFORMANCE

Selection Summary

*** York Model OM Units are rated in accordance with ARI Standard 550/590 ***
Materials and construction per Mechanical Specifications - Form 160.72-EG1

Description	Units	Data	Description	Units	Data
Specified Capacity	Tons	2470	Sound	dBA	-----
Condensing Steam Turbine	Bhp	2307	Refrigerant	-----	R-134A
Bhp / Capacity	Bhp/Ton	0.934	Charge	Lb	6833
NPLV	Bhp/Ton	0.906			

Liquid Injection = NO

Heat Exchanger Data	Units	Evaporator	Condenser
Flow	GPM	3750	7500
Fluid	-----	WATER	WATER
Concentration	% Wt	0.0	0.0
Entering Fluid Temp.	Deg F	57.8	85.0
Leaving Fluid Temp.	Deg F	42.0	94.5
Fouling Factor	HrSqFt Deg F/Btu	0.00010	0.00025
Velocity in Tubes	Ft/Sec	7.8	9.2
Pressure Drop	Ft	28.1	20.0
Saturated Refrigerant Temp.	Deg F	38.0	98.8
Shell Data			
Inside Diameter (ID)	In	62	52
Nominal Tube Length	Ft	16	16
Tube Count	-----	1910	2162
Tube Identification No.	-----	3	3
Passes	-----	3	2
Type of Waterbox	Marine	Integral	Integral
Water Connection Size	In	-----	-----

Description	Units	Data	Description	Units	Data
Subcooler	-----	No	Compressor Model	-----	238.A8.A6
Intercooler Type	-----	Horizontal	RPM	-----	5385
Intercooler Size	-----	4000	Thrust Bearing	-----	Tilting Pad
Suction Line Size	In	14	Coupling Size	In	2.500
Discharge Line Size	In	10			
Interstage Line Size	In	-----	**Refrig Pumpout Unit**		
Auto Hotgas Valve	-----	4	Transfer Unit (RTU-10)	Hp	10
			Receiver (D x L)	In x Ft	42x16

Weight (lb)	Evaporator	Condenser	Intercooler	Driveline	Misc	Total	Pumpout
Shipping	42599	35131	6162	25224	7477	116594	4550
Operating	99047	< included	8445	27056	4456	139004	13934

Dimensions (ft)	Length	Width	Height	Tube Pull (add to length)
Chiller Size (Excludes Pumpout Unit)	-----	-----	-----	-----
Refrigerant Pumpout Unit	16-2	3-6	7-8	-----

Non-Standard Tube Count - RATING ONLY

Partload Report

* York Model OM Units are rated in accordance with ARI Standard 550/590 *

Description	Units	Evaporator	Condenser
Flow	GPM	3750	7500
Fluid	-----	WATER	WATER
Concentration	% Wt	0.0	0.0
Fouling Factor	HrSqFt Deg F/Btu	0.00010	0.00025
Leaving Water Temp	Deg F	42.0	94.5

$$NPLV = \frac{1}{\frac{0.01}{A} + \frac{0.42}{B} + \frac{0.45}{C} + \frac{0.12}{D}}$$

Where:

Bhp/Ton	%Capacity	Weight
A	100	0.01
B	75	0.42
C	50	0.45
D	25	0.12

ECWT Deg F	Load %	Capacity Tons	Fluid Evap Entering Deg F	Fluid Cond Leaving Deg F	Refrig ETP Deg F	Refrig CTP Deg F	Condensing Steam Turbine Input Bhp	Bhp/Ton	NPLV Wgt. Tons/Bhp
85.00	100.0	2470	57.80	94.46	37.97	98.80	2307	0.934	0.011
75.00	75.0	1852	53.85	82.09	38.81	85.23	1498	0.809	0.519
65.00	50.0	1235	49.90	69.73	39.66	71.74	1110	0.899*	0.501
65.00	25.0	618	45.95	67.36	40.57	68.37	1017	1.648*	0.073
								NPLV =	0.906

CHILLER 4 DESIGN PERFORMANCE

DESIGN LOAD CONDITIONS

1. LOAD 5000 TONS
2. COOL 8000 G.P.M. OF WATER FROM 56.0°F TO 41.0°F.
3. CONDENSER 12000 G.P.M. OF WATER FROM 85.0°F TO 96.8°F.
4. COMPRESSOR SHAFT HORSEPOWER 4117
5. COMPRESSOR FULL LOAD SPEED 3702 R.P.M.
6. APPROX. EVAPORATOR TEMP. 38.5°F.
7. APPROX. CONDENSER TEMP. 101.7°F.
8. EVAPORATOR LIQUID PRESS. DROP 29.3 FT. WATER
9. CONDENSER WATER PRESS. DROP 6.0 FT. WATER
10. DESIGN TUBE SIDE FOULING FACTOR EVAPORATOR 0.00010 HR-FT^2-°F/BTU
11. DESIGN TUBE SIDE FOULING FACTOR CONDENSER 0.00025 HR-FT^2-°F/BTU
12. CONTROL POWER REQUIREMENTS: ELECTRIC 7.5 KVA, 120 V., 1 PH., 60 HZ.
    PNEUMATIC 5 S.C.F.M. AT 80 TO 100 PSIG
13. AUX. OIL PUMP POWER REQUIREMENTS: COMPR. 3.0 H.P. / TURBINE 3.0 H.P. } 460 V., 3 PH., 60 HZ.
    TOTAL 6.0 H.P.
15. AUX. WATER REQUIREMENTS OIL COOLERS: COMPRESSOR 26.0 G.P.M., 4.6 FT. P.D.
    ENTERING WATER TEMP. 85.0°F — TURBINE 25.5 G.P.M., 4.1 FT. P.D.
    TUBE SIDE FOULING FACTOR 0.0005 HR-FT^2-°F/BTU — TOTAL 51.5 G.P.M.
16. TURBINE STEAM REQUIREMENTS: 62,784 LBS./HR. CLEAN AND DE-AERATED WITH INLET AT 50 PSIG AND 310°F, EXHAUST 3.0 IN. Hg. ABSOLUTE
17. BRAKE HORSEPOWER / CAPACITY: 0.823 BHP/TON

PLATE & FRAME HEAT EXCHANGER SELECTION

Bell & Gossett GPXTM

Gasketed Plate Heat Exchanger Specification Sheet

175 Standard Parkway
Cheektowaga, New York 14227
1-800-447-7700
www.bellgossett.com

Customer		Date	Thursday, February 05, 2015
Inquiry Number	2015-2-5-CR	Item Number	

Performance of One Unit: P188 PN: BY5427 **Units Connected in Parallel:** 1

Fluid Name	Water	Water
Total Flow	7,230.0 GPM	7,230.0 GPM
Inlet Temperature	52.0 °F	39.5 °F
Outlet Temperature	42.0 °F	49.5 °F
Operating Pressure	0.0 PSIG	0.0 PSIG
Pressure Drop, Allow./Calc	10.0/10.0 PSIG	10.0/10.0 PSIG
Density	62.4 lb/ft3	62.4 lb/ft3
Viscosity	1.3 cp	1.4 cp
Specific Heat	1.00 Btu/lbm,°F	1.00 Btu/lbm,°F
Thermal Conductivity	0.33 Btu/ft,h,°F	0.33 Btu/ft,h,°F
Specified Fouling Factor	0.00000 hr,ft2,°F/Btu	0.00000 hr,ft2,°F/Bt

Total Heat Exchanged	36,307,634.7 Btu/h
LMTD	2.5 °F
Overall Heat Transfer Coefficient, Clean/Dirty	1,020.2/1,020.2 Btu/hr,ft2,°F
Overall Heat Transfer Coefficient, Service	1,002.3 Btu/hr,ft2,°F
Effective Surface Area	14,472.7 ft2
Excess Surface	1.8 %

Construction

Number of Passes * Channels	1*343	1*344
Total Number of Plates	688	
Pressure, Design/Test	150/195(PSIG)	150/195(PSIG)
Design Temperature, min/max	32/284(°F)	32/284(°F)
Internal Volume	88.4(ft3)	88.7(ft3)
Inlet Connection(Location) flange	F1, steel studded port for 150# ansi flange 12.0"	F3, steel studded port for 150# ansi 12.0"
Outlet Connection(Location) flange	F4, steel studded port for 150# ansi flange 12.0"	F2, steel studded port for 150# ansi 12.0"

Plate Material	304
Plate Thickness	0.5 mm
Plate Mix	TKTL-48
Gasket Material	NITRILE HT
Empty/Flooded Weight	20,949 / 32,005 lb
Frame Size / Max. Frame Capacity	196.9 inch / 859 plates
Approvals	ASME Sect VIII Div 1 w/U stamp.

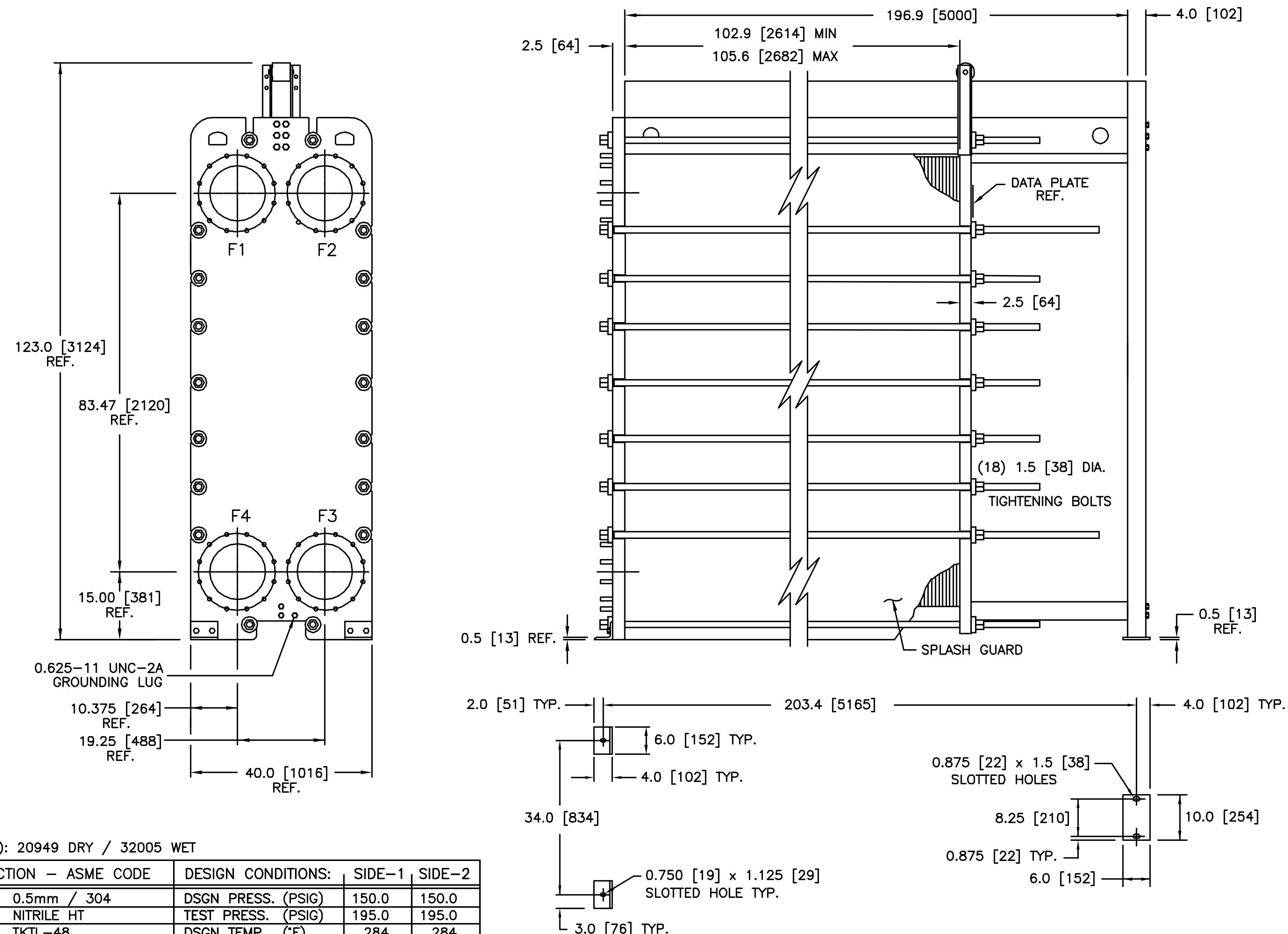
Note: Customer to verify fluid/material compatibility.

Performance evaluation is dependent on customers' ability to provide sufficiently accurate measurements.

Version No:
0001/01/01

NOTE(s): 1. Dimensions shown in inches (in.)

REV NO.	REVISION NOTE	REV BY	REV DATE



EST. WEIGHT (LBS): 20949 DRY / 32005 WET

UNIT CONSTRUCTION – ASME CODE	
THERMAL PLATES:	0.5mm / 304
GASKET TYPE:	NITRILE HT
PLATE MIXTURE:	TKTL–48
MIX (SIDE 1):	1 x 343
MIX (SIDE 2):	1 x 344
PLATE QUANTITY:	688 / 859 MAX
CARRY BAR LGTH:	5000mm
CARRY BAR MTL:	STL
TIE BOLTS:	ZINC PLATED CARBON
SPLASH GUARD:	ALUMINUM

DESIGN CONDITIONS:	SIDE–1	SIDE–2
DSGN PRESS. (PSIG)	150.0	150.0
TEST PRESS. (PSIG)	195.0	195.0
DSGN TEMP. (°F)	284	284
MIN. TEMP. (°F)	32	32

PORT IDENTIFICATION		CONNECTION – TYPE AND DESCRIPTION
F1	SIDE–1 IN	12" PORT, STUDDED FOR 150# ANSI FLANGE, STEEL UNLINED
F4	SIDE–1 OUT	12" PORT, STUDDED FOR 150# ANSI FLANGE, STEEL UNLINED
F3	SIDE–2 IN	12" PORT, STUDDED FOR 150# ANSI FLANGE, STEEL UNLINED
F2	SIDE–2 OUT	12" PORT, STUDDED FOR 150# ANSI FLANGE, STEEL UNLINED

MADE: ITT 02/05/15
CHK: HT 02/05/15

NA
ORDER NO.

Bell & Gossett
Buffalo, NY 14227 USA

MODEL P188

2015–2–5–CR –00
REV.

COOLING TOWER DESIGN CRITERIA

CELLS #1, #2, #3

First Name: Univ. of Last Name: Massachusetts - Medical Center

Street: 55 Lake Street

City: Worcester ST: MA Zip: 01605 Terr: 6

Contact: Steve Blair Phone: 508-856-2153

Order#: 12-0065-70 Model#: 6611-03

GPM: 28350 HW: 103 CW: 85 WB: 75

HP: 50.0 Volts: 460/3/60 SPD: 2 WND: SF:

Encl: TEFC Gear: 34.3T Ratio: 0 Shaft/cplng: 225

Fan: 264H312 Pitch: Noz:

Fill: P Elims: P

Proposal Data Sheets

Customer:	Manufacturer:
Skanska University of Massachusetts Medical School Worcester, MA	SPX Cooling Technologies 7401 West 129 Street Overland Park, KS 66213

All data set forth herein is in accordance with definitions and standards published by the Cooling Technology Institute.

GENERAL:

Selection	Class F400
Marley Model	F466-4.0-2
Tower Type	Industrial Fiberglass Counterflow Mechanical Draft

DESIGN & OPERATING CONDITIONS:

Circulating Water Flow	18,900 gpm
Hot (Inlet) Water Temp.	100 °F
Cold (Outlet) Water Temp.	85 °F
Wet Bulb Temp., Inlet	75 °F
Tower Pump Head	32 ft
Total Fan Brake Horsepower, (Driver Output)	385 Hp
Drift Loss, % of Circulating Flow	0.0005 %
Evaporation Loss (at Design)	257 gpm
Design Wind Load	100 mph
Design Seismic Load	Per Spec.
Tower Site (Ground Level, Roof, etc.)	Ground Level
Elevation Above Sea Level	480 ft
Tower Exposure	One Side Louvered and one side, two ends closed

STRUCTURAL DETAILS:

Number of Cells	2
Fans per Cell	1
Total Number of Fans	2
Nominal Cell Dimension, LxW (no sound attn)	36 ft x 36 ft
Overall Tower Dimension, LxW (no sound attn)	72.67 ft x 39.17 ft
Overall Tower Dimension, LxW (with sound attn)	72.67 ft x ~ 43 ft
Height-Basin Curb to Fan Deck	38.59 ft
Fan Stack Height	12 ft
Overall Tower Height	50.59 ft
Inside Basin Dimensions, LxW	73 ft x 37 ft

COOLING TOWER INQUIRY & BID FORM

Mechanical Draft Water Cooling Tower

Customer:	Manufacturer:
University of Massachusetts Medical Center Cooling Tower Expansion Worcester, Massachusetts	The Marley Cooling Tower Company 7401 W 129 Street Overland Park, KS 66213
Inquiry No.: 98-8420-002-40-0-D	**Proposal No.:** 1780
Date: **Rev.:**	**Date:** June24 , 1998 **Rev.:** 1

All data set forth herein is in accordance with definitions and standards published by the Cooling Tower Institute.

GENERAL:

Selection	
Tower Model	6610-0-1
Tower Type	Industrial Wood Crossflow Mechanical Draft

DESIGN & OPERATING CONDITIONS:

Circulating Water Flow, U.S. GPM	8065
Hot (Inlet) Water Temp. deg F	103.5
Cold (Outlet) Water Temp. deg F	85.0
Wet Bulb Temp. deg F., Inlet	75.0
Wet Bulb Temp. deg F., Ambient	
Tower Pump Head, Ft.	Later
Total Fan BHP, (Driver Output)	150.0
Drift Loss, % of Circulating Flow	0.005%
Evaporation Loss (at Design), U.S. GPM	137
Design Wind Load, Lbs./Sq.Ft.	30 PSF
Design Wind Load, Mi./Hr.	100 MPH
Design Seismic Load	UBC Zone 2A
Tower Site (Ground Level, Roof, etc.)	Ground Level
Elevation Above Sea Level, Ft.	0
Tower Exposure	Enclosed by Louvered Wall

STRUCTURAL DETAILS:

Number of Cells	1
Fans per Cell	One (1)
Total Number of Fans	1
Nominal Cell Dimension, LxW, Ft.	**36' x 47'**
Overall Tower Dimension, LxW Ft.	**36' x 47'**
Height-Basin Curb to Fan Deck, Ft.	**41'-1 1/4**
Fan stack height, Ft.	**12'-0**
Overall Tower Height, Ft.	**53'- 1/1/4**
Inside Basin Dimensions, LxW Ft.	37' x 31'-4